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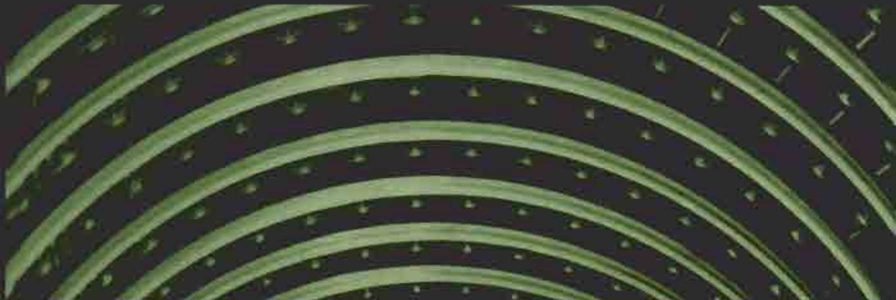
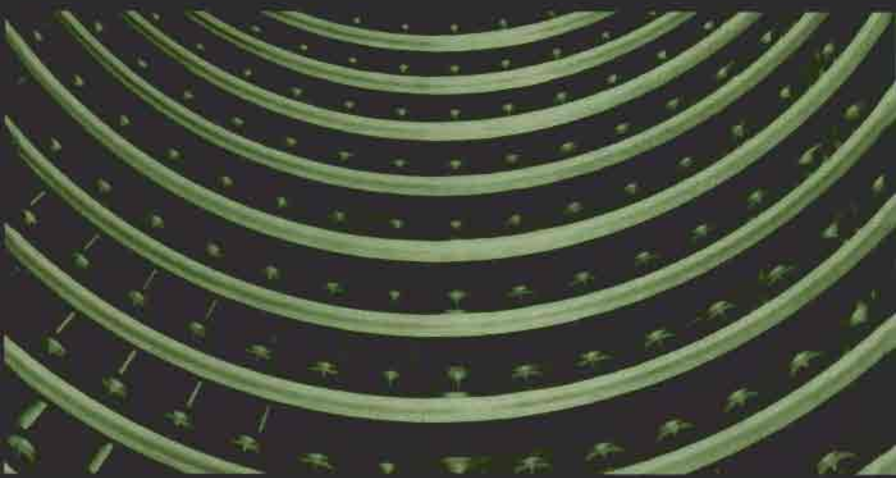


COMPRESSORS

SELECTION AND SIZING



ROYCE N. BROWN

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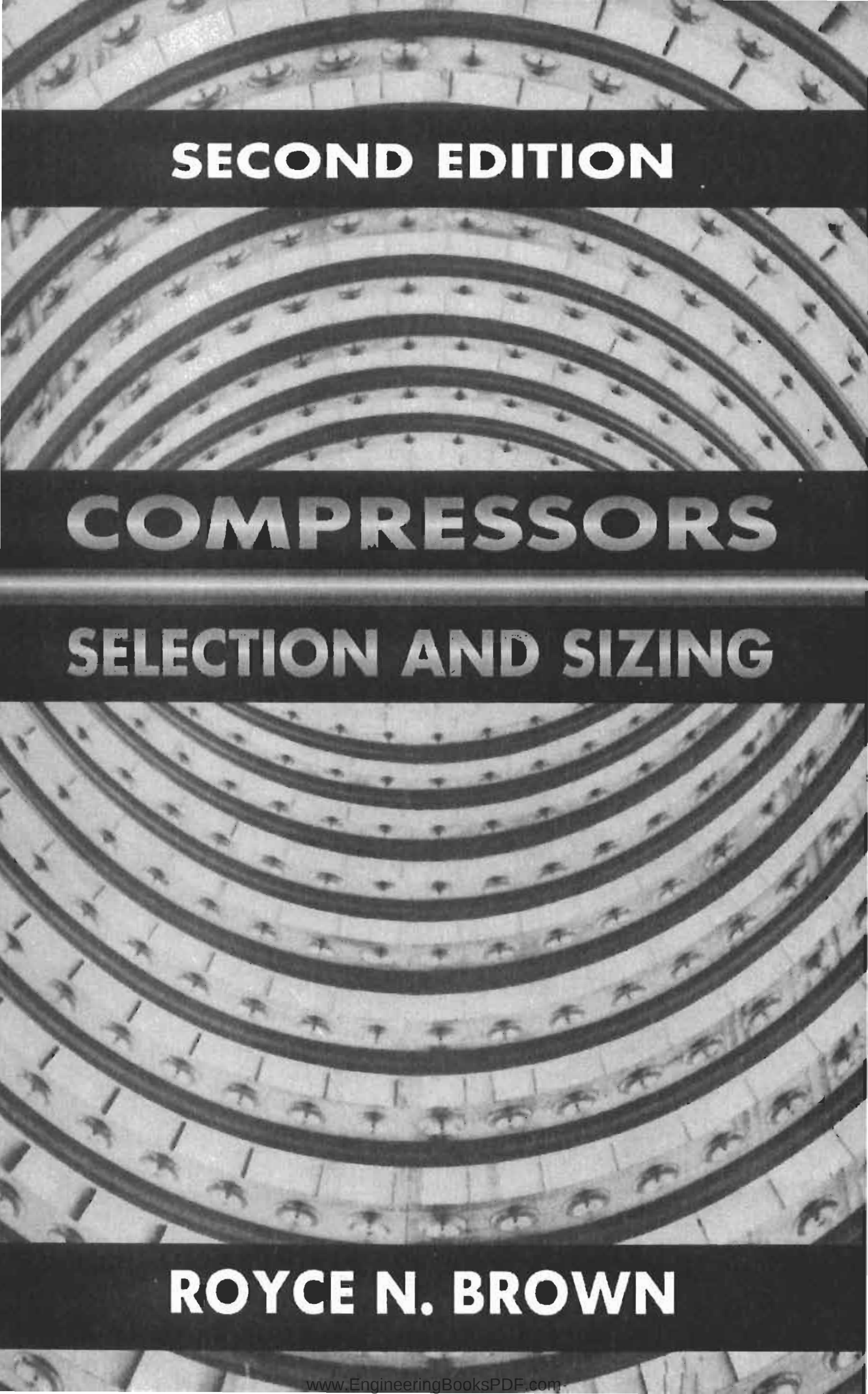

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COMPRESSORS

SELECTION AND SIZING

ROYCE N. BROWN

Dedication

To June,
for her love and encouragement
to keep me moving.

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Preface to Second Edition

About the time the first edition was written, the process industries, which represent a large part of the compressor market, were at a low ebb. As a result, the activity in the compressor world was almost at a standstill. Development at best was relatively slow. Currently, however, activity level has increased significantly. A look at the credit lines on many of the suppliers will tell of the many changes that have taken place. Even many of the companies whose names have not changed are now under different ownership than they were at the time of the first edition. Large investments have been made in facilities, in terms of new or remodeled factory buildings and the addition of new improved machine tools. Development funds are being expended and improved designs are becoming available. Management styles have changed and the theme of continuous improvement is quite prevalent. With all this activity, it seemed appropriate to offer an updated edition of this book.

Many of the readers of the first edition have commented that the book was easy to read. I have attempted to maintain that tone in this new edition. The major change to the book is the addition of a chapter on reliability. As in the other chapters, this one also leaves the high power statistics for someone else and instead uses a "common sense" approach. It probably has a "do and don't" flavor, which just seemed appropriate as I was writing it. Because the subject of reliability is so important and so much can be written about it, the chapter had to be limited to what I felt was the more pertinent information. I had to remind myself that the subject of the book was compressors, not just their reliability. It is hoped that a proper balance was obtained.

Another area that is addressed in the new edition is the dry gas seal. The subject of dry gas seals, which are now widely used by the industry, was expanded considerably in Chapter 5, and a discussion of dry gas seal systems has been added to Chapter 8. Also in Chapter 5, I added a section on magnetic bearings, which are emerging in the industry although they are not as quick to catch on. Chapter 8 expands the discussion of dry flexible element couplings to reflect current industry practice. The section on gear couplings was left because gear couplings are still used and I felt the information would provide some useful background.

I touched up some of Chapter 3 by reworking the valve section, and I hope it does a better job of describing the currently available valves. I also expanded the area of unloaders to more adequately cover the different styles available to the industry.

Where current practice seemed to dictate I updated curves, and added a table in Chapter 4 to help with the sizing of the oil-free helical lobe compressors. Instrumentation was updated to take rod-drop monitoring of reciprocating compressors into consideration. Improvements in torque monitoring are also included.

In general, wherever I felt the organization of the material could be improved, I did it. The most notable of this are the changes to the testing chapter to aid in clarity.

Royce N. Brown

Acknowledgments Second Edition

I would like to thank Alex and Linda Atkins of Alta Systems for coming to my assistance when I got overloaded with the chore of scanning my photographs and line illustrations. They helped get the illustrations organized and kept them in the proper order. Linda also helped with debugging the text and keeping the format consistent. Alex put the finishing touches on the figures and then put them on a CD Rom so they could be transported to the publisher. They were very flexible and made themselves available to fit my schedule.

I also want to thank Dan Beard and his son Sean for computer support and some tedious image editing.

Thanks go to Brown and Root for scanning the first edition, and for giving me an electronic form on which to build the revised edition. Thanks also to Buddy Wachel of EDI for giving me an assist at the reciprocating compressor acoustics, and to Susan Dally, Terryl Matthews, Rick Powell, Kelly Fort, Rich Lewis, Carl Fredericks, and Mary Rivers of Dow Chemical for their reviews of the revised chapters.

Finally, a sincere thanks to all the suppliers who provided material for the figures.

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1

Overview

Introduction

A compressor is a device used to increase the pressure of a compressible fluid. The inlet pressure level can be any value from a deep vacuum to a high positive pressure. The discharge pressure can range from subatmospheric levels to high values in the tens of thousands of pounds per square inch. The inlet and outlet pressure are related, corresponding with the type of compressor and its configuration. The fluid can be any compressible fluid, either gas or vapor, and can have a wide molecular weight range. Recorded molecular weights of compressed gases range from 2 for hydrogen to 352 for uranium hexafluoride. Applications of compressed gas vary from consumer products, such as the home refrigerator, to large complex petrochemical plant installations.

The compressors to be covered in this book are those using mechanical motion to effect the compression. These types of compressors are commonly used in the process and gas transport/distribution industries. A partial list of these industries includes chemical, petrochemical, refinery, pulp and paper, and utilities. A few typical applications are air separation, vapor extraction, refrigeration, steam recompression, process and plant air.

Compression Methods

Compressors have numerous forms, the exact configuration being based on the application. For comparison, the different types of compressors can be subdivided into two broad groups based on compression mode. There are two basic modes: intermittent and continuous. The *intermittent* mode of compression is cyclic in nature, in that a specific quantity of gas is ingested by the compressor, acted upon, and discharged, before the cycle is repeated. The *continuous* compression mode is one in which the gas is moved into the compressor, is acted upon, moved through the compressor, and discharged without interruption of the flow at any point in the process.

Compressors using the intermittent compression mode are referred to as positive displacement compressors, of which there are two distinct types: reciprocating and rotary. Continuous-mode compressors are also characterized by two fundamental types: dynamic and ejector.

This chapter will give a brief overview of each of the different compressors commonly used in the process industries. Subsequent chapters will then cover each of the mechanical types in depth. (The ejector, which does not use mechanical action, will not be covered in detail.) Figure 1-1

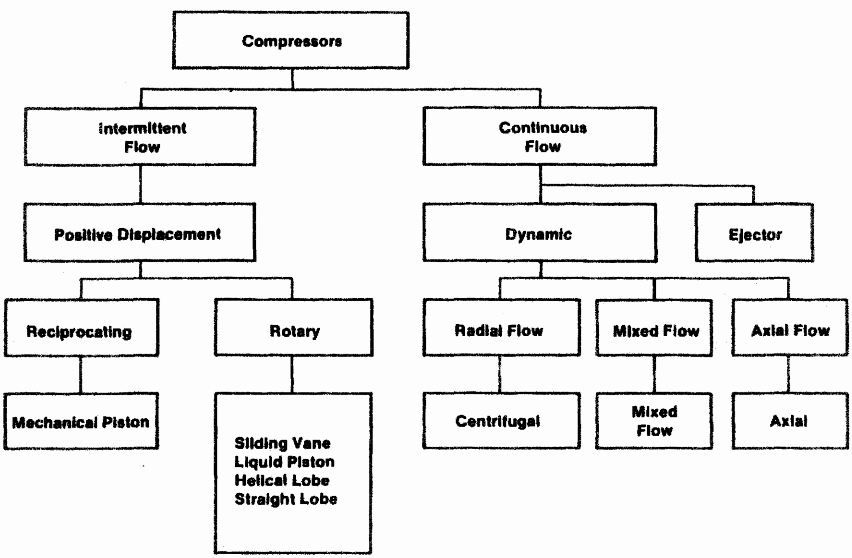

Figure 1-1. Chart of compressor types.

diagrams the relationship of the various compressors by type. Figure 1-2 shows the typical application range of each compressor, and Figure 1-3 compares the characteristic curves of the dynamic compressors, axial and centrifugal, with positive displacement compressors.

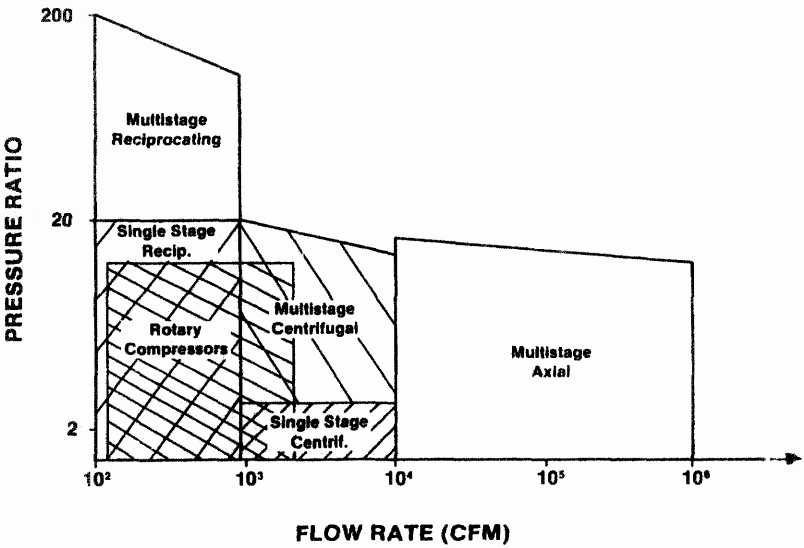

Figure 1-2. Typical application ranges of compressor types.

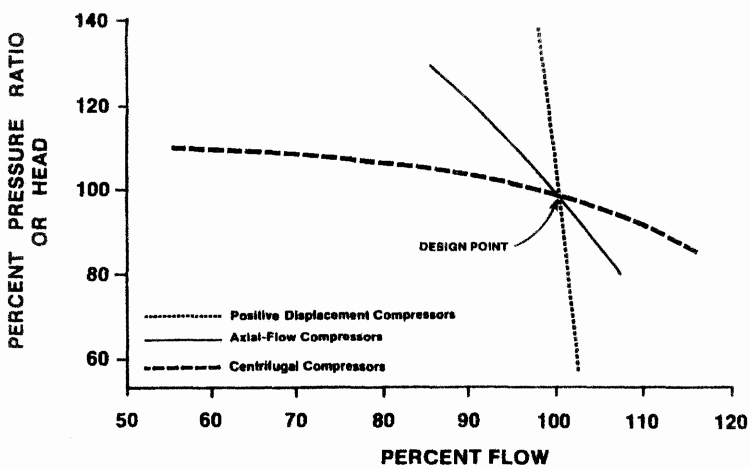

Figure 1-3. General performance curve for axial flow, centrifugal, and positive displacement.

Intermittent Mode Compressors

Reciprocating Compressors

The reciprocating compressor is probably the best known and the most widely used of all compressors. It consists of a mechanical arrangement in which reciprocating motion is transmitted to a piston which is free to move in a cylinder. The displacing action of the piston, together with the inlet valve or valves, causes a quantity of gas to enter the cylinder where it is in turn compressed and discharged. Action of the discharge valve or valves prevents the backflow of gas into the compressor from the discharge line during the next intake cycle. When the compression takes place on one side of the piston only, the compressor is said to be single-acting. The compressor is double-acting when compression takes place on each side of the piston. Configurations consist of a single cylinder or multiple cylinders on a frame. When a single cylinder is used or when multiple cylinders on a common frame are connected in parallel, the arrangement is referred to as a *single-stage compressor.* When multiple cylinders on a common frame are connected in series, usually through a cooler, the arrangement is referred to as a *multistage compressor.* Figures 1-4 and 1-5 are typical reciprocating compressor arrangements, beginning with the single-stage and ending with a more complex multistage.

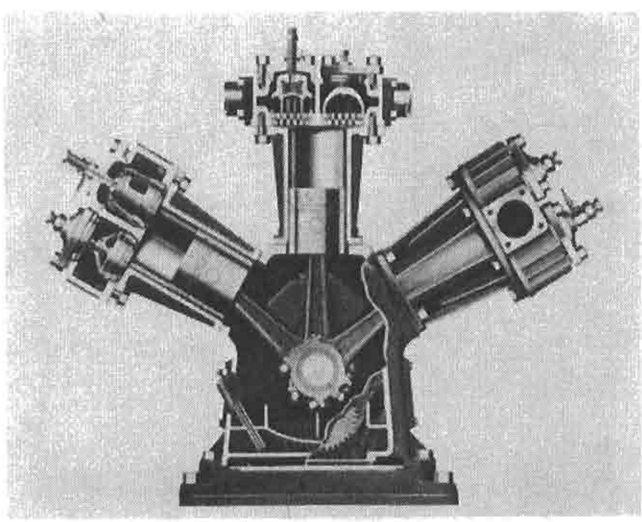

Figure 1-4. A three-stage single-acting reciprocating compressor. (*Courtesy of Ingersoll Rand*)

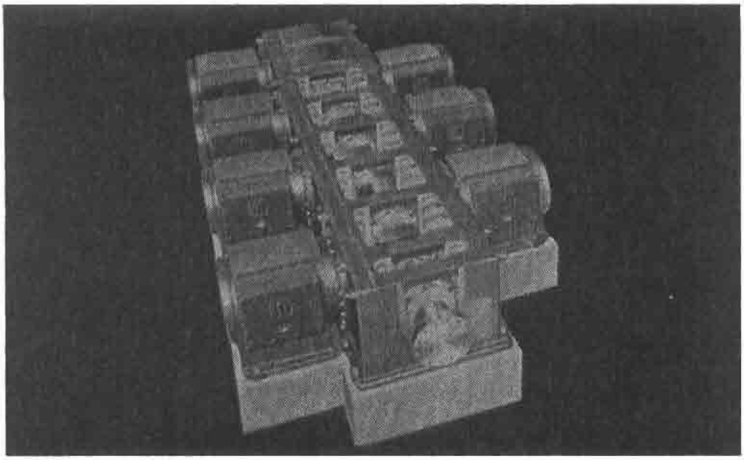

Figure 1-5. Cutaway of the frame end of a large multistage reciprocating compressor. (*Courtesy of Dresser-Rand*)

The reciprocating compressor is generally in the lower flow end of the compressor spectrum. Inlet flows range from less than 100 to approximately 10,000 cfm per cylinder. It is particularly well-suited for high-pressure service. One of the highest pressure applications is at a discharge pressure of 40,000 psi. Above approximately a 1.5-to-1 pressure ratio, the reciprocating compressor is one of the most efficient of all the compressors.

Rotary Compressors

The rotary compressor portion of the positive displacement family is made up of several compressor configurations. The features these compressors have in common are:

1. They impart energy to the gas being compressed by way of an input shaft moving a single or multiple rotating element.
2. They perform the compression in an intermittent mode.
3. They do not use inlet and discharge valves.

The helical and spiral-lobe compressors are generally similar and use two intermeshing helical or spiral lobes to compress gas between the lobes and the rotor chamber of the casing. The compression cycle begins as the open part of the spiral form of the rotors passes over the inlet port and traps a quantity of gas. The gas is moved axially along the rotor to the discharge port where the gas is discharged into the discharge nozzle of the casing. The volume of the trapped gas is decreased as it moves toward the outlet, with the relative port location controlling the pressure ratio. Figure 1-6 shows a cutaway view of a helical-lobe compressor. The spiral-lobe version is the more limited of the two and is used only in the lower pressure applications. Therefore, only the helical-lobe compressor will be covered in depth in this book (see Chapter 4).

The helical-lobe compressor is further divided into a dry and a flooded form. The dry form uses timing gears to hold a prescribed timing to the relative motion of the rotors; the flooded form uses a liquid media to keep the rotors from touching. The helical-lobe compressor is the most sophisticated and versatile of the rotary compressor group and operates at the highest rotor tip Mach number of any of the compressors in the rotary family. This compressor is usually referred to as the "screw compressor" or the "SRM compressor."

The application range of the helical-lobe compressor is unique in that it bridges the application gap between the centrifugal compressor and the reciprocating compressor. The capacity range for the dry configuration is approximately 500 to 35,000 cfm. Discharge pressure is limited to 45 psi in single-stage configuration with atmospheric suction pressure. On

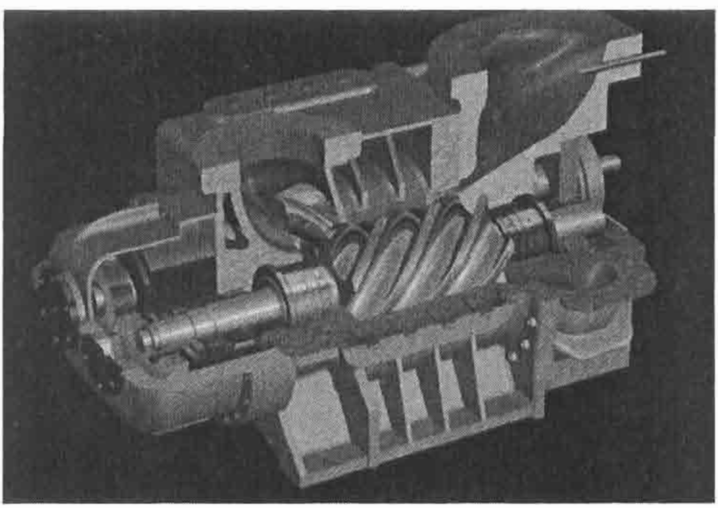

Figure 1-6. Cutaway of an oil-free helical-lobe rotary compressor. (*Courtesy of A-C Compressor Corporation*)

supercharged or multistage applications, pressures of 250 psi are attainable. The spiral-lobe version is limited to 10,000 cfm flow and about 15 psi discharge pressure.

The *straight-lobe compressor* is similar to the helical-lobe machine but is much less sophisticated. As the name implies, it has two untwisted or straight-lobe rotors that intermesh as they rotate. Normally, each rotor pair has a two-lobe rotor configuration, although a three-lobe version is available. All versions of the straight-lobe compressor use timing gears to phase the rotors. Gas is trapped in the open area of the lobes as the lobe pair crosses the inlet port. There is no compression as gas is moved to the discharge port; rather, it is compressed by the backflow from the discharge port. Four cycles of compression take place in the period of one shaft rotation on the two-lobe version. The operating cycle of the straight-lobe rotary compressor is shown in Figure 1-7.

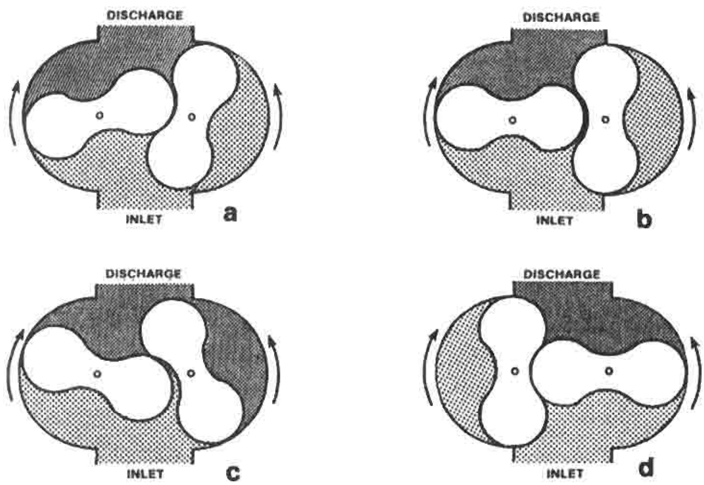

Figure 1-7. Operating cycle of a straight-lobe rotary compressor. (*Modified, courtesy of Ingersoll-Rand*)

Volume range of the straight-lobe compressor is 5 to 30,000 cfm. Pressure ranges are very limited with the maximum single-stage rating at 15 psi. In a few applications, the compressors are used in two-stage form where the discharge pressure is extended to 20 psi.

The *sliding-vane compressor* uses a single rotating element (see Figure 1-8). The rotor is mounted eccentric to the center of the cylinder portion of the casing and is slotted and fitted with vanes. The vanes are free to

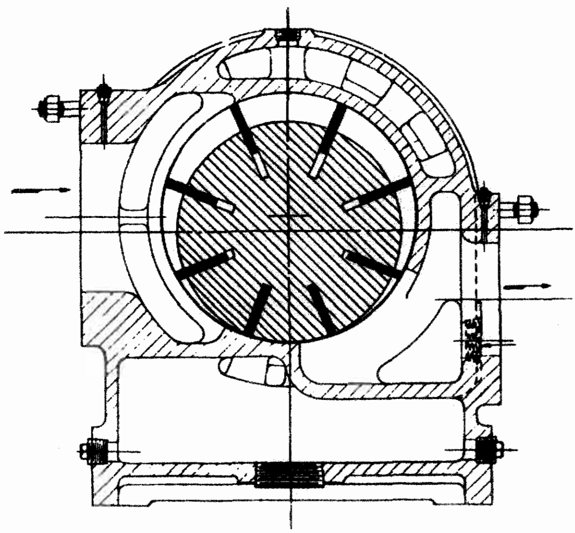

Figure 1-8. Cross section of a sliding vane compressor. (*Courtesy of A-C Compressor Corporation*)

move in and out within the slots as the rotor revolves. Gas is trapped between a pair of vanes as the vanes cross the inlet port. Gas is moved and compressed circumferentially as the vane pair moves toward the discharge port. The port locations control the pressure ratio. (This compressor must have an external source of lubrication for the vanes.)

The sliding-vane compressor is widely used as a vacuum pump as well as a compressor, with the largest volume approximately 6,000 cfm. The lower end of the volume range is 50 cfm. A single-stage compressor with atmospheric inlet pressure is limited to a 50 psi discharge pressure. In booster service, the smaller units can be used to approximately 400 psi.

The *liquid piston compressor,* or liquid ring pump as it is more commonly called, uses a single rotor and can be seen in Figure 1-9. The rotor consists of a set of forward-curved vanes. The inner area of the rotor contains sealed openings, which in turn rotate about a stationary hollow inner core. The inner core contains the inlet and discharge ports. The rotor turns in an eccentric cylinder of either a single- or double-lobe design. Liquid is carried at the tips of the vanes and moves in and out as the rotor turns, forming a liquid piston. The port openings are so located as to allow gas to enter when the liquid piston is moving away from center. The port is then closed as rotation progresses and compression takes place, with the discharge port coming open as the liquid piston approaches the innermost part of the travel. As with some of the other rotary com-

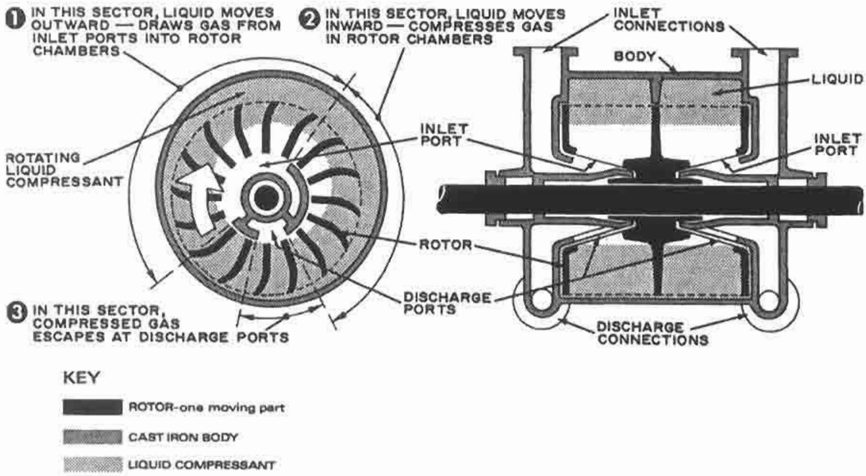

Figure 1-9. A sectional and end view of a liquid piston compressor. (*Courtesy of Nash Engineering Co.*)

pressors, the exact port locations must be tailored to the desired pressure ratio at time of manufacture. In the two-lobe design, two compression cycles take place during the course of one rotor revolution.

The capacity range is relatively large, ranging from 2 to 16,000 cfm. Like the sliding-vane compressors, the liquid piston compressor is widely used in vacuum service. The compressor is also used in pressure service with a normal range of 5 to 80 psi with an occasional application up to 100 psi. Because of the liquid piston, the compressor can ingest liquid in the suction gas without damage. This feature helps offset a somewhat poor efficiency. The compressor is used in multiple units to form a multistage arrangement.

Continuous Compression Compressors

Ejectors

Continuous compression compressors are of two types: ejector and dynamic.

The ejector can first be identified as having no moving parts (see Figure 1-10). It is used primarily for that feature as it is not as efficient as most of the mechanical compressors. Simplicity and the lack of wearing parts contribute to the unit's inherent reliability and low-maintenance expense.

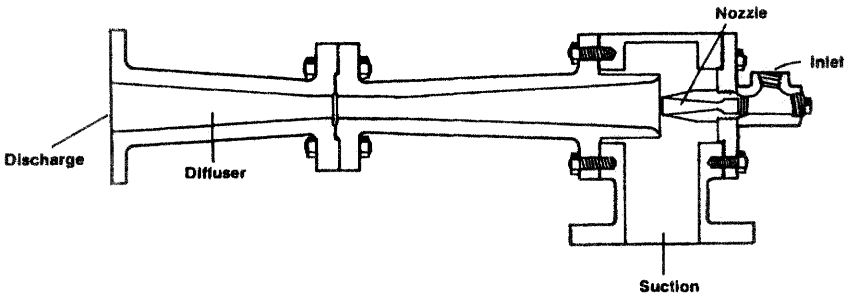

Figure 1-10. Cross section of an ejector. (*Courtesy of Graham Manufacturing Co., Inc.*)

The ejector is operated directly by a motive gas or vapor source. Air and steam are probably the two most common of the motive gases. The ejector uses a nozzle to accelerate the motive gas into the suction chamber where the gas to be compressed is admitted at right angles to the motive gas direction. In the suction chamber, also referred to as the mixing chamber, the suction gas is entrained by the motive fluid. The mixture moves into a diffuser where the high velocity gas is gradually decelerated and increased in pressure.

The ejector is widely used as a vacuum pump, where it is staged when required to achieve deeper vacuum levels. If the motive fluid pressure is sufficiently high, the ejector can compress gas to a slightly positive pressure. Ejectors are used both as subsonic and supersonic devices. The design must incorporate the appropriate nozzle and diffuser compatible with the gas velocity. The ejector is one of the few compressors immune to liquid carryover in the suction gas.

Dynamic Compressors

In dynamic compressors, energy is transferred from a moving set of blades to the gas. The energy takes the form of velocity and pressure in the rotating element, with further pressure conversion taking place in the stationary elements. Because of the dynamic nature of these compressors, the density and molecular weight have an influence on the amount of pressure the compressor can generate. The dynamic compressors are further subdivided into three categories, based primarily on the direction of flow through the machine. These are radial, axial, and mixed flow.

The *radial-flow,* or *centrifugal compressor* is a widely used compressor and is probably second only to the reciprocating compressor in usage in the process industries. A typical multistage centrifugal compressor can be seen in Figure 1-11. The compressor uses an impeller consisting of

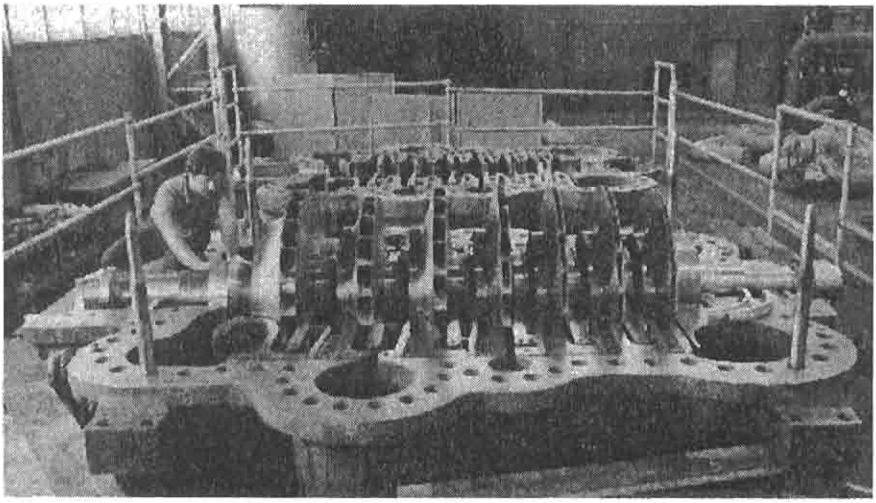

Figure 1-11. Radial-flow horizontally split multistage centrifugal compressor. (***Courtesy of Nuovo Pignone***)

radial or backward-leaning blades and a front and rear shroud. The front shroud is optionally rotating or stationary depending on the specific design. As the impeller rotates, gas is moved between the rotating blades from the area near the shaft and radially outward to discharge into a stationary section, called a diffuser. Energy is transferred to the gas while it is traveling through the impeller. Part of the energy converts to pressure along the blade path while the balance remains as velocity at the impeller tip where it is slowed in the diffuser and converted to pressure. The fraction of the pressure conversion taking place in the impeller is a function of the backward leaning of the blades. The more radial the blade, the less pressure conversion in the impeller and the more conversion taking place in the diffuser. Centrifugal compressors are quite often built in a multistage configuration, where multiple impellers are installed in one frame and operate in series.

Centrifugal compressors range in volumetric size from approximately 1,000 to 150,000 cfm. In single-wheel configuration, pressures vary considerably. A common low pressure compressor may only be capable of 10 to 12 psi discharge pressure. In higher-head models, pressure ratios of 3 are available, which on air is a 30-psi discharge pressure when the inlet is at atmospheric conditions.

Another feature of the centrifugal is its ability to admit or extract flow to or from the main flow stream, at relatively close pressure intervals, by means of strategically located nozzles. These flows are referred to as sidestreams. Pressures of the multistage machine are quite varied, and difficult to generalize because of the many factors that control pressure. Centrifugals are in service at relatively high pressures up to 10,000 psi either as a booster or as the result of multiple compressors operating in series.

Axial compressors are large-volume compressors that are characterized by the axial direction of the flow passing through the machine. The energy from the rotor is transferred to the gas by blading (see Figure 1-12). Typically, the rotor consists of multiple rows of unshrouded blades. Before and after each rotor row is a stationary (stator) row. For example, a gas particle passing through the machine alternately moves through a stationary row, then a rotor row, then another stationary row, until it completes the total gas path. A pair of rotating and stationary blade rows define a stage. One common arrangement has the energy transfer arranged to provide 50% of the pressure rise in the rotating row and the other 50% in the stationary row. This design is referred to as 50% reaction.

Axial compressors are smaller and are significantly more efficient than centrifugal compressors when a comparison is made at an equivalent flow rating. The exacting blade design, while maintaining structural integrity, renders this an expensive piece of equipment when compared to centrifugals. But it is generally justified with an overall evaluation that includes the energy cost.

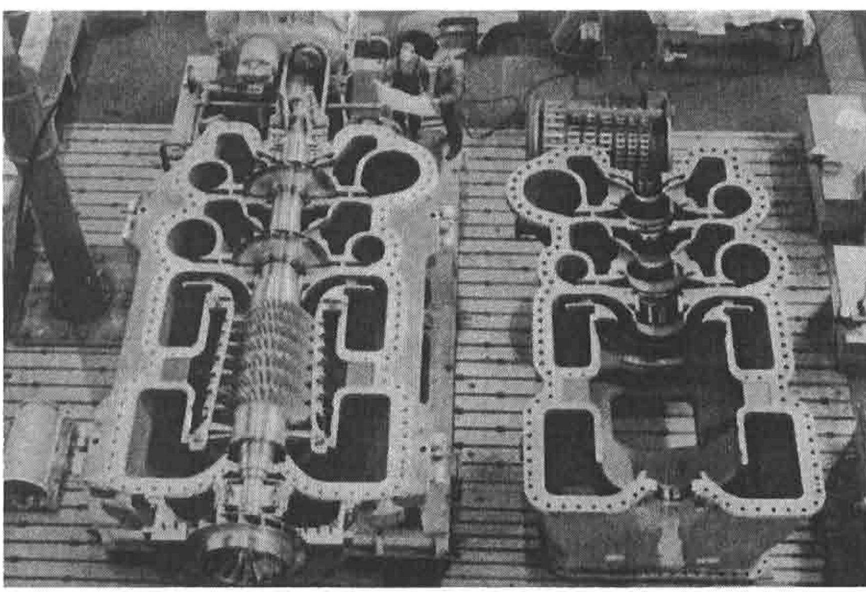

Figure 1-12. Axial-flow compressor. (*Courtesy of Demag Delaval Turbomachinery Corp.*)

The volume range of the axial starts at approximately 70,000 cfm. One of the largest sizes built is 1,000,000 cfm, with the common upper range at 300,000 cfm. The axial compressor, because of a low-pressure rise per stage, is exclusively manufactured as a multistage machine. The pressure for a process air compressor can go as high as 60 psi. Axial compressors are an integral part of large gas turbines where the pressure ratios normally are much higher. In gas turbine service, discharge pressures up to 250 psi are used.

The mixed-flow compressor is a relatively uncommon form, and is being mentioned here in the interest of completeness. At first glance, the mixed-flow compressor very much resembles the radial-flow compressor. A bladed impeller is used, but the flow path is angular in direction to the rotor; that is, it has both radial and axial components (see Figure 1-13). Because the stage spacing is wide, the compressor is used almost exclusively as a single-stage machine. The energy transfer is the same as was described for the radial-flow compressor.

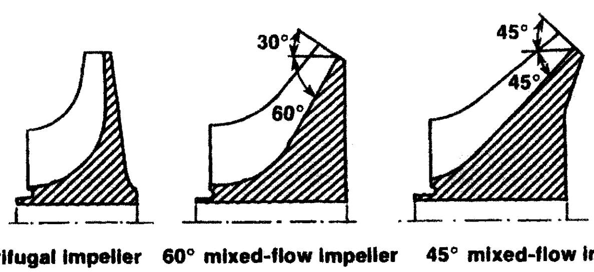

Figure 1-13. Comparison of radial- and mixed-flow compressor impellers.

The compressor size is flexible and covers the centrifugal compressor flow range, generally favoring the higher flow rates. The head per stage is lower than available in the centrifugal. The compressor finds itself in the marketplace because of the unique head-capacity characteristic, which can be illustrated by its application in pipeline booster service. In this situation the pressure ratio needed is not high, and as a result the head required is low. However, because of the high inlet pressure of the gas, a relatively high pressure rise is taken across the machine. Thus, there is a real need for a more rugged and less expensive alternative to the axial compressor.

2

Basic Relationships

Introduction

This chapter presents some basic thermodynamic relationships that apply to all compressors. Equations that apply to a particular type of compressor will be covered in the chapter addressing that compressor. In most cases, the derivations will not be presented, as these are available in the literature. The references given are one possible source for additional background information.

The equations are presented in their primitive form to keep them more universal. Consistent units must be used, as appropriate, at the time of application. The example problems will include conversion values for the units presented. The symbol g will be used for the universal gravity constant to maintain open form to the units.

Gas and Vapor

A gas is defined as the state of matter distinguished from solid and liquid states by very low density and viscosity, relatively great expansion and contraction with changes in pressure and temperature, and the ability to diffuse readily, distributing itself uniformly throughout any container.

A vapor is defined as a substance that exists below its critical temperature and that may be liquefied by application of sufficient pressure. It may be defined more broadly as the gaseous state of any substance that is liquid or solid under ordinary conditions.

Many of the common "gases" used in compressors for process plant service are actually vapors. In many cases, the material may change states during a portion of the compression cycle. Water is a good example, since a decrease in temperature at high pressure will cause a portion of the water to condense. This is a common occurrence in the first intercooler of a plant air compressor. Conversely, lowering the pressure in a reservoir of liquid refrigerant at a fixed temperature will cause the vapor quantity to increase.

Perfect Gas Equation

Charles and Gay-Lussac, working independently, found that gas pressure varied with the absolute temperature. If the volume was maintained constant, the pressure would vary in proportion to the absolute temperature [1]. Using a proportionality constant R, the relationships can be combined to form the equation of state for a perfect gas, otherwise known as the perfect gas law.

$$Pv = RT \tag{2.1}$$

where

P = absolute pressure
v = specific volume
R = constant of proportionality
T = absolute temperature

If the specific volume v is multiplied by mass m, the volume becomes a total volume V. Therefore, multiplying both sides of Equation 2.1 by m, yields

$$PV = mRT \tag{2.2}$$

In process engineering, moles are used extensively in performing the calculations. A mole is defined as that mass of a substance that is numerically equal to its molecular weight. Avogadro's Law states that identical volumes of gas at the same temperature and pressure contain equal numbers of molecules for each gas. It can be reasoned that these identical volumes will have a weight proportional to the molecular weight of the gas. If the mass is expressed as

$$m = n \times mw \tag{2.3}$$

where

n = number of moles
mw = molecular weight

then,

$$PV = n\ mw\ RT \tag{2.4}$$

If the value mw R is the same for all gases, the universal gas constant Ugc is defined and R becomes the specific gas constant.

$$R = \frac{Ugc}{mw} \tag{2.5}$$

Another useful relationship can be written using Equation 2.2.

$$\frac{P_1V_1}{T_1} = mR = \frac{P_2V_2}{T_2} \tag{2.6}$$

If in Equation 2.2 both sides are divided by time, the term V becomes Q, volumetric flow per unit time, and the mass flow per unit time becomes w.

$$PQ = wRT \tag{2.7}$$

Compressibility

A term may now be added to Equation 2.1 to correct it for deviations from the ideal gas or perfect gas law.

$$Pv = ZRT \tag{2.8}$$

Solving for Z:

$$Z = \frac{Pv}{RT} \tag{2.9}$$

Equation 2.7 may be modified in a similar manner by the addition of the compressibility term Z as follows:

$$PQ = wZRT \tag{2.10}$$

Generalized Compressibility Charts

The vapor definition introduces another concept, that of critical temperature. Critical temperature is defined as that temperature above which a gas will not liquefy regardless of any increase in pressure. Critical pressure is defined as the pressure required at the critical temperature to cause the gas to change state.

The following two equations are used to define reduced temperature and reduced pressure:

$$T_r = \frac{T}{T_c} \tag{2.11}$$

$$P_r = \frac{P}{P_c} \tag{2.12}$$

The generalized compressibility charts may be used with values obtained in the use of Equations 2.7 and 2.8 to determine the compressibility of a wide range of gases. The charts were derived from experimental data and are a good source of information for use in compressor calculations [1].

Partial Pressure

Avogadro's Law states that equal volumes of gas at identical pressure and temperature contain equal numbers of molecules. Avogadro's Law can be used in a similar manner to develop gas mixture relationships. A mixture of gases occupying a given volume will have the same number of molecules as a single gas. The weight will be a sum of the proportionate parts of the gases in the mixture. If the gas proportion is presented as a mole percent, this value is the same as a volume percent.

When one pure liquid exists in the presence of another pure liquid, where the liquids neither react nor are soluble in each other, the vapor pressure of one liquid will not affect the vapor pressure of the other liquid. The sum of the partial pressures P_n is equal to the total pressure P. This relationship is formalized in Dalton's Law, which is expressed as

$$P = P_1 + P_2 + P_3 + \ldots \tag{2.13}$$

Gas Mixtures

If the total pressure of a mixture is known, the partial pressure of each component can be calculated from the mole fraction. The total number of moles in the mixture M_m is the sum of the individual component moles.

$$M_m = M_1 + M_2 + M_3 + \ldots \tag{2.14}$$

The mole fraction x_n is

$$x_1 = \frac{M_1}{M_m}; x_2 = \frac{M_3}{M_m}; x_3 = \frac{M_3}{M_m} \tag{2.15}$$

The partial pressure can be calculated by use of the following:

$$P_1 = x_1P; P_2 = x_2 P; P_3 = x_3P \tag{2.16}$$

Specific Heat Ratio

The value k is defined as the ratio of specific heats.

$$k = \frac{c_p}{c_v} \tag{2.18}$$

where

c_p = specific heat at constant pressure
c_v = specific heat at constant volume

$$\text{Also, } k = \frac{Mc_p}{Mc_p - 1.99} \tag{2.19}$$

where

Mc_p = molal specific heat at constant pressure.

$$Mc_{pm} = x_1 Mc_{p1} + x_2Mc_{p2} + x_3Mc_{p3} + \ldots \tag{2.20}$$

Substitute into Equation 2.19

$$k_m = \frac{Mc_{pm}}{Mc_{pm} - 1.99} \tag{2.21}$$

Molecular Weight

To calculate the mixture molecular weight (mw_m) use the following equation:

$$mw_m = x_1mw_1 + x_2mw_2 + x_3mw_3 \tag{2.22}$$

The weight fraction y_n of the mixture is

$$y_1 = \frac{x_1mw_1}{mw_m}; y_2 = \frac{x_2mw_2}{mw_m}; y_3 = \frac{x_3mw_3}{mw_m} \tag{2.23}$$

$$y_1 + y_2 + y_3 + \ldots = 1.0 \tag{2.24}$$

Specific Gravity

The specific gravity, SG, is the ratio of the density of a given gas to the density of dry air at the same temperature and pressure. It can be calculated from the ratio of molecular weights if the given gas is a perfect gas.

$$SG = \frac{mw}{28.96} \tag{2.25}$$

Mixture Compressibility

The simplest and most common method of establishing pseudocriticals for a mixture is Kay's Rule.

$$T_{cm} = x_1T_{c1} + x_2T_{c2} + x_3T_{c3} + \ldots \tag{2.26}$$

$$P_{cm} = x_1P_{c1} + x_2P_{c2} + x_3P_{c3} + \ldots \tag{2.27}$$

Substituting Equations 2.26 and 2.27 into Equations 2.11 and 2.12:

$$T_{rm} = \frac{T}{T_{cm}} \tag{2.28}$$

$$P_{rm} = \frac{P}{P_{cm}} \tag{2.29}$$

Humidity

Although air is a mixture of gases, it is generally treated as an individual gas with accounting made only for other components such as moisture when present.

When a mixture is saturated, the proper terminology is that the volume occupied by the mixture is saturated by one or more of the components. For air space, which is partially saturated by water vapor, the actual partial pressure of the water vapor may be determined by multiplying the saturation pressure at the space temperature by the relative humidity.

Relative humidity can be calculated from the following:

$$RH = \frac{P_v}{P_{satv}} \times 100 \tag{2.30}$$

Specific humidity, which is the weight of water vapor to the weight of dry air, is given by the following ratio:

$$SH = \frac{W_v}{W_a} \tag{2.31}$$

Psychrometric charts plot wet bulb and dry bulb data for air-water vapor mixtures at atmospheric pressure. These charts are quite useful for moisture corrections in air compressors with atmospheric inlets (see Figures B-2 and B-3 in Appendix B).

Flow

There are several different flow terminology conventions in common use. The following discussion is presented in order to eliminate any confusion this may cause.

The most important thing to remember in compressor calculations is that compressor flow is a volumetric value based on the flowing conditions of pressure, temperature, relative humidity (if moisture is present), and gas composition at the compressor inlet nozzle. The flow units are inlet cubic feet per minute (icfm).

Process calculations, where material balances are performed, normally produce flow values in terms of a weight flow. The flow is generally stated as pounds per hour. Equation 2.10 can be used either with a single-component gas or with a mixture.

Pipeline engineers use the flow value stated as standard cubic feet per day. This is an artificial weight flow because flowing conditions are referred to a standard pressure and temperature. The balance of the flow specification is then stated in terms of specific gravity.

A common method of stating flow is standard cubic feet per minute where the flowing conditions are referred to an arbitrary set of standard conditions. Unfortunately, standard conditions are anything but standard. Of the many used, two are more common. The ASME standard uses 68°F and 14.7 psia. The relative humidity is given as 36%. The other standard that is used by the gas transmission industry and the API Mechanical Equipment Standards is 60°F at 14.7 psia. As can be seen from this short discussion, a flow value must be carefully evaluated before it can be used in a compressor calculation.

Example 2-1

A pipeline is flowing 3.6 standard million cubic feet per day. The gas is made up of the following components: 85% methane, 10% ethane, 4% butane, 1% nitrogen. The values are given as a mole percent. The flowing temperature is 80°F and the pressure is 300 psig.

The problem is to calculate the suction conditions for a proposed booster compressor. Values to calculate are flow in cfm at the flowing conditions, the mixture molecular weight, mixture specific heat ratio, and the compressibility of the mixture.

Step 1. Convert the flow to standard cfm using 24 hours per day and 60 minutes per hour.

$$Q_{std} = \frac{3.6 \times 10^6}{24 \times 60}$$

$$Q_{std} = 2500$$

Step 2. Convert scfm to flowing conditions using Equation 2.6.
Standard conditions:

$$P_2 = 14.7 \text{ psia}$$

$$T_2 = 60°F + 460°R = 520°R$$

Flowing conditions:

$$P_1 = 300 + 14.7 = 314.7 \text{ psia}$$

$$T_1 = 80°F + 460°R = 540°R$$

Step 3. Substituting into Equation 2.6, using Q_1 for V_1 and solving for Q_1.

$$Q_1 = \frac{14.7}{314.7} \times \frac{540}{520} \times 2500$$

$Q_1 = 121.3$ cfm (flow at the compressor inlet)

Step 4. Change the molal percentages to fractions and substitute for x_n, then use Equations 2.20, 2.22, 2.26, and 2.27 to construct Table 2-1.

Step 5. Solve for mixture specific heat ratio k_m, using Equation 2.21.

$$k_m = \frac{9.59}{9.59 - 1.99}$$

$$k_m = 1.26$$

Table 2-1
Gas Mixture Data

Gas	x_n	m_{cp}	$x_n m_{cp}$	mw	$x_n$mw	T_c	$x_n T_c$	p_c	$x_n p_c$
Methane	.85	8.60	7.31	16.04	13.63	344	292.4	673	572.1
Ethane	.10	12.64	1.26	30.07	3.01	550	55.0	708	70.8
Butane	.04	23.82	.95	58.12	2.33	766	30.6	551	22.0
Nitrogen	.01	6.97	.07	28.02	0.28	227	2.3	492	4.9
Mixture	1.00		9.59		19.25		380.		670.

Step 6. Using T_{cm} = 380°R and P_{cm} = 670 psia, substitute into Equations 2.28 and 2.29.

$$T_{rm} = \frac{540}{380}$$

$$T_{rm} = 1.42$$

$$P_{rm} = \frac{314.7}{670}$$

$$P_{rm} = .47$$

Step 7. From the general compressibility charts in the Appendix, Z = .95.

Example 2-2

Determine the volumetric flow to use in sizing a compressor to meet the following suction requirements:

Weight flow = 425 lb/min dry air
Inlet pressure = 14.7 psia ambient air
Inlet temperature = 90°F
Inlet relative humidity = 95%

Step 1. Determine the total moist air flow to provide the dry air needed. Because the air is at atmospheric pressure, psychrometric charts may be used to determine the amount of water vapor contained in the dry air (see Figures B-2 and B-3 in Appendix B).

From the psychrometric chart, for a dry bulb temperature of 90°F with a relative humidity of 95%,

Specific humidity = .0294 lbs of water vapor/lb of dry air
For the 425 lb/min of dry air, the water vapor content is

$$w_2 = 425 \times .0294$$

$$w_2 = 12.495 \text{ lb/min water vapor}$$

Therefore,

$$w_m = 425 + 12.495$$

$$w_m = 437.5 \text{ lb/min total weight flow}$$

Step 2. Determine the molecular weight of the moist air mixture using Equation 2.3.

$$M_1 = \frac{425 \text{ lb/min}}{28.95 \text{ lb/lb-mol}}$$

$$M_1 = 14.68 \text{ lb-mols/min dry air}$$

$$M_2 = \frac{12.495 \text{ lb/min}}{18.02 \text{ lb/lb-mol}}$$

$$M_2 = .693 \text{ lb-mols/min water vapor}$$

$$M_m = 14.68 + .693$$

$$M_m = 15.373 \text{ total mols/min mixture}$$

Step 3. Using Equation 2.15, calculate the mol fraction of each component.

$$x_1 = \frac{14.68}{15.373}$$

$$x_1 = .955 \text{ mol fraction dry air}$$

$$x_2 = \frac{.693}{15.373}$$

$$x_2 = .045 \text{ mol fraction dry air}$$

Step 4. Calculate the molecular weight using Equation 2.22.

$$mw_m = .955 \times 28.95 + .045 \times 18.02$$

$$mw_m = 28.46 \text{ mol weight mixture}$$

Step 5. Calculate the compressor inlet volume using Equation 2.10. First use Equation 2.5 to calculate the specific gas constant.

$$R_m = 1545/28.46$$

$$R_m = 54.29$$

Convert to absolute temperature.

$$T_1 = 460 + 90$$

$$T_1 = 550°R$$

Substitute into Equation 2.10 and, using 144 in^2/ft^2,

$$Q_1 = 437.5 \frac{1 \times 54.29 \times 550}{14.7 \times 144}$$

$$Q_1 = 6171 \text{ cfm air mixture}$$

For comparison, assume the moisture had been ignored.

$$R_m = 1545/28.95$$

$$R_m = 53.37$$

$$Q_1 = 425 \frac{1 \times 53.37 \times 550}{14.7 \times 144}$$

$$Q_1 = 5893 \text{ cfm}$$

The calculation would indicate that the volume would have been short by approximately 5% if the moisture in the air was ignored.

Acoustic Velocity

A relationship that is useful in compressor and compressor systems is the speed of sound of the gas at the flowing conditions. The *acoustic velocity*, a, can be calculated using the following equation:

$$a = \sqrt{kRgT} \tag{2.32}$$

where

k = ratio of specific heats
R = specific gas constant
g = gravitational constant
T = absolute temperature of the fluid

The Mach number is given by

$$M_a = \frac{V}{a} \tag{2.33}$$

The relationship for uniform flow velocity V in a cross-sectional area, A, such as a compressor flow channel or nozzle is

$$V = \frac{Q}{A} \tag{2.34}$$

where

Q = volumetric flow.

Equations of State

Gases can be treated individually or as mixtures by the methods just outlined for most applications including evaluation of vendor proposals. More sophisticated equations of state can be used for real gas applications when large deviations from the perfect gas law are anticipated. For mixtures, more sophisticated mixing rules can be paired with the equation of state when required. For hydrocarbons, the most widely used equation of state is the Benedict-Webb-Rubin (BWR) equation [2]. For a gas mixture, the pseudocritical constants used in the BWR equation may be developed using Kay's mixing rule. If the application is outside Kay's Rule guidelines, a more complex rule such as Leland-Mueller may be substituted [3]. An alternate approach is the Starling BWR implementation [4]. Starling includes gas mixing in formulation of the equation of state. Another, the Redlich-Kwong equation, is widely used because of its simplicity. Finally, for chlorinated compounds and halocarbon refrigerants, the Martin-Hou equation yields results generally superior to the previously mentioned equations, which were developed primarily for hydrocarbons [5]. The equations of state discussed are by no means a complete list, but they have proved to be especially accurate in direct application.

The equations of state will not be further described or presented in more detail as they are unfortunately somewhat difficult to solve without the use of a computer. Full details are available in the referenced material for those wishing to pursue this subject further. In the past, these equations required the use of a mainframe computer not only to solve the equations themselves, but to store the great number of constants required. This has been true particularly if the gas mixture contains numerous components. With the power and storage capacity of personal computers increasing, the equations have the potential of becoming more readily available for general use.

Mollier Charts

Another form in which gas properties are presented is found in plots of pressure, specific volume, temperature, entropy, and enthalpy. The most common form, the Mollier chart, plots enthalpy against entropy. A good example of this is the Mollier chart for steam. Gases are generally plotted as pressure against enthalpy (P-h charts). These are also sometimes referred to as Mollier charts. The charts are readily available for a wide range of pure gases, particularly hydrocarbons and refrigerants. Some of the more common charts are included in Appendix B.

First Law of Thermodynamics

The first law of thermodynamics states that energy cannot be created or destroyed, although it may be changed from one form to another. Stated in equation form, it is written as follows:

$$Q_h - W_t = \Delta E \tag{2.35}$$

where

Q_h = heat supplied to a system
W_t = work done by the system
ΔE = change in energy of the system

If the change in energy to the system is expanded, then

$$\Delta E = \Delta U + \Delta PE + \Delta KE \tag{2.36}$$

where

ΔU = change in internal energy
ΔPE = change in potential energy
ΔKE = change in kinetic energy

If the work term W_t is expanded to breakdown shaft work done to or from the system and the work done by the system, then

$$W_t = W + (pv\Delta m)_{out} - (pv\Delta m)_{in} \tag{2.37}$$

where

W = shaft work in or out of the system
p = fluid pressure in the system
v = specific volume of the fluid in the system
Δm = mass of fluid working in the system

If Equation 2.36 is rewritten in a general form using specific energy notation,

$$e = u + \frac{V^2}{2g} + z \tag{2.38}$$

where

u = specific form of internal energy
V = velocity of the gas
z = height above some arbitrary reference

By substituting Equations 2.37 and 2.38 into Equation 2.35, maintaining the specific energy form, and regrouping, the following equation can be written:

$$u_1 + P_1v_1 + \frac{V_1^2}{2g} + z_1 + Q_h = u_2 + P_2v_2 \frac{V_2^2}{2g} + z_2 + W \tag{2.39}$$

By defining enthalpy as

$$h = u + Pv \tag{2.40}$$

and substituting into Equation 2.39,

$$h_1 + \frac{V_1^2}{2g} + z_1 + Q_h = h_2 + \frac{V_2^2}{2g} + z_2 + W \tag{2.41}$$

Equation 2.41 is the general energy equation for a steady flow process.

Second Law of Thermodynamics

The second law of thermodynamics was actually postulated by Carnot prior to the development of the first law. The original statements made concerning the second law were negative—they said what would *not* happen. The second law states that heat will not flow, in itself, from cold to hot. While no mathematical relationships come directly from the second law, a set of equations can be developed by adding a few assumptions for use in compressor analysis. For a reversible process, entropy, s, can be defined in differential form as

$$ds = \frac{dQ_h}{T} \tag{2.42}$$

It is recognized that a truly reversible process does not exist in the real world. If it is further recognized that real processes result in an increase in entropy, the second law can be stated.

$$\Delta s \geqq 0 \tag{2.43}$$

If work done in a system is distributed over an area, for example, pressure P is acting through volume v, then in specific notation and in differential form the Equation 2.44 results.

$$dW_t = Pdv \tag{2.44}$$

If further $\Delta U = \Delta E$ when the kinetic and potential energies in Equation 2.36 do not change, Equation 2.35 can be rewritten, substituting U for E, changing to the specific notation and putting the equation in differential form.

$$du = dQ_h - W_t \tag{2.45}$$

Combining Equations 2.42, 2.44 and 2.45 yields

$$du = Tds - Pdv \tag{2.46}$$

Theoretical Work

Theoretical work or compressor head is the heart and substance of compressor design. Some basic form of understanding must be developed even if involvement with compressors is less than that of design of the machine itself. Proper applications cannot be made if this understanding is absent. The following theoretical evaluations will be abbreviated as much as possible to reduce the length and still present the philosophy. For the reader with the ambition and desire, the presentation will be an outline to which the reader can fill in the spaces.

In deriving the head equation, the general energy Equation 2.41 will be used. The equation can be modified by regrouping and eliminating the z terms, as elevation differences are not significant with gas.

$$\left(h_2 + \frac{V_2^2}{2g}\right) - \left(h_1 + \frac{V_1^2}{2g}\right) = -W + Q_h \tag{2.47}$$

The velocity term can be considered part of the enthalpy if the enthalpy is defined as the stagnation or total enthalpy. The equation can be simplified to

$$h_2 - h_1 = -W + Q_h \tag{2.48}$$

If the process is assumed to be adiabatic (no heat transfer), then

$$Q_h = 0$$

For the next step the enthalpy equation is written in differential form:

$$dh = du + Pdv + vdP \tag{2.49}$$

Recalling Equation 2.46,

$$du = Tds - Pdv \tag{2.46}$$

and substituting Equation 2.46 into 2.49,

$$dh = Tds + vdP \tag{2.50}$$

The process is assumed reversible. This defines entropy as constant and therefore ds = 0, making Tds = 0. The enthalpy equation is simplified to

$$dh = vdP \tag{2.51}$$

For an isentropic, adiabatic process,

$$Pv^k = \text{constant} = C \tag{2.52}$$

solving for P,

$$P = Cv^{-k} \tag{2.53}$$

Taking the derivative of P with respect to v yields,

$$dP = C(-k)v^{-k-1}dv \tag{2.54}$$

Substituting into the enthalpy Equation 2.51,

$$dh = C(-k)v^{-k}dv \tag{2.55}$$

Integrating from state point 1 to 2 and assuming k is constant over the path yields,

$$h_2 - h_1 = C\,\frac{v_2^{1-k} - v_1^{1-k}}{(k-1)/k} \tag{2.56}$$

Substitute

$$C = P_1 v_1{}^k = P_2 v_2{}^k \tag{2.57}$$

into Equation 2.55, which yields

$$h_2 - h_1 = \frac{P_2 v_2 - P_1 v_1}{(k-1)/k} \tag{2.58}$$

Using the perfect gas Equation 2.1 and substituting into Equation 2.58 yields

$$h_2 - h_1 = \frac{R(T_2 - T_1)}{(k-1)/k} \qquad (2.59)$$

As a check on the assumptions made, a comparison can be made to a different method of checking the derivation of the head. Enthalpy difference, as a function of temperature change, for an adiabatic process is expressed by

$$h_2 - h_1 = c_p\,(T_2 - T_1) \qquad (2.60)$$

Specific heat c_p can be calculated using specific gas constant R and specific heat ratio k.

$$c_p = \frac{Rk}{k-1} \qquad (2.61)$$

Substitute Equation 2.61 into Equation 2.60 with the result,

$$h_2 - h_1 = \frac{R(T_2 - T_1)}{(k-1)/k} \qquad (2.59)$$

This equation is identical with Equation 2.59 previously derived, giving a check on the method.

By regrouping Equation 2.59, substituting into Equation 2.48, and maintaining the adiabatic assumption $Q_h = 0$, Equation 2.62 is developed.

$$h_2 - h_1 = \frac{RT_1k}{k-1}\left[\frac{T_2}{T_1} - 1\right] = -W \qquad (2.62)$$

The $-W$ signifies work done to the system, a driven machine, as contrasted to $+W$, which would indicate work done by the system as with a driver.

If the adiabatic head is defined by the following equation:

$$H_a = h_2 - h_1 \qquad (2.63)$$

and the term r_p is introduced as the ratio of discharge pressure to inlet pressure,

$$r_p = \frac{P_2}{P_1} \qquad (2.64)$$

Next, the temperature ratio relationship in Equation 2.65 will be used. This relationship is the result of combining Equations 2.6 and 2.57 as well as a half dozen algebraic steps:

$$\frac{T_2}{T_1} = r_p^{\frac{k-1}{k}} \tag{2.65}$$

When substituting Equation 2.65 into Equation 2.62, the result is the classical form of the adiabatic head equation.

$$H_a = RT_1 \frac{k}{k-1}(r_p^{\frac{k-1}{k}} - 1) \tag{2.66}$$

An interesting note is that if in Equation 2.58, Equation 2.8 were used in place of 2.1, the result would be

$$h_2 - h_1 = \frac{R(T_2 Z_2 - T_1 Z_1)}{(k-1)/k} \tag{2.67}$$

Since the compressibility does not change the isentropic temperature rise, it should be factored out of the ΔT portion of the equation. To achieve this for moderate changes in compressibility, an assumption can be made as follows:

$$Z_{avg} = (Z_2 + Z_1)/2 \tag{2.68}$$

By replacing the values of Z_2 and Z_1, with Z_{avg} in Equation 2.67 and factoring, Equation 2.67 is rewritten as

$$h_2 - h_1 = \frac{Z_{avg} R\,(T_2 - T_1)}{(k-1)/k} \tag{2.69}$$

Now with the same process used to obtain Equation 2.66, the final form of the head equation with compressibility is

$$H_a = Z_{avg}\, RT_1 \frac{k}{k-1}(r_p^{\frac{k-1}{k}} - 1) \tag{2.70}$$

For a polytropic (reversible) process, the following definitions need to be considered:

$$\frac{n-1}{n}=\frac{k-1}{k}\times\frac{1}{\eta_p} \tag{2.71}$$

where

ηp = polytropic efficiency
n = polytropic exponent

By regrouping Equation 2.71, a polytropic expression can be

$$\eta_p=\frac{n/(n-1)}{k/(k-1)} \tag{2.72}$$

By substituting n for k, the head equation becomes

$$H_p=Z_{avg}\,RT_1\,\frac{n}{n-1}\left(r_p^{\frac{n-1}{n}}-1\right) \tag{2.73}$$

One significant practical difference in use of polytropic head is that the temperature rise in the equation is the actual temperature rise when there is no jacket cooling. The other practical uses of the equation will be covered as they apply to each compressor in the later chapters.

Real Gas Exponent

About the time it appears that there is some order to all the chaos of compressible flow, there comes another complication to worry about. It has been implied that k is constant over the compression path. The sad fact is that it is not really true. The k value has been defined in Equation 2.18 as

$$k=\frac{c_p}{c_v} \tag{2.18}$$

It has played a dual role, one in Equation 2.18 on specific heat ratio and the other as an isentropic exponent in Equation 2.53. In the previous calculation of the speed of sound, Equation 2.32, the k assumes the singular specific heat ratio value, such as at compressor suction conditions. When a non-perfect gas is being compressed from point 1 to point 2, as in the head Equation 2.66, k at 2 will not necessarily be the same as k at 1. Fortunately, in many practical conditions, the k doesn't change very much. But if one were inclined to be a bit more judicious about it and calculate a k at both state points, and if the values differed by a small amount, then one could average the two and never look back. This could not be done, however, with a gas near its critical pressure or one that's somewhat unruly like ethylene, where the k value change from point 1 to 2 is highly nonlinear. For a situation like this, the averaging approach is just not good enough and the following modification will be presented to help make the analysis more accurate.

To calculate a single compression exponent to represent the path from point 1 to point 2, the following equations will be used. Substitute γ for k and Equation 2.64 into Equation 2.65

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{2.74}$$

where γ = compression path exponent.

The expression in Equation 2.52 can be modified to Equation 2.75 to show the basic relationship for the exponent.

$$\left(P\frac{v}{Z}\right)^{\gamma} = C \tag{2.75}$$

To solve for γ use the following equation:

$$\gamma = \frac{\ln(P_2/P_1)}{\ln(P_2/P_1) - \ln(T_2/T_1)} \tag{2.76}$$

To solve for the compression exponent, use a Mollier diagram to establish the T_2 temperature value. By establishing a starting point at P_1, and T_1, and taking a path of constant entropy to P_2, the T_2 value can be read from the diagram. For a gas mixture or gas with no convenient Mollier diagram available, the problem becomes more acute. There are two alternatives: one is to use an equation of state and the other is to use a method suggested by Edmister and McGarry [6]. The latter is somewhat tedious, making the equation of state the preferred method.

Power

Input shaft power is the head of the compressor multiplied by the weight flow and divided by an appropriate efficiency with the result being added to the mechanical losses. The head portion covers the fluid or thermodynamic portion of the cycle, whereas the mechanical losses cover items such as bearings and liquid seals that are not directly linked to the fluid process. The form shown here is generalized. Each compressor type has its own unique considerations and will be covered in the appropriate chapter. The adiabatic shaft work can be expressed as

$$W_a = \frac{wH_a}{\eta_a} + \text{mech losses} \tag{2.77}$$

For polytropic shaft work,

$$W_p = \frac{wH_p}{\eta_p} + \text{mech losses} \tag{2.78}$$

Velocity Head

The determination of pressure losses at compressor nozzles and other peripheral points must be made when performing an analysis of the system. It is common in the compressor industry to state the losses as a function of velocity head. An expression for velocity head may be derived from Equation 2.39 and the following: (1) Assume flow is incompressible, which is reasonable since the change in density is negligible; therefore, $v_1 = v_2$, (2) because there is no heat added or work done, u, W, Q, = 0. When these assumptions are factored into Equation 2.39,

$$vP_2 = \frac{V_2^2}{2g} = vP_1 + \frac{V_1^2}{2g} \tag{2.79}$$

Equation 2.79 contains two pairs of head terms, the Pv head terms and the $V^2/2g$ or velocity head terms. When a flow stream passes through a nozzle, the flow is accelerated. This flow phenomenon can be further examined by regrouping Equation 2.79.

$$v(P_1 - P_2) = \frac{V_2^2}{2g} - \frac{V_1^2}{2g} \tag{2.80}$$

The left term of Equation 2.80 represents a head drop required to accelerate the flow from an initial velocity to the final velocity V_2. If the initial velocity is low it can be assumed negligible and if density $\rho = 1/v$ is substituted into Equation 2.80, it can be written as

$$P_1 - P_2 = \rho \frac{V_2^2}{2g} \tag{2.81}$$

When gas flows through pipe, casing openings, valves, or fittings, a pressure drop is experienced. This pressure drop can be defined in terms of an equivalent velocity head. The velocity head is, therefore, the pressure drop necessary to produce a velocity equal to the flowing stream velocity. The term K will be used to describe the pressure dropping potential of various restrictive elements, regardless of density or velocity. The term K is a multiplier equal to one at a value of one velocity head and can be greater than or less than one. Typical values of K are presented in Table 2-2. By substituting $\Delta P = P_1 - P_2$ and dropping the subscript on the velocity term, the working equation to use in the calculation of pressure drop for K velocity heads is

$$\Delta P = K\rho \frac{V^2}{2g} \tag{2.82}$$

Example 2-3

An example will help illustrate one use of velocity head. A compressor is being considered for reuse in another application, and the question was raised as to the size of the inlet nozzle. The original conditions are stated as follows:

Inlet nozzle size: 18 inches
Inlet flow: 10,000 CFM
Inlet pressure: 25 psia
Inlet temperature: 80°F
Molecular weight: 29
Specific heat ratio: 1.35

The new conditions:

Inlet flow: 11,000 CFM
Inlet pressure: 31 psia
Inlet temperature: 40°F
Molecular weight: 31
Specific heat ratio: 1.30

Table 2-2
Velocity Head Multipliers

Description	K-Factor
Reducer contraction	
0.75	0.2
0.50	0.3
Reducer enlargement	
0.75	0.5
0.50	0.6
0.25	0.9
Gate valve	
Fully open	0.15
0.25 open	25.0
Elbow	
Long radius	0.15
Short radius	0.25
Miter	1.10
Close return bend	0.5
Swing check or ball valve	2.2
Tee flow through bull-head	1.8
Angle valve, open	3.0
Globe valve, open	5.0
Filters	
Clean	4.0
Foul	20.0
Intercoolers	17.0
Gas separators	7.0
Surge bottles	
No choke tube	4.0
With choke tube	12.0
Casing inlet nozzle	0.5
Sidestream inlet nozzle (diaphragm)	1.0
Sidestream inlet nozzle (stage space)	0.8
Casing discharge nozzle	0.5
Extraction nozzle	0.8

Source: Modified from [16].

By consultation with the original equipment maker, it has been determined that the vendor used a value of .2 velocity heads in the original design. From this information, K = .2. The effect of the rerate conditions on the inlet will be

$$V = \frac{10{,}000 \times 144}{233.7 \times 60}$$

$$V = 102.7 \text{ fps}$$

Calculate the sonic velocity using Equation 2.32

where

$$R = 1545/29 = 53.3$$

$$T = 80 + 460 = 540°F$$

$$a = \sqrt{1.35T \times 53.3 \times 32.2 \times 540}$$

$$a = 1118.3 \text{ fps}$$

Using Equation 2.33, calculate the Mach number

$$M_a = \frac{102.7}{1118.3}$$

$$M_a = .09$$

This is a low value, therefore, the possibility exists of an up-rate relative to any nozzle flow limits. At this point, a comment or two is in order. There is a rule of thumb that sets inlet nozzle velocity limit at approximately 100 fps. But because the gases used in the examples have relatively high acoustic velocities, they will help illustrate how this limit may be extended. Regardless of the method being used to extend the velocity, a value of 150 fps should be considered maximum. When the sonic velocity of a gas is relatively low, the method used in this example may dictate a velocity for the inlet nozzle of less than 100 fps. The pressure drop due to velocity head loss of the original design is calculated as follows:

$$v = \frac{53.3 \times 540}{25 \times 144}$$

$$v = 7.99 \text{ ft}^3/\text{lb}$$

$$\rho = \frac{1}{7.99}$$

$$\rho = .13 \text{ lb/ft}^3$$

$$\Delta P = .2 \times .13 \frac{(102.7)^2}{64.4 \times 144}$$

$$\Delta P = .03 \text{ psi}$$

This is also a low value, so the proposed rerate conditions will be:

$$V = \frac{11{,}000 \times 144}{233.7 \times 60}$$

$$V = 113 \text{ fps}$$

$$R = 1545/31 = 49.8$$

$$T = 40 + 460 = 500°F$$

$$a = \sqrt{1.30 \times 49.8 \times 32.2 \times 500}$$

$$a = 1020 \text{ fps}$$

$$M_a = \frac{113}{1020}$$

$$M_a = .11$$

$$v = \frac{49.8 \times 500}{31 \times 144}$$

$$v = 5.58 \text{ ft}^3/\text{lb}$$

$$\rho = \frac{1}{5.58}$$

$$\rho = .18 \text{ lb/ft}^3$$

$$\Delta P = .2 \times .18 \frac{(113)^2}{64.4 \times 144}$$

$$\Delta P = .05 \text{ psi}$$

The up-rate looks feasible considering that none of the inlet nozzle guidelines have been exceeded, the Mach number is still a low value, and the pressure drop is not significant. If the pressure drop had been significant, the effect of the drop could have been evaluated with respect to the compressor head and possibly a usable compromise worked out.

Intercooling

Cooling between compressor stages limits the value of the discharge temperature and reduces energy demands. Normally there would be no argument against intercooling because of the obvious operating cost saving. However, in some process applications, the higher temperature of the gas leaving the compressor can have additional uses such as driving a reboiler. Since heat would have to be added to the gas anyway, it is more economical to use the heat in the gas at the compressor discharge and forego the benefit of the intercooling. However, each application must be evaluated if there is a temperature limit for the gas, or the power savings from cooling overshadows the alternate heat sources available to drive the reboiler.

The capital cost of the coolers, the piping, and the installation must become a part of any evaluation. Figure 2-1 shows a two-intercooler compressor. Cooling water must be added as an operating expense. Air cooling is an alternative, subject, however, to higher outlet temperature and higher capital cost. The extremes in ambient temperature and their effect on operation should not be ignored. An additional consideration is to observe the gas stream for possible condensation of components during cooling. If there is an objection to these components coming out of the stream, then some form of temperature control must be provided. If the condensation is acceptable, provision must still be made to remove this liquid fraction from the coolers before it enters the compressor. Most compressors are quite sensitive to liquid with some more so than others. In the sizing procedure, the loss of the fraction must also be considered in the resulting gas properties for the succeeding stages.

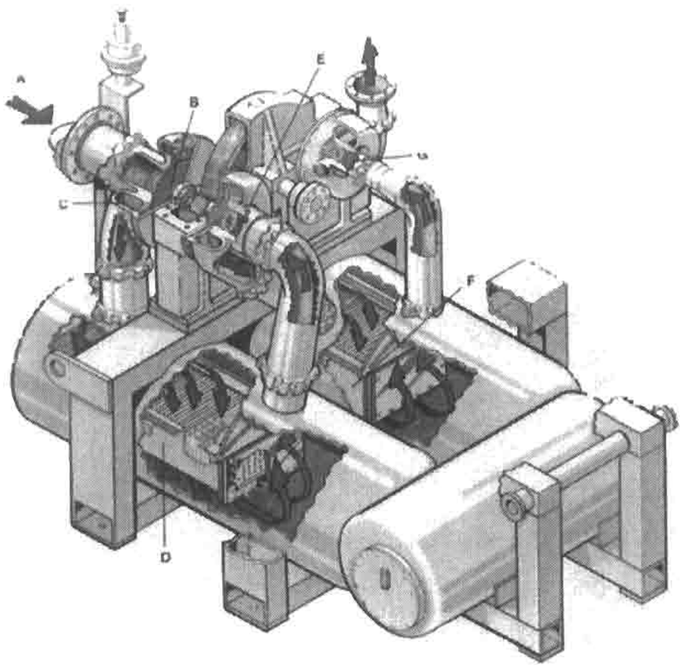

Figure 2-1. Two-water cooled intercoolers on a three-stage air compressor. (*Courtesy of Elliott Company*)

Isothermal Compression

Isothermal compression is presented here to represent the upper limits of cooling and horsepower savings. It is the equivalent of an infinite number of intercoolers and is not achievable in the practical types of compressors described in this book. For an isothermal process,

$$Pv = C \tag{2.83}$$

From this, a theoretical value for the power used by the compressor can be found.

$$W_{it} = wRT\ \mathrm{In}(P_2/P_1) \tag{2.84}$$

This equation is useful when evaluating the benefit of multiple intercoolers, because it establishes the theoretical power limit that can be achieved by cooling. Figure 2-2 is a plot comparing the effect of different numbers of intercoolers in terms of uncooled horsepower. Note the diminishing effect as the number of coolers is increased.

An example illustrates the benefits of intercooling.

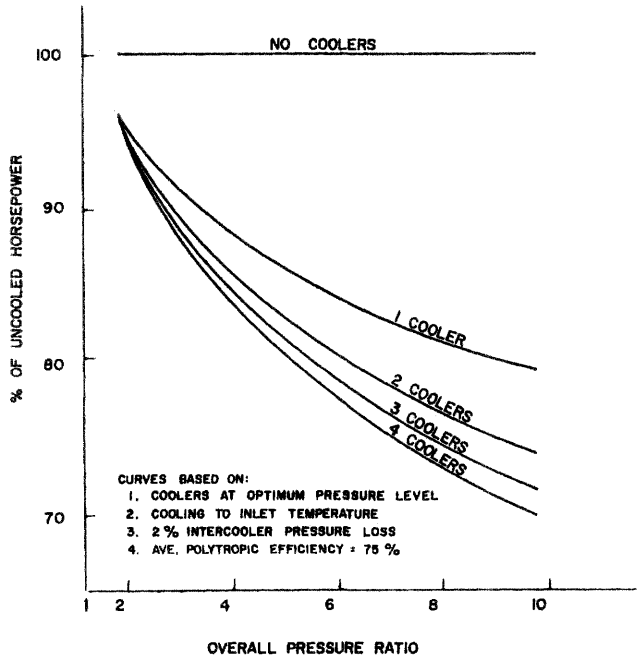

Figure 2-2. Percent of uncooled horsepower required with intercoolers. (*Courtesy of Elliott Company*)

Example 2-4

To keep from complicating the example with real world considerations, a few simplifying assumptions will be made. In all cases, the compressor will be considered to be 100% efficient. Intercooling will be perfect, that is, no pressure drop will be considered and the cooler return gas will be the same temperature to the first stage of the compressor.

Gas: nitrogen
Molecular weight: 29
Ratio of specific heats: 1.4
Inlet pressure: 20 psia
Outlet pressure: 180 psia
Inlet temperature: 80°F
Weight flow: 100 lb/min

Calculate the theoretical power for each case: (1) no intercooling, (2) one intercooler, (3) two intercoolers, (4) isothermal compression.

Because of the assumption that efficiency is 100%, Equations 2.70 and 2.73 yield the same results.

$R = 1545/29$

$R = 53.3$ ft lb/lb °R

$T_1 = 80 + 460$

$T_1 = 540°R$

$k/(k - 1) = 1.4/.4$

$k/(k - 1) = 3.5$

$(k - 1)/k = .4/1.4$

$(k - 1)/k = .286$

$H_a = 1.0 \times 53.3 \times 540 \times 3.5(9^{.286} - 1)$

$H_a = 88{,}107$ ft lb/lb (no intercooler)

Before proceeding with the power calculation, the head for each cooled case will be calculated. In the idealized case, the most efficient division of work for minimum power is achieved by taking the nth root of the pressure ratio, where n is the number of uncooled sections or compression stages in the parlance of the process engineer. For one cooler, n = 2.

$r_p = 9^{1/2}$

$r_p = 3$

For two coolers n = 3.

$r_p = 9^{1/3}$

$r_p = 2.08$

$H_a = 100737(3^{.286} - 1)$

$H_a = 37{,}189$ ft lb/lb (one cooler)

$H_a = 100{,}737(2.08^{.286} - 1)$

$H_a = 23{,}473$ ft lb/lb (two coolers)

Calculate the power using Equation 2.77 and setting $\eta_a = 1.0$ and mechanical losses = 0.

$$W_a = \frac{100 \times 87,983}{33,000 \times 1.0}$$

$$W_a = 267.0\text{hp (no cooler)}$$

Multiply the head by n for the cooled cases.

$$W_a = \frac{100 \times 37,189}{33,000 \times 1.0} \times 2$$

$$W_a = 225.4 \text{ hp (one cooler)}$$

$$W_a = \frac{100 \times 23,473}{33,000 \times 1.0} \times 3$$

$$W_a = 213.3 \text{ hp (two coolers)}$$

Using Equation 2.84 to establish the theoretical limit of isothermal compression,

$$W_{it} = \frac{100 \times 53.3 \times 540 \times \ln(9)}{33,000}$$

$$W_{it} = 191.6 \text{ hp}$$

Taking the horsepower values due to cooling, comparing them to the uncooled case, and converting to percentage,

$$= \frac{225.4}{267.0} \times 100$$

$$= 84.4\% \text{ of the uncooled hp, one cooler}$$

$$= \frac{213.3}{267.0} \times 100$$

$$= 79.9\% \text{ of the uncooled hp, two coolers}$$

$$= \frac{191.6}{267.0} \times 100$$

$$= 71.8\% \text{ of the uncooled hp, isothermal case}$$

It can be seen by comparing the percentages that the benefit of cooling diminishes as each cooler is added. This is particularly noticeable in light of the comparatively small horsepower reduction brought about by isothermal compression since this represents the effect of an infinite number of coolers. The first step was a 15.6% decrease in power, while the second cooler only reduced it another 4.5 percentage points. Even the addition of an infinite number of coolers (isothermal case) added just 12.6 percentage points, a decrease less than the percentage achieved with the first cooler. While the economic impact must be evaluated in each case, this illustration does demonstrate that intercooling does save horsepower. In a practical evaluation, some of the idealized values used in the illustration must be replaced with anticipated actual values, such as real efficiency for the compressor, pressure drops in the coolers and piping, and the true outlet temperature expected from the coolers, based on cooling medium temperature. Because the point was made about the decrease in outlet temperature, these values will be calculated to make the example complete. Note that with temperature as with horsepower, the first increment of cooling yields the largest return. The Equation 2.65 is used to make the calculation.

$$t_2 = 540(9)^{.286} - 460$$

$$t_2 = 552°F \text{ (no cooler)}$$

$$t_2 = 540(3)^{.286} - 460$$

$$t_2 = 279°F \text{ (one cooler)}$$

$$t_2 = 540(2.08)^{.286} - 460$$

$$t_2 = 205°F \text{ (two coolers)}$$

The equations presented in this chapter should have general application to most compressors, particularly the ones to be discussed in the following chapters. As each compressor is covered, additional equations will be introduced.

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3

Reciprocating Compressors

Description

The reciprocating compressor is the patriarch of the compressor family. In the process industry, the reciprocating compressor is probably the oldest of the compressors with wide application ranging from consumer to industrial usage. This compressor is manufactured in a broad range of configurations and its pressure range is the broadest in the compressor family extending from vacuum to 40,000 psig. The reciprocating compressor declined in popularity from the late 1950s through the mid 1970s. Higher maintenance cost and lower capacity, when compared to the centrifugal compressor, contributed to this decline. However, recent rises in energy cost and the advent of new specialty process plants have given the more flexible, higher efficiency, though lower capacity, reciprocating compressor a more prominent role in new plant design.

The reciprocating compressor is a positive displacement, intermittent flow machine and operates at a fixed volume in its basic configuration.

One method of volume variations is by speed modulation. Another, more common method, is the use of clearance pockets, with or without valve unloading. With clearance pockets, the cylinder performance is modified. With valve unloading, one or more inlet valves are physically open. Capacity may be regulated in a single- or double-acting cylinder with single or multiple cylinder configuration.

A unique feature of the reciprocating compressor is the possibility of multiple services on one compressor frame. On a multistage frame, each cylinder can be used for a separate gas service. For example, one cylinder may be dedicated to propane refrigeration, while the balance of the cylinders may be devoted to product gas.

Lubrication of compressor cylinders can be tailored to the application. The cylinders may be designed for normal hydrocarbon lubricants or can be modified for synthetic lubricants. The cylinder may also be designed for self lubrication, generally referred to as nonlubed. A compromise lubrication method that uses the nonlubed design but requires a small amount of lubricant is referred to as the mini-lube system.

An unusual nonlubed compressor is a labyrinth piston compressor, shown in Figure 3-1. The piston does not touch the sides of the cylinder

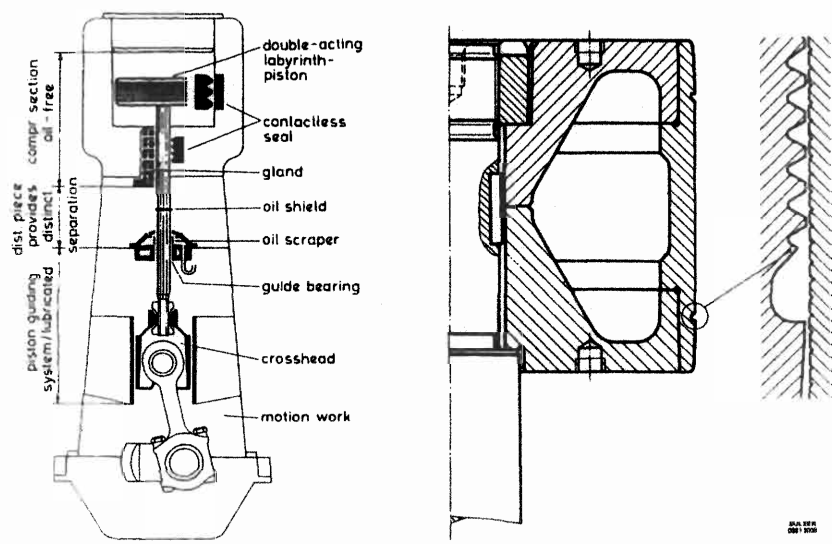

Figure 3-1. Labyrinth piston compressor. This non-lubed piston's circumferential labyrinths operate with a close clearance to the cylinder wall instead of rubbing. (*Courtesy of Sulzer*)

that is equipped with a series of circumferential labyrinths operating with a close clearance to the cylinder wall. Efficiency is sacrificed (due to gas bypass) in order to obtain a low maintenance cylinder. This design is mentioned primarily because it is unique and not widely manufactured.

Another feature necessary to the reciprocating compressor is cylinder cooling. Most process compressors are furnished with water jackets as an integral part of the cylinder. Alternatively, particularly in the smaller size compressors, the cylinder can be designed for air cooling.

Classification

Reciprocating compressors can be classified into several types. One type is the trunk or automotive piston type (see Figure 3-2). The piston is connected to a connecting rod, which is in turn connected directly to the crankshaft. This type of compressor has a single-acting cylinder and is limited to refrigeration service and to smaller air compressors. Most of the smaller packaged refrigeration system compressors are of this type. The compressors may be single or multistage. Approximate capacity is 50 tons in water-chilled refrigeration service and 75 scfm in air service.

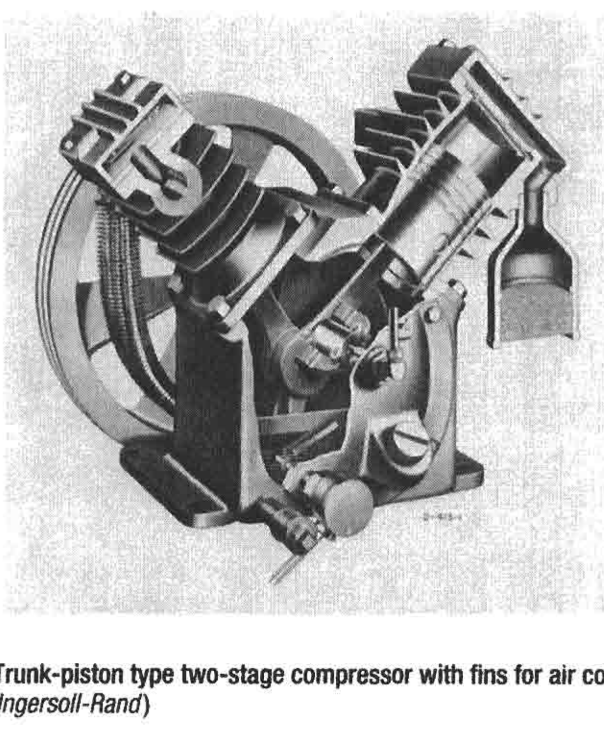

Figure 3-2. Trunk-piston type two-stage compressor with fins for air cooling. (*Courtesy of Ingersoll-Rand*)

The more common type of compressor used in process service is the crosshead type, as shown in Figure 3-3. The piston is driven by a fixed piston rod that passes through a stuffing or packing box and is connected to a crosshead. The crosshead, in turn, is connected to the crankshaft by a connecting rod. In this design, the cylinder is isolated from the crankcase by a distance piece. A variable length or double distance piece is used to keep crankcase lubrication from being exposed to the process gas. This design has obvious advantages for hazardous material. The cylinder can be either single- or double-acting. The double-acting construction uses both sides of the piston and compresses on both strokes of the piston during one revolution. Except for very small compressors, most reciprocating compressors furnished to the process industry use the double-acting configuration.

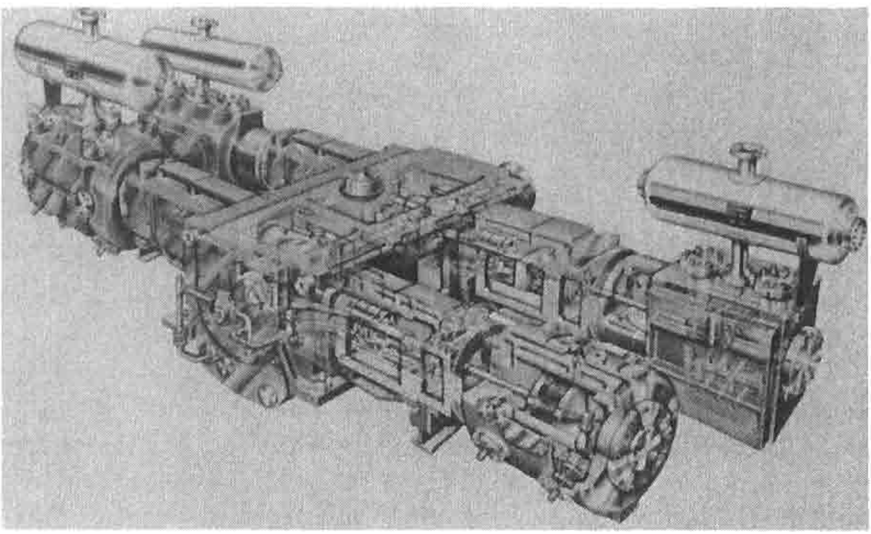

Figure 3-3. Typical multistage crosshead type compressor. (*Courtesy of Nuovo Pignone*)

Arrangement

The trunk type compressor is generally arranged with the cylinder vertical in the basic single-stage arrangement. In the vertical, "in line," multistage configuration, the number of cylinders is normally limited to two. Most multi-cylinder arrangements are in pairs in the form of a V, usually at 45° from the vertical. These compressors usually have up to eight cylinders and are normally used in compressing organic refrigerants.

The few single-acting crosshead compressors are normally single-stage machines with vertical cylinders. The more common double-acting type, when used as a single-stage, commonly has a horizontal cylinder. The double-acting cylinder compressor is built in both the horizontal and the vertical arrangement. There is generally a design trade-off to be made in this group of compressors regarding cylinder orientation. From a ring wear consideration, the more logical orientation is vertical; however, taking into account size and the ensuing physical location as well as maintenance problems, most installations normally favor the horizontal arrangement.

There is wide variation in multistage configuration. The most common is the horizontally opposed. Probably the next most common is the vertical arrangement. Other variations include V, Y, angle or L type. These later arrangements are not too common and are mentioned only to complete possible configurations. Another modification is the tandem-cylinder arrangement, which is almost always horizontal. In this configuration, the cylinders are oriented in line with one another with the innermost cylinder having a piston rod protruding from both ends. This outboard rod in turn drives the next cylinder. While somewhat compact and more competitive in price than the side-by-side arrangements, it is not too popular with maintenance people.

Drive Methods

Another feature of reciprocating compressors that is somewhat unique when compared to the rest of the compressor family is the number of available drive arrangements, which is almost as complex as the cylinder arrangements. In single and multistage arrangement small compressors, particularly the trunk type, are usually V-belt driven by electric motors. The single-acting crosshead type and the small, double-acting, single-stage compressor are also driven in a similar manner. Larger, multistage, trunk type compressors can be sized to operate at common motor speeds and therefore are direct coupled. The larger, crosshead, double-acting, multistage compressors present the most variations in drive arrangements. If it has an integral electric motor sharing a common shaft with the compressor, it is called an engine type. These compressors can also be directly coupled to a separate electric motor in a more conventional manner. Gear units may be involved in the drive train where speed matching is required. Multiple frames are sometimes used with a common crankshaft in a compound arrangement to use a common driver.

Variable frequency motor drives are becoming more popular because of the ability to provide capacity control.

Reciprocating compressors are available with a large variety of other drivers, which include the piston engine, steam turbine, or, in rare cases, a gas turbine. Next in popularity to the electric motor is the piston engine. The arrangement lends itself to skid mounting, particularly with the semi-portable units found in the oilfield. The unit is also popular as a "lease" unit, which may be lifted onto a flat bed trailer and moved from one location to another as needed. The engine is either direct-coupled or, as with smaller compressors, it may be belt-connected.

A variation of the smaller, skid-mounted, engine-driven compressor is a larger, engine-driven version in the form of the integral engine compressor (see Figure 3-4). The compressor and the engine share a common frame and crankshaft. When the engine cylinders are vertical or in a V configuration and the compressor cylinders are horizontal, the machine is called an angle engine compressor.

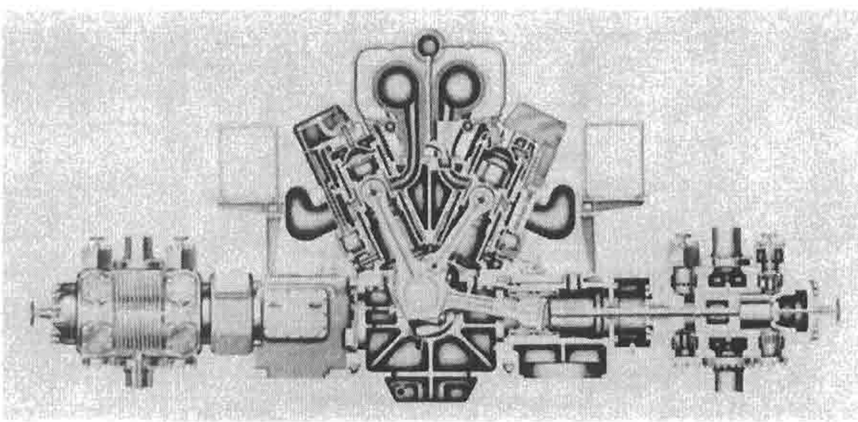

Figure 3-4. Cutaway of a two-stage piston engine driven compressor. (*Courtesy of Dresser-Rand*)

A more rare form of driver is the steam cylinder. Most arrangements combine the steam driver and compressor on the same frame with the steam cylinder opposite the compressor cylinder. Each cylinder's connecting rod is connected to a common throw on the crankshaft. A flywheel is used to provide inertia. For air service, the units are built as single- and two-stage units, with other combinations available for process service.

Performance

Compression Cycle

For the following discussion, refer to Figure 3-5, which shows an ideal indicator diagram followed by a series of cylinder illustrations depicting piston movement and valve position. The figure shows in diagram form one

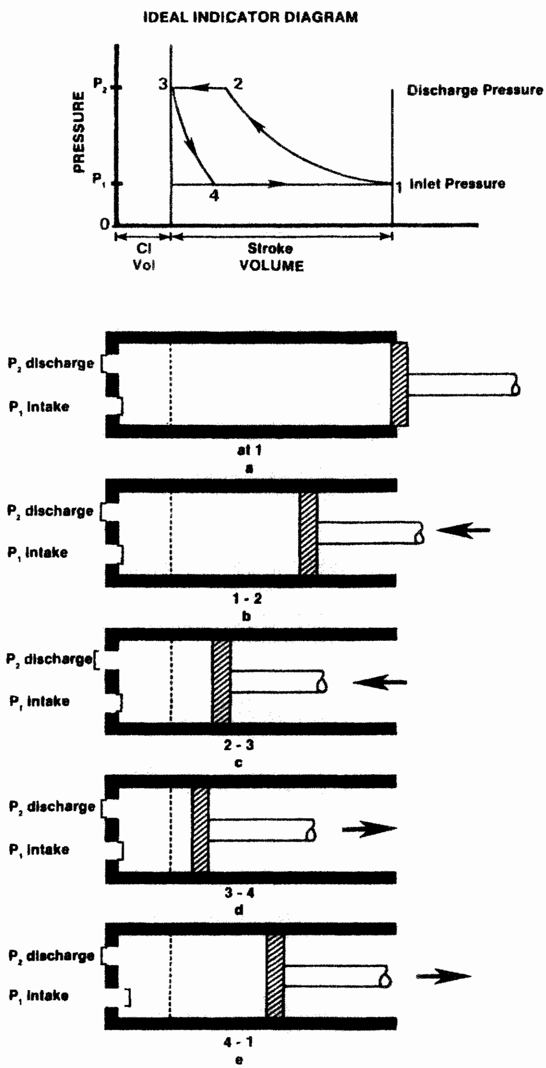

Figure 3-5. Steps in the cycle of reciprocating compressor.

complete crankshaft revolution and encompasses a complete compression cycle. To begin the cycle, refer to the figure at (a) the location where the piston is at the lower end of the stroke (bottom dead center) and is at path point 1 on the indicator diagram. At this point, the cylinder has filled with gas at intake pressure P_1. Note that the valves are both closed. At (b), the piston has started to move to the left. This is the compression portion of the cycle and is illustrated by Path 1-2. When the piston reaches point 2 on the indicator diagram, the exhaust valve starts to open. The discharge portion of the cycle is shown at (c). This is shown on the indicator diagram Path 2-3. Note that the discharge valve is open during this period while the intake valve is closed. The gas is discharged at the discharge line pressure P_2. When the piston reaches point 3, it has traveled to the upper end of its stroke (top dead center). Physically, at this point in the stroke, there is a space between the piston face and the head. This space results in a trapped volume and is called the clearance volume. Next in the cycle, the piston reverses direction and starts the expansion portion of the cycle, as illustrated at (d) in the figure. Path 3-4 shows this portion of the cycle. Here the gas trapped in the clearance volume is re-expanded to the intake pressure. Note that the discharge valve has closed, and the intake valve is still closed. At point 4, the expansion is complete and the intake valve opens. The intake portion of the cycle is shown at (e). This is indicated by Path 4-1 on the indicator diagram. The cylinder fills with gas at intake line pressure P_I. When the piston reaches point 1, the cycle is complete and starts to repeat.

Cylinder Displacement

The calculation of the cylinder displacement is a straightforward geometric procedure. It is the product of three factors, namely, the piston area minus rod area (when appropriate), the stroke, and the number of strokes in a given time. There are four options, which can be covered by three equations.

For a single-acting cylinder compressing at the outer end of the cylinder.

$$Pd = S_t \times N \times \frac{\pi D^2}{4} \tag{3.1}$$

where

Pd = piston displacement
S_t = stroke
N = speed of the compressor
D = cylinder diameter

For a double-acting cylinder without a tail rod,

$$Pd = S_t \times N \times \frac{\pi(2D^2 - d^2)}{4} \tag{3.2}$$

where

d = piston rod diameter

For a double-acting cylinder with a tail rod,

$$Pd = S_t \times N \times \frac{2\pi(D^2 - d^2)}{4} \tag{3.3}$$

For the application requiring a single-acting cylinder compressing on the frame end only, use Equation 3.3 deleting the 2 in the expression.

Volumetric Efficiency

To determine the actual inlet capacity of a cylinder, the calculated displacement must be modified. There are two reasons why modification is needed. The first is because of the clearance at the end of the piston travel.

Earlier in the chapter, when the compression cycle was described, a portion of the indicator, Path 3-4, was referred to as the expansion portion of the cycle. The gas trapped in the clearance area expands and partly refills the cylinder taking away some of the capacity. The following equation reflects the expansion effect on capacity and is referred to as the theoretical volumetric efficiency E_{vt}.

$$E_{vt} = 1.00 - [(1/f)r_p^{1/k} - 1]c \tag{3.4}$$

where

f = ratio of discharge compressibility to inlet compressibility as calculated by Equation 3.6
r_p = pressure ratio
c = percent clearance
k = isentropic exponent

The limit of the theoretical value can be demonstrated by substituting zero for the clearance c, which results in a volumetric efficiency multiplier of 1.0.

The second reason for modification of the displaced volume is that in real world application, the cylinder will not achieve the volumetric performance predicted by Equation 3.4. It is modified, therefore, to include empirical data. The equation used here is the one recommended by the Compressed Air and Gas Institute [1], but it is somewhat arbitrary as there is no universal equation. Practically speaking, however, there is enough flexibility in guidelines for the equation to produce reasonable results. The 1.00 in the theoretical equation is replaced with .97 to reflect that even with zero clearance the cylinder will not fill perfectly. Term L is added at the end to allow for gas slippage past the piston rings in the various types of construction. If, in the course of making an estimate, a specific value is desired, use .03 for lubricated compressors and .07 for nonlubricated machines. These are approximations, and the exact value may vary by as much as an additional .02 to .03.

$$E_v = .97 - [(1/f)r_p^{1/k} - 1]c - L \tag{3.5}$$

$$f = Z_2/Z_1 \tag{3.6}$$

The inlet capacity of the cylinder is calculated by

$$Q_1 = E_v \times Pd \tag{3.7}$$

Piston Speed

Another value to be determined is piston speed, PS. The average piston speed may be calculated by

$$PS = 2 \times S_t \times N \tag{3.8}$$

The basis for evaluation of piston speed varies throughout industry. This indicates that the subject is spiced with as much emotion as technical basics. An attempt to sort out the fundamentals will be made. First, because there are so many configurations and forms of the reciprocating compressor, it would appear logical that there is no one piston speed limit that will apply across the board to all machines. The manufacturer is at odds with the user because he would like to keep the speed up to keep the size of the compressor down, while the user would like to keep the speed down for reliability purposes. As is true for so many other cases, the referee is the economics. An obvious reason to limit the speed is maintenance expense. The lower the piston speed, the lower the maintenance and the higher the reliability. The relationship given by Equation 3.1 defines the size of the cylinder. Therefore, if the speed is reduced to lower the piston speed, then the diameter of the cylinder must increase to compensate for the lost displacement to maintain the desired capacity. As cylinder size goes up, so does the cost of the cylinder. It is not difficult to see why the user and manufacturer are at somewhat of a cross purpose. If the user's service requires a high degree of reliability and he wants to keep cylinder and ring wear down, he must be aware of the increase in cost.

To complicate the subject of piston speed, look at Equations 3.1 and 3.8. Note the term S_t (stroke). The piston speed can be controlled by a shorter stroke, but because of loss of displacement, the diameter and/or the speed must be increased. If only speed is increased, the whole exercise is academic as the piston speed will be back up to the original value. If, however, diameter alone or both diameter and speed are increased, the net result can be a lower piston speed. Another factor comes to bear at this point concerning valve life, that decreases with the increase in the number of strokes and can negate the apparent gain in maintenance cost if not adequate. It would appear that the engineer trying to evaluate a compressor bid just can't win. The various points are not tendered just to frustrate the user but rather are given to help show that this is another area that must have a complete evaluation. All facets of a problem must be considered before an intelligent evaluation can be made.

After all the previous statements, it would seem very difficult to select a piston speed. For someone without direct experience, the following guidelines can be used as a starting point. Actual gas compressing experience should be solicited when a new compressor for the same gas is being considered. These values will apply to the industrial process type of compressor with a double-acting cylinder construction. For horizontal compressors with lubricated cylinders, use 700 feet per minute (fpm) and for nonlubricated cylinders use 600 fpm. For vertical compressors with lubricated cylinders, use 800 fpm and for nonlubricated cylinders use 700 fpm.

Another factor to consider is the compressor rotative speed relative to valve wear. The lower the speed, the fewer the valve cycles, which contribute to longer valve life. A desirable speed range is 300 to 600 rpm.

Discharge Temperature

While head is normally not a particularly significant value in the selection of the reciprocating compressor, it is used for comparison with other types of compressors. Equation 2.66, the equation for adiabatic head, is recalled as

$$H_a = RT_1 \frac{k}{k-1}(r_p^{\frac{k-1}{k}} - 1) \tag{2.66}$$

The *discharge temperature* can be calculated by rewriting Equation 2.65.

$$T_2 = T_1 \left(r_p^{\frac{k-1}{k}} \right) \tag{3.9}$$

where

T_1 = absolute inlet temperature
T_2 = absolute discharge temperature

Why use an adiabatic relationship with a compressor whose cylinder is almost always cooled? An assumption made in Chapter 2 on adiabatic isentropic relationships was that heat transfer was zero. In practical applications, however, the cooling generally offsets the effect of efficiency. As a side note, cylinder cooling is as much cylinder stabilization for the various load points as it is heat removal.

Power

The work-per-stage can be calculated by multiplying the adiabatic head by the weight flow per stage.

$$\text{Work} = H_a \times w \tag{3.10}$$

then,

$$W_{cyl} = wRT_1 \frac{k}{k-1}\left(r_p^{\frac{k-1}{k}} - 1 \right) \tag{3.11}$$

Substituting P_1Q_1 for wRT_1 from Equation 2.7,

$$\text{Work} = P_1 Q_1 \frac{k}{k-1}\left(r_p^{\frac{k-1}{k}} - 1 \right) \tag{3.12}$$

For two stages, the above equation can be expanded to add the interstage conditions for the second stage. Note the subscript i is added to the second set of terms to reflect the second-stage inlet.

$$\text{Work} = P_1 Q_1 \frac{k}{k-1}\left(r_p^{\frac{k-1}{k}} - 1\right) + P_i Q_i \frac{k}{k-1}\left(r_{pi}^{\frac{k-1}{k}} - 1\right) \tag{3.13}$$

For a first trial at sizing or for estimates, the Equation 3.13 can be differentiated and solved for P_i, with the result,

$$P_i = \sqrt{P_1 \times P_2} \tag{3.14}$$

This expression can be changed to

$$\frac{P_i}{P_1} = \frac{P_2}{P_i} \tag{3.15}$$

Substituting the term r for the pressure ratio, the following results

$$r_{p1} = r_{pi} \tag{3.16}$$

Equation 3.16 can be generalized for optimum work division by dividing the pressure ratio into a set of balanced values,

$$r_{p\text{-stage}} = (r_{p\text{-overall}})^{1/n\text{-stage}} \tag{3.17}$$

The values for pressure ratio in a practical case must include allowance for pressure drop in the interstage piping. In the sizing procedure used by manufacturers, certain adjustments must be made to the ideal for incremental cylinder sizes and allowable rod loading. Efficiency is represented by η_{cyl}.

$$W_{cyl} = \frac{P_1 Q_1}{\eta_{cyl}} \frac{k}{k-1} (r_p^{\frac{k-1}{k}} - 1) \tag{3.18}$$

To assist the engineer in making estimates, the curve in Figure 3-6 gives values of efficiency plotted against pressure ratios. The values on the curve include a 95% mechanical efficiency and a valve velocity of 3,000 feet per minute. Table 3-1 and Table 3-2 are included to permit a correction to be made to the compressor horsepower for specific gravity and low inlet pressure. They are included to help illustrate the influence of these factors to the power required. The application of these factors to

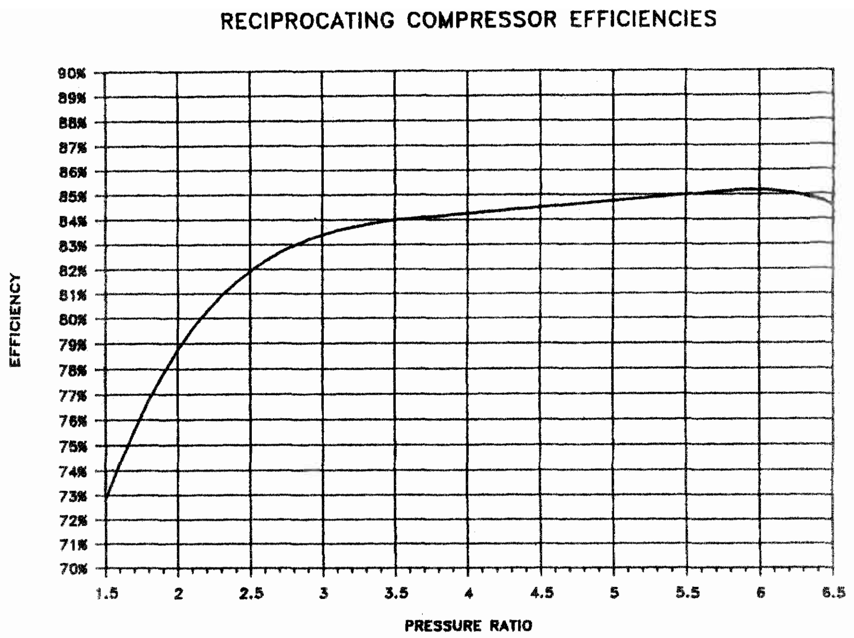

Figure 3-6. Reciprocating compressor efficiencies plotted against pressure ratio with a valve velocity of 3,000 fpm and a mechanical efficiency of 95%.

Table 3-1
Efficiency Multiplier for Specific Gravity

	SG				
r_p	1.5	1.3	1.0	0.8	0.6
2.0	0.99	1.0	1.0	1.0	1.01
1.75	0.97	0.99	1.0	1.01	1.02
1.5	0.94	0.97	1.0	1.02	1.04

Source: Modified courtesy of the Gas Processors Suppliers Association.

Table 3-2
Efficiency Multiplier for Low Pressure

	Pressure Psia							
r_p	10	14.7	20	40	60	80	100	150
3.0	.990	1.00	1.00	1.00	1.00	1.00	1.00	1.00
2.5	.980	.985	.990	.995	1.00	1.00	1.00	1.00
2.0	.960	.965	.970	.980	.990	1.00	1.00	1.00
1.5	.890	.900	.920	.940	.960	.980	.990	1.00

Source: Modified courtesy of the Gas Processors Association and Ingersoll-Rand.

efficiency value is arbitrary. While it is recognized that the efficiency is not necessarily the element affected, the desire is to modify the power required per the criteria in the tables.

The efficiency correction accomplishes this. These corrections become more significant at the lower pressure ratios.

Valve Loss

The efficiency values are affected by several losses: ring slippage, packing leakage, and valve losses. Valve losses are generally the most significant and are made of several components such as channel loss, loss in the valve opening, and leakage. Also, because of inertia and imperfect damping properties of the gas, the valve may have transient losses due to *bounce.* The manufacturer, therefore, modifies the valve lift to suit the gas specified. For example, an air compressor might be furnished with a lift of .100 inch. The same compressor being furnished for a low molecular service such as a hydrogen-rich gas, might use a lift of .032 inches. The problem with the higher lift is that hydrogen lacks the damping properties of air and, as a result, the valve would experience excessive bounce. The effect on the compressor would be loss in efficiency and higher valve maintenance.

The valve porting influences volumetric efficiency by contributing to the minimum clearance volume. If the porting must be enlarged to reduce the flow loss, it is done at the expense of minimum clearance volume.

This is just one example of the many compromises the engineer is faced with while designing the compressor. The subject of valve design is involved and complex. For individuals wishing to obtain more information on the subject, references discussing additional aspects of valves are included at the end of the chapter [2, 3, 4, 5, 6].

To calculate the valve velocity for evaluation purposes, use the following equation. This equation is based on the equation given in API 618.

$$v = 144\,\frac{Pd}{A} \tag{3.19}$$

where

v = average gas velocity, fpm
Pd = piston displacement per cylinder, ft^3/min
A = total inlet or discharge valve area per cylinder, in.^{2}

To calculate Pd, use Equation 3.1 for single-acting cylinders, Equation 3.2 for double-acting cylinders without a tail rod, and Equation 3.3 for double-acting cylinders with a tail rod.

The area, A, is the product of actual lift and the valve opening periphery and is the total for all inlet or discharge valves in a cylinder. The lift is a compressor vendor-furnished number.

Example 3-1

Calculate the suction capacity, horsepower, discharge temperature, and piston speed for the following single-stage double acting compressor.

Bore: 6 inches
Stroke: 12 inches
Speed: 300 rpm
Rod diameter: 2½ inches
Clearance: 12%
Gas: CO_2
Inlet pPressure: 1,720 psia
Discharge pressure: 3,440 psia
Inlet temperature: 115°F

Calculate the piston displacement using Equation 3.2 and dividing by 1,728 in.^{3} per ft^3 to convert the output to cfm.

$$\text{Pd} = \frac{12 \times 300 \times \pi\,[2(6)^2 - (2.5)]}{1{,}728 \times 4}$$

$$\text{Pd} = 107.6\ \text{cfm}$$

Step 1. Calculate volumetric efficiency using Equations 3.5 and 3.6. To complete the calculation for volumetric efficiency, the compressibilities are needed to evaluate the f term of Equation 3.6. Using Equations 2.11 and 2.12 for the inlet conditions,

$$T_1 = 460 + 115$$

$$T_1 = 575°\text{R}$$

$$T_r = \frac{575}{548}$$

$$T_r = 1.05$$

$$P_r = \frac{1,720}{1,073}$$

$$P_r = 1.6$$

From the generalized compressibility charts (see Appendix B),

$$Z_1 = .312$$

Step 2. At this point the discharge temperature must be calculated to arrive at a value for the discharge compressibility.

$$r_p = \frac{3,440}{1,720}$$

$$r_p = 2.0$$

$$T_2 = 575\ [2^{(1.3-1)/1.3}]$$

$$T_2 = 674\ 7°R$$

$$t_2 = 674.7 - 460$$

$t_2 = 214.7°F$ discharge temperature

Calculate the discharge compressibility:

$$T_r = \frac{674.7}{548}$$

$$T_r = 1.23$$

$$P_r = \frac{3,440}{1,073}$$

$$P_r = 3.21$$

From the generalized compressibility charts,

$$Z_2 = 0.575$$

From Equation 3.6, calculate f:

$$f = \frac{.575}{.312}$$

$$f = 1.842$$

Calculate the volumetric efficiency using Equation 3.5. Use .05 for L because of the high differential pressure:

$$E_v = .97 - [(1/1.842)(2)^{1/1.3} - 1].12 - 0.5$$

$$E_v = .93 \text{ volumetric efficiency}$$

Now calculate suction capacity using Equation 3.7:

$$Q_1 = .93 \times 107.6$$

$$Q_1 = 100.1 \text{ cfm suction capacity}$$

Step 3. Piston speed is calculated using Equation 3.8 converting the stroke to feet by dividing the equation by 12 inches per foot:

$$PS = \frac{2 \times 12 \times 300}{12}$$

$$PS = 600 \text{ fpm piston speed}$$

Step 4. Calculate the power required. Refer to Figure 3-6 and select the efficiency at a pressure ratio of 2.0. The value from the curve is 79%. Equation 3.18 is used to calculate power. The constants 144 in^2ft^2 and 33,000 ft-lbs/min/hp have been used to correct the equation for the unit from the example.

$$W_{cyl} = \frac{144 \times 1,720 \times 100.1}{33,000 \times .79} \times \frac{1.3}{.3}(2^{.3/1.3} - 1)$$

$$W_{cyl} = 714.8 \text{ hp cylinder horsepower}$$

Application Notes

There are several items regarding the application of reciprocating compressors that must be considered. These items are minor, but if neglected may cause a great deal of concern when the inevitable problem occurs.

Reciprocating compressors are not fond of liquids of any sort, particularly when delivered with the inlet gas stream. For any application, a good-sized suction drum with a drain provision is in order. It may be a part of the pulsation control if properly done. The pulsation control will be covered in more detail later in the chapter. If the stream is near saturation or has a component near saturation, consideration should be given to using a horizontally oriented cylinder configuration, with the discharge nozzle on the bottom side of the cylinder. While on the subject of condensation, for the same gas near saturation, cylinder cooling must be monitored and controlled. It would not do to let the gas condense inside the cylinder after all the care has been taken not to let it condense outside the cylinder. A rule of thumb is to keep the cooling water temperature 10°F above the gas inlet temperature.

It would appear obvious for startup, and in some cases full-time operation, that a suction strainer or filter is mandatory. The reason for the strainer is to keep junk and pipe scale out of the compressor. Fines from pipe scale and rust will make short work of the internal bore of a cylinder and are not all that good for the balance of the components. In some severe cases, cylinders have been badly damaged in a matter of a few weeks. The strainer should be removable in service for cleaning, particularly when it is intended for permanent installation. Under all circumstances, provision must be made to monitor the condition of the strainer. Much frustration has been expended because a compressor overheated or lost capacity and no one knew if the strainer had fouled or blinded.

The discharge temperature should be limited to 300°F as recommended by API 618. Higher temperatures cause problems with lubricant coking and valve deterioration. In nonlube service, the ring material is also a factor in setting the temperature limit. While 300°F doesn't seem all that hot, it should be remembered that this is an average outlet temperature, whereas the cylinder will have "hot" spots exceeding this temperature.

Finally, planning may save money and time if process changes are foreseeable. For instance, capacity increase, or an increase in molecular weight due to a catalyst change, results in decreased volumetric flow. Although the cylinders must be sized for economical operation at the present rate, the frame can be sized for future applications. When the future conditions become a reality, the cylinders can be changed while keeping the same frame. This saves the investment cost and delivery time of a complete new compressor without the penalty of oversizing and its inherent inefficient operation.

Mechanics

Cylinders

Cylinders for compressors used in the process industries are separable from the frame. They are attached to the frame by way of an intermediate part known as the distance piece and can be seen in Figure 3-3. Piloting is provided to maintain alignment of all moving elements. A requirement of API 618 is for the cylinders to be equipped with replaceable liners. The purpose of the liner is to provide a renewable surface to the wearing portion of the cylinder. This saves the cost of replacing a complete cylinder once the bore has been worn or scored. In the larger, more complex compressors, this feature is standard or readily available as an option. On the smaller frames, particularly the single-stage models, the smaller cylinder size is such that the replaceable liner is not economical and may not be available.

All cylinders are equipped for cooling, usually by means of a water jacket. Those not having a water jacket are finned to provide air cooling. The latter method is limited to either small or special purpose machines.

The most common material used in cylinder construction is cast iron for the larger, low-pressure cylinders and steel for the smaller, high pressure cylinders. In some cases, nodular or ductile iron can be used in lieu of cast iron. For hydrocarbon service, steel is most desirable, although not universally available.

Larger cylinders normally have enough space for clearance pockets. An additional location is the head casting on the outboard end of the cylinder. Figure 3-7 is an illustration of a cylinder with an unloading pocket in the head. On smaller cylinders, this feature must be provided external to the cylinder.

Pistons and Rods

The lowly piston, one of the more simple items, has one of the most important functions of the entire compressor. The piston must translate the energy from the crankshaft to the gas in the cylinder. The piston is

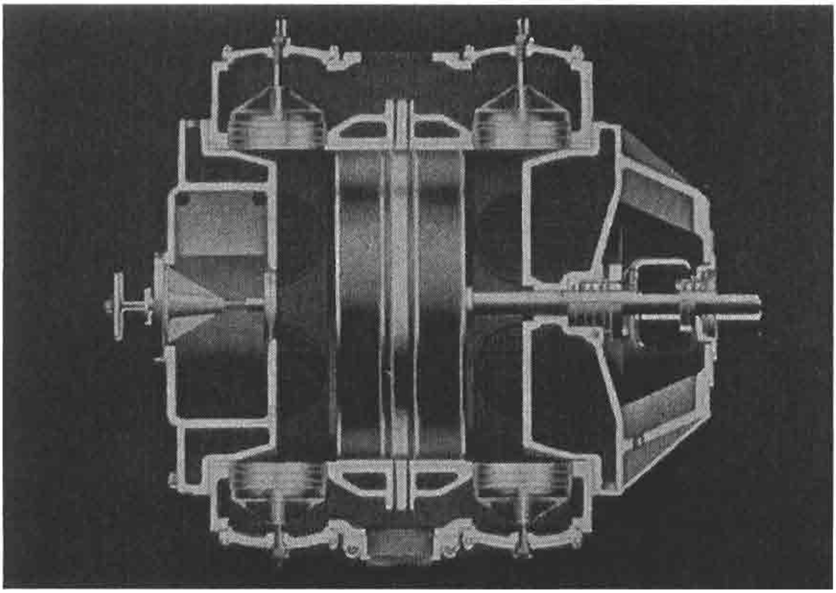

Figure 3-7. Cylinder with clearance pocket. (*Courtesy of Dresser-Rand*)

equipped with a set of sliding seals referred to as piston rings. Rings are made of a material that must be reasonably compliant for sealing, yet must slide along the cylinder wall with minimum wear. Different rings are used for lubricated or nonlubricated service, with the rings in the nonlubed cylinders needing good dry lubricating qualities. For lubricated service, metallic rings such as cast iron or bronze as well as nonmetallic materials such as filled nylon are used. The nonmetallic materials are becoming more common. For nonlubricated service, the ring material is nonmetallic, ranging from carbon to an assortment of fluorocarbon compounds. Horizontal cylinder pistons feature the addition of a wear band, sometimes referred to as a rider ring (see Figure 3-8).

Pistons may be of segmented construction to permit the use of one-piece wear bands. One-piece wear bands are a requirement in API 618. Pistons have a problem in common with humans—a weight problem. Weight in a piston contributes directly to the compressor shaking forces and must be controlled. For this reason, aluminum pistons are often found in larger low pressure cylinders. Hollow pistons are used but can pose a hazard to maintenance personnel if not properly vented. If trapped, the gas will be released in an unpredictable and dangerous manner when the piston is dismantled.

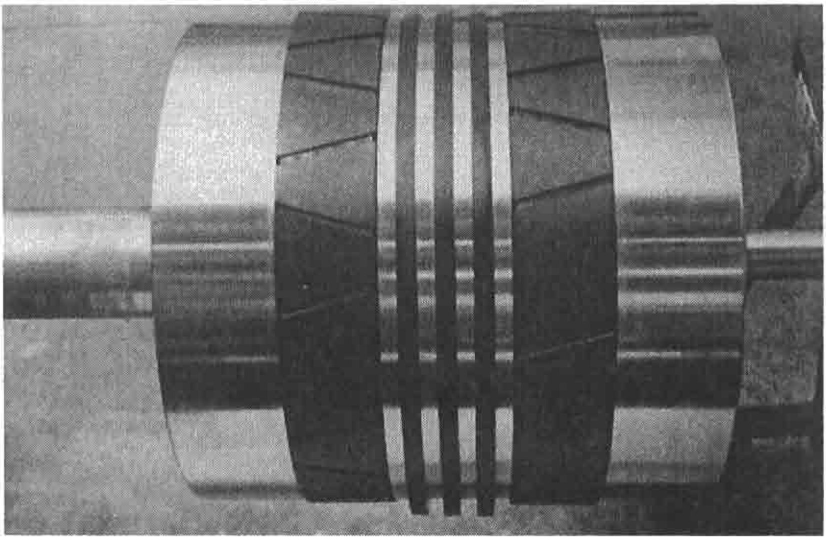

Figure 3-8. Piston rings and wear band. (*Courtesy of Nuovo Pignone*)

The piston rod is threaded to the piston and transmits the reciprocating motion from the crosshead to the piston. The piston rod is normally constructed of alloy steel and must have a hardened and polished surface, particularly where it passes through the cylinder packing (double-acting cylinders). Rod loading must be kept within the limits set by the compressor vendor because overloading can cause excess runout of the rod resulting in premature packing wear. This in turn leads to leakage, reduced efficiency, and increased maintenance expense.

In unloaded or part-load operation, rod reversals must be of sufficient magnitude to provide lubrication to the crosshead bearings. The bearings are lubricated by the pumping action of the opening and closing of the bearing clearance area.

Tail rods are dummy rods that protrude from the head end of the cylinder (see Figure 3-9). The purpose of the rod is to pressure-balance a piston or to stabilize a particular piston design. Because of the personnel hazard, a guard must be specified and provided. In a tandem cylinder arrangement, the outboard cylinders are driven with a rod similar to the tail rod.

Valves

The compressor cylinder valves are of the spring-loaded, gas-actuated type in all but a limited number of portable compressors. This kind of

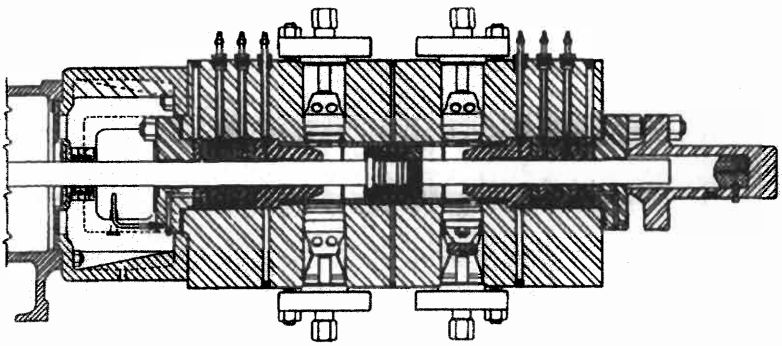

Figure 3-9. Diagram of cylinder with piston tail rod. (*Courtesy of Dresser-Rand*)

valve is used in contrast to the cam-actuated poppet type normally found in piston engines. Reciprocating compressors generally use one of four basic valve configurations:

- rectangular element
- concentric ring
- ported plate
- poppet

The rectangular element valve, as the name implies, uses rectangular-shaped sealing elements. These valves are the feather valve, channel valve, and the reed valve. These valves are applied to the industrial air machines for the most part. A channel valve is shown in Figure 3-10.

The concentric ring valve uses one or more relatively narrow rings arranged concentrically about the centerline of the valve (see Figure 3-11). These valves have the advantage of a low stress level due to the lack of stress concentration points. The disadvantage is that it is difficult to maintain uniform flow control with the independent rings. These valves work well with plug type unloaders. Space for the unloader is obtained by eliminating one or more of the innermost rings.

The ported plate valves, as shown in Figure 3-12, are similar to the concentric ring valve except that the rings are joined into a single element. The advantage is that the valve has a single element making flow control somewhat easier. Because of the single element, the number of edges available for impact is reduced. The valve may be mechanically damped, as this design permits the use of damping plates. It has the disadvantage that because of the geometry used, the stress is higher due to the potential of higher stress concentrations. This valve element is probably one of the most commonly used in process reciprocating compressors.

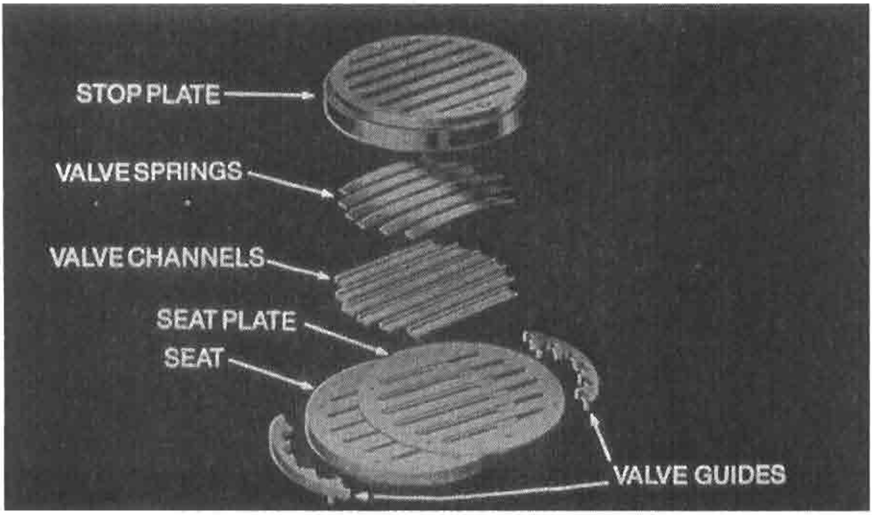

Figure 3-10. An exploded view of a cushioned channel valve. (*Courtesy of Dresser-Rand*)

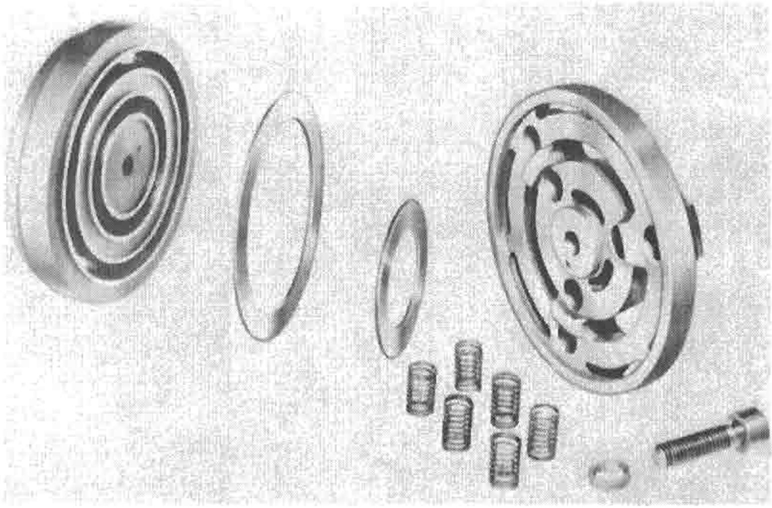

Figure 3-11. Exploded view of a concentric plate valve. (*Courtesy of Dresser-Rand*)

The poppet valve (see Figure 3-13) consists of multiple, same-size ports and sealing elements. The advantage of the valve is that has a high flow efficiency due to the high lift used and the streamlined shape of the sealing element. The disadvantage is that the valve is not tolerant of

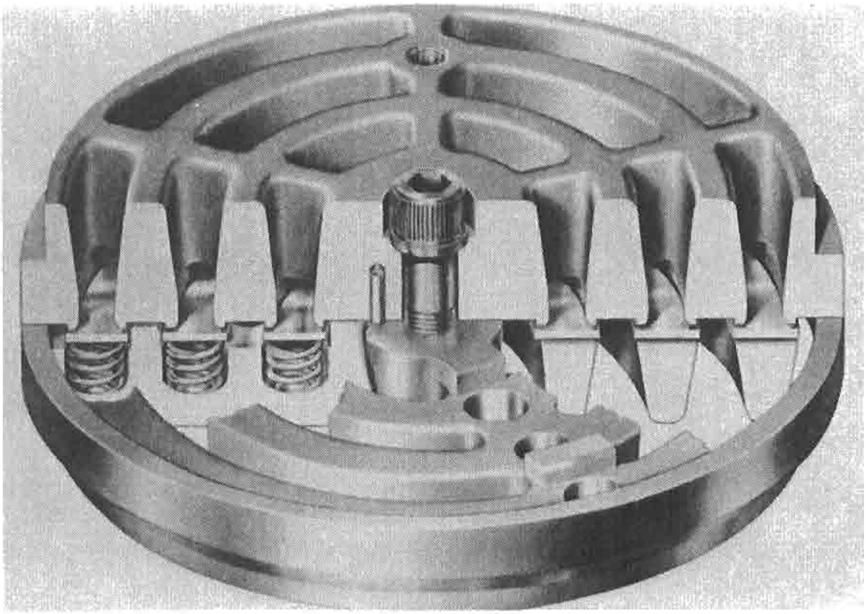

Figure 3-12. Cutaway of a ported plate valve. (*Courtesy of Dresser-Rand*)

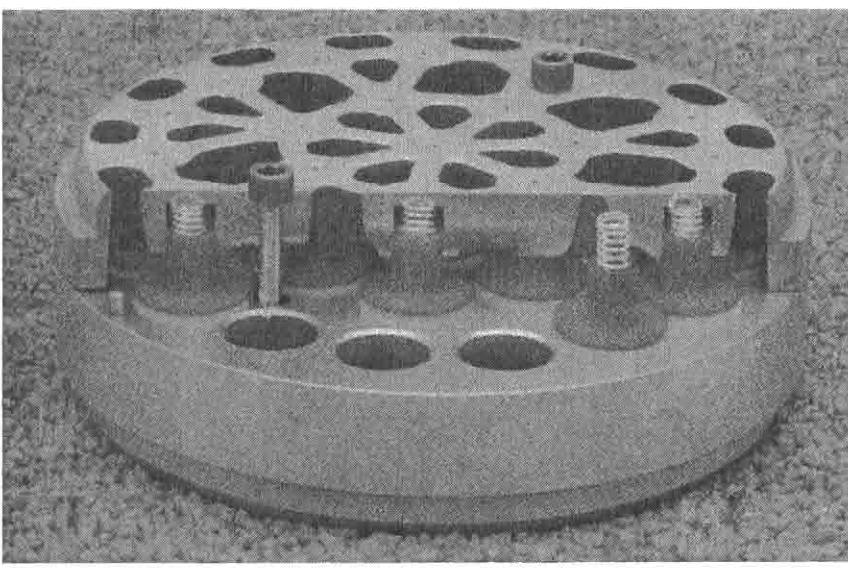

Figure 3-13. Cutaway of a poppet valve. (*Courtesy of Dresser-Rand*)

uneven flow distribution. The valve is most commonly used in gas transmission service and in low speed, low-to-medium compression ratio compressors. There appears to be an increase in the use of poppet valves in hydrocarbon process service because of the ease of maintenance.

Valve materials must be selected for durable, long-term operation and must also be compatible with the gas being handled. The use of polymer nonmetallic sealing elements is quite common. The valves are symmetrically placed around the outer circumference of the cylinder and can normally be removed and serviced from outside the cylinder without dismantling any other portion. A good design will have the valve and associated parts so arranged that an assembly cannot be installed backwards. The inlet and discharge valves should not be physically interchangeable and should be so constructed as to keep the valve assembly or its parts from entering the cylinder should they become unbolted or break.

Distance Piece

The distance piece is a separable housing that connects the cylinder to the frame. The distance piece may be open or closed and may have multiple compartments. It may be furnished as single, double, or extra long. The purpose of a longer distance piece is to isolate that part of the rod entering the crankcase and receiving lubrication from the part entering the cylinder and contacting the gas. This prevents lubricant from entering the cylinder and contaminating the gas, particularly necessary in nonlubricated cylinders. It can also keep a synthetic lubricant in a cylinder from being corrupted by the crankcase lubricant.

Compartments in the distance piece collect and control packing leakage when the gas is toxic or flammable. Today, the toxic category covers many of the gases that were allowed to freely escape into the atmosphere not many years ago. With the pollution laws becoming more stringent, leakage control takes on a much greater significance. The leakage can be directed to a flare or other disposal point and, as with multiple compartments, a buffer of inert gas can be used together with the collection compartment to further prevent gas leakage.

Rod Packing

A packing is required on double-acting cylinders to provide a barrier to leakage past the rod where it passes through the crank end cylinder closure. The same arrangement is needed at the head end if a tail rod or tandem cylinder is used. The packing may consist of a number of rings of packing material and may include a lantern ring (see Figure 3-14). The lantern ring provides a space into which a gas or liquid may be injected to aid in the sealing process. If cooling of the packing is required, the packing box may be jacketed for liquid coolant.

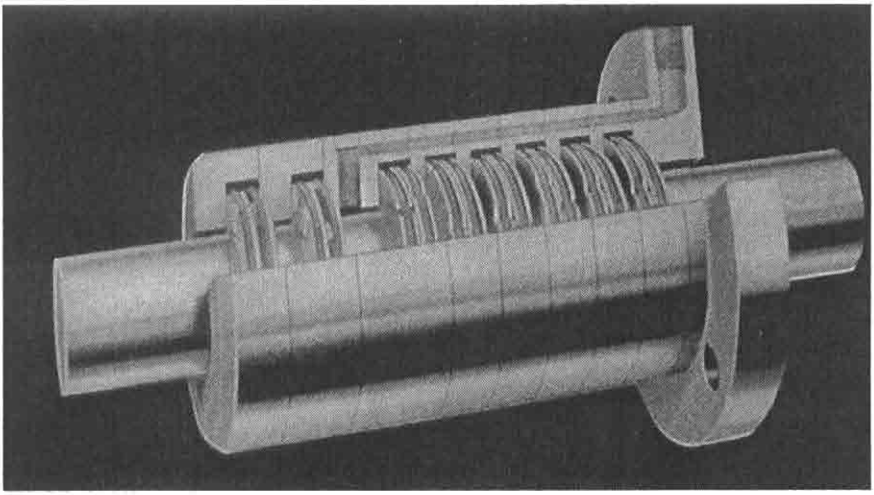

Figure 3-14. Rod packing box. Lantern rings in packing provide space into which a buffer may be injected to aid in sealing. (*Courtesy of Dresser-Rand*)

Crankshaft and Bearings

Larger compressors, normally above 150 to 200 horsepower, have forged steel crankshafts. Cast crankshafts are used in medium-size machines. Crankshafts should have removable balance weights to compensate for rotary unbalance as well as reciprocating unbalance. The crankshaft should be dynamically balanced when above 800 rpm.

When pressure lubrication is used, the crankshaft oil passages should be drilled rather than cored in the cast construction. Figure 3-15 shows a drilled crankshaft. On machines above 150 horsepower, the main and connecting rod bearings should be split-sleeve, steel-backed, babbitted-insert type. Figure 3-16 shows a connecting rod. The main bearings of smaller compressors are the rolling element type. Crosshead pins should

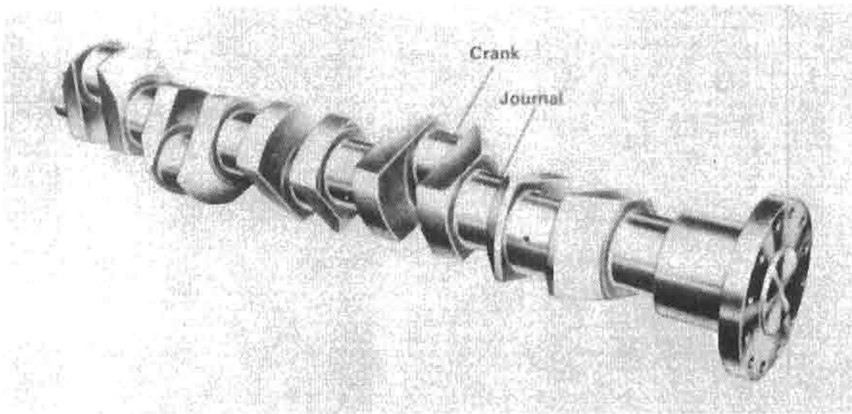

Figure 3-15. A five throw crankshaft with drilled oil passages. (*Courtesy of Dresser-Rand*)

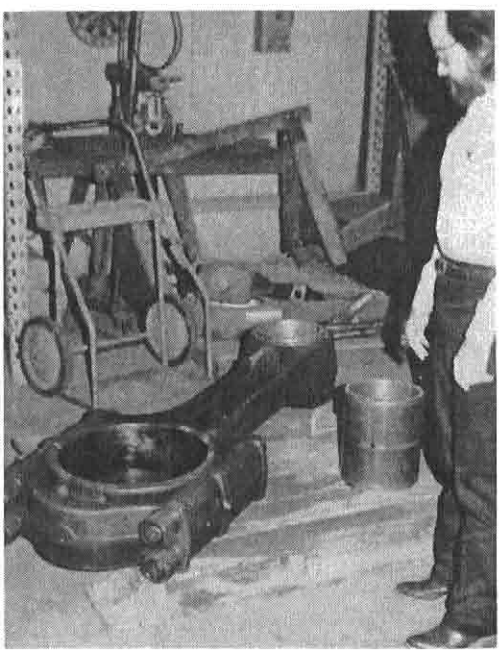

Figure 3-16. A large connecting rod.

have replaceable bushings if available. See Figure 3-17 for some typical crossheads. Figure 3-18 shows split sleeve main bearing caps. Replaceable bushings are standard on larger, multistage compressors and option-

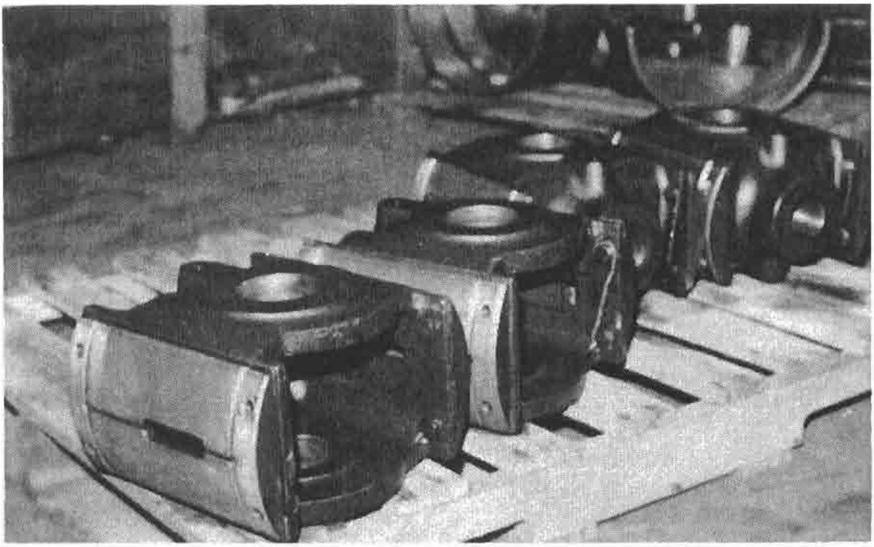

Figure 3-17. Crossheads.

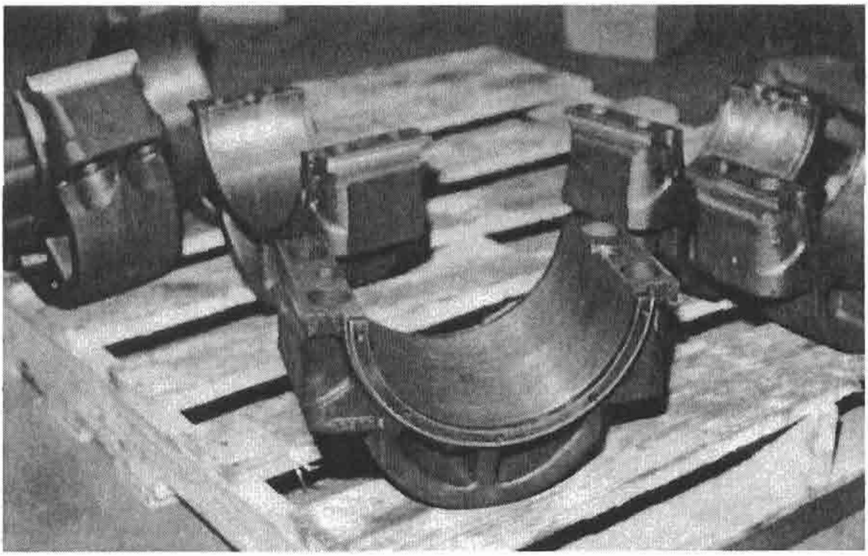

Figure 3-18. Split sleeve bearing caps.

al as the size decreases. On the smaller, standardized single-stage machines, they are not available at all. On large multistage compressors, flywheels are sometimes used to dampen torque pulsations, minimize transient torque absorbed by the driver, and to tune torsional natural frequencies. In most applications however, flywheels are not used and the driver inertia must absorb torque pulses.

Frame Lubrication

Frame lubrication is integral on most reciprocating compressors. The small, horizontal, single-stage compressors, particularly 100 horsepower and smaller, use the splash lubrication system. This system distributes lubricating oil by the splashing of the crankthrow moving through the lubricant surface in the sump. Dippers may be attached to the crankshaft to increase this effect.

The pressurized lubrication system is a more elaborate lubrication method (see Figure 3-19). The system has a main oil pump, either crankshaft or separately driven, a pump suction strainer, a cooler when needed, a

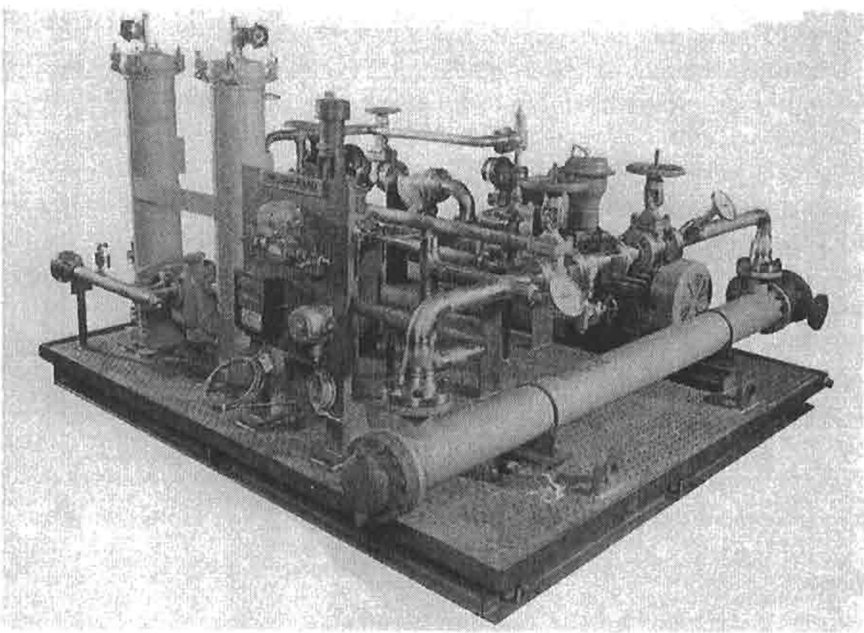

Figure 3-19. Pressurized lubrication system for a multistage reciprocating compressor. (*Courtesy of Dresser-Rand*)

full-flow oil filter and safety instrumentation. Options that should be considered when purchasing a new compressor are an auxiliary oil pump, which can also be used for startup, and dual oil filters with a non-shutoff type of transfer valve. Safety instrumentation should include a crankcase low-oil-level switch, a low-oil-pressure switch, and a high-oil-temperature switch. The switches can be duplicated and set for different operating points to provide an alarm, or early warning signal, and a shutdown signal. In this system, as in the splash system, the crank case acts as the oil sump.

Cylinder and Packing Lubrication

Lubricated cylinders use a separate mechanical lubricator to force feed, in metered droplet form, a very precise amount of lubricant to specified points. This minimizes the amount of lubricant in the cylinder and allows a lubricant most compatible with the gas to be selected without compromising the frame lubrication system. Lubricant is fed to a point or points on the cylinder to service the piston rings and the packing when required. In a few cases, as in air compressors, the packing is lubricated from the crankcase. On some applications involving wet CO_2 or H_2S in the gas stream, special materials may be avoided if one of the lubrication points is connected to the suction pulsation dampener.

One type of mechanical lubricator is the multiplunger pump, which has a plunger dedicated to each feed point (see Figure 3-20). This arrangement normally includes a sight glass per feed point. Another type of mechanical lubricator is the single metering pump sized for total flow, with a divider block arrangement to separate the lubricants going to different feed points (see Figure 3-21). While there are pros and cons to each system, the compressor vendor will normally recommend a system for any given application.

Because of the small amount of lubricant dispensed, the divider block system must be employed with the mini-lube method of lubrication previously discussed. The special divider block is usually connected to one plunger on a multiplunger pump, taking advantage of the smaller output to do the initial flow reduction. The balance of the pump's plungers may be used where a more conventional quantity of lubricant can be used.

Cooling

Three methods of cooling are in common use, the pressurized cooling fluid system, the thermosyphon, and the static system. The *static* system is used on smaller compressors and is probably the least common. Cool-

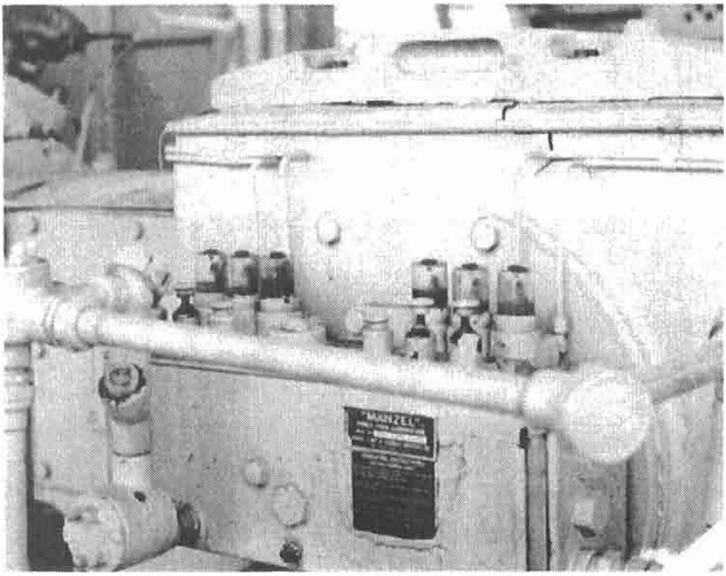

Figure 3-20. A multiplunger lubrication pump.

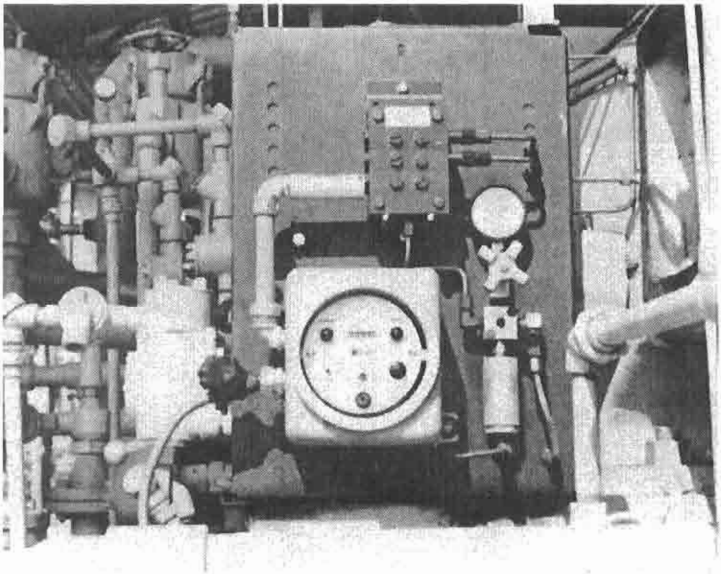

Figure 3-21. A single lubrication pump with a divider block.

ing fluid is used as a static heat sink and can be thought of more as a heat stabilizer than a cooling system. There is some heat transferred from the system by normal conduction to the atmosphere.

The *thermosyphon* is a good system for remote areas where utilities are limited, but requires some careful design to ensure proper operation. This is a circulating system with the motive force derived from the change in density of the cooling fluid from the hot to the cold sections of the system. API 618 permits this system for discharge gas temperatures below 210°F or a temperature rise across the compressor of 150°F or less.

The most common system is the *pressurized cooling fluid* system. In a plant or refinery environment where cooling tower water is available, this system has the highest heat removal capability. In locations where cooling water is not available, a self-contained, closed-cooling fluid may be used. The system consists of a circulating pump, a surge tank, and a fan-cooled radiator or air-to-liquid heat exchanger. The radiator may have multiple sections, one for frame oil cooling and another for inter- or after-cooling. The cooling fluid is either water or an ethylene glycol and water mixture. Allowance must be made in the design to accommodate the inherently higher temperature coming from the air-cooled radiator and also ambient temperature variations.

On all systems using water, the obvious is overlooked all too often. A method of draining the equipment during periods when the equipment is idle and freezing temperatures are a possibility should be provided. The consequences of failing to provide this feature are obvious.

Capacity Control

The reciprocating compressor is a fixed displacement compressor in its basic configuration; however, several methods are used to overcome this limitation to permit running at multiple operating points. In the discussion on cylinders, mention was made of clearance pockets. By use of the clearance pockets, the cylinder capacity can be lowered (see Equation 3.5). If the pocket is connected directly to the clearance area, the clearance term c can be increased. Increasing the clearance reduces the capacity by lowering the volumetric efficiency. Control of the pocket addition is by either a manual valve or by a remotely operated valve. If multiple pockets are used, a step unloading system can be designed (see Figure 3-22). The variable volume clearance pocket can provide an alternate unloading method. This device is normally attached to the outboard head. It consists of a piston-cylinder arrangement where the piston rod is threaded and

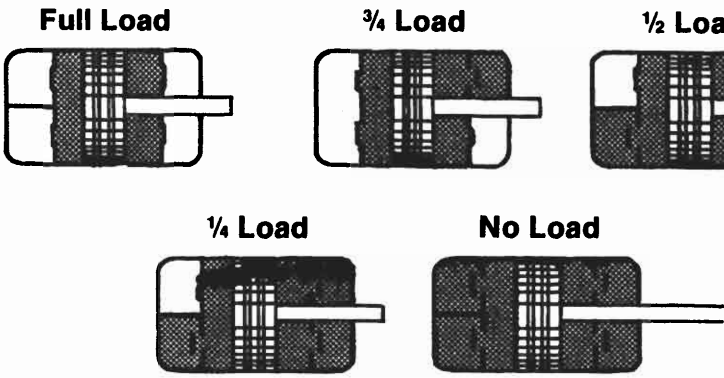

Figure 3-22. A 5-step clearance pocket unloading scheme. (*Courtesy of Dresser-Rand*)

attached to a handwheel. Turning the handwheel changes the clearance volume in an infinite number of steps up to the total pocket volume.

On cylinders lacking the physical space for pockets, the same effect can be achieved by using external bottles and some piping. Care must be taken to keep the piping close-coupled and physically strong enough to prevent accidental breakage. Remotely operated valves permit the capacity reduction to be integrated into an automatic control system.

An additional capacity control method is the *unloader.* This method can be used in conjunction with clearance pockets to extend the range of control to zero capacity. On double-acting cylinders, unloading the individual sides one at a time will provide a two-step unloading of the cylinder. On multicylinder arrangements, the cylinders can be unloaded one at a time providing as many steps as cylinders operating in parallel. The unloaders can also be used to totally unload the compressor, as is necessary for electric motor driver startup.

Three types of unloaders will be described, the plug type, the port type, and the plunger type. The plug type, shown in Figure 3-23, is normally used on all inlet valves for the unloaded end. The center of the valve is used for the unloader plug and port. The port type, shown in Figure 3-24, is used to replace one of the inlet valves on multiple inlet valve cylinders. It is normally used with low molecular weight applications. This unloader consists of a plug and port using the entire space of the valve it replaces. The plunger type, shown in Figure 3-25, is used on heavier gas applications where the maximum unloaded flow area is needed. The unloader operates by using the plunger fingers to hold the valve

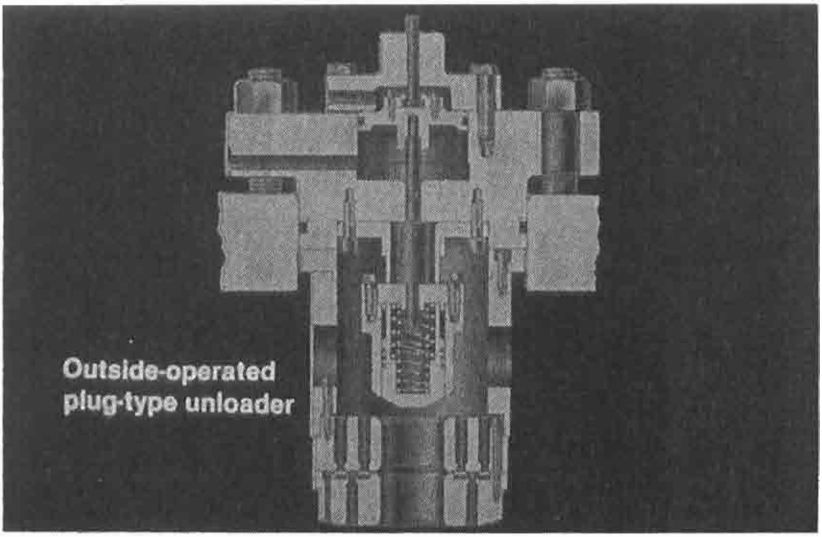

Figure 3-23. Plug type unloader. (*Courtesy of Dresser-Rand*)

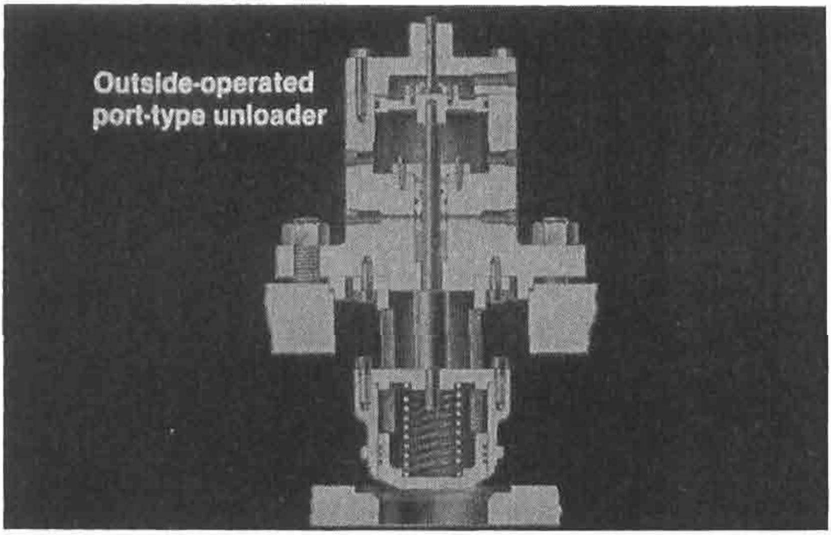

Figure 3-24. Port type unloader. (*Courtesy of Dresser-Rand*)

plates open. Control of all the described unloaders is the same, in that a piston operator is used. Additional control may be obtained by using a cooled bypass line from the discharge to the compressor suction. The bypass is normally used with discrete unloading steps.

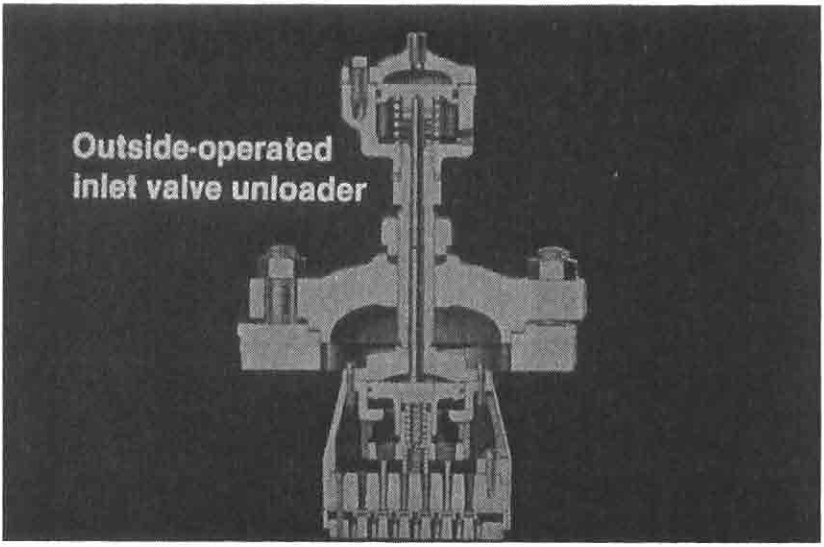

Figure 3-25. Plunger type unloader. Note the plunger finger used to hold the valve open when energized. (*Courtesy of Dresser-Rand*)

A few words of caution when using the valve unloading method: A problem arises with the possible loss of rod load reversals. Rod reversals are needed to provide lubrication to some of the bearings, as discussed earlier in this chapter. While the reversal problem is generally associated with unloading a double-acting cylinder from one side, it should be checked for all unloaded cases, including pocket unloading. If operation without rod reversals is absolutely mandatory, auxiliary lubrication must be brought to the bearings affected. The second caution is the anticipated duration of a totally unloaded condition. While the capacity has been reduced to zero, the gas in the outer end of the cylinder is being moved about in a reciprocating manner following piston movement. The movement of uncompressed gas will generate heat, and prolonged unloaded operation without proper cooling may cause severe overheating. In any case, investigation of potential problems should be undertaken with the equipment manufacturer.

From the foregoing discussion, it should be clear that cylinder capacity can be controlled. While the automatic control is normally limited to certain finite steps, the steps can be selected in size or number to minimize any adverse effect especially in conjunction with prudent use of the variable volume pocket.

Pulsation Control

The intermittent personality of the reciprocating compressor becomes evident when the subject of pulsations is broached. Because discharge flow is interrupted while the piston is on the suction stroke, pressure pulses are superimposed on the discharge system's mean pressure. At the suction side of the system, the same type of interruption is going on, causing the suction pressure to take on a non-steady component. The frequency of the pulses is constant when the speed is constant, which is the most normal condition. The pulses are literally that, not sinusoidal in characteristic; therefore, if the frequency spectrum is analyzed, it will be found to contain the fundamental frequency and a rich content of harmonics. When a forcing phenomenon is superimposed on a system with elastic and inertial properties (a second order system), a resonant response is likely to occur. This is particularly true when the band of exciting frequencies is as broad as the type of system under consideration. The gas system meets the criteria of the second order system, as gas is compressible (elastic) and has inertia (mass). If left unchecked, and a resonant response were to occur, the pressure peaks could easily reach a dangerous level. Because the oscillations are waves, standing waves will form, and interference with valve action may occur, adversely affecting the cylinder performance.

While a single, low pressure compressor may require little or no treatment for pulsation control, the same machine with an increased gas density, pressure, or operational changes may develop a problem with pressure pulses or standing wave performance deterioration. As an installation becomes more complex, such as with an increase in the number of cylinders connected to one header and the use of multiple stages, the possibility of a problem can increase.

When an installation is being planned, it is recommended that the API Standard 618 be reviewed in detail. The pulsation level for API 618 at Design Approach 1, the outlet side of any pulsation control device regardless of type, should be no larger than 2% peak-to-peak of the line pressure, or the value given by the following equation, whichever is less.

$$P\% = \frac{10}{P_{line}^{1/3}} \tag{3.20}$$

where

$P\%$ = maximum allowable peak-to-peak pulsation level at any discrete frequency, as a percentage of average absolute pressure.

P_{line} = average absolute line pressure.

The objective of this approach is to improve the reliability of the system without having to design acoustical filters. For many systems, this is all that is needed. API 618 contains a chart that recommends the type of analysis that should be performed, based on horsepower and pressure.

The pulsation control elements can have several forms, such as plain volume bottles, volume bottles with baffles, bottles and orifices, and proprietary acoustical filters. See Figure 3-26 for an example of a compressor with a set of attached volume bottles. Regardless of which device or element is selected, a pressure loss evaluation must be made before the selection is finalized because each of these devices causes a pressure drop.

For those installations where a detailed pulsation analysis, API 618 Design Approach 2 or 3, is required, several consulting companies offer these services. Until the 1980s, the most common method was to perform the pulsation analysis on the analog simulator of the Pipeline and Compressor Research Council of the Southern Gas Association. The

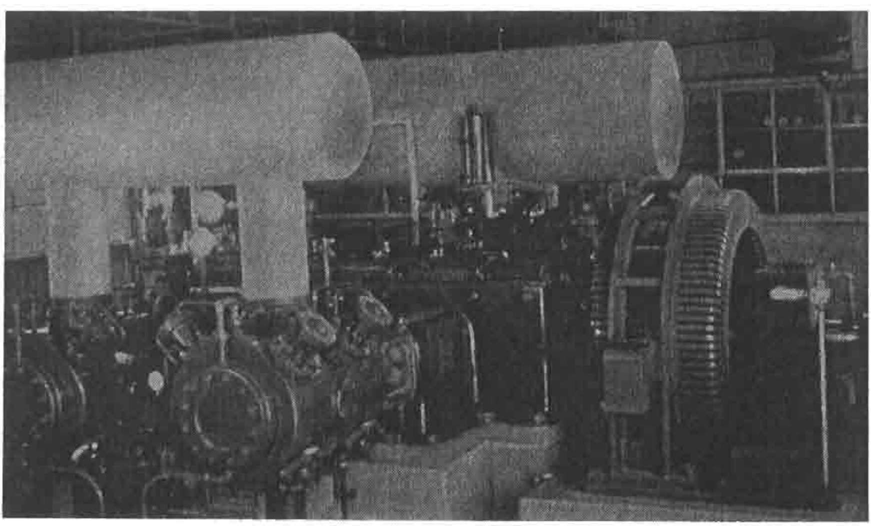

Figure 3-26. Manifold-type volume bottles are used where cylinders are operated in parallel, as on this two-stage, motor driven compressor. (*Courtesy of Dresser-Rand*)

procedures used in planning a new installation were to include the pulsation study in the contract with the compressor vendor. During the analog pulsation study, the isometric piping drawings were used to create a lumped model of the piping connected to the compressor cylinders. The purchaser's representative was required to be present for the analog study and had to be familiar with the piping requirements for the compressor area. This was necessary so that decisions relative to the space available and location for the bottles, as well as feasibility of piping modifications could be made during the study. The representative helped to expedite the completion of a final configuration for the piping system and bottle location since the analog components were disassembled after the study was completed. The analog method is still used, although much less frequently.

With the advent of modern workstations and faster PC computers, the solution of the differential equations of motion for acoustical waves in piping system on a digital computer has become feasible. In current practice, pulsation design studies using digital computer technology can produce the same results as obtained with a dynamic simulation on the analog system. The results from digital simulation satisfy the requirements of API 618. The digital computer has the advantage of data file storage. With storage capability and the ability to readily manipulate the data, it is not as necessary to have immediate decisions made. Piping changes that are recommended for acoustical control can be evaluated in a more comprehensive manner taking into account safety, cost, maintenance, and operational considerations. An additional benefit is realized if system changes are anticipated at a later time. The data files can be retrieved and the system rerun with the changes to the thermophysical properties or in the piping system itself without the need to remodel the entire system.

The interpretation of the results and the quality of the design from the pulsation study, whether performed on the analog simulator or with digital computer simulation, depends quite heavily on the experience and skill of the analyst performing the study. A purchaser of a compressor system who may be a novice at this type of analysis should give serious consideration to using the services of a competent consultant.

For the purpose of quick estimates or field evaluation of existing systems, consider the curve in Figure 3-27. This curve is not meant to supersede a comprehensive analysis as previously discussed. It should be used in checking vendor proposals or in revising existing installations where a single cylinder is connected to a header without the interaction of multiple cylinders. While not a hard rule, the curve should be conservative for

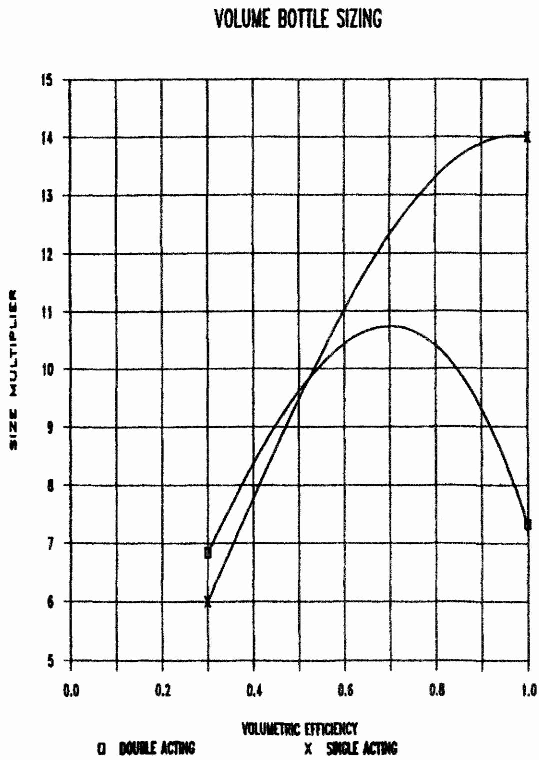

Figure 3-27. Volume bottle sizing graph.

compressors under 1,000 psi and 500 hp. The volume bottle defined is of the simple, unbaffled type.

To calculate the discharge volumetric efficiency necessary to use the curve for discharge volume bottles, use the following equation. Use Equation 3.5 to obtain the inlet volumetric efficiency and Equation 3.6 to calculate the factor.

$$E_{vd} = \frac{E_v}{(r_p)^{1/k}} \times f \tag{3.21}$$

Once the inlet and discharge volumetric efficiencies are determined, bottles for the inlet and discharge may be sized. Begin the process of sizing the bottle of interest using the appropriate volumetric efficiency (inlet or discharge) and determine a multiplier from Figure 3-27. Use Equation 3.1, 3.2, or 3.3 to determine the piston displacement. In the calculation, use a 1 for the speed and N to determine the piston displacement for a single revolution. Apply the multiplier to the piston displacement per revolution. The product is the bottle volume, Vol, for use in Equation 3.22. This equation will yield the bottle diameter.

$$d_b = .86(\text{Vol})^{1/3} \tag{3.22}$$

To complete the solution for the volume bottle dimensions, assume 2:1 elliptical heads and use the following relationship:

$$L_b = 2d_b \tag{3.23}$$

where:

L_b = volume bottle length
d_b = volume bottle diameter

Example 3-2

Approximate the size of a suction and a discharge volume bottle for a single-stage, single-acting, lubricated, reciprocating compressor. The gas being compressed is natural gas at the following conditions:

Cylinder bore:	9 in.
Cylinder stroke:	5 in.
Rod diameter:	2.25 in.
Suction temperature:	80°F
Discharge temperature:	141°F
Suction pressure:	514 psia
Discharge pressure:	831 psia
Isentropic exponent:	1.28
Specific gravity:	.60
Percent clearance:	25.7%

Step 1. Find the suction and discharge volumetric efficiencies using Equations 3.5 and 3.21 with $r_p = 831/514 = 1.617$. The natural gas compressibility values can be obtained by using the gravity/compressibility charts (see Appendix B-29 through B-35) for a specific gravity of .60. Both Z_1 and Z_2 values are .93. Applying Equation 3.6, the value of f may be obtained as follows:

$f = .93/.93$

$f = 1.0$

Using the equation for the suction volumetric efficiency,

$E_v = .97 - [(1/1)1.617^{1/1.28}\ 1]\ .257 - .03$

$E_v = .823$ suction volumetric efficiency

For the discharge volumetric efficiency, use Equation 3.21.

$$E_{vd} = \frac{.823 \times 1}{(1.617)^{1/1.28}}$$

$E_{vd} = .565$ discharge volumetric efficiency

Step 2. Find the total volume displaced per revolution using Equation 3.1 for a single-acting compressor.

$$Pd/Rev = 5 \times \frac{\pi \times 9^2}{4}$$

$Pd/Rev = 318.1\ in.^3$

Step 3. Using the volumetric efficiencies found in Step 1, find the size multiplier from the volume bottle sizing chart, Figure 3-27.

Suction multiplier = 13.5

Discharge multiplier = 10.4

Step 4. Find the required bottle volume from the displacement and the multiplier.

$Vol = 13.5 \times 318.1$

$Vol = 4{,}294.2\ in.^3$ suction bottle volume

$Vol_d = 10.4 \times 318.1$

$Vol_d = 3,308.1$ in.^{3} discharge bottle volume

Step 5. Find the bottle dimensions from Equations 3.22 and 3.23 for vessels with 2:1 elliptical heads. Use Equation 3.22 to calculate the diameter.

$d_{bs} = .86\ (4294.2)^{1/3}$

$d_{bs} = 16.3$ in. diameter of suction bottle

$d_{bd} = .86\ (3308.1)^{1/3}$

$d_{bd} = 14.9$ in. diameter of discharge bottle

Use Equation 3.23 to calculate the length.

$L_{bs} = 2 \times 16.3$

$L_{bs} = 32.6$ in. length of suction bottle

$L_{bd} = 2 \times 14.9$

$L_{bd} = 29.8$ in. length of the discharge bottle

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4

Rotary Compressors

Common Features

Rotary compressors as a group make up the balance of the positive displacement machines. This group of compressors has several features in common despite differences in construction. Probably the most important feature is the lack of valves as used on the reciprocating compressor. The rotary is lighter in weight than the reciprocator and does not exhibit the shaking forces of the reciprocating compressor, making the foundation requirements less rigorous. Even though rotary compressors are relatively simple in construction, the physical design can vary widely. Both multiple- and single-rotor construction is found. Rotor design is one of the main items that distinguishes the different types. Size and operating range is another area unique to each type of rotary. The following sections cover some of the more common rotary compressors in detail.

Arrangements and Drivers

Rotary compressors are frequently arranged as single units with a driver. Occasionally the compressors are also used in series arrangements, with or without an intercooler. The series configurations may use a form of tandem drive or multiple pinion gears to permit the use of a common driver.

For most of the rotary compressors in process service, the driver is an electric motor. Compressors in portable service, however, particularly the helical-lobe compressor, use internal combustion engines. Many of the rotary compressors require the high speed that can be obtained from a direct-connected motor. The dry type helical-lobe compressor is probably the main exception as the smaller units operate above motor speed and require a speed increasing gear which may be either internal or external (see Figure 4-1). The helical-lobe compressor is the most likely candidate for a driver other than the electric motor. Aside from the portables already mentioned, engines are used extensively as drivers for rotaries located in the field in gas-gathering service. Steam turbines, while not common, probably comprise most of process service alternate drive applications.

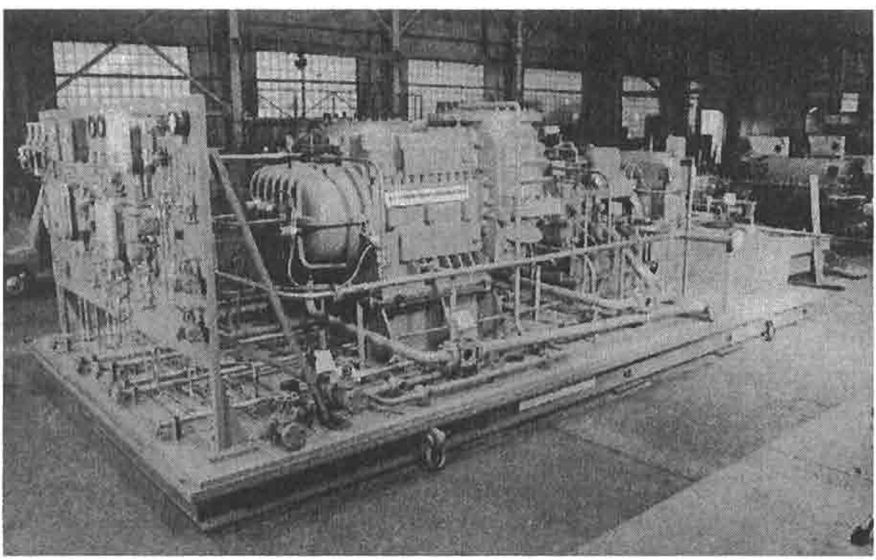

Figure 4-1. A skid-mounted oil-free helical-lobe compressor. (*Courtesy of A-C Compressor Corporation*)

Helical Lobe

History

While a form of helical-rotor compressor was invented in Germany in 1878 [1], the helical-lobe compressor as used today is credited to Alf Lysholm, the chief engineer of Svenska Rotor Maskiner AB (SRM). Mr. Lysholm conceived the idea in 1934 as part of gas turbine development at SRM. The original compressor was an oil-free design using timing gears to synchronize the rotors. Three male and four female rotor lobes were used, and a steeper helix angle permitted higher built-in compression ratios and improved operation at higher pressures. The pressures were in the 20 to 30 psig range. Unfortunately, the profile created a trapped pocket where the gas was overcompressed prior to being released. This led to lower efficiency and high noise levels. Despite the disadvantages, the compressor was licensed and used in varying applications. Because of Mr. Lysholm's contributions, the helical-lobe compressor is sometimes referred to as the Lysholm compressor. It also goes by the name of SRM, the company controlling the development and licensing.

Hans Nilson became chief engineer of SRM in the late 1940s and later became president of the company. He made numerous contributions to the technical and commercial growth of the compressor, such as the circular profile invented in 1952. The profile used the four male lobe and six female lobe rotors. The design eliminated the trapped pocket, permitting a steeper helix angle. The resulting higher, built-in pressure ratios also improved efficiency.

The next significant event in the evolution of the SRM compressor was the application of a Holroyd rotor cutting machine to the production of the rotors. Prior to this event, producing the rotors was both slow and costly. In 1952, the first special Holroyd machine was delivered to Howden Company, a licensee in the U.K., who later contributed to the development of the oil-flooded compressor.

The slide valve was invented in the early 1950s, giving the SRM compressor a new dimension by providing a means of flow control. Capacity control had been a limiting factor for applications needing a range of flows. The slide valve provided infinite capacity control while still retaining built-in compression during flow reduction. The slide valve became widely used with the advent of the oil-flooded compressor.

The development and subsequent patent of the oil-flooded compressor was the result of a joint effort on the part of Howden and SRM. The oil-injected prototype was run at SRM on July 4, 1954 and proved to be 8 to 10% better in performance than the dry compressor with timing gears. Performance at low speeds was improved, permitting the use of direct drive motors. The flooding provided both cooling, which permitted higher pressure ratios and lubrication allowing the elimination of the timing gears. The male rotor drives the female rotor through the oil film. The first commercial application was introduced by Atlas Copco in 1957 for air compression. The slide valve was incorporated into the flooded design in the 1960s and was originally used in refrigerant service. More recently, it has also been incorporated into gas compressor service.

Lars Schibbye became chief engineer of SRM in 1950 and contributed to the technical advancement of the compressor. Most significant was the invention of the asymmetric rotor profile, which was introduced commercially by Sullair in the U.S. in 1969. The asymmetric rotor profile reduces the leakage path area and sealing line length resulting in increased efficiency.

The more recent developments have been in the area of manufacture. Rune Nilson of SRM worked with the machine tool manufacturers to develop precision carbide form cutting tools. These permit the rotors to be cut in two to three passes with high accuracy. The machining time has been reduced significantly.

The development of the helical-lobe compressor is quite unique in that it has been controlled primarily by one company, with the cooperation of licensees who, in turn, provide application expertise [2].

Operating Principles

Another name for the helical-lobe compressor is the *screw compressor.* This is probably the most common, even though all the names are used quite interchangeably. While the screw compressor originally fit the area between the centrifugal and the reciprocating compressor, the application areas have expanded. The larger, screw type machines now range to 40,000 cfm, and definitely cross into the centrifugal area. The smaller ones, particularly the oil-flooded type, are being considered for automotive air conditioning service, therefore, completely overlapping the reciprocating compressor in volume. The dry variety generally stops in the 50 cfm area.

Compression is achieved by the intermeshing of the male and female rotor. Power is applied to the male rotor and as a lobe of the male rotor starts to move out of mesh with the female rotor a void is created and gas is taken in at the inlet port (see Figure 4-2). As the rotor continues to turn, the intermesh space is increased and gas continues to flow into the

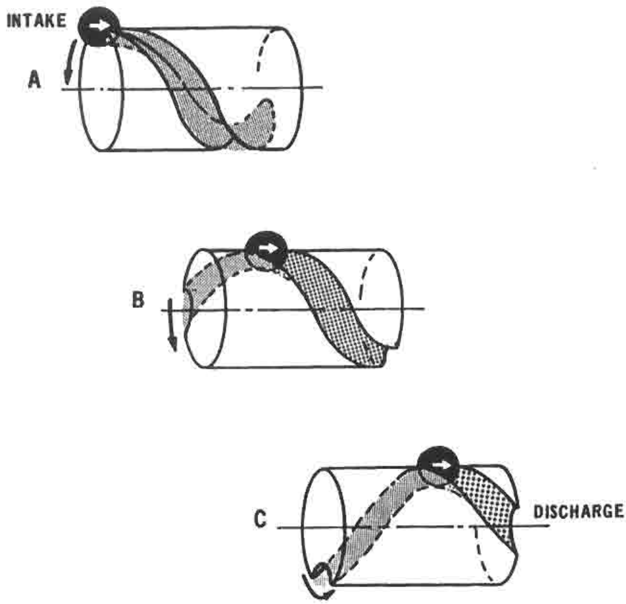

Figure 4-2. The compression cycle of a helical-lobe compressor.

compressor until the entire interlobe space is filled. Continued rotation brings a male lobe into the interlobe space compressing and moving the gas in the direction of the discharge port. The volume of the gas is progressively reduced, increasing the pressure. Further rotation uncovers the discharge port, and the compressed gas starts to flow out of the compressor. Rotation then moves the balance of the trapped gas out while a new charge is drawn into the suction of the unmeshing of a new pair of lobes as the compression cycle begins.

The compressor porting is physically arranged to match the application pressure ratio. To maintain the best efficiency, it is important that the matching be as close as possible.

Figure 4-3 includes four diagrams to show two cases, (A) a low-ratio compressor and (B) a high-ratio compressor. The lower diagram (C) demonstrates a low-volume ratio compressor in a higher-than-design application. Because the gas arriving at the discharge port has not been sufficiently compressed, the resulting negative ratio across the discharge port causes a backflow and resulting loss. Diagram (D) shows a compressor with too high a volume ratio for the process. Here the gas is compressed higher than needed to match the pressure of the gas on the outlet side of the discharge port, resulting in energy waste.

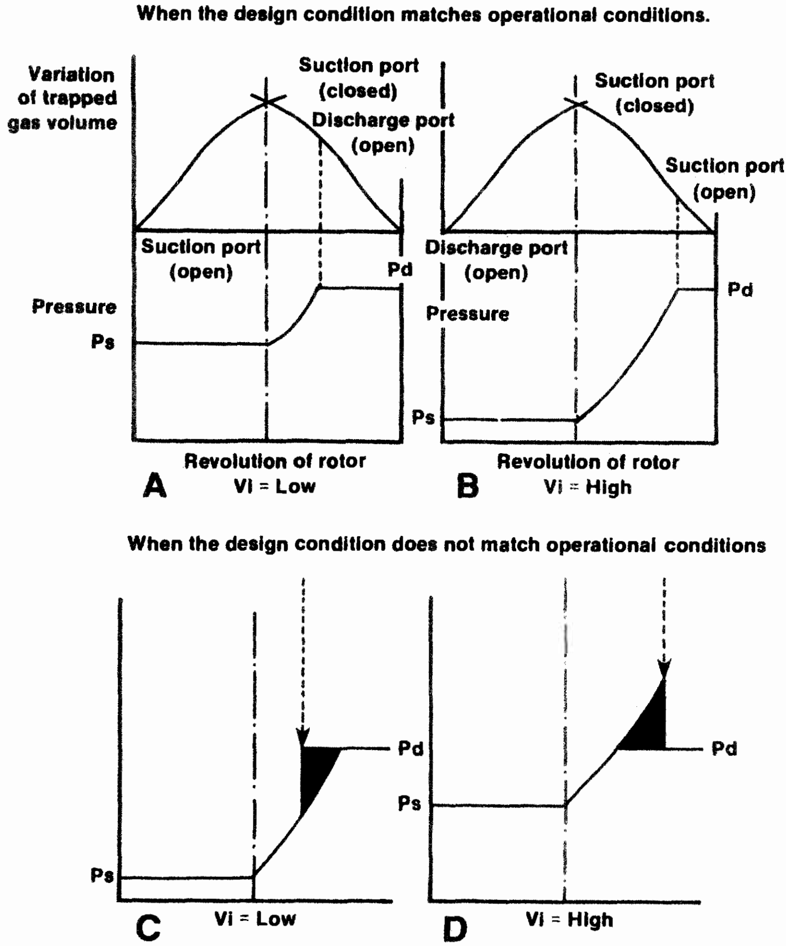

Figure 4-3. Effects of low- and high-volume ratios on the cycle of a screw compressor. (*Courtesy of Mayekawa Manufacturing Company, Ltd.*)

The terms *pressure ratio* and *volume ratio* are used interchangeably in the literature on these machines. To prevent confusion, *volume ratio* r_v, is defined as the volume of the trapped gas at the start of the compression cycle divided by the volume of the gas just prior to the opening of the discharge port. *Pressure ratio* is defined, in Equation 2.64, as the discharge pressure divided by the suction pressure. Their relationship is given in the following equation.

$$r_p = r_v^{\,k} \tag{4.1}$$

where

r_p = pressure ratio
k = isentropic exponent
r_v = volume ratio

Displacement

The displacement of the screw compressor is a function of the interlobe volume and speed. The interlobe volume is a function of rotor profile, diameter, and length. Table 4-1 provides some typical rotor diameters and corresponding L/d ratios. The interlobe volume can be expressed by the following equation.

$$Q_r = \frac{d^3 (L/d)}{C} \tag{4.2}$$

where

Q_r = displacement per revolution
d = rotor diameter
L = rotor length
C = typical profile constant, for 4 + 6 rotor arrangement
C = 2.231 circular profile
= 2.055 asymmetric profile [3]

Table 4-1
Rotor Diameters
with Available L/d Ratios

Rotor Diameter Inches	L/d 1.0	L/d 1.5
6.75	X	—
8.50	X	X
10.50	X	X
13.25	X	X
16.50	X	X
20.00	—	X
24.80	—	X

Available sizes (X)
Data for table courtesy of A-C Compressor.

$$Q_d = Q_r \times N \tag{4.3}$$

where

Q_d = displacement
N = compressor speed

$$Q_i = Q_d \times E_v \tag{4.4}$$

where

Q_i = actual inlet volume
E_v = volumetric efficiency

Because there is no clearance volume expansion, as in the reciprocating compressor, the volumetric efficiency is a function of the rotor slip. This is the internal leakage from the higher pressure to the lower pressure side, reducing potential volume capacity of the compressor.

Dry Compressors

The nonflooded compressor rotor leakage can be related to the rotor tip Mach number. The rotor tip velocity can be calculated by

$$u = \pi \times d \times N \tag{4.5}$$

The optimum tip speed is .25 Mach at a pressure ratio of 3. The value shifts slightly for other built-in pressure ratios, as shown in Figure 4-4.

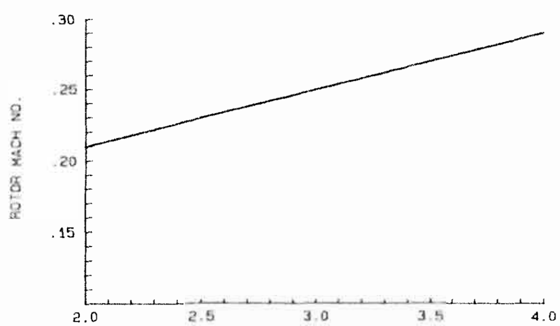

Figure 4-4. Optimum tip speed vs. pressure ratio.

Besides affecting the volumetric efficiency, the leakage also has an effect on the adiabatic efficiency. Figure 4-5 is a plot of the tip speed ratio, u/u_o, (operating to optimum) against the efficiency ratio, off-peak-to-peak effi-

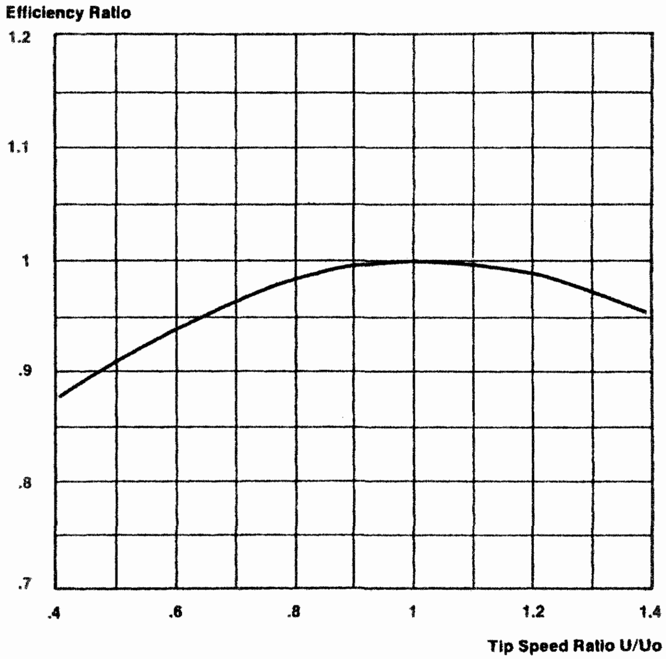

Figure 4-5. Tip speed ratio vs. efficiency ratio.

ciency. Figures 4-6 and 4-7 show a set of typical volumetric and adiabatic efficiency curves for three built-in ratios.

The adiabatic efficiency should be corrected for molecular weight. Generally the efficiency decreases with lower molecular weight and increases with increased molecular weight. As an arbitrary rule of thumb, a straight line relationship can be assumed. The correction is a –3 percentage points at a molecular weight of 2, 0 at 29, and +3 at the molecular weight of 56. For example, a compressor with an air efficiency of 78% would have an adiabatic optimum tip speed efficiency of 75% when operating on hydrogen.

The screw compressor can be evaluated using the adiabatic work equation. Discharge temperature can be calculated by taking the adiabatic temperature rise and dividing by the adiabatic efficiency then multiplying by the

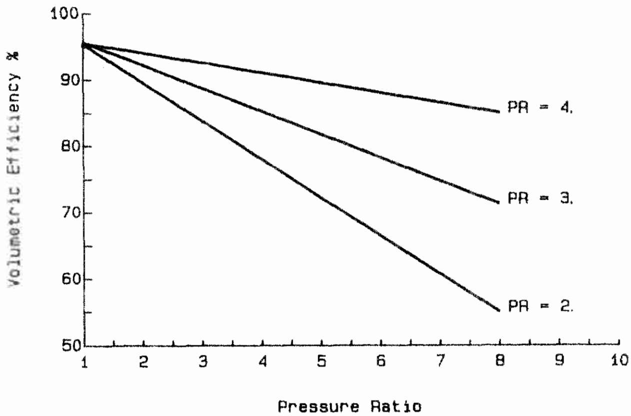

Figure 4-6. Pressure ratio vs. volumetric efficiency for an SRM compressor.

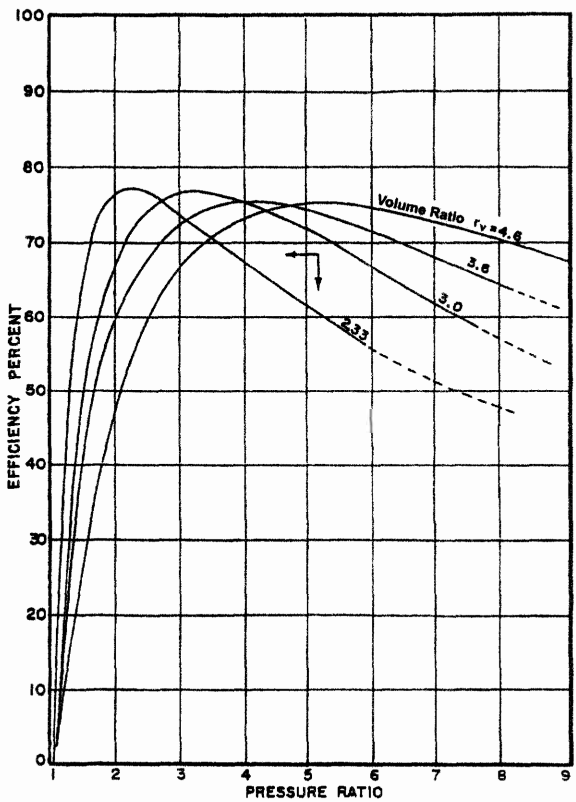

Figure 4-7. Pressure ratio vs. adiabatic efficiency for an SRM compressor. (Modified from [4].)

temperature rise efficiency to account for cooling. To obtain the discharge temperature, add the inlet temperature to the temperature rise. The work equation was developed in Chapter 3 and repeated here for convenience.

$$W_a = P_1 Q_1 \frac{k}{\eta_a (k-1)} (r_p^{\frac{k-1}{k}} - 1) \tag{3.18}$$

where

W_a = adiabatic work input
P_1 = inlet pressure
Q_1 = inlet volume
η_a = adiabatic efficiency

For the discharge temperature, t_2,

$$t_2 = t_1 + \frac{T_1 (r_p^{\frac{k-1}{k}} - 1)}{\eta_a} \times \eta_t \tag{4.6}$$

where

t_1 = inlet temperature
T_1 = absolute inlet temperature
η_a = adiabatic efficiency
η_t = temperature rise efficiency

A typical value for the temperature rise efficiency is .9. For shaft power, W_s,

$$W_s = W_a + \text{mech loss} \tag{4.7}$$

where mech loss = .07 × W_a for estimating purposes.

Example 4-1

Calculate the performance of a compressor using air for the following conditions:

d = 10.5 in. rotor diameter

L/d = 1.5 length-diameter ratio

mw = 23

Q_1 = 2,500 acfm inlet volume

t_1 = 100°F inlet temperature

P_1, = 14.5 psia inlet pressure

P_2 = 43.5 psia discharge pressure

r_p = 3.0 pressure ratio

k = 1.23

w = 138.8 lbs/min weight flow

Step 1. Using Equation 4.2, solve for the displaced volume per revolution. Convert the units using 1728 in.^{3}/ft^3.

$$Q_r = \frac{10.5^3 \times 1.5}{1728 \times 2.231}$$

$$Q_r = .450 \text{ ft}^3/\text{ rev}$$

From Figure 4-6, read a volumetric efficiency for pressure ratio, 3, where E_v = 89%. Use Equation 4.4 and solve for the total displaced volume.

Q_d = 2,500/.89

Q_d = 2,809 cfm total displacement volume

Now calculate the required speed by substituting into Equation 4.3.

N = 2,809/.450

N = 6,242 rpm rotor speed

Step 2. Find the velocity of sound for air at the inlet conditions given, using Equation 2.32 from Chapter 2.

R = 1,545/23

R = 67.17 specific gas constant

$a = (1.23 \times 67.17 \times 32.2 \times 560)^{1/2}$

a = 1,220.6 fps velocity of sound

Compute the rotor tip velocity using Equation 4.5 and the unit conversions of 12 in./ft and 60 sec/min.

$$u = \frac{\pi \times 10.5 \times 6242}{60 \times 12}$$

$u = 286.0$ fps rotor tip velocity

Refer to Figure 4-4 with pressure ratio = 3.0 and read the rotor Mach number $u_o/a = .25$. Calculate u_o, the optimum tip velocity.

$u_o = .25 \times 1220.6$

$u_o = 305$ fps optimum tip velocity

Then calculate the tip speed ratio.

$u/u_o = 286.0/305.0$

$u/u_o = .937$ tip speed ratio

Step 3. Refer to Figure 4-7 and select an efficiency at a pressure ratio of 3 and a volume ratio, r_v of 2.44. The adiabatic efficiency is 74%. Now, from Figure 4-5, select a value of efficiency ratio using the tip speed ratio just calculated. Because the value is .99+, round off to an even 1.0. With a multiplier of 1.0, the final adiabatic efficiency is the value read directly off the curve or $\eta_a = 74$. The molecular weight correction for efficiency, per rule of thumb, is 0.6 for a final efficiency of 73.4.

Step 4. The adiabatic power can be solved by substituting into Equation 3.18.

$k/(k-1) = 1.23/.23$

$k/(k-1) = 5.34$

$(k-1)/k = .187$

Calculate the power using the conversions of 144 in.^{2}/ft^2 and 33,000 ft lbs/min/hp for a net value of 229.

$$W_a = \frac{14.5 \times 2500 \times 5.34\,(3^{.187} - 1)}{.734 \times 229}$$

$W_a = 262.6$ hp

Substitute into Equation 4.6 for the discharge temperature.

$$t_2 = 100 + \frac{560\,(3^{.187} - 1)}{.734} \times .9$$

$t_2 = 256.6°F$ discharge temperature

Step 5. Solve for the shaft power substituting into Equation 4.7.

$P_S = 262.6 + 18.4$

$P_S = 281.0$ hp shaft horsepower

Example 4-2

Rerate the compressor considered in Example 4-1 for an alternate set of conditions given below. Use all other conditions from the previous example.

$P_2 = 50.75$ psia new discharge pressure

$r_p = 3.5$ new pressure ratio

$r_v = 2.77$ new volume ratio

Step 1. Calculate a new inlet volume using the value of displaced volume from the previous example.

$Q_d = 2{,}809$ cfm displaced volume

Refer to Figure 4-6 and, at a pressure ratio = 3.5, read the volumetric efficiency = 87%. Use Equation 4.4 to develop the inlet volume.

$Q_i = 2{,}809 \times .87$

$Q_i = 2{,}444$ cfm inlet volume

By proportion, obtain a new inlet weight flow.

$w = 2{,}444/2{,}500 \times 138.8$

$w = 135.7$ lb/min new weight flow

Step 2. Reuse the rotor tip speed and sonic velocity from Example 4-1 as the conditions used in their development that have not changed.

$a = 1220.6$ fps sonic velocity

$u = 286.0$ fps rotor tip speed

Refer to Figure 4-4 and, at a pressure ratio = 3.5, read the Mach number, $u_o/a = 0.27$. Calculate the optimum tip speed u_o.

$u_o = .27 \times 1220.6$

$u_o = 329.6$ fps optimum tip speed

Now calculate the tip speed ratio.

$u/u_o = 286.0/329.6$

$u/u_o = .868$ tip speed ratio

Step 3. Use Figure 4-7 to obtain the adiabatic efficiency for pressure ratio = 3.5 and volume ratio of 2.77. From the curve, adiabatic efficiency = 73%. Next, look up the efficiency ratio on Figure 4-5 for the tip speed ratio just developed and obtain a value of .98. Use this value as a multiplier to derate the adiabatic efficiency for operation at other than the optimum tip speed.

$\eta_a = .98 \times 73.0$

$\eta_a = 71.5\%$

Step 4. Solve for the adiabatic power, making the same conversions used in Step 4 of Example 4-1.

$$W_a = \frac{14.5 \times 2444 \times 5.34(3.5^{.187} - 1)}{.715 \times 229}$$

$W_a = 326.2$ hp

Use Equation 4.6 to solve for the discharge temperature.

$$t_2 = 100 + \frac{560\,(3.5^{.187} - 1)}{.715} \times .9$$

$t_2 = 399°F$ discharge temperature

Step 5. For the final step, compute the new shaft power value using Equation 4.7.

$W_S = 325.0 + 21.4$

$W_s = 346.4$ hp new shaft horsepower

The example demonstrates that operating the compressor off the built-in pressure ratio means operating at a lower efficiency. This could be anticipated from Figure 4-3. The optimum port configuration for the various types of screw compressors was determined from a series of prototype tests. The change in volumetric efficiency is not a result of the built-in volume ratio, but is due to the increased slip (internal leakage) from the higher operating pressure ratio. In the last example, a slight loss of efficiency was shown for operation at other than the optimum tip speed. While, in the example, the penalty was not too severe, it does give a directional indication of the potential problems with off-design operation.

Figure 4-8 shows a comparison of the two currently used rotor profiles. Figure 4-8a shows the circular profile used in the past for both the dry and flooded compressor. The newer asymmetric profile shown in Figure 4-8b is being adopted for use in both dry and flooded service by various vendors because of the improved efficiency due to a lower leakage in the discharge area of the compressor. Because size is a factor, the improvement in efficiency is more dramatic in the smaller compressors.

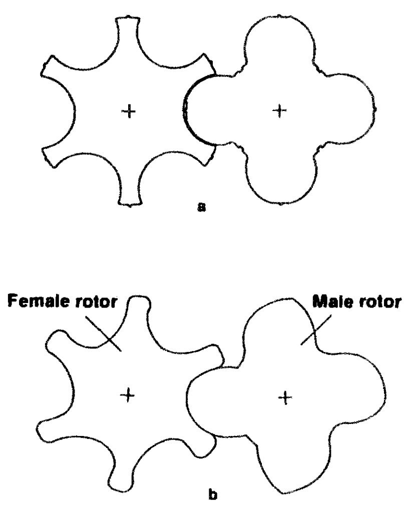

Figure 4-8. The two rotor profiles of helical screw compressors.

Flooded Compressors

The oil-flooded version is an increasingly popular variation of the screw compressor and is seeing a variety of applications. This type of compressor is less complex than the dry version because of the elimination of the timing gears. It also has the advantage of the oil acting as a seal to the internal clearances, which means a higher volumetric and overall efficiency. The sealing improvement also results in higher efficiency at lower speeds. This means quiet operation and the possibility of direct connection to motor drivers, eliminating the need for speed increasing gears. (When gears *are* needed they are available as internal on some models, see Figure 4-9.) Higher pressure ratios can also be realized because of the direct cooling from the injected oil. Pressure ratios as high as 21 to 1 in one casing are possible [3]. Besides the inherently quiet operation from lower speed, the oil dampens some of the internal pulses aiding the suppression of noise. The timing gears can be eliminated because the female rotor is driven by the male through the oil film. To take advantage of the 3-to-2 speed increase, development work is in progress to drive the female rotor. Alterations to the 90-to-10 power division for male and female rotor must be made to shift more of the power

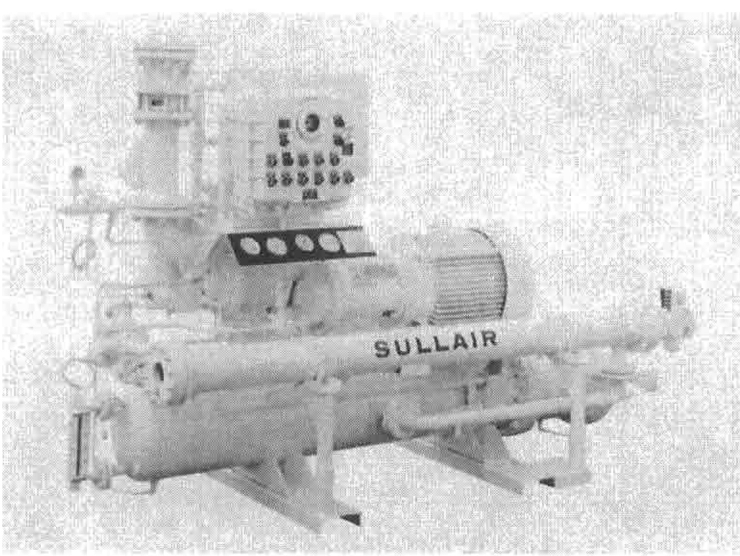

Figure 4-9. An oil-flooded, integrally geared screw compressor package. (*Courtesy of Sullair*)

to the female rotor. The contact surfaces should also be improved to better transfer the additional power.

The injected oil is sheared and pumped in the course of moving through the compressor. These losses can be minimized by taking advantage of the slower speed performance. Figure 4-10 shows the operating speed plotted against the shaft input power. There is an optimum operating speed where the improvement in operation from the oil offsets the potential losses. The points of injection are quite important for efficient operation. The oil is injected in the casing wall at or near the intersection of the rotor bores on the discharge side of the machine. The orifices are lined up axially in the region where compression is taking place. Also, oil enters from each bearing. Good drainage control will keep oil from recycling back to contact the inlet charge and transferring unwanted heat to the uncompressed gas. The inlet port, as well, must be designed to prevent slip oil (oil traveling in the rotor clearance area) from heating the inlet gas.

Test data indicate that for the *pumpless oil system* compressor, the discharge temperature remains constant over a wide range of operation, at varying pressure ratios, staying close to 176°F (80°C)[1]. In contrast, on a *pumped system,* the outlet temperature can be maintained at a desired level. The amount of oil injected must be carefully controlled, admitting enough for operation and not too much to cause high pumping losses. For an application of 100 psig air compression, rates of 6 to 7 gpm are used

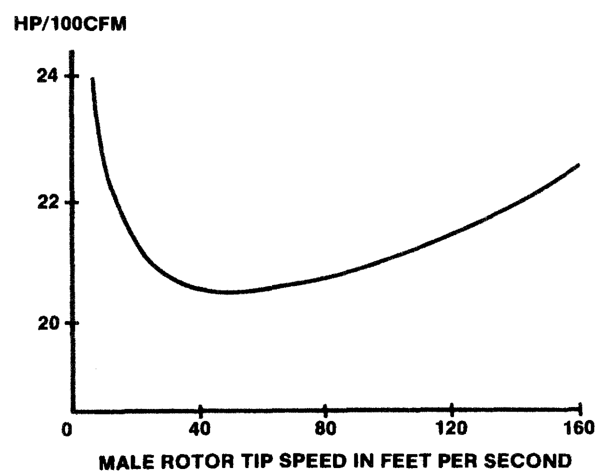

Figure 4-10. Tip speed vs. shaft input power for an oil-flooded screw compressor discharging at 100 psig [3].

per 100 cfm. This results in about 450 BTU/min heat rejection per 10 hp of input energy [3]. Because heat is rejected to the oil and the oil is recirculated in the flooded compressor, a larger lubrication system is required, along with more cooling water than would be needed for a dry compressor. There may be a potential trade-off in some applications such as refrigeration or air compression where dry compressors use inter and aftercoolers. In the case of refrigeration, the heat load is reduced to the refrigerant condenser and, in the air compression, the load is less for an aftercooler.

For areas where cooling water is either scarce or not available, direct liquid injection may be a possibility. The liquid coolant should be injected near the discharge end of the compressor to minimize lubricant dilution. Alternatively, the liquid can be flashed in a separate exchanger and used to cool the lubricant. While the cooling may appear to decrease the power to the compressor, the net effect is an increase in the power due to the additional weight flow of the extra refrigerant needed to perform the cooling.

The following equation provides a way of estimating the discharge temperature if the shaft power is known, or it can be used to estimate the shaft power if the temperature rise and quantity of lubricant is known. The equation assumes 85% of the heat of the compressor is absorbed by the lubricant.

$$.85W_s = q_L \times \rho_L \times c_{pL} \times \Delta t \tag{4.8}$$

where

W_s = work input to the shaft
q_L = volume of lubricant
ρ_L = specific weight of the lubricant
c_{pL} = specific heat of the lubricant
Δt = lubricant temperature rise

Flooded compressors use the asymmetric profile rotor extensively because the rotor's efficiency is most apparent in this size range. Flooded compressor size has, over the more recent times, been increased. The upper range is in the 7000 cfm range. While most applications are in air and refrigeration, certain modifications can make it applicable for process gas service. One of the considerations is the liquid used for the flooding.

Flooding Fluid

The fluid used in the compressor is normally a petroleum based lubricating oil, but this is not universal. Factors to consider when selecting the lubricant include the following:

1. Oxidation
2. Condensation
3. Viscosity
4. Outgassing in the inlet
5. Foaming
6. Separation performance
7. Chemical reaction

Some of the problems can be solved with specially selected oil grades. Another solution is synthetic oils, but cost is a problem particularly with silicone oils. Alternatives must be reviewed to match service life of the lubricant with lubrication requirements in the compressor.

For chemical service, some lubrication qualities may be sacrificed in order to obtain a fluid compatible with the process gas. In these applications, alternate bearing materials such as graphite or silver have been required. While the requirements may make the operation somewhat special and require considerable care, the life of the compressor and service can be greatly improved.

Application Notes—Dry Compressors

Screw compressors of the dry type generate high frequency pulsations that move into the system piping and can cause acoustic vibration problems. These would be similar to the type of problems experienced in reciprocating compressor applications, except that the frequency is higher. While volume bottles will work with the reciprocator, the dry type screw compressor would require a manufacturer-supplied proprietary silencer that should take care of the problem rather nicely.

There is one problem the dry compressor can handle quite well. Unlike most other compressors, this one will tolerate a moderate amount of liquid. Injection of liquids for auxiliary cooling can be used, normally at a lower level than would be used in the flooded compressor. The compressor also takes reasonably well to fouling service, if the material is not abrasive. The foulant tends to help seal the compressor and, in time, may improve performance. One other application for which the dry machine is particularly well-suited is for hydrogen-rich service, where the molecular weight is low, with a resulting high adiabatic head. For larger flow streams, within the centrifugal compressor's flow range, the screw compressor is a good alternative. While the high adiabatic head requires expensive, multiple centrifugal casings, the positive displacement characteristic of the screw compressor is not compromised by the low molecular weight. For very low molecular weight gas, such as pure hydrogen or helium, a good seal is important to keep the slip in control. This can be tedious and, in extreme cases, a liquid injection is used for leakage control to maintain performance.

Application Notes—Flooded Compressor

One consideration for the flooded compressor is the recovery of the liquid. In the conventional arrangement, the lubricating oil is separated at the compressor outlet, cooled, filtered, and returned to the compressor. Figure 4-11 is a diagram of a typical oil-flooded system. This is fine for air service where oil in the stream is not a major problem, but when oil-free air is needed, the separation problem becomes more complex. Because the machine is flooded and the discharge temperature is not high, separation is much easier than with compressors that send small amounts of fluid at high temperature downstream. Usually part of the lubricant is in a vaporized form and is difficult to condense except where it isn't wanted. To achieve quality oil free air, such as that suitable for a desiccant type dryer, separators to the tertiary level should be considered (see Figure 4-12). Here, the operator must be dedicated to separator maintenance, because these units require more than casual attention. Separation in refrigeration is not as critical if direct expansion chillers are used. In these applications, the oil moves through the tubes with the

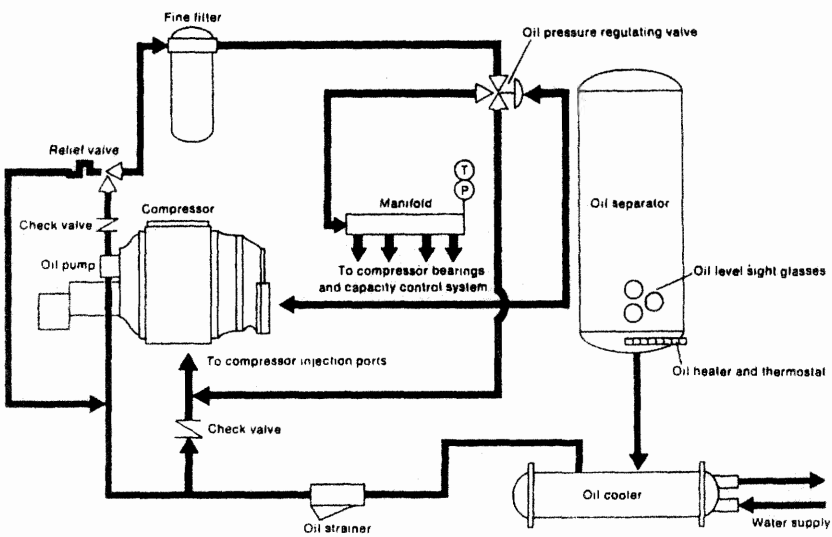

Figure 4-11. Lubrication diagram for a flooded screw compressor. (*Courtesy of Sullair*)

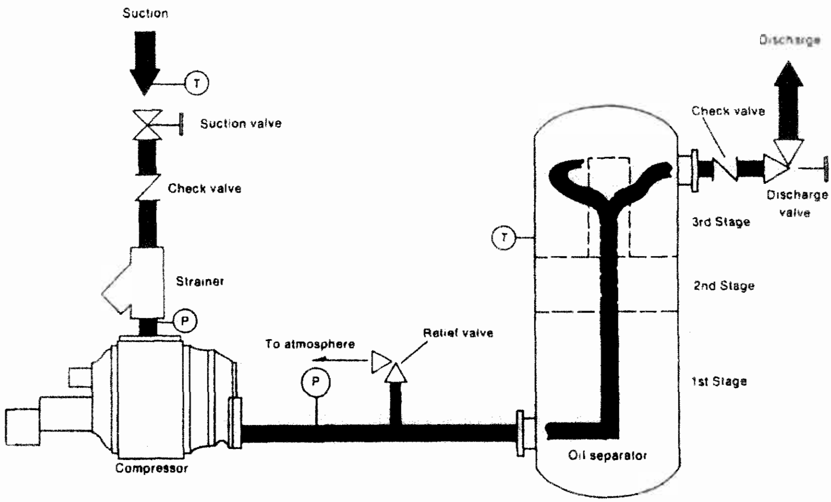

Figure 4-12. Oil and gas separation system for a flooded screw compressor. (*Courtesy of Sullair*)

refrigerant and comes back to the compressor with no problem, if the temperature is not too low for the lubricant. In the case of kettle type chillers, things tend to be more complex. Oil can be removed from a kettle with a *skimmer* arrangement, but getting the skimmer working on a wildly boiling and foaming fluid tends to make the refrigerant maintenance people somewhat testy. Again, it is necessary to match the lubricant to the temperature to get it back at all. No lubricant will return if it's frozen to the walls of an exchanger. If the evaporator is not field-located and can be made a part of the refrigeration package, some of these problems of oil handling can be deferred to the compressor vendor.

A nice feature of the flooded or dry screw compressor is its ability to achieve the required outlet pressure regardless of the molecular weight. The compressor process that starts on nitrogen and then gradually brings in a hydrogen-rich gas mixture does not change in performance as a centrifugal compressor would.

Casings

Most casings on both flooded and dry compressors are cast, normally of grey cast iron. API 619 [5] limits the use of cast iron by specifying steel for services in excess of 400 psig, discharge temperatures in excess of 500°F, and for flammable or toxic gases. While rare, austenitic and high nickel casings have been furnished. On dry compressors, the casing normally includes a water jacket. While referred to as a cooling jacket, the cooling water or alternative fluid is used as a heat sink or casing stabilizer to help control distortions and clearance changes. While castings are used for the iron casings, steel casings may be fabricated or cast and fabricated. Nozzle connections and allowable forces and moments are specified by the API standard, using the NEMA steam turbine equations for the force and moment basis. Most casings are vertically split, using end closures and withdrawing the rotors axially for maintenance. On the larger dry machines, the casing is horizontally split, to facilitate the removal of the heavier rotors.

Rotors

The rotor is the working portion of the compressor and, as was pointed out, is machined to both generate the helix and form the profile (see Figure 4-13). Some dry compressors are furnished with hollow rotors

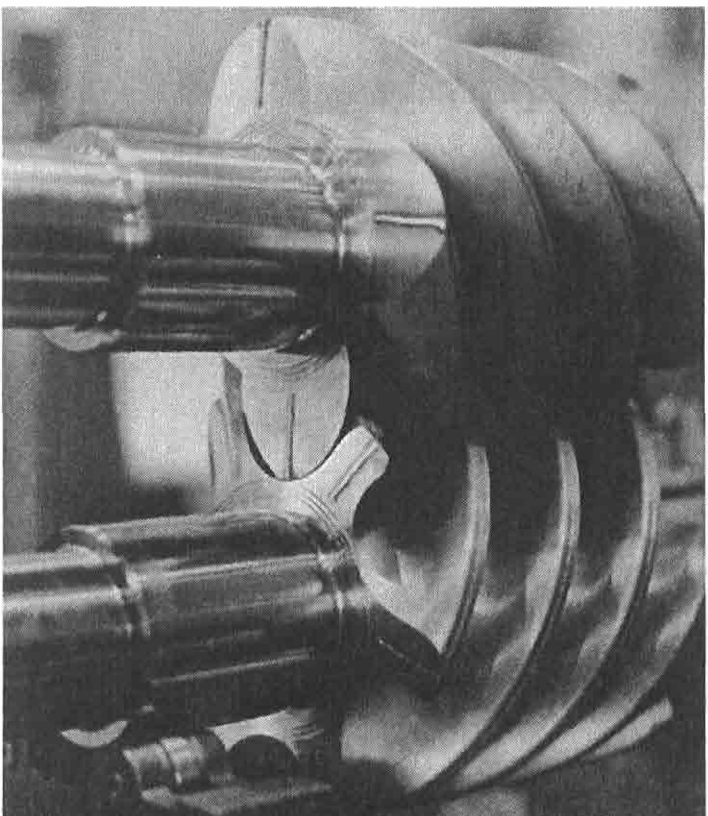

Figure 4-13. A rotor set for an oil-free helical-lobe compressor. (*Courtesy of A-C Compressor Corporation*)

through which cooling fluid is circulated. As with the cooling jacket, the cooling is somewhat misnamed because the more important aspect of the fluid is to help stabilize the rotor. Materials of construction are steel in most applications. The material may be either a forging or bar stock, based on size availability of the bar stock in the quality needed. Other materials are used whenever carbon steel is not compatible with the gas being compressed. These range from stainless, either of the austenitic or 12 chrome type, to more exotic nickel alloys.

Some vendors furnish coatings for the rotors in order to keep the rotor from wearing and losing seal clearance. One such coating is TFE. There is wide diversity of opinion on the value of coatings in light of varying performance. For the purpose of renewing clearances at the time of maintenance, some vendors use a renewable seal strip. This is a very good feature on dry compressors where seal strip clearance is .0005 to .001 inch per inch of rotor diameter at 350°F. If the compressor is designed for 450°F and run in at 500°F, the loss in area would result in an efficiency loss of ½ to 1%.

Rotor speeds are such that dynamic balancing is required for proper vibration control. Also, while the critical speeds are generally above the operating speed, review of the rotor dynamics should not be ignored, particularly for the dry type.

Bearings and Seals

For a general discussion of bearings and seals refer to Chapter 5. The coverage at this point will be limited to the identification of the various types used on the screw compressor.

In the larger, dry process compressors, the radial bearings are of the *sleeve* or *tilting pad* type. Bearing surfaces use a high tin babbitt on a steel backing. API 619 requires the bearings to be removable without removing the rotors or the upper half on the horizontally split machine. Thrust bearings are generally tilt pad type, though not necessarily symmetric. On standardized compressors for air or refrigeration, the bearings are normally the *rolling element* type. Some standardized dry compressors use a *tapered land* thrust bearing. Most of the flooded compressors and some of the standardized dry compressors use rolling element thrust bearings. In all cases, the bearings are pressure-lubricated with some compressors using the gas differential pressure to circulate the lubricant and thus pressurize the bearings. For difficult services mentioned previously, unusual bearings may be used, such as graphite with a sulfuric acid flooding medium.

In dry compressors, shaft end seals are generally one of five types. These are labyrinth, restrictive ring, mechanical contact, liquid film, and dry gas seal. The *labyrinth* type is the most simple but has the highest leakage. The labyrinth seal is generally ported at an axial point between the seals in order to use an eductor or ejector to control leakage and direct it to the suction or a suitable disposal area. Alternatively, a buffer gas is used to prevent the loss of process gas. Appendix D presents a calculation method for use with labyrinth seals.

Probably the most common seal is the *restrictive ring type,* normally used in the form of *carbon rings*. This seal controls leakage better than the non-floating labyrinth type, although it wears faster. The carbon ring seal does not tolerate dirt as well as the labyrinth seal. The carbon ring seal and the labyrinth seal may be ported for gas injection, ejection, or a combination of both. Any injection gas should be clean.

The *mechanical contact seal* is a very positive seal. The seal is normally oil-buffered. The mechanical seal, which is the most complex and expensive, is used where gas leakage to the atmosphere cannot be tolerated. This may be due to the cost of the gas, as in closed-loop refrigeration, or where the process gas is toxic or flammable. The mechanical contact seal requires more power than the other seals, which is a deterrent to its use on lower power compressors.

The *liquid film seal* uses metallic sealing rings and is liquid buffered to maintain a fluid film in the clearance area and thereby preclude gas leakage. It is not unusual in the screw compressor to find the radial bearing and seal combined.

The *dry gas seal* is a variation of the mechanical contact seal. It differs in that it uses a microscopically thin layer of gas to separate and lubricate the faces. The seal is configured in a tandem or double-opposed seal arrangement. More complete details are covered in Chapter 5 under Dry Gas Seals.

Timing Gears

In screw compressors of the dry type, the rotors are synchronized by timing gears. Because the male rotor, with a conventional profile, absorbs about 90% of the power transmitted to the compressor, only 10% of the power is transmitted through the gears. The gears have to be of good quality both to maintain the timing of the rotors and to minimize noise. Because the compressor will turn in reverse on gas backflow, keeping gear backlash to a minimum is important. A check valve should be included in the compressor installation to prevent gas backflow. To control the backlash in the gears, a split-driven gear is used to provide adjustment to the gear lash and maintain timing on reverse rotation. To provide timing adjustment, the female rotor's timing gear is made to be movable relative to its hub. A close-up of a timing gear set is shown in Figure 4-14.

Timing gears are machined from low alloy steel, normally consisting of a chrome, nickel, and molybdenum chemistry. API 619 mandates an AGMA quality 12 gear, which is commonly used. The gears are of the helical type, which also help control noise. The pitch line runout must be minimized to control torsional excitation. The gears are housed in a chamber outboard from the drive end and are isolated from the gas being compressed.

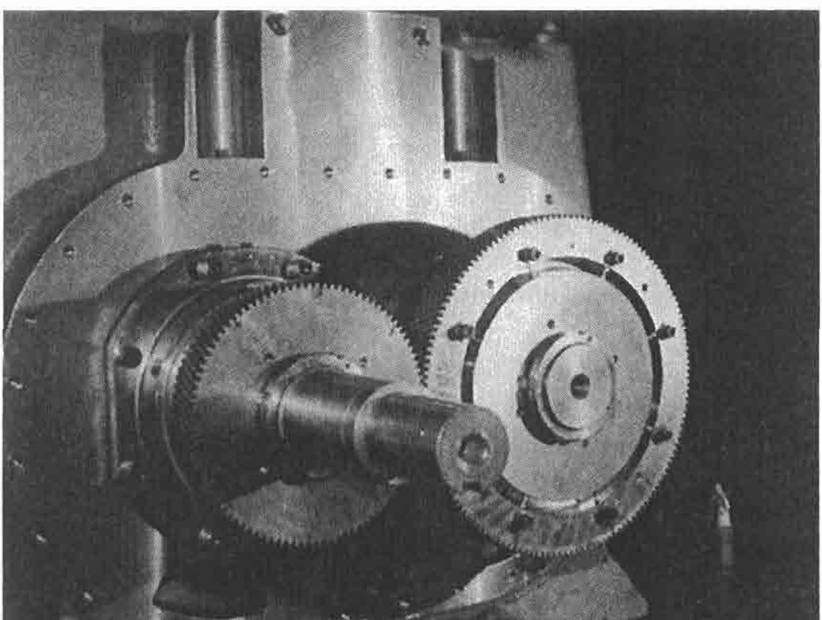

Figure 4-14. Timing gears for an oil-free helical-lobe compressor. Note the timing adjustment capability on the right side gear. (*Courtesy of A-C Compressor Corporation*)

Capacity Control

Until the slide valve came on the scene, suction throttling was about the only capacity control available to the fixed speed flooded screw compressor. Suction throttling is generally not appropriate on positive displacement compressors because of potential high pressure ratios and resulting high temperatures. The previous examples showed that off-design operation is not energy efficient. Control by use of a bypass is not satisfactory due to the lack of power turndown with the net capacity reduction. If a variable speed driver can be justified, speed control can be used to provide good control in an energy efficient manner. The slide valve offers a more economical alternative to speed control, particularly for small compressors.

The slide valve moves parallel to the rotor axis and changes the area of the opening in the bottom of the rotor casing. This lengthens or shortens the region of compression of the rotor and returns gas to the suction side before compression has taken place. Figure 4-15 shows this in diagram

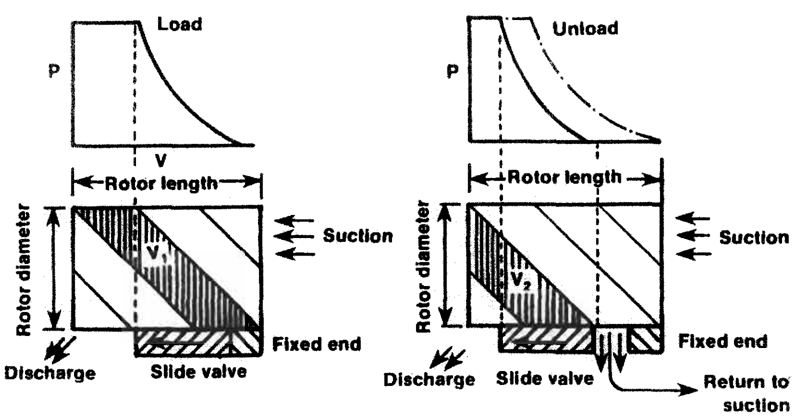

Figure 4-15. Operation of a slide valve. (*Courtesy of Mayekawa Manufacturing Company, Ltd.*)

form and makes it somewhat easier to follow the logic. Figure 4-16 is a cross section of a flooded screw compressor with a slide valve. The slide valve is readily adaptable to the flooded compressor because of ease of lubrication. It can be used in the dry compressor, as well, if provision is made for lubrication of the slide. Figure 4-17 compares the slide valve, labeled as capacity-controlled compressor, with the ideal and a fixed-capacity machine using suction throttling. The energy savings are signifi-

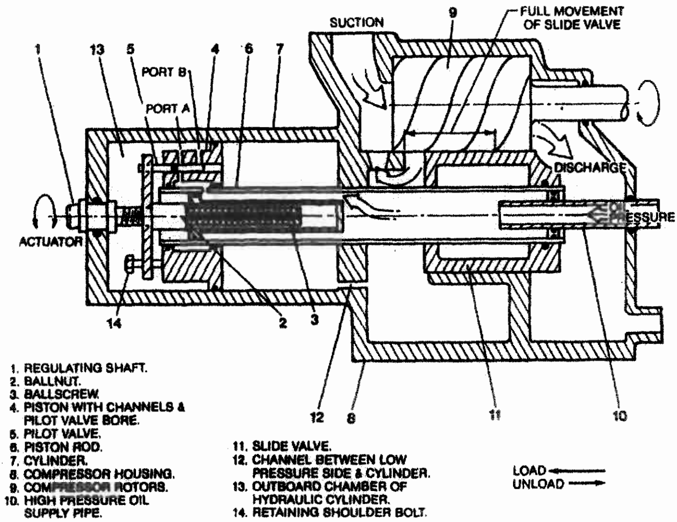

Figure 4-16. Cross section of a screw compressor. (*Courtesy of Sullair*)

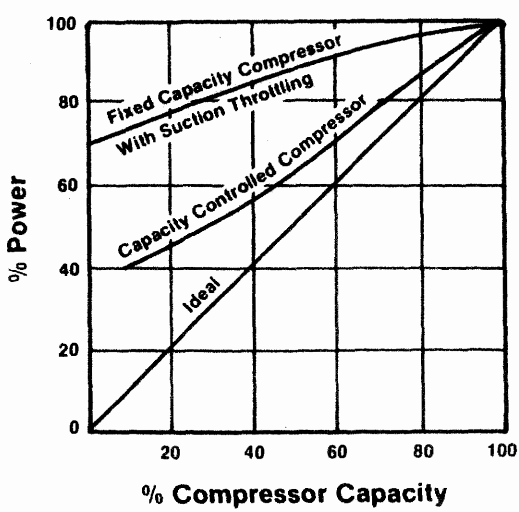

Figure 4-17. Power demand for flooded screw compressors using slide valve and suction throttling [1]. (*Reprinted by permission of the Council of the Institution of Mechanical Engineers from "The Place of the Screw Compressor in Refrigeration"*)

cant. Figure 4-18 is a schematic of the oil system and the slide valve with a piston operator.

Variation to the slide valve is the *turn valve.* The valve functions by turning rather than sliding but has the same effect as the slide valve. The plug is

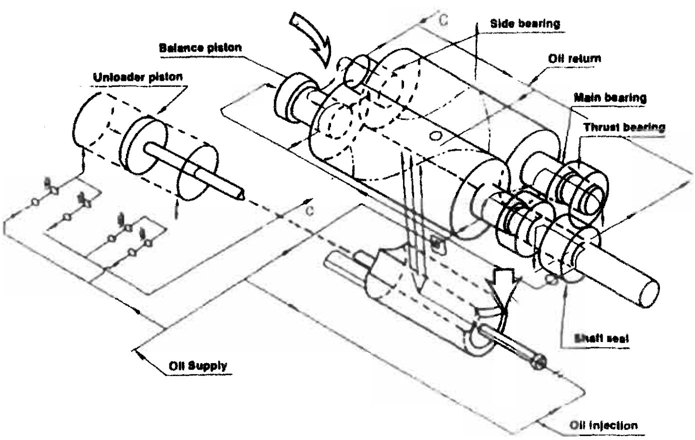

Figure 4-18. Schematic of oil system and slide valve with piston operator. *(Courtesy of Mayekawa Manufacturing Company, Ltd.)*

grooved and the compressor casing is slotted. As the valve turns, the grooves move away from the slots, passing a quantity of gas. The leakage in this valve is greater because of the imperfect matching of grooves and slots.

A control innovation that has potential to broaden the screw compressor's application range is the variable-volume-ratio compressor.[4] A diagram of a low- and a high-volume-ratio compressor is shown together with a *movable slide stop,* which is used with the slide valve to form a variable volume ratio compressor in Figure 4-19. While the "volume ratio" terminology is more descriptive, it is more commonly known as "built-in pressure ratio." By controlling inlet volume with the slide valve and the discharge volume by use of the slide stop, an infinite number of volume ratios can be achieved (see Figures 4-20 and 4-21).

Straight Lobe

Compression Cycle

Straight-lobe compressors, or blowers, as they are commonly called are low-pressure machines. The feature unique to these compressors is that the machines do not compress the gas internally as do most of the other rotaries. The straight-lobe compressor uses two rotors that intermesh as they rotate (see Figure 1-7). The rotors are timed by a set of gears. The lobe shape is either an involute or cyclodial form. A rotor may have either two or three lobes. As the rotors turn and pass the inlet port, a volume of gas is trapped and carried between the lobes and the outer

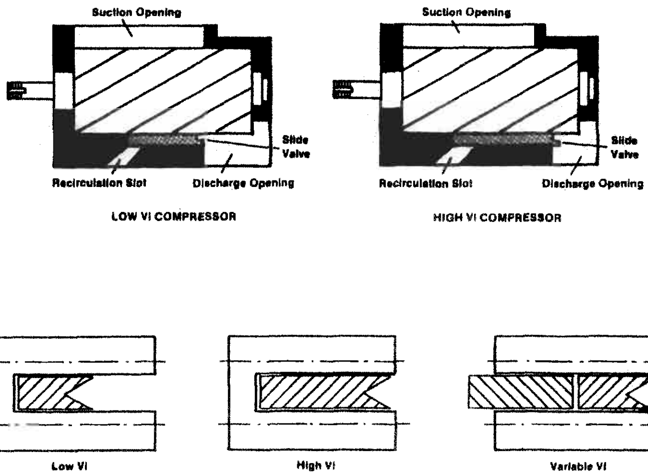

Figure 4-19. Side and top view of slide valve arrangements for low- high- and variable-volume ratio compressors [4].

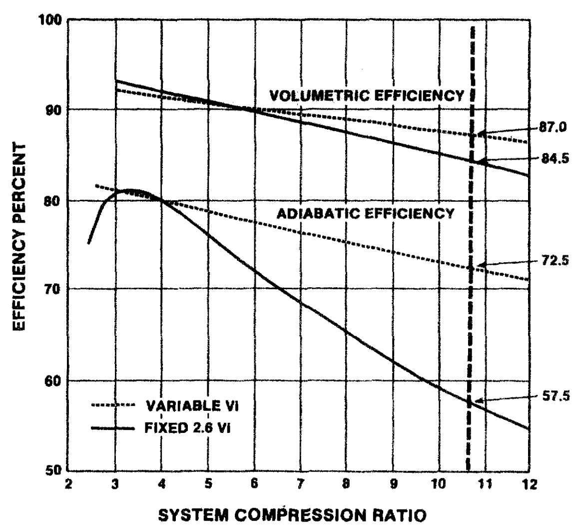

Figure 4-20. Efficiency comparison between a variable-volume ratio and a fixed-volume ratio compressor. These compressors have an asymmetric rotor profile [4].

cylinder wall. When the lobe pushes the gas toward the exit, the gas is compressed by the back pressure of the gas in the discharge line.

Volumetric efficiency is determined by the tip leakage past the rotors, not unlike the rotary screw compressor. The leakage is referred to as slip.

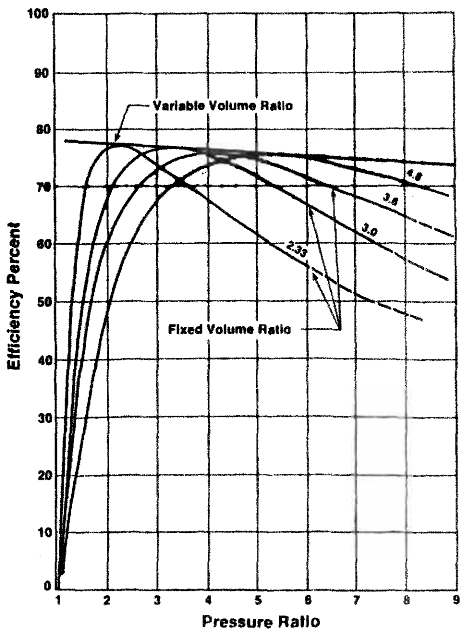

Figure 4-21. Efficiency improvement with variable volume ratio screw compressors [4].

Slippage is a function of the rotor diameter and differential pressure for a given gas. Slippage is determined by test. For the test, the differential pressure is imposed on the blower and the speed gradually increased until the point is reached where the output just matches the slip leakage. This point is detected by watching for the machine to just begin to give a positive output. The speed at which this occurs is called the slip speed. A slip speed is determined for each of several pressure differentials.

Output volume is calculated from displaced volume multiplied by the difference between a desired speed and the slip speed, rather than using the concept of volumetric efficiency as in the other positive displacement compressors. Output volume is expressed by

$$Q_1 = Q_r (N - N_s) \qquad (4.9)$$

where

Q_1 = delivered volume
Q_r = displaced volume per revolution
N = rated speed
N_s = slip speed

While volumetric efficiency is not commonly used, the following relationship is an approximation for comparison purposes.

$$E_v = 1.0 - N_s/N \tag{4.10}$$

Sizing

Sizing for the straight-lobe compressor is normally done using catalog data. Rotor lengths range from approximately one to two times the rotor diameter. Individual frame sizes within a given vendor's line may exceed these limits. Maximum tip speeds are in the 125 fps range with some units approaching 140 fps. The following relationship will permit size approximation if no catalog is available. It should be remembered that this is a rough size and that it may not be a standard with any of the vendors. It is helpful, however, to know at least one dimension from the vendor's line to make an estimate more meaningful.

$$Q_r = d^2 \times l_r/c \tag{4.11}$$

where

d = rotor diameter
l_r = rotor length
c = sizing constant

A sizing constant of 1.2 can be used to make a reasonable approximation of many commercial sizes. The constant, c, varied from 1.11 to 1.27 for a number of the frames investigated. With the displaced size approximated, the delivered volume can be calculated. Use Equation 4.10 and an assumed volumetric efficiency of .90. This is arbitrary, as the actual volumetric efficiency varies from .95 to .75 or lower for the higher differential pressure applications. Once a slip speed has been determined, Equation 4.9 can be used to complete the calculation. The tip speed should stay near 125 fps.

Discharge temperature and horsepower can be calculated using the expressions given in the rotary screw compressor section, Equations 4.6 and 4.7. Efficiencies are a function of the differential pressure. For differential pressures of 5 psi and less, the efficiency is .80 or higher. It falls off fast with increased pressure rise, dropping to .60 in the 10 to 12 psi differential range. Mechanical losses are near .07 of the gas horsepower in the smaller sizes and .05 to .06 in the larger sizes. Larger is considered as over 5000 cfm.

Applications

The straight-lobe blowers are used both in pressure and vacuum service. Larger units are directly connected to their drivers and the smaller units are belt-driven. The drivers are normally electric motors. Some of the larger models offer an internal gear arrangement to permit the direct connection of a two- or four-pole electric motor. These blowers would normally operate at less than the nominal four-pole speed on a 60 Hz system of 1800 rpm. The main limitation to this rotary compressor is the differential pressure with the longer rotors where deflection is large. For a two-lobe machine, caution should be used when the rotor is more than 1.5 times the rotor diameter at pressures in excess of 8 psi differential. The three-lobe compressors inherently have a stiffer rotor and can sustain a higher differential with less difficulty. The practical upper limit should be 10 psi differential for units above 3000 cfm and 12 psi differential for the smaller units.

Mechanical Construction

Straight-lobe compressor casings, also called housings or cylinders by different manufacturers, are furnished in cast iron by all vendors. There is an optional aluminum construction available for special applications. Inlet and outlet are suitable for a 125 pound standard ANSI flanged connection.

Rotors are cast from ductile iron. Again, the exception is the aluminum construction. Shafts are steel and are cast into the rotors or are pinned to the rotor in a stub shaft construction arrangement. An alternate design has the rotors drilled for through shafts. Rotors are supported by a set of rolling element bearings on the outboard end of each rotor. The timing gears are forged steel on the more competitively priced models and are low alloy steel on the more rugged models. Also on the competitive models, the gears are straight spur, while helical gear teeth are furnished on the heavy duty models.

The most common seals are the lip type and the labyrinth type. Mechanical seals are available where seal leakage must be controlled.

Lubrication is splash type and grease is used on the more competitive models. There are variations available with internal pressure lubrication systems. Some models can be equipped with an external lube system, and for rare cases, API 614 lubrication systems have been proposed.

Sliding Vane

Compression Cycle

The *sliding vane* compressor consists of a single rotor mounted eccentrically in a cylinder slightly larger than the rotor. The rotor has a series of radial slots that hold a set of vanes. The vanes are free to move radially within the rotor slots. They maintain contact with the cylinder wall by centrifugal force generated as the rotor turns.

The space between a pair of vanes and the rotor and the cylinder wall form crescent shaped cells (see Figure 1-8). As the rotor turns and a pair of vanes approach the inlet, gas begins to fill the cell. The rotation and subsequent filling continue until the suction port edge has been passed by both vanes. Simultaneously, the vanes have passed their maximum extension and begin to be pushed back into the rotor by the eccentricity of the cylinder wall. As the space becomes smaller, the gas is compressed. The compression continues until the leading vane crosses the edge of the discharge port, at which time the compressed gas is released into the discharge line.

The port location must be matched to the pressure ratio dictated by the application for efficient compression to take place. Figure 4-22 is an indicator diagram of a compression cycle. If the port has been optimized for a ratio of P_2/P_1, the compression line is a smooth curve, as can be seen as compression proceeds from point 1 to 2. Note point 2 is at the same pressure level as P_2. If the external pressure is higher than the pressure for which the port was cut, so that $P_o/P_1 > P_2/P_1$, then, when the port opens at point 2, discharge gas will return to the compressor from the line and must again be expelled from the compressor. This energy waste is depicted by the shaded area to the left of the line o-2. Conversely, if the external pressure ratio is lower than the pressure ratio for which the port was cut, where $P_U/P_1 < P_2/P_1$, then the gas will be overcompressed to point 2 and when the port opens, it will expand to point u. The lost energy is represented by the shaded area to the right of the line u-2.

Sizing

The displaced volume of the sliding vane compressor can be calculated if certain geometric data are available. Unfortunately the vendor catalog data generally will not be very useful in establishing the geometry because frame designations are not directly related to any convenient measurement. The vendor's literature does give the expected capacity for air at various pressures, which include the displaced volume and the vol-

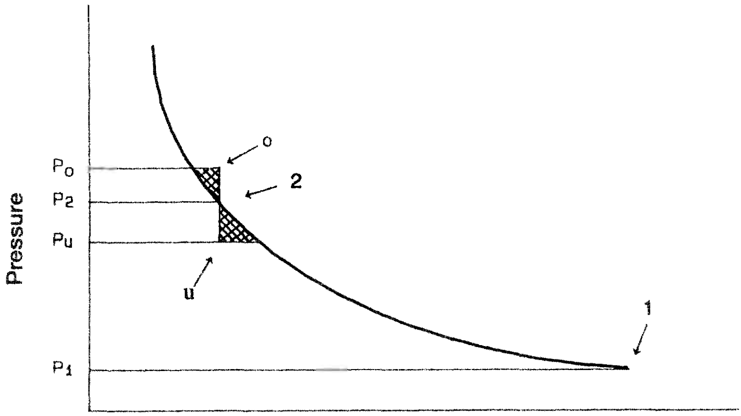

Figure 4-22. Pressure-volume diagram of the compression cycle of a sliding vane compressor.

umetric efficiency. Geometric information might be available, such as may occur with a unit being considered for reuse where measurements can be made. The name plate, of course, would provide the rated conditions. The design ratio would be available and could be used to make a decision regarding the suitability for alternate service. With the bore, the rotor diameter, the cylinder length, and vane number and thickness, a calculation to determine the displaced capacity per revolution may be made. By applying an estimate of volumetric efficiency to the displacement value, the speed needed for a given output can be calculated. If the speed is within the allowable limits, and the pressure ratio is in the range of the original value, the compressor can be reapplied.

The following provides an estimate of Q_r, the displaced volume per revolution.

$$Q_r = 2\, e\, L\, (\pi D - m\, s) \tag{4.12}$$

where

e = eccentricity, $R - r$
R = radius of cylinder bore
r = radius of rotor
D = diameter of cylinder bore
m = number of vanes
s = vane thickness
L = cylinder length

Some typical values of geometry are r/R = .88 and e = .12 R. The L/R ranges from 4.5 to 5.8, increasing with the size of the compressor. Volumetric efficiency ranges from approximately .90 at 10 psig to .85 at 30 psig for air service. Volumetric efficiency is slightly better for heavier gases and lower for the lighter gases. Typical maximum vane speed, calculated using the cylinder bore as the diameter, is 50 fps.

Power requirements and discharge temperatures are calculated using the same relationships as used with the other rotary compressors already discussed. The efficiency is .80 for air service and pressure in the 30 psig range. The mechanical losses are higher than the other rotaries. The mechanical loss is variable and dependent on gas, lubrication, and other factors. For an estimate, use .15 of the gas horsepower. This approximation should be close enough for an estimate.

Application Notes

The sliding vane compressor can be used to 50 psig in single-stage form and when staged can be used to 125 psig. An often overlooked application for the sliding vane machine is that of vacuum service where, in single-stage form, it can be used to 28 in hg. Volumes in vacuum service are in the 5000 cfm range. For pressure service, at the lower pressures, volumes are just under 4000 cfm and decrease to around 2000 cfm as the discharge pressure exceeds 30 psig.

The sliding vane compressor is used in gas gathering and gas boosting applications in direct competition with the reciprocating compressor. Efficiency is not as good, but the machine is rugged and light and doesn't have the foundation or skid weight requirement of the reciprocator. The compressor is also widely used as a vapor extraction machine in a wide variety of applications, which include steam turbine condenser service for air extraction.

Vane wear must be monitored in order to schedule replacement before the vanes become too short and wear the rotor slots. If the vanes are permitted to become too worn on the sides or too short, the vane may break and wedge between the rotor and the cylinder wall at the point of eccentricity, possibly breaking the cylinder. Shear pin couplings or equivalent torque limiting couplings are sometimes used to prevent damage from a broken vane under sudden stall conditions.

As in most jacket-cooled compressors, the cooling acts as a heat sink to stabilize the cylinder dimensionally. The jacket outlet temperature should be around 115°F and be controlled by an automatic temperature regulator if the load or the water inlet temperature are prone to change.

Most of the drivers used with the sliding vane compressor are electric motors. Variable speed operation is possible within the limits of vane speed requirements. The vanes must travel fast enough to seal against the cylinder wall but not so fast that they cause excessive wear. For the smaller units, under 100 hp, V-belts are widely used. Direct connection to a motor, however, is possible for most compressors and is used throughout the size range.

Mechanical Construction

The cylinder is generally constructed of cast iron and includes the water jacket. The bore is machined and brought to a good finish to reduce the vane sliding friction. The inlet and outlet connections are flanged. The heads, which also house the bearings and stuffing box, are also made of cast iron.

The rotor and shaft extension are machined from a single piece of bar stock or from a forging in all but the largest sizes, where the rotor and shaft may be made as two separate parts. The material is carbon steel for the single-piece models. The larger compressors, using the two-piece rotor arrangement, use carbon steel for the shaft and cast iron for the rotor body. The rotor body is attached to the shaft using a press fit. Keys are used to lock the rotor body to the shaft. Vanes attach to the rotor body by means of milled slots.

For the lubricated machines, vanes are made of a laminated material impregnated with phenolic resin. For a non-lubricated design, carbon is used. The vane number influences the differential pressure between adjacent vane cells. This influence becomes less as the number of vanes increases.

Rolling element bearings are widely used, generally the roller type. Seals are either a packing or mechanical contact type. Packing and bearings are lubricated by a pressurized system. For the non-flooded, lubricated compressor a multiplunger pump, similar to the one used with reciprocating compressors, is used. Lubrication is directed from the lubricator to drilled passages in the compressor cylinder and heads. One feed is directed to each of the bearings. Other feeds meter lubrication onto the cylinder wall. As the vanes pass the oil injection openings, lubricant is spread around the cylinder walls to lubricate the vane tips and eventually the vanes themselves. The oil entering the gas stream is separated in the discharge line. Because of the high local heat, the lubricant may have broken down and, therefore, is not suitable for recycling.

Flooded compressors pressure feed a large amount of lubricant into the compressor where it both cools the gas and lubricates the compressor. It is separated from the gas at the discharge line and recycled.

Liquid Piston

Operation

The liquid piston compressor is a unique type of rotary compressor in that it performs its compression by use of a liquid ring acting as a piston. Refer to Figure 1-9 for a cross section of this compressor. As with the sliding vane compressor, the single rotor is located eccentrically inside a cylinder or stator. Extending from the rotor is a series of vanes in a purely radial or radial with forward curved tip orientation. Gas inlet and outlet passages are located on the rotor. A liquid compressant partially fills the rotor and cylinder and orients itself in a ring-like manner as the rotor turns. Because of the eccentricity, the ring moves in an oscillatory manner. The center of the ring communicates with the inlet and outlet ports and forms the gas pocket. As the rotor turns, and the pocket is moving away from the rotor, the gas enters through the inlet and fills the pocket. As the rotor turns, it carries the gas pocket with it. Further turning takes the liquid ring from the maximum clearance area toward the minimum side. The ring seals off the inlet port and traps the pocket of gas. As the liquid ring is taken into the minimum clearance area, the pocket is compressed. When the ring uncovers the discharge port, the compressed pocket of gas is discharged.

Performance

Efficiency of the liquid piston is about 50%, which is not very good compared to the other rotary compressors. But because liquid is integral to the liquid piston compressor, taking in liquid with the gas stream does not affect its operation as it would in other types of compressors. Because of significant differences in the construction of the various competitive models of this compressor, no universal sizing data are available. The process engineer will therefore have to rely on catalog data for sizing estimates. The liquid ring compressor is most often used in vacuum service; although, it can also act as a positive pressure compressor. The liquid piston machine can be staged when the application requires more differential pressure than can be generated by a single stage. The liquid piston compressor can be used to compress air to 100 psig. Vacuums of 26 in hg are possible. Flow capacity ranges from 2 cfm to 16,000 cfm.

Mechanical Construction

Standard materials for the compressor are cast iron for the cylinder and carbon steel for the shaft. The rotor parts are steel. The liquid piston compressor has another feature that compensates for low efficiency. By using special materials of construction and compatible liquid compressant, unusual or difficult gases may be compressed. By using titanium internal materials and water as a compressant, gases containing wet chlorine can be compressed. This is a very difficult application for most of the other compressor types.

Both rolling element and split sleeve bearings are used. Normally, packing is used for shaft sealing or for special services. Mechanical contact seals can be used.

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5

Centrifugal Compressors

Introduction

Centrifugal compressors are second only to reciprocating compressors in numbers of machines in service. In the process plant arena, the leader in numbers is too close to call with any degree of certainty. Where capacity or horsepower rather than numbers is considered as a measure, the centrifugal, without a doubt, heads the compressor field. During the past 30 years, the centrifugal compressor, because of its simplicity and larger capacity/size ratio, compared to the reciprocating machine, became much more popular for use in process plants that were growing in size. The centrifugal compressor does not exhibit the inertially induced shaking forces of the reciprocator and, therefore, does not need the same massive foundation. Initially, the efficiency of the centrifugal was not as high as that of a well-maintained reciprocating compressor. However, the centrifugal established its hold on the market in an era of cheap energy, when power cost was rarely, if ever, evaluated.

The centrifugal compressor has been around for quite a long time. Originally, it was used in process applications at relatively low-pressure, high-volume service. In the early 1930s, the main application was in the steel industry, where it was used chiefly as an oxidation air compressor for blast furnaces. The centrifugal displaced the reciprocating blowing engines that were being used at the time. The centrifugal was employed in the coal-to-coke conversion process, where it was used to draw off the gas from the coke ovens. In the late 1930s, the beginning of air conditioning for movie theaters, department stores, and later office buildings, gave birth to a generation of small centrifugals, which gained the advantage because of smaller size and absence of shaking forces. These forces were difficult to contain when a comparable capacity reciprocating compressor was used in a populated environment. It was the smaller compressor design that was able to penetrate the general process plant market, which had historically belonged to the reciprocating compressor. As stated previously, the growth of plant size and low-cost energy helped bring the centrifugal compressor into prominence in the 1950s. As the compressor grew in popularity, developments were begun to improve reliability, performance, and efficiency. With the increase in energy cost in the mid 1970s, efficiency improvements moved from last to first priority in the allocation of development funds. Prior to this turn of events, most development had concentrated on making the machine reliable, a goal which was reasonably well achieved. Run time between overhauls currently is three years or more with six-year run times not unusual. As plant size increased, the pressure to maintain or improve reliability was very high because of the large economic impact of a nonscheduled shutdown. This being the case, even with an increase in the efficiency emphasis, there is no sympathy for an energy versus reliability trade-off. The operating groups tend to evaluate reliability first, with the energy cost as secondary.

The centrifugal compressor has been applied in an approximate range of 1,000 cfm to 150,000 cfm. Plant air package centrifugals are available somewhat lower in capacity but have problems competing because of other more efficient compressors that are available in the lower ranges. Pressure ratios and pressure levels are difficult to describe in general terms because of the wide range of applications. Pressure ratio is probably the best parameter for comparing the centrifugal compressor to other types of compressors. Polytropic head, as defined in Chapter 2, is much more definitive to the dynamic machine but does not mean much numerically to a user. Pressure ratios of up to 3 and higher are available for single-stage compressors, operating on air or nitrogen. Multistage machines, of the process type, generally operate at a pressure ratio of less than 2 per impeller.

Classification

A better definition of a compressor stage can be made here to prevent confusion later on. Up to this point, in the positive displacement compressors, a compressing entity and a stage were one and the same; for example, a cylinder is a stage in the reciprocating compressor. The centrifugal and the other dynamic compressors to be discussed have the problem of a dual vernacular, one used by the machine design engineer and the one used by the process engineer. To the machine builder, a stage is an impeller-diffuser pair, whereas the process designer tends to think of a stage as a process block that equates to an uncooled section of one or more impeller and diffuser sets. There is no problem with the single impeller machine as the two are synonymous. The confusion comes with the use of the multiple impeller machine. To make everyone equally happy or unhappy, as the case may be, hereinafter, a process compression stage will be referred to as an *uncooled section.* Whenever the term *stage* must be used in the process connotation, it will be called a *process stage.* The multiwheeled machine will retain the name of multistage, and the individual impeller and diffuser pairs will be called a stage.

With the foregoing discussion as an entreé to the types of centrifugal compressors, it seems redundant to classify them as single and multistage. A cross classification can be established by the manner in which the machine casing is constructed, whether it has an axial or radial joint. More commonly, this type of construction is referred to as horizontal and vertical split. For simplicity, the second terminology will be used. The *overhung* style of single stage is an example of the vertical split type of compressor (see Figure 5-1). An example of the horizontally split compressor is the common multistage. Maintenance of the horizontally split compressor is very simple and straightforward, as the rotor may be removed without disturbing the impellers. When the pressure is too high to maintain a proper joint seal or for low molecular weight service, another style commonly used is referred to as a barrel compressor (see Figure 5-2). The barrel uses a vertical split construction. In the multistage configuration, it is constructed with a removable, horizontally split, inner barrel that permits the removal of the rotor without removing the impellers. Many overhung compressors do not permit the removal of the rotor without first removing the impeller.

Another common type of compressor is manufactured in an integrally geared configuration. It is basically an overhung style machine mounted on a gear box and uses the gear pinion shaft extension to mount an

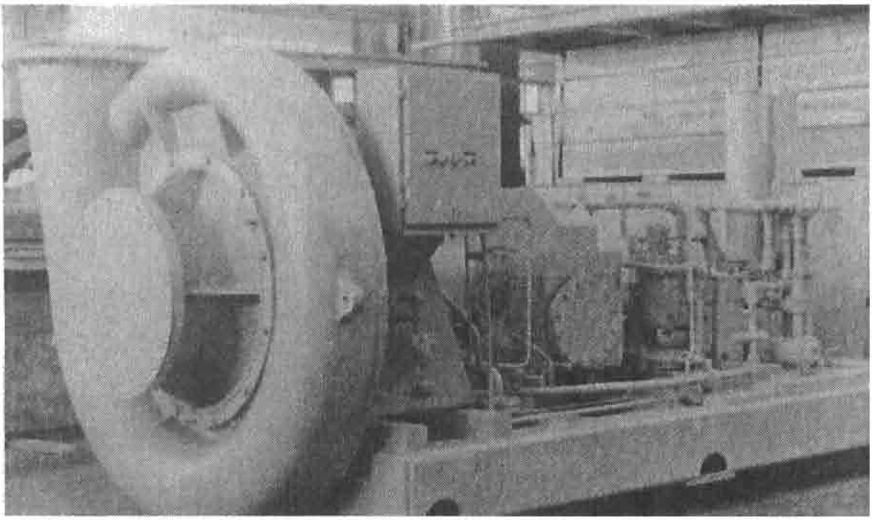


Figure 5-1. Single-stage, vertically split, overhung style centrifugal compressor. (*Courtesy of Elliott Company*)

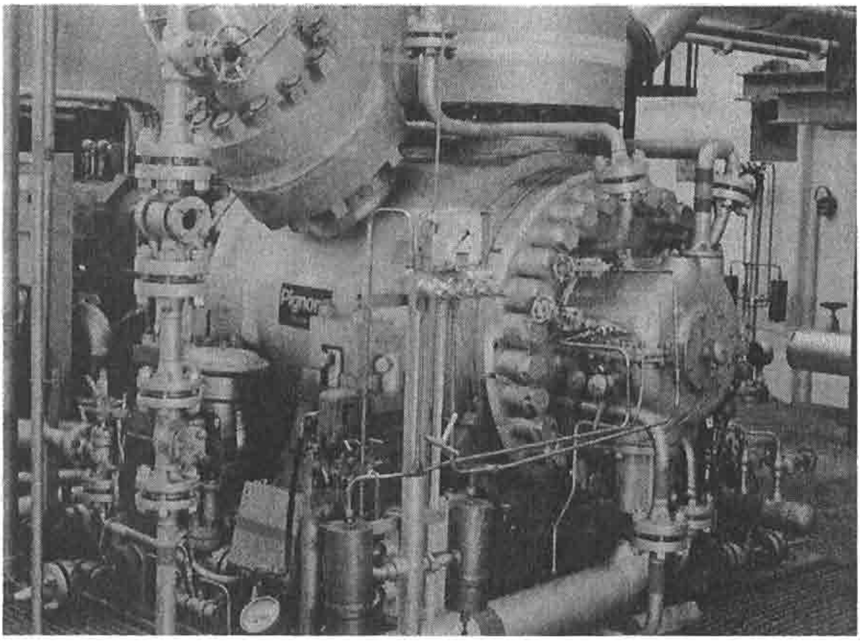


Figure 5-2. Multistage barrel type compressor. (*Courtesy of Nuovo Pignone*)

impeller (see Figure 5-3). The casing is also attached to the gear box. This style is built in both the single and multistage configuration. The most common form of multistage is the plant air compressor, which also has intercoolers included as part of the machine package.

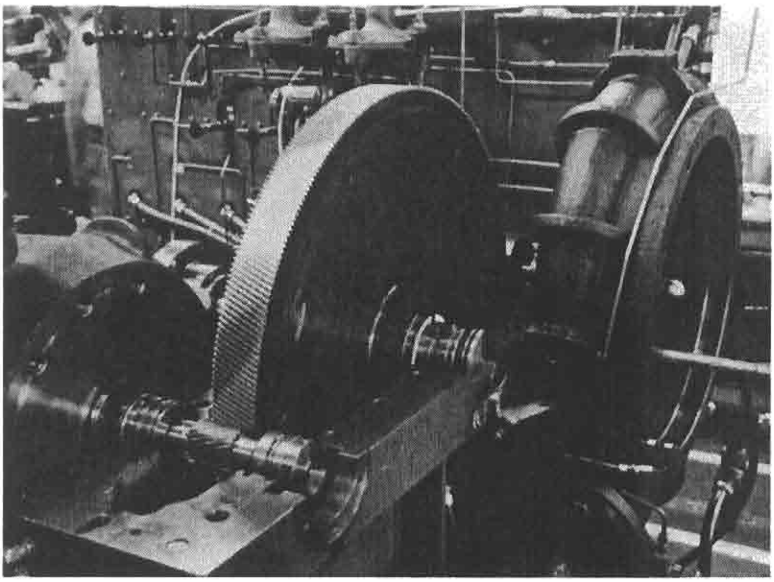


Figure 5-3. Overhung, gearbox mounted centrifugal compressor. (*Courtesy of Atlas Copco Comptec Inc.*)

Arrangement

The single stage can be arranged, as has been discussed in the previous paragraphs, in the overhung style. Figure 5-4 shows a schematic of the compressor. Note that the flow enters axially and exits in a tangential direction. For a comprehensive discussion, it should be mentioned that the overhung style is, on very rare occasions, constructed in the multistage form, usually overhanging no more than two impellers. The overhung compressor is generally more competitively priced than the between-bearing design. Careful application must be made because the overhung impeller configuration is more sensitive to unbalance than the between-bearing design. If impeller fouling is anticipated, this design may not be acceptable.

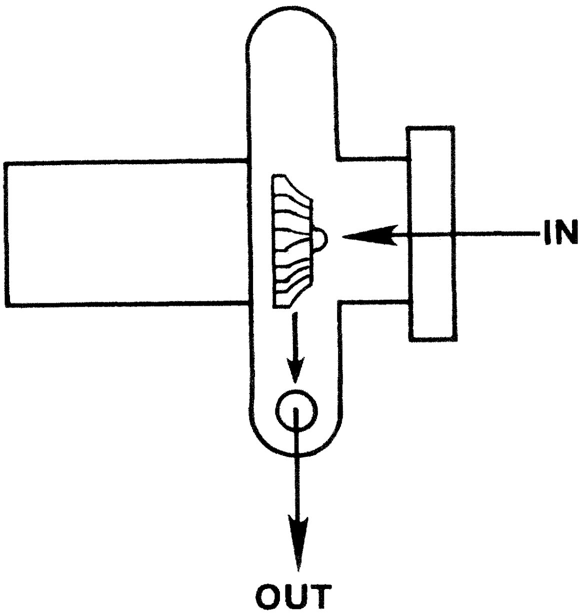


Figure 5-4. Diagram of a single-stage overhung type centrifugal compressor.

A less common form of the single stage is shown in Figure 5-5. In this form, the impeller is located between two bearings, as is the multistage. This type of compressor is sometimes referred to as a *beam type* single stage. The flow enters and leaves in a tangential direction with the nozzles located in the horizontal plane. The between-bearing single stage is found most commonly in pipe line booster service where the inherent rigidity of the two outboard bearings is desirable.

Figure 5-6 is a flow diagram and schematic layout of the integrally geared compressor, and Figure 5-7 shows exploded view. It consists of three impellers, the first located on one pinion, which would have a lower speed than the other pinion that has mounted the remaining two impellers. This arrangement is common to the plant air compressor. Configurations such as this are used in process air and gas services, with the number of stages set to match the application.

Figure 5-8 shows the multistage arrangement. The flow path is straight through the compressor, moving through each impeller in turn. This type of centrifugal compressor is probably the most common of any found in process service, with applications ranging from air to gas. The latter includes various process gases and basic refrigeration service.

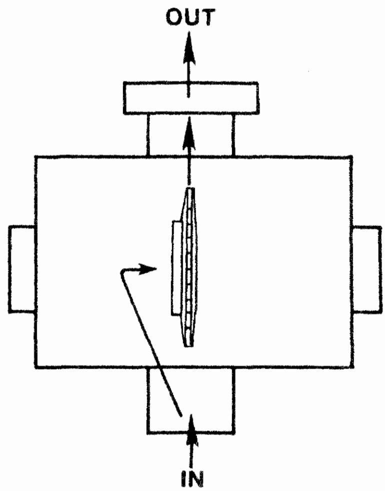


Figure 5-5. Diagram of a beam type single-stage compressor.

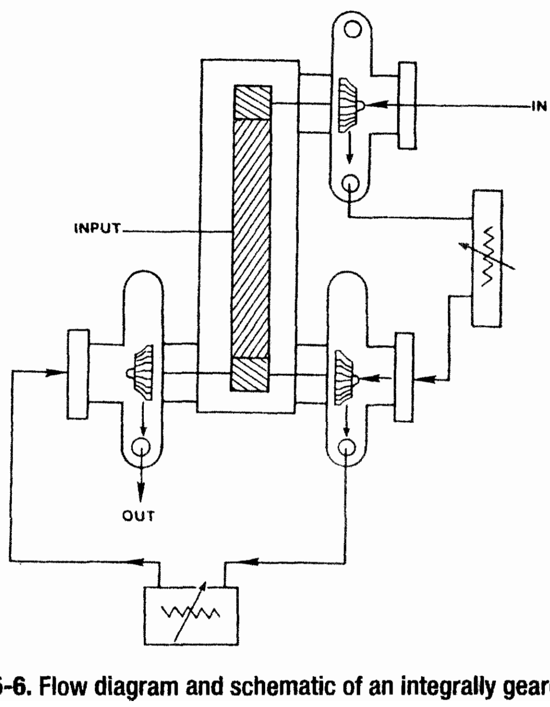


Figure 5-6. Flow diagram and schematic of an integrally geared compressor.

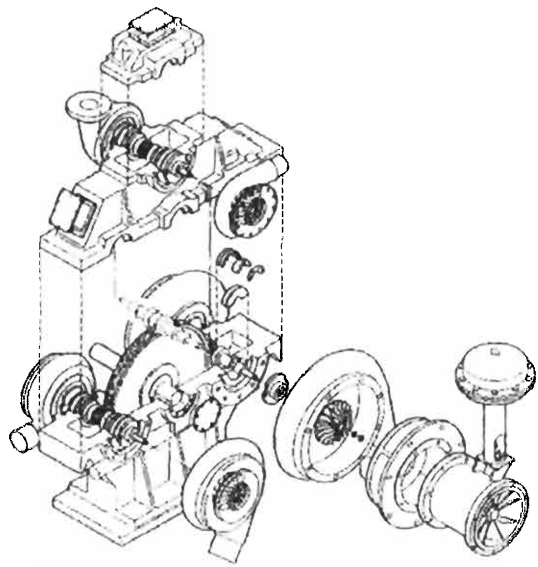


Figure 5-7. An exploded view of an integrally geared compressor. (*Courtesy of Cooper Turbocompressor*)

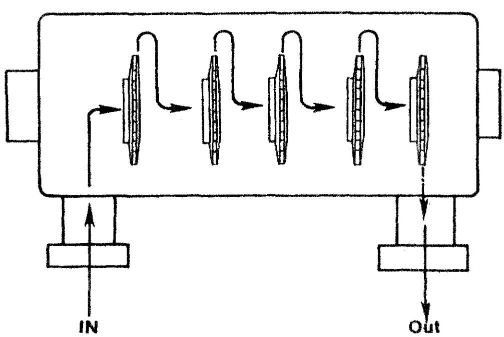


Figure 5-8. Diagram of a multistage centrifugal compressor with a straight-through flow path.

Figures 5-9 and 5-10 depict the two most common forms of in-out arrangements. This arrangement is also referred to as a compound compressor. In these applications, the flow out of the compressor is taken through an intercooler and back to the compressor. The arrangement is not limited to cooling because some services use this arrangement to remove and scrub the gas stream at a particular pressure level. Provision for liquid removal must be made if one of the gas components reaches its saturation

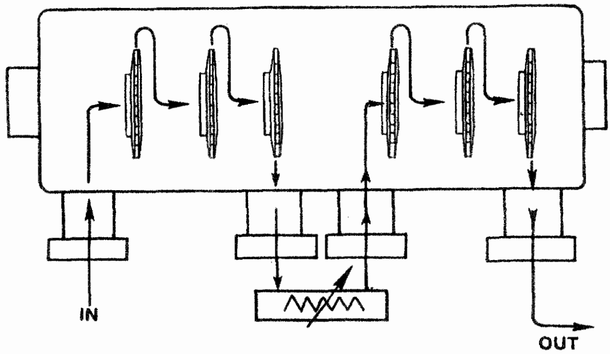


Figure 5-9. Diagram of an in-out arrangement with intercooling.

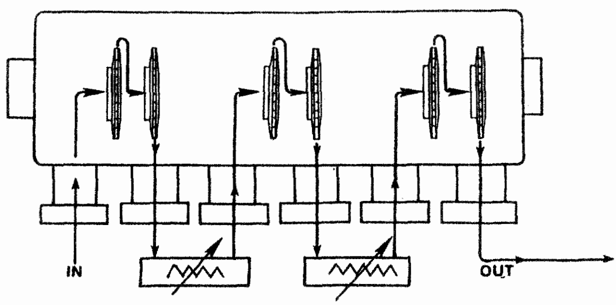


Figure 5-10. Diagram of a double-cooled centrifugal compressor.

temperature in the process of cooling. Figure 5-10 shows a double-cooled or double compound compressor. This arrangement is used mostly when the gas being compressed has a temperature limit. The limit may be imposed by the materials of construction or where the gas becomes more reactive with an increase in temperature and thus sets the limit in a given application. Polymer formation is generally related to temperature and may form the basis for an upper temperature limit. However, with the external cooling, the amount of compression needed can be accomplished in a single case. The physical space needed to locate the multiple nozzles normally limits the number of in-out points to the two shown.

The arrangement shown in Figure 5-11 is referred to as a double-flow compressor (see also Figure 5-12). As indicated in the figure, the flow enters the case at two points, is compressed by one or more stages at each end, and then enters the double-flow impeller. The flow passes through each individual section of the double-flow impeller and joins at

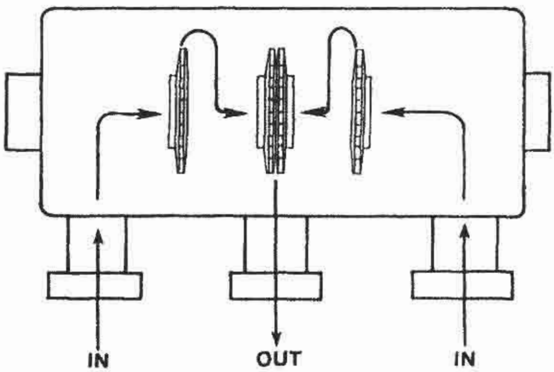


Figure 5-11. Diagram of a double-flow compressor with two inlets.

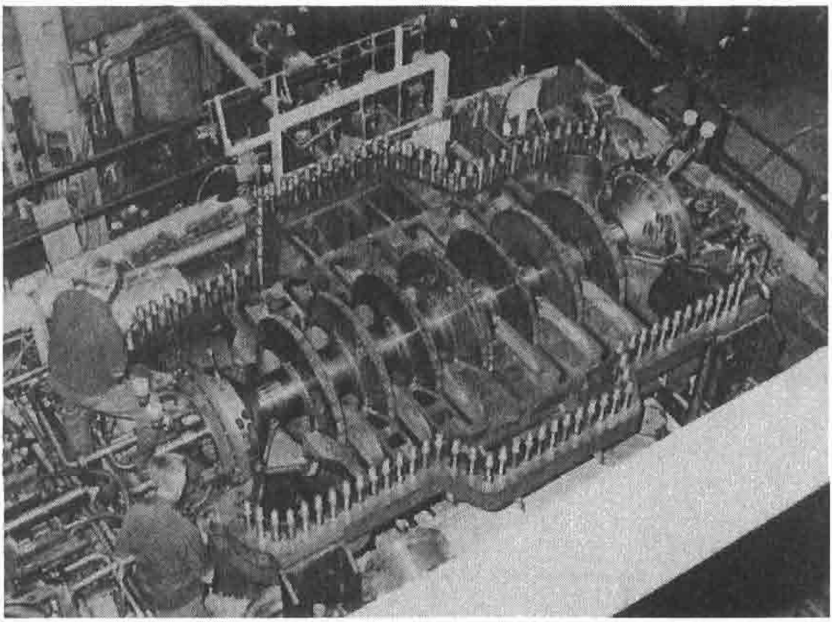


Figure 5-12. A double-flow compressor with inlets on each end and a common center discharge. (*Courtesy of Elliott Company*)

the diffuser. There are various physical arrangements to accomplish the double-flow compression. One variation is to use two back-to-back stages for the final compression and join the flow either internally, prior to leaving the case, or join two separate outlet nozzles outside the case.

From a process point of view, the flow should be joined prior to exiting the discharge nozzle.

Another variation of this arrangement is to use it in the single-stage configuration, where only a single inlet and outlet nozzle is used. The flow enters the case and is divided to each side of the double-flow impeller and then joins at the impeller exit prior to entering the diffuser. Figure 5-13 shows a schematic diagram of the flow in this machine. The advantage of the double-flow arrangement is, of course, that in the same casing size, it doubles the flow. However, the realization of the advantage is more complex. The losses in the flow paths through the double-flow impeller must, in theory, be identical. In practice, of course, this is not possible. The sensitivity is a function of the total head level. The lower the levels, the more nearly the paths must be the same.

The single-stage configuration, the lower head compressor, will exhibit the highest degree of sensitivity to the flow imbalance and have its performance most adversely affected. The multistage configuration, while not as sensitive to the flow anomalies because of the higher head generated, will benefit from careful flow path design to keep the flow balanced to each section of the double flow inlets. If a number of options are open for a given application, the double-flow option should not be the first choice; although, it should be evaluated because successful applications in service indicate that with careful design the compressor will perform satisfactorily.

The arrangement in Figure 5-14, generally called "back to back," is normally considered useful in solving difficult thrust balance problems where the conventional thrust bearing and balance drum size are inade-

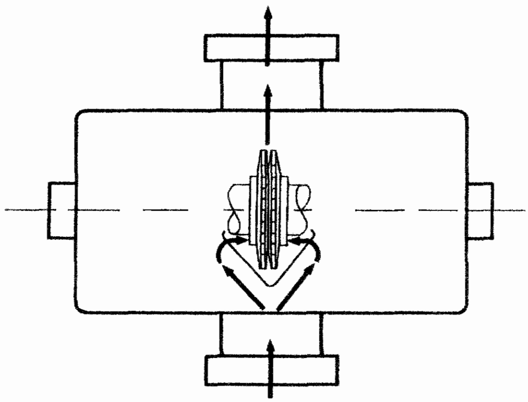


Figure 5-13. Diagram of a double-flow compressor with flow split internally.

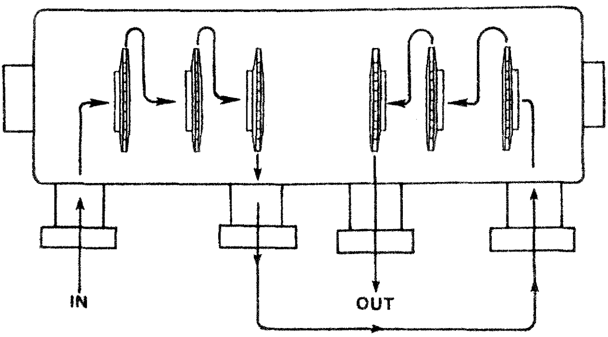


Figure 5-14. Diagram of an arrangement used to overcome a thrust balance problem.

quate or become excessively large. The balance drum will be described in detail in a following section. The flow is removed part way through the compressor and reintroduced at the opposite end, then allowed to exit at the center. Because centrifugal impellers inherently exhibit a unidirectional thrust, this arrangement can be used to reduce the net rotor thrust. The obvious use is for applications generating high thrusts, higher than can be readily controlled by a normal size thrust bearing and balance drum. An evaluation of the cross leakage between the two discharge nozzles must be made and compared to the balance drum leakage to determine the desirability of the "back to back." It can be combined with the sidestream modes, discussed in the next paragraph, to possibly help sway a close evaluation. In some rare cases, this design has been used for two different services. Unfortunately, it is difficult to totally isolate the two streams because of the potential cross leakage. In cases where the two services may have a common source, or the mixing of the streams does not cause a problem, it is possible to generate savings by using only one compressor case.

A very common compressor design used in the chemical industry, particularly in large refrigeration systems, is the *sidestream* compressor (see Figure 5-15). Gas enters the first impeller and passes through two impellers. As the main stream approaches the third impeller, it is joined by a second stream of gas, mixed, and then sent through the third impeller. The properties of the gas stream are modified at the mixing point, as the sidestream is rarely at the same temperature as the stream from the second impeller. In refrigeration service, this stream is taken from an exchanger where it is flashed to a vapor, resulting in a stream temperature near saturation. As such, the sidestream would act to cool

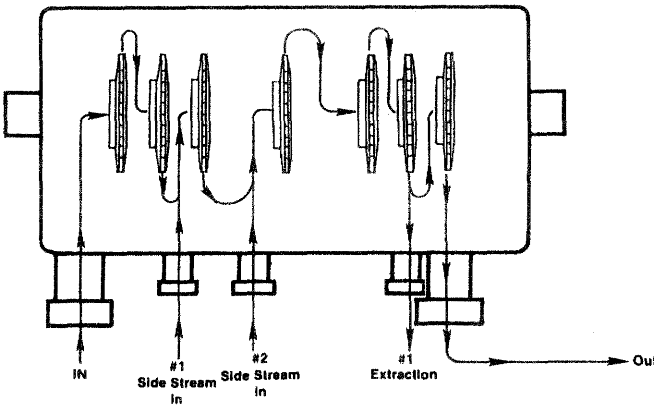


Figure 5-15. Diagram of flow path through a sidestream compressor.

the total stream. The weight flow to the third impeller is the combined weight flow of the two streams.

The second sidestream follows the same logic. To show the flexibility of the arrangement, the last sidestream is indicated as an extraction. This stream could be used where heated gas at less than discharge pressure is required. Using the extraction saves the energy needed to compress this quantity of gas to the full discharge pressure and then throttling for the heating service. One potential application of an extraction stream is for use in a reboiler. The arrangement shown was arbitrarily chosen to illustrate the available options. The total number of sidestream nozzles is limited only by the physical space required to locate them on the case. Three nozzles are not uncommon.

When applications are more complex than can be accommodated by a single-case compressor, multiple cases can be used. The most frequently used is the tandem-driven series flow arrangement using a common driver (see Figure 5-16). A gear unit may be included in the compressor train, either between cases or between the driver and the compressors. The individual compressor cases may take the form of any of the types described before. The maximum number of compressors is generally limited to three. Longer, tandem-driven series-connected compressor trains tend to encounter specific speed problems. In the longer trains, the double-flow arrangement can be useful in permitting more compressors to run at the same speed. At the inlet, where flow is the highest, the gas stream is divided into parallel streams and the volume is reduced by compression to a value within the specific speed capability of a single-flow compressor. The

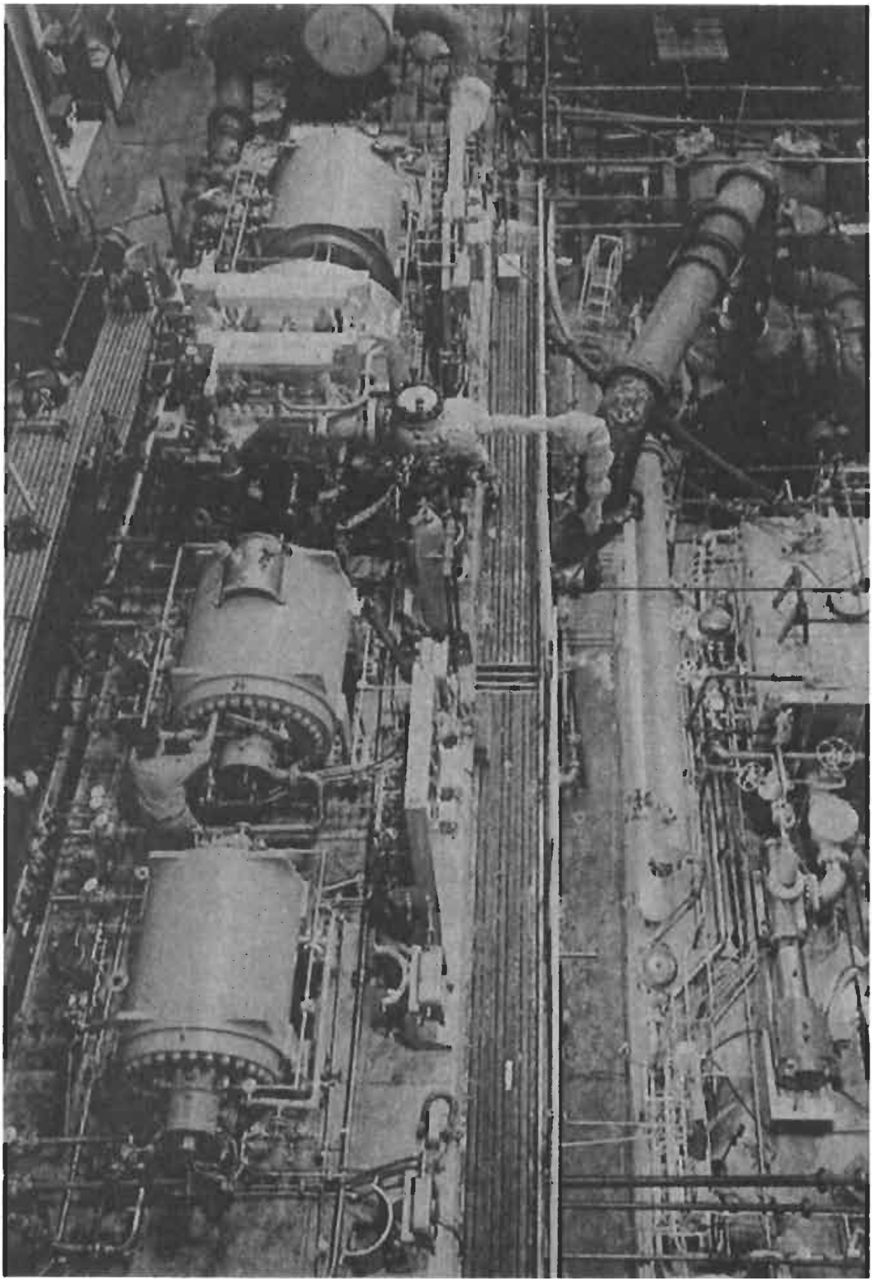


Figure 5-16. A tandem driven multi-body centrifugal compressor train with a steam turbine driver. (*Courtesy of Demag Delaval Turbomachinery Corp.*)

alternative to the double-flow arrangement is the use of a speed increasing gear between compressor bodies to permit the flow matching of downstream stages. This is one case where the double-flow compressor should be considered first. When longer trains are needed, the cases are grouped with several individual drivers, maintaining the series flow concept. One installation that can be recalled used nine individual cases, separately driven and series connected, for a very high pressure air application.

Drive Methods

Historically, the most popular driver for the centrifugal compressor has been the steam turbine. Steam turbines can readily be speed matched to the compressor. Prior to the upsurge in energy costs, reliability, simplicity, and operational convenience were the primary factors in driver selection. The steam turbine, with its ability to operate over a relatively wide speed range, was ideal for the centrifugal compressor, which could be matched to the process load by speed modulation.

With the advent of energy as a more significant consideration in driver selection, the electric motor received a higher degree of attention. While motors were probably second to the steam turbine in general industry usage, the limitations imposed by a constant speed driver tended to discourage their use in process plants. But because fossil fuel can be more efficiently converted to electricity in large central generating stations, the cost of electrical energy for motors became such that they began to displace the more convenient steam turbines. Local steam generation cannot be accomplished at a competitive energy cost in many instances. While large electric drivers using variable frequency conversion to provide for variable speed are relatively new, they provide an alternative to the steam turbine. Two primary factors that have prevented universal acceptance of the variable frequency system are cost and experience. As more units are furnished, and with the passing of time, the negative factors will undoubtedly begin to diminish.

Electric motors, whether speed controlled or not, are either *induction* or *synchronous* in design. Size and plant electric system requirements set the parameters for motor selection. Synchronous motors normally receive consideration only for the larger drives, with the individual plant setting the minimum size at which the synchronous machine is used. Regardless of which motor type is selected, a speed increasing gear will be needed, because motor speed is rarely high enough to match the necessary centrifugal compressor speed.

As an alternate to the drivers mentioned, a gas turbine may be selected as the driver. If exhaust heat recovery or regeneration is used, the efficiency of the gas turbine is quite attractive. Unfortunately, the gas turbine is expensive and in some cases has demonstrated high maintenance cost. It should be understood that gas turbines are relatively standardized even though they cover a wide range of power and speed. They are not custom engineered to the specific application for a power and speed as is customary with steam turbines. In many applications, a speed matching gear must be included, which adds the complication of another piece of equipment, subsequently higher capital cost, and potentially decreased reliability. This gear also inherently has a high pitch-line velocity making for one of the more difficult applications. Despite some of the hurdles just mentioned, the gas turbine is widely used in offshore installations because of its superior power-to-weight ratio over other drivers. It is quite popular for use in remote locations where the package concept minimizes the need for support equipment. As an example, the north slope of Alaska is estimated to have in excess of 1.5 million horsepower in gas turbine powered compressors.

The remaining driver is the *gas expander*, which can only be considered if the process stream has the potential for energy recovery. The expander can be either *cryogenic* or *hot gas* in design depending on the application. Normally the cryogenic expanders are relatively small in size and may be integral with the compressor. These are relatively special purpose and do not have a wide range of application. The hot-gas expander tends to be a larger machine and makes an excellent driver in that it can be speed matched to the compressor and may have variable speed capability. The expander must operate at high temperatures to have sufficient energy for a reasonable output power level. The high temperature does make the supply piping design somewhat complex and also makes the cost of the expander higher than a comparably sized steam turbine. Alignment maintenance is more difficult than with other drivers. It would seem fairly obvious that the economic return of this driver would have to be quite favorable to entice someone to consider it. There are numerous successful installations using the expander, so it is a viable alternative to consider under proper circumstances.

Performance

Compression Cycle

Figure 5-17 is a section of a typical multistage compressor, which should aid the reader in following the flow path through the machine.

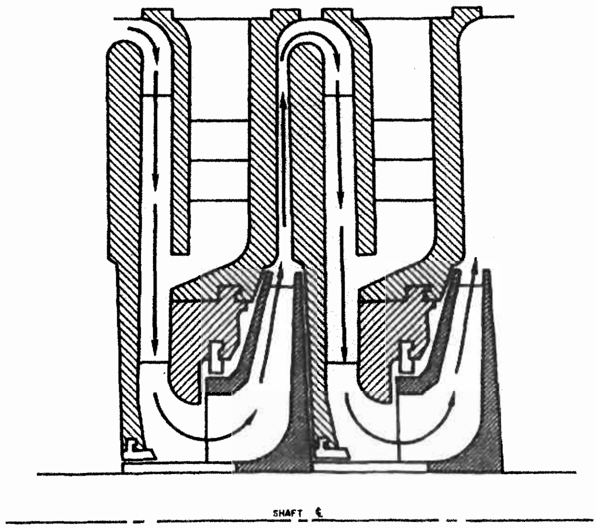


Figure 5-17. Flow path through typical stages on a multistage unit. ***(Courtesy of Elliott Company)***

Gas enters the impeller from one of several sources. In the case of the first impeller of a multistage, the flow has moved through an inlet nozzle and is collected in a plenum from which it is then directed into the first impeller. Another possible path occurs when the flow has passed through one or more stages and approaches the impeller through a channel referred to as a return passage. In the return passage, the flow stream passes through a set of vanes. The vanes are called *straightener vanes,* if the flow is directed axially at the impeller entrance (eye), or *guide vanes,* if the flow is modified by the addition of prerotation. The final possible path occurs when the flow comes into the compressor from a sidestream nozzle. This stream is directed into the flow stream to mix and be directed into the impeller eye using one of two alternative methods as shown on Figure 5-18. One method is by way of a blank section between the stages where the stream mixing point is immediately ahead of the impeller inlet. This method is used if the sidestream flow is large in comparison to the through flow. The alternative is used when the flow is small compared to the through flow, and consists of injecting the flow into the return passage from the previous stage. The latter has better mixing, and takes less axial space, but has a higher pressure drop. For the former, the opposite is true. It has a lower pressure drop, but exhibits somewhat poorer mixing and uses more axial space, normally at least a

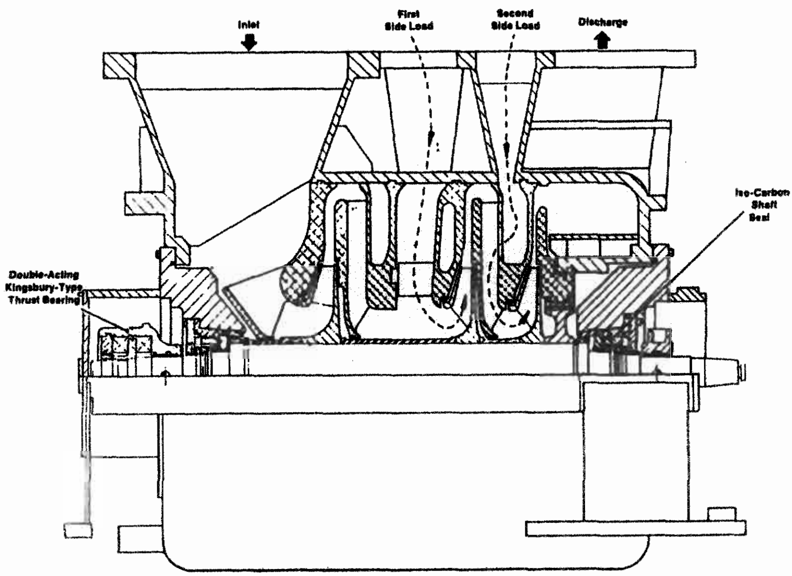


Figure 5-18. Two methods of directing sidestream flows into through flows. (*Courtesy of Elliott Company*)

full-stage pitch in length. A *stage pitch* is defined as the axial distance measured from the entrance of one impeller to the same location on the following impeller. Stage pitch may be a constant, as on low-volume ratio staging, or variable, as may be found in higher-volume ratio stages. The variable stage pitch is commonly used on higher flow coefficient stages using the 3D impeller designs. The importance of physical length will become apparent as the entire compressor is explored, but at this point, it will suffice to say that there never seems to be enough.

Generally, there are no vanes in the inlet of an axial entry compressor (see Figure 5-19). Normally there is no more than the plenum divider vane in the inlet section of the typical multistage compressor, although there are designs that use vanes in this area. These are externally movable and are used to provide flow control for constant speed machines. The use of these vanes will be explored further in the section on capacity control.

After the flow has been introduced into the compressor and has been acted on by one or more stages, it must be extracted. Because there is a relatively large amount of velocity head available in the stream, care must be used when designing the discharge section to keep the head loss low and maintain overall efficiency. The flow from the last stage is gath-

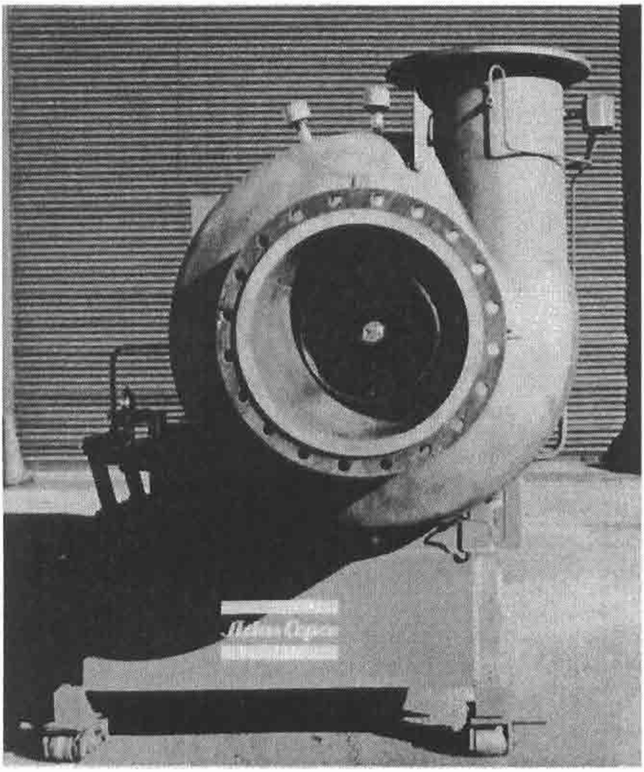


Figure 5-19. The impeller blades can be seen in this view through the inlet of a single-stage compressor. (*Courtesy of Atlas Copco Comptec, Inc.*)

ered in some form of collector, normally a scroll, in an effort to convert as much of the remaining velocity head as possible into pressure. With intermediate extraction, or for some of the in-out designs, a compromise must be made, reducing large passages to preserve axial length.

Having gotten the flow in and out of the machine, a closer examination of just how the compression takes place is needed. An important concept to maintain throughout the following discussion is that all work done to the gas must be done by the active element, the impeller. The stationary element is passive, that is, it cannot contribute any additional energy to the stage. It can only convert the energy and unfortunately contribute to the losses. Figure 5-20 is a schematic diagram of an impeller and the basic inlet and outlet flow vector triangles.

The impeller will be covered in detail in the following sections; therefore, a brief review of the various impeller components is in order. The

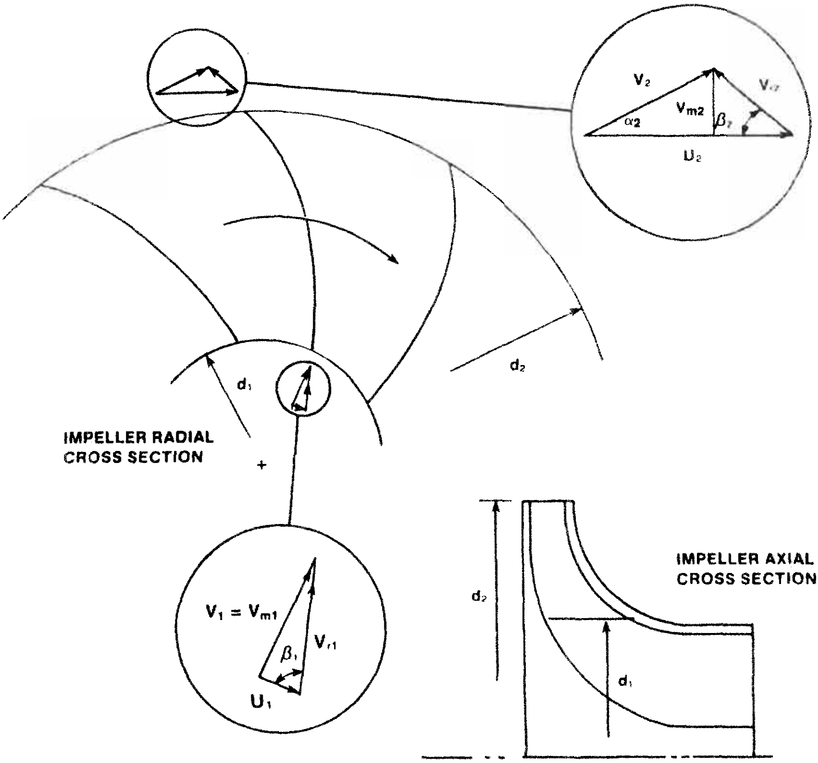


Figure 5-20. Impeller inlet and outlet flow vector triangles.

impeller consists of a set of vanes radially oriented on a hub. The vanes are enclosed either by a rotating or stationary front and rear shroud. If both front and rear shroud are stationary, the impeller is referred to as an *open impeller.* If the rear shroud is attached to the vanes and rotates as a part of the impeller assembly, it is referred to as *semi-open.* If the front shroud is also attached to the vanes and rotates with the assembly, it is referred to as a *closed impeller.* The vanes may be forward curved, radial, or backward curved, as shown diagramatically in Figure 5-21. Forward curved vanes are normally only used in fans or blowers, and rarely, if ever, used in centrifugal compressors.

Figure 5-21 includes an outlet velocity vector triangle for the various vane shapes. Figure 5-20 shows a backward curved impeller that includes the inlet and outlet velocity vector triangle. Because most of the compressors used in process applications are either backward curved or radial, only these two types will be covered in detail.

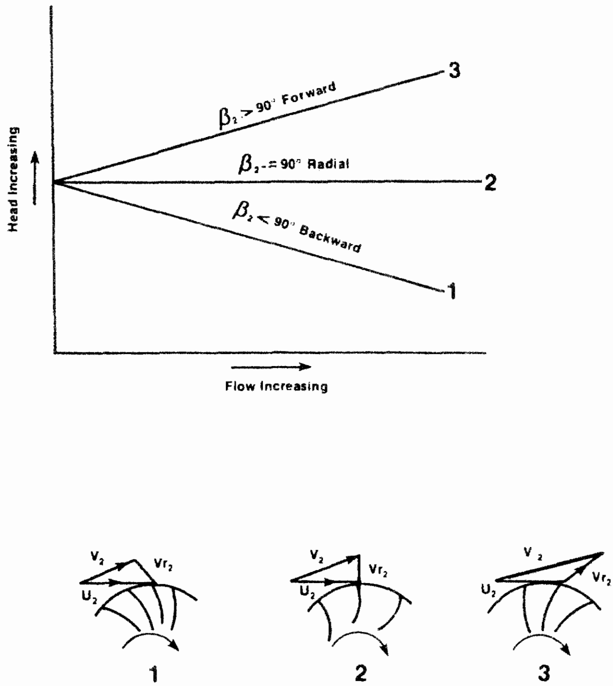


Figure 5-21. Diagram depicting backwards, radial and forward curved blades.

Vector Triangles

Gas enters the impeller vanes at the diameter d_1. The absolute gas velocity approaching the vanes is V_1. As shown in Figure 5-20, the gas approaches the vane in a radial direction after entering the impeller in an axial direction and makes the turn to a radial direction inside the impeller. The vane leading edge velocity is represented by the velocity vector u_1. The net velocity is the relative velocity V_{r1}. It should be noted for this basic example that the relative velocity vector aligns itself with the vane angle β_1, resulting in zero incidence. In this idealized case, the *meridional flow vector* V_{m1} is aligned with and equal to the absolute velocity. After passing between the vanes, the gas exits the impeller at the diameter d_2. The velocity of the gas just prior to leaving the impeller is the relative velocity V_{r2} and leaves at the vane angle β_2 in the idealized example. By the addition of the impeller tip velocity vector u_2, the absolute leaving velocity V_2 is generated. The angle of the absolute flow vector is α_2. This is the velocity and direction which the gas assumes as it leaves the impeller and enters the diffuser. The meridional velocity V_{m2} is shown by the radial vector passing through the apex of the outlet velocity triangle. If the vane was radial, rather than backward leaning, $\beta_2 = 90°$, the relative velocity and the meridional velocity would be equal and aligned.

Slip

In real world application, the gas leaving the impeller will not follow the vane exit angle. The deviation from the geometric angle is referred to as *slip*. The leaving angle will be referred to as the gas angle β'_2. Figure 5-22 shows the discharge velocity vector triangle, including the effect of slip. The terms on the ideal triangle are the same as those used in Figure 5-20. Superimposed over the ideal triangle is the velocity triangle, including the effect of slip. Note that the terms are indicated with the prime (′) symbol. While there are numerous papers written on the subject of slip, none seem to present a complete answer. One of the better papers, which summarizes the field and brings the subject into focus, is the one by Wiesner [7]. In this book, for the purpose of understanding the workings of the centrifugal compressor, the Stodola slip equation will be used. It is probably one of the oldest and has been used in practical design prior to the advent of some of the more sophisticated methods available now. Returning to the triangle under discussion, the gas angle, β'_2, is always less than the geometric angle, β_2. In Figure 5-22, projections are

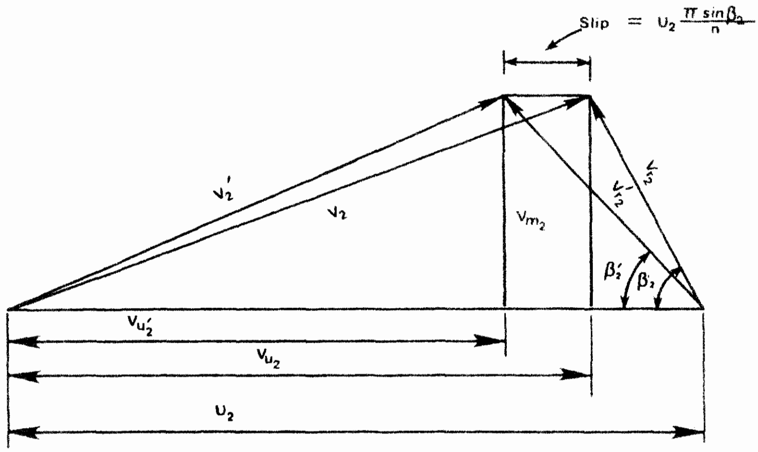


Figure 5-22. Discharge velocity vector triangle showing the effect of slip.

made onto the tip velocity vector from the absolute gas vectors, V_2 and V'_2. These are labeled as V_{u2} and V'_{u2}, respectively, and have the designation of tangential component of the absolute velocity. From these vectors, some simple relationships can be presented that will give a reasonable explanation of how the centrifugal compressor geometry relates to its ability to compress gas. The ideal work input coefficient, ζ_i, is given by the following expression:

$$\zeta_i = \frac{V_{u2}}{u_2} \tag{5.1}$$

where

V_{u2} = tangential component of the absolute velocity
u_2 = impeller tip velocity

The ideal head input to the stage is given by

$$H_{in\ ideal} = (1/g)\zeta_i\, u_2^2 \tag{5.2}$$

The Stodola slip factor is defined as

$$\text{Slip} = u_2 \frac{\pi \sin\beta_2}{n_v} \tag{5.3}$$

where

β_2 = geometric vane exit angle
n_v = number of vanes in the impeller

The slip factor SF follows.

$$SF = \frac{V'_{u2}}{V_{u2}} \tag{5.4}$$

Reference is made to Figure 5-22, where

$$V'_{u2} = V_{u2} - \text{slip} \tag{5.5}$$

Substituting into Equation 5.4 yields the following slip factor equation:

$$SF = 1 - \frac{u_2}{V_{u2}}\left(\frac{\pi \sin\beta_2}{n_v}\right) \tag{5.6}$$

The actual work input coefficient, ζ, is written by taking the ideal work input coefficient, Equation 5.1, and modifying by the addition of the slip factor, SF. The geometric relationship of the Stodola slip function is shown in Figure 5-23.

$$\zeta = \frac{V_{u2}}{u_2}(SF) \tag{5.7}$$

By replacing the ideal work input coefficient with actual work input coefficient, the actual head input can be written as

$$H_{in} = (1/g)\zeta u_2^2 \tag{5.8}$$

If the head coefficient is written as

$$\mu = \eta\zeta \tag{5.9}$$

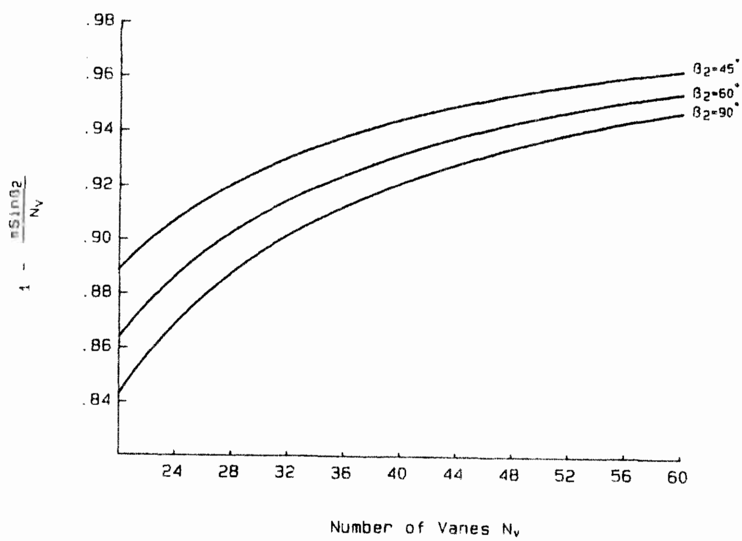


Figure 5-23. Geometric relationship of Stodola slip function.

where

η = stage efficiency, then

$$H_{out} = (1/g)\mu u_2^2 \tag{5.10}$$

For adiabatic head, the head coefficient is defined as μ_a and Equation 2.70 is recalled. The geometric and the thermodynamic head relationships for a stage may be equated.

$$H_a = \frac{\mu_a u_2^2}{g} = Z_{avg}\ RT_1 \frac{k}{k-1}(r_p^{\frac{k-1}{k}} - 1) \tag{5.11}$$

Similarly, for polytropic head, the head coefficient is defined as μ_p and Equation 2.73 is recalled, the geometric and thermodynamic head relationships, on a per-stage basis, may be equated as above.

$$H_p = \frac{\mu_p u_2^2}{g} = Z_{avg}\ RT_1 \frac{n}{n-1}(r_p^{\frac{n-1}{n}} - 1) \tag{5.12}$$

In the previous paragraphs, the term *specific speed* has been used. This is a generalized turbomachinery term used quite successfully with pumps and to some extent with turbines. It can be used with turbocompressors to help delimit the various kinds of machines. It is also used as a general term to describe the need for a correction on multistage machines when the wheel geometry at the current speed will no longer support a reasonable efficiency. For compressors, specific speed is paired with specific diameter to include the geometric factors. In centrifugal compressors, attempts have been made to correlate efficiency directly to these parameters. Most designers feel the relationships, while satisfactory to set bounds, are not adequate for describing impeller efficiency with good resolution. Definitions for specific speed, N_s, and specific diameter, D_s, are

$$N_s = \frac{NQ_1^{1/2}}{H_a^{3/4}} \tag{5.13}$$

$$D_s = \frac{DH_a^{1/4}}{Q_1^{1/2}} \tag{5.14}$$

Reaction

The outlet vane angle for the normal centrifugal compressor varies from radial to a backward leaning angle. An ideal vector tip triangle, with no slip, is shown in Figure 5-24. Three angles are illustrated to show the effect of varying the vane outlet angle.

Reaction is defined as the ratio of the static head converted in the impeller to the total head produced by the stage. Restating in a more philosophical sense, the object of the compressor stage is to increase the pressure of the gas stream, and reaction gives the relationship of the division of effort between the impeller and the diffuser.

Ideal reaction, R_i, is defined as

$$R_i = \frac{2 + \cot \beta_2}{4} \tag{5.15}$$

One of the practical aspects of reaction is that for a well-proportioned stage, the higher the reaction, the higher the efficiency. Again, using a philosophical approach to explain, for a given stage the impeller is more efficient than the diffuser. This is particularly true for the typical process compressor that uses a simple vaneless diffuser. If the radial vane impeller is used for the reference, it will have an ideal reaction of 50%, as calculated using Equation 5.15. Because the static head conversion is evenly divided between the impeller and the diffuser, the net stage effi-

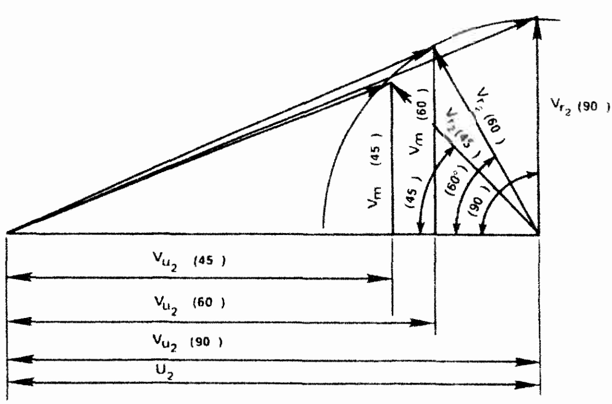


Figure 5-24. Vector tip triangle without slip, showing the effect of different exit vane angles.

ciency is the numeric average of the impeller efficiency and the diffuser efficiency. Figure 5-25 shows that as the vane angle decreases, the reaction increases. If the efficiency is evaluated for the lower angle, the net stage efficiency is now the weighted average of the two component individual efficiencies, with the higher impeller efficiency contributing a greater influence. A numeric example may help to illustrate the idea.

Example 5-1

Assume

Impeller efficiency = .90

Diffuser efficiency = .60

Calculate an ideal stage efficiency for a radial and a 45° backward leaning impeller.

For the radial impeller, using Equation 5.15,

$R_i = .50$

$.50 \times .90 = .45$

$.50 \times .60 = .30$

$.45 + .30 = .75$ net stage efficiency

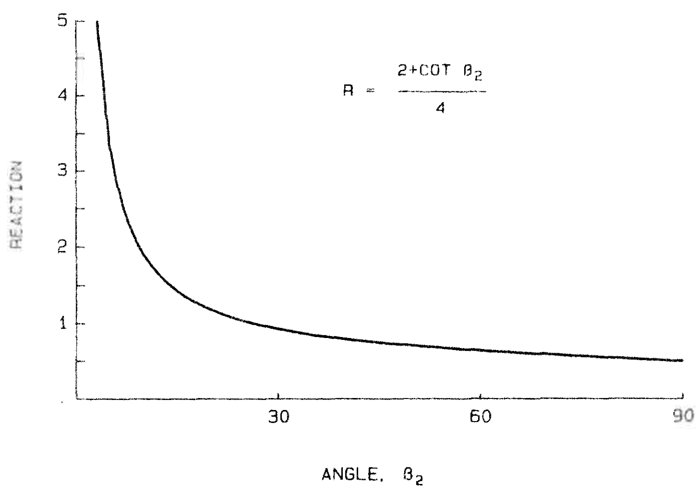


Figure 5-25. Theoretical reaction without slip.

For a 45° backward leaning impeller,

$R_i = .75$

The diffuser then converts $1 - R_i$ or .25

$$.75 \times .90 = .68$$

$$.25 \times .60 = .15$$

$.68 + .15 = .83$ net stage efficiency

The example indicates that an improvement of seven percentage points was achieved by backward leaning the vanes 45°. The obvious question arises. Why not make all impellers high reaction? Maybe this can be put into a good/bad analogy. The good is better efficiency. The bad is a lower head produced by the stage. To see why the head is less, review Figure 5-24. It can be seen that as the outlet angle, β_2, is decreased, the tangential component of the absolute velocity, V_{u2}, is decreased. If Equation 5.1 is recalled, it should be noted that a decrease in V_{u2} will decrease the value of the head input coefficient, ζ_i. By carrying a lower value of ζ_i into Equation 5.9, the head coefficient, μ, is decreased. In Equation 5.10, it is obvious that for a lower μ the output head is decreased. There is some relief in that in Equation 5.9, the stage efficiency η increases to offset the lowered ζ. However, in real life, this is not enough to make up the difference and the output head of a higher reaction stage is indeed lower. There are several effects that influence a commercial design and, again, the designer is faced with trade-offs. Equation 5.10 indicates that increase in the tip velocity u_2 would offset the loss in μ. Impeller stresses and rotor dynamics must also be considered and may act to limit the amount of correction that can be made. Another possibility is using additional stages. A well-proportioned stage is assumed, which brings to light the fact that the high reaction stage tends to use more axial length. This tends to counter the addition of extra stages, especially where the length of the rotor is beginning to cause critical speed problems. Despite the conflicts, changing reaction can sometimes aid the designer in achieving a higher efficiency. Another benefit is a steeper head-capacity curve. Also in some cases, the higher reaction stage seems to perform better where fouling is evident.

Sizing

Many of the steps used in sizing estimates are also useful for checking bids or evaluating existing equipment. In the latter two endeavors, there is one advantage: someone else has established the initial evaluation criteria. When working from a material balance flow sheet as a starting point, it is sometimes difficult to envision what the compressor should look like. Except for the addition of a few rules of thumb, most of the tools needed have already been established. The method outlined is based on the more conventional multistage compressors used in process service. Earlier, integrally geared, as well as direct expander driven compressors, were briefly described. These compressors may also be sized by the method outlined, but because they are tailored for higher head service, modifications to the method regarding the head per stage and the head coefficient are necessary.

To start, convert the flow to values estimated to be the compressor inlet conditions. Initially, the polytropic head equation (Equation 2.73) will be used with n as the polytropic compression exponent. If prior knowledge of the gas indicates a substantial nonlinear tendency, the real gas compression exponent (Equation 2.76) should be substituted. As discussed in Chapter 2, an approximation may be made by using the linear average of the inlet and outlet k values as the exponent or for the determination of the polytropic exponent. If only the inlet value of k is known, don't be too concerned. The calculations will be repeated several times as knowledge of the process for the compression cycle is developed. After selecting the k value, use Equation 2.71 and an estimated stage efficiency of 75% to develop the polytropic compression exponent n.

The molecular weight, inlet temperature, and inlet pressure are combined with the compressibility and discharge pressure in Equation 2.73 to estimate the polytropic head. The average of inlet and outlet compressibility should be used, using the polytropic discharge temperature calculated by the following equation to evaluate the discharge compressibility.

$$T_2 = T_1 \, r_p^{\frac{n-1}{n}} \tag{5.16}$$

where

T_2 = absolute discharge temperature of the uncooled section
T_1 = absolute inlet temperature of the uncooled section

To determine the number of stages, using the impeller and diffuser defined as the stage, assume 10,000 ft-lb/lb of head per stage. This value can be used if the molecular weight is in the range of 28 to 30. For other molecular weights, this initial value must be modified. As a rule of thumb, lower the head per stage by 100 ft-lb/lb for each unit increase in molecular weight. Conversely, raise the allowable head per stage 200 ft lb/lb for a unit decrease in molecular weight. The rule of thumb gives the best results for a molecular weight range of 2 through 70. Because this sizing procedure is being used only to establish the rough size of the compressor, the upper range may be extended with some loss in accuracy.

Once the head per stage has been established, the number of stages can be estimated by taking the total head, as calculated by the head equation, and dividing by the head per stage value. A fraction is usually rounded to the next whole number. However, if the fraction is less than .2, it may be dropped. The stage number should be used to calculate a new head value per stage. This method assumes an uncooled or no sidestream compressor. If either of the two are involved, the uncooled sections can be estimated, taken one at a time. Assumptions for between-section pressure drop or sidestream mixing can be added to the calculation as appropriate to account for all facets of the process. When all calculations are completed, the compressor sections can be arranged to form a complete unit.

Before proceeding, a few limits need to be considered. The temperature, if not limited by any other consideration, should not exceed 475°F. This limitation is arbitrary, as centrifugals may be built to higher limits, but the estimator is cautioned not to venture too far into this region without additional considerations. The number of stages per casing should not exceed 8 for rotor dynamics considerations. Also, knowledge of auxiliary nozzle stage pitch would be needed to evaluate exactly how far to venture in this direction. Vendor literature advertises the availability of as many as ten stages; however, an estimate should never go to the edge without a background of considerable experience. These limits can also be used to evaluate proposals and help to determine a series of questions for the vendor skirting the upper limits.

The next step begins by assuming a head coefficient equal to .48. Equation 5.12 can be used to calculate the tip speed, u_2. Figure 5-26 can be used to get an impeller diameter estimate from the inlet volume calculated earlier. The diagonal line on the diagram marks the right extremity of each impeller's flow range to guide the user in making the first selection. The tip speed and diameter can be used to calculate an approximate speed, N, by

$$N = \frac{u_2}{\pi d_2} \tag{5.17}$$

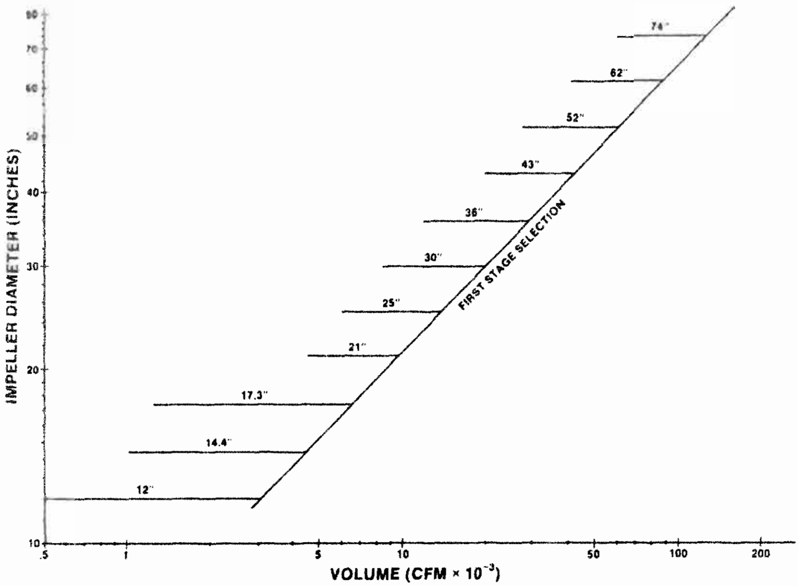


Figure 5-26. Estimation of impeller diameter using inlet volume.

where

d_2 = impeller outside diameter

To summarize the sizing to this point: the inlet volume, an overall head, number of stages, head per stage, impeller tip speed, and impeller diameter have been established. The one parameter of interest still missing is the efficiency. To obtain an estimate of efficiency without empirical data, a generalized form may be used. As in the previous chapters, where estimates were involved, the data presented is just one way to approach the problem, and any other reasonable source such as specific vendor data may be used. To use the generalized curve, Figure 5-26, the volume for the first and last stage must be developed. The volume for the first stage is the inlet volume. The volume for the last stage, Q_{ls}, can be estimated by

$$Q_{ls} = \frac{Q_{in}}{\left(r_p^{\,1-\frac{1}{z}}\right)^{\frac{1}{n}}} \tag{5.18}$$

where

Q_{in} = inlet volume
r_p = pressure ratio for an uncooled section
z = number of stages in the uncooled section

Use the inlet and the last stage volume for the uncooled section and use the following equation to calculate the inlet flow coefficient δ.

$$\delta = 700 \frac{Q_i}{Nd_2^3} \tag{5.19}$$

where

Q_i = volumetric flow, ft^3/min
N = rotational speed, rpm
d_2 = impeller diameter, in.

Note: This equation is not in the primitive form. While δ is basically dimensionless, the constant 700 is not easily derived; therefore, units were assigned.

The value for the first-stage flow coefficient should not exceed .1 for a 2D type impeller and for a 3D design, the upper value can be as high as .15. The value for the last stage should be no less than .01. If the flow coefficients should fall outside these limits, another impeller diameter should be selected. It may be necessary to interpolate to obtain a reasonable diameter from Figure 5-26. This can be done because this is an estimate and not bound to an arbitrary line of compressor frames. The diagram was set up to give the user an idea of how a compressor line might be organized. A vendor may quote values outside the guidelines due to the constraints of his available frame sizes. For estimates, values as close as possible to the given guidelines are recommended. At the time of a proposal, the benefit of stages beyond either extreme value of flow coefficient can be evaluated. It should be noted at this point that not all vendors report their flow coefficients on the same basis. If necessary, the parameters for flow coefficient should be obtained to permit evaluation with Equation 5.19. An average of efficiency can be calculated from two efficiencies selected from Figure 5-27. The figure includes efficiency values for 2D and 3D impeller designs. While it would appear obvious that only 3D impellers should be used, there is a caveat. Generally, 3D impellers require more space, that is, the axial stage spacing (stage pitch)

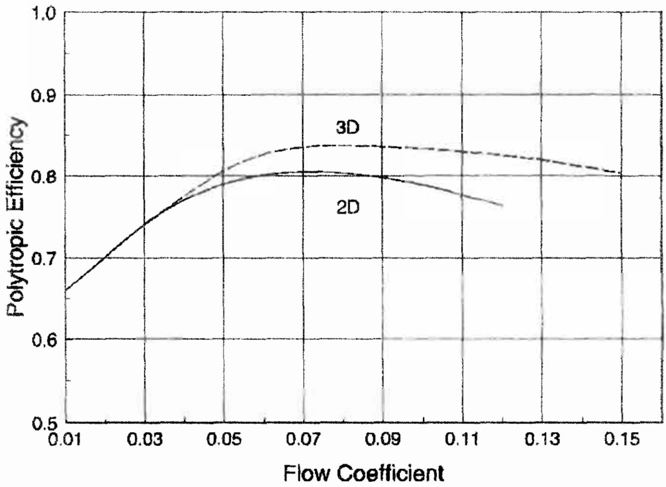


Figure 5-27. Centrifugal stage efficiency vs. flow coefficienct for 2D and 3D blading.

is longer. This will result in a longer compressor, which makes for possible rotor dynamic problems and does also increase cost. Also, it should be pointed out that the increase in efficiency begins above flow coefficients of .04. The increase in stage pitch can vary from approximately 1.1 to 1.3 times a 2D stage pitch with the values increasing with increased flow coefficient. For the 2D impeller, it should be noted that the peak efficiency occurs at a flow coefficient of approximately .07. The 3D impeller peak efficiency value curve is broader and occurs in the range of from .07 to values as high as 1.3.

At this point, after a first pass through the calculation, a new polytropic exponent should be calculated. All values calculated to this point should be rechecked to see if original estimates were reasonable. If the deviation appears significant, a second pass should be made to improve the accuracy. Equation 2.78 can be used to calculate the power for the uncooled section. For an estimate, use a value of 1% for the mechanical losses.

If time permits and a more accurate estimate is desired, particularly if the compressor is intercooled or has sidestreams, the velocity head losses through the nozzles can be estimated using the values from Table 2-2. This is possible where the nozzle sizes are available or can readily be estimated. When coolers are involved, the drop through the cooler should be included. Subtract the pressure drop from the inlet pressure (of the stage following the element) and recalculate a modified pressure ratio for the section. The cooler pressure drop can be approximated by using 2%

of the absolute pressure at the entrance to the cooler. Because the percentage gives unrealistic values at the lower pressures, a lower limit of 2 psi should be used. Compressors with in-out nozzles used to take gas from the compressor for external cooling and return to the compressor can experience some temperature crossover in the internal sections of the machine. Unless the design has specifically provided for a heat barrier, heating of the return gas can be expected. For a first estimate, a 10°F rise should be used. Balance pistons will be described in the mechanical section of this chapter. Briefly, the balance piston contributes a parasitic loss to the compressor not accounted for in the stage efficiency. The weight flow passing from the balance piston area, normally the discharge, and entering the suction must be added to the flow entering the first stage or the stage receiving the balance piston flow. Unfortunately, the flow is not the only problem, as the return flow also acts to heat the inlet gas. For discharge pressures of 150 psia or less, a value of 1% can be used. For pressures higher than 150 psia but under 1,000 psia, a value of 2% is a reasonable starting point. An equation for the heating is

$$t_w = \frac{t_i + t_d BP}{1 + BP} \tag{5.20}$$

where

t_w = impeller inlet temperature
t_i = nozzle inlet temperature
t_d = temperature at the balance piston
BP = balance piston leakage fraction

The relationships are given to help the user size a compressor from scratch. The same relationships can be used in the bid evaluation process. The vendor-provided geometry and performance values can be compared to the original sizing, which should have been performed prior to going out for the bid. The vendor's results can be evaluated using some of the rules of thumb or guidelines provided. Any deviations can be used as a focus for additional discussions. Also, some insight can be gained into the vendor's sizing techniques, particularly the way the vendor trims out a selection. Incremental wheel sizing is fairly universal. Some vendors also offer fixed guide vane sections as part of a stage to aid in the achievement of a particular performance specification.

Example 5-2

Using the results of Example 2-2, size a centrifugal air compressor using the sizing procedure. A summary of the results is:

$Q1 = 6{,}171$ cfm inlet volume

$w_m = 437.5$ lbs/min

$mw = 28.46$ molecular weight

$P_1 = 14.7$ psia inlet pressure

$t_1 = 90.0°F$ inlet temperature

$T_1 = 550°R$ absolute inlet temperature

$Rm = 54.29$ specific gas constant

Add the following conditions to complete the application:

$k = 1.395$ isentropic exponent for air

$P_2 = 40$ psia discharge pressure

Assumed polytropic efficiency

$\eta_p = .75$

Step 1. Calculate the polytropic exponent using Equation 2.71.

$$\frac{n-1}{n} = \frac{1.395-1}{1.395} \times \frac{1}{.75}$$

$$\frac{n-1}{n} = .378$$

$$\frac{n}{n-1} = 2.646$$

$$n = 1.608$$

From Equation 2.64,

$r_p = 40/14.7$

$r_p = 2.721$ pressure ratio

Step 2. Calculate the total required polytropic head using Equation 2.73, assuming a value for $Z_{avg} = 1$:

$H_p = 1 \times 54.29 \times 550 \times 2.646\ (2.721^{.378} - 1)$

$H_p = 36{,}338.4$ ft-lb/lb overall polytropic head

Step 3. Determine the number of stages, z, required using the recommended 10,000 ft-lb/lb head per stage.

$z = 36{,}338.4/10{,}000$

$z = 3.63$ stages, round off to 4

Calculate a new head per stage using four stages:

$H_p = 36{,}338.4/4$

$H_p = 9{,}085$ ft-lb/lb

Step 4. Use the geometric form of Equation 5.12 to calculate a tip speed to produce the head per stage just calculated. Also, use the recommended head coefficient $\mu = .48$ in the equation.

$u_2 = (9{,}085 \times 32.2/.48)^{.5}$

$u_2 = 780.7$ fps impeller tip speed

From Figure 5-26 and the inlet volume, select an initial impeller diameter.

$d_2 = 17.3$ inches impeller diameter

Use Equation 5.17 to calculate the initial speed, N, and use the conversion factors of 12 in/ft and 60 secs/min.

$$N = \frac{60 \times 12 \times 780.7}{\pi \times 17.3}$$

$$= 10,342 \text{ rpm shaft speed}$$

Step 5. The volume into the last impeller, in this example stage 4 inlet, is calculated using the Equation 5.18.

$$Q_{ls} = \frac{6,171}{(2.721^{1-1/4})^{1/1.608}}$$

$$Q_{ls} = 3,869 \text{ cfm volume at last stage}$$

To obtain an efficiency for the geometry selected, the value of the flow coefficient must be calculated using Equation 5.19 for the first inlet and the last stage flow.

$$\delta = (700 \times 6,171)/(10,342 \times 17.3^3)$$

$$\delta = .081 \text{ first stage flow coefficient}$$

$$\delta = (700 \times 3,869)/(10,342 \times 17.3^3)$$

$$\delta = .051 \text{ last stage flow coefficient}$$

Using the flow coefficients just calculated and Figure 5-26, the corresponding efficiencies may be looked up:

$$\delta = .081, \eta_p = .79$$

$$\delta = .051, \eta_p = .79$$

The average is rather easy to calculate.

$$\eta_p = .79 \text{ the average efficiency}$$

Step 6. Recalculate the polytropic exponent using Equation 2.71 and the new efficiency.

$$\frac{n-1}{n} = \frac{1.395-1}{1.395} \times \frac{1}{.79}$$

$$\frac{n-1}{n} = .359$$

$$\frac{n}{n-1} = 2.787$$

$$n = 1.559$$

Using the new polytropic exponent, calculate the discharge temperature using Equation 5.16.

$$T_2 = 550 \times 2.721^{.359}$$

$$T_2 = 787.8°R$$

$$t_2 = 787.8 - 460$$

$t_2 = 327.8°F$ discharge temperature

Calculate the power required using Equation 2.78 and the recommended 1% for mechanical losses.

$$W_p = \frac{437.5 \times 36{,}338.4}{33{,}000 \times .79} + .01W_p$$

$$W_p = 609.8 + 6.1$$

$W_p = 615.9$ hp total for the compressor

Note, the polytropic head was not recalculated as the change in efficiency only made an approximate 1% difference in original value and is well within the accuracy of an estimate.

Example 5-3

For a sample problem that will include some of the additional losses that are normally encountered in an actual situation, size a compressor to the following given conditions for a hydrocarbon gas:

$mw = 53.0$

$k_1 = 1.23$

$Z_1 = 0.97$

$t_1 = 85°F$

$P_1 = 40$ psia

$P_2 = 120$ psia

$w = 2{,}050$ lb/min

Step 1. Use Equation 2.5 to calculate the specific gas constant.

$$R = 1{,}545/53$$

$$R = 29.15$$

Step 2. Convert the inlet temperature to absolute.

$$T_1 = 85 + 460$$

$$T_1 = 545°R$$

Step 3. Calculate the polytropic exponent using Equation 2.71. Assume an efficiency of $\eta_p = .75$. Use as $k_{avg} = k_1 = 1.23$.

$$\frac{n-1}{n} = \frac{1.23-1}{1.23} \times \frac{1}{.75}$$

$$\frac{n-1}{n} = .249$$

$$\frac{n}{n-1} = 4.011$$

$$n = 1.332$$

Step 4. From Equation 2.64,

$r_p = 120/40$

$r_p = 3.0$ pressure ratio

Step 5. Calculate the estimated discharge temperature using Equation 5.14.

$T_2 = 545 \times 3^{.249}$

$T_2 = 716.7°R$ absolute discharge temperature estimate

Convert to °F:

$t_2 = 716.7 - 460$

$t_2 = 256.7°F$

Correct for the balance piston leakage using 1% for pressures of 150 psia and under. The weight flow into the impeller must be increased to account for the leakage.

$w = 1.01 \times 2{,}050$

$w = 2{,}070.5$ lb/min net flow to the impeller.

The temperature at the entrance to the impeller is increased because of the hot leakage. Calculate the corrected impeller inlet temperature using Equation 5.20.

$$t_w = \frac{85 + 256.7(.01)}{1.01}$$

$t_w = 86.7°F$ corrected impeller inlet temperature

Convert to absolute:

$T_w = 86.7 + 460$

$T_w = 546.7°R$

Step 6. Substitute into Equation 2.10 and using 144 in^2/ft^2.

$$Q_1 = \frac{.97 \times 29.15 \times 546.7}{40 \times 144} \times 2{,}070.5$$

$Q_1 = 5{,}557$ cfm inlet flow to the impeller

Step 7. Calculate the total required polytropic head using Equation 2.73, assuming the average value of $Z_{avg} = .97$.

$$H_p = 0.97 \times 29.15 \times 546.7 \times 4.011(3^{.249} - 1)$$

$H_p = 19{,}508$ ft-lb/lb total polytropic head required

Step 8. Determine the number of stages required using the modified rule of thumb on head per stage, H_{stg}.

$$H_{stg} = 10{,}000 - ((53 - 29)100)$$

$H_{stg} = 7{,}600.$ ft-lb/lb

$$z = \frac{19{,}508}{7{,}600}$$

$z = 2.57$ stages, round off to 3

Calculate a new head per stage using three stages.

$$H_p = 19{,}508/3$$

$H_p = 6{,}502.7$ ft-lb/lb head per stage

Step 9. Use the geometric portion of Equation 5.12 to calculate a required tip speed, which will produce the head per stage. Use the recommended head coefficient $\mu = .48$ for the calculation.

$$u_2 = (6{,}502.7 \times 32.2/.48)^{1/2}$$

$u_2 = 660.5$ fps impeller tip speed

Step 10. From Figure 5-26 and the inlet volume, select an initial impeller diameter.

$d_2 = 17.3$ in. initial impeller diameter

Use Equation 5.17 to calculate the initial speed, N.

$$N = \frac{60 \times 12 \times 660.5}{\pi \times 17.3}$$

$N = 8{,}750$ rpm compressor shaft speed

Step 11. The volume into the last impeller is calculated with the use of Equation 5.18.

$$Q_{ls} = \frac{5,557}{(3^{1-1/3})^{1/1.332}}$$

$Q_{ls} = 3,206$ cfm volume into last stage

With the volumes just calculated, calculate the inlet flow coefficient for each of the two stages using Equation 5.19.

$\delta = (700 \times 5,557)/(8,750 \times 17.3)^3$

$\delta = .086$ first stage flow coefficient

$\delta = (700 \times 3,206)/(8,750 \times 17.3)^3$

$\delta = .050$ last stage flow coefficient

Look up the efficiencies for the two flow coefficients on Figure 5-27.

$\eta_p = .79$ first stage efficiency

$\eta_p = .79$ last stage efficiency

$\eta_p = .79$ average of the two efficiencies

Step 12. Recalculate the polytropic exponent using Equation 2.71 and the new average efficiency.

$$\frac{n-1}{n} = \frac{1.23-1}{1.23} \times \frac{1}{.793}$$

$$\frac{n-1}{n} = .236$$

$$\frac{n}{n-1} = 4.24$$

With the new polytropic exponent, calculate the discharge temperature by substituting into Equation 5.16.

$T_2 = 546.7 \times 3.236$

$T_2 = 708.5°R$ absolute discharge temperature

$t_2 = 708.5 - 460$

$t_2 = 248.5°F$ discharge temperature

Step 13. Calculate the power required using Equation 2.78, allowing 1% for the mechanical losses. Use the conversion 33,000 ft-lb/min/hp.

$$W_p = \frac{2,070.5 \times 19,508}{33,000 \times .793} + .01W_p$$

$$W_p = 1,543.1 + 15.4$$

$$W_p = 1,558.5 \text{ hp shaft horsepower}$$

There is no need to recalculate the polytropic head for the changed efficiency because the head difference from the original value is negligible. Another item to note is that the horsepower is 1.5% higher than if the balance piston had been neglected. The interesting part is not the value itself, but the fact that the slight temperature addition at the impeller inlet is responsible for .5% of the increase and the remainder is the 1% weight flow increase through the compressor. As the small, but significant "real life" items are included, the actual efficiency is being eroded. If the calculation had been made with only the original weight flow, the equivalent efficiency would prorate to .781.

Example 5-4

This example presents a gas with a temperature limit and is typically found in a halogen mixture. A multi-section compressor is required to accommodate the limit. This example illustrates one approach for the division of work between the sections to achieve a discharge temperature within the specified bound.

$$mw = 69$$

$$k_1 = 1.35$$

$$k_2 = 1.33$$

$$Z_1 = .98$$

$$Z_2 = .96$$

$$t_1 = 80°F$$

$$P_1 = 24 \text{ psia}$$

$$P_2 = 105 \text{ psia}$$

$$w = 3,200 \text{ lbs/min}$$

The temperature, t_2, is limited to a value of 265°F.

Step 1. Use Equation 2.5 to calculate the specific gas constant.

$$R = 1{,}545/69$$

$$R = 22.39$$

Step 2. Convert the inlet temperature to absolute.

$$T_1 = 80 + 460$$

$$T_1 = 540°R$$

Substitute into Equation 2.10 and using the conversion constant of 144 in.^{2}/ft^2, calculate the inlet volume.

$$Q_1 = \frac{.98 \times 22.39 \times 540}{24 \times 144} \times 3200$$

$$Q_1 = 10{,}971 \text{ cfm inlet flow}$$

Step 3. Calculate the overall poltropic exponent using Equation 2.71 and an assumed polytropic efficiency of $\eta_p = .75$.

$$k_{avg} = (1.35 + 1.33)/2$$

$$k_{avg} = 1.34$$

The average was used in evaluating k because the values were not all that different.

$$\frac{n-1}{n} = \frac{1.34-1}{1.34} \times \frac{1}{.75}$$

$$\frac{n-1}{n} = .338$$

$$\frac{n}{n-1} = 2.956$$

$$n = 1.511$$

Step 4. From Equation 2.64,

$r_p = 105/24$

$r_p = 4.375$ overall pressure ratio

Step 5. Calculate the discharge temperature for the total pressure ratio to check against the stated temperature limit, using the assumed efficiency, $\eta p = .75$ and the polytropic exponent. Apply Equation 5.14.

$T_2 = 540 \times 4.375^{.338}$

$T_2 = 889.3°R$

$t_2 = 889.3 - 460$

$t_2 = 429.3°F$ discharge temperature

Since the limit is 265°F and the overall temperature is obviously in excess of this limit, intercooling is required.

Intercooler outlet temperature must be determined. If cooling water at 90°F and an approach temperature of 15°F are assumed, the gas outlet from the cooler returning to the compressor will be 105°F.

If Equation 3.12 is borrowed from the reciprocating compressor chapter and used for an uncooled section, the pressure ratio per section may be calculated assuming an approximate equal-work division. For the first trial, assume the limit of temperature may be achieved in two sections.

$r_p = 4.375^{1/2}$ pressure ratio per section

$r_p = 2.092$

$P_2 = 2.092 \times 24$

$P_2 = 50.2$ psia first section discharge pressure

From the rule of thumb given for estimating intercooler pressure drop, a value of 2 psi is used because it is larger than 2% of the absolute pressure at the cooler. The pressure drop must be made up by the compressor by additional head, and can be added to the first or second section pressure ratio. By applying a little experience, the guessing can be improved. The front section has a lower inlet temperature and is generally more efficient, so the best location for additional pressure would be in the first section. The first section discharge pressure is 50.2 + 2 = 52.2 psia. A new pressure ratio for the first section must be evaluated.

$$r_p = 52.2/24$$

$$r_p = 2.175$$

Step 6. Evaluate the discharge temperature, continuing the use of the previously calculated polytropic exponent.

$$T_2 = 540(2.175)^{.338}$$

$$T_2 = 702.2°R$$

$$t_2 = 702.2 - 460$$

$t_2 = 242.2°F$ first section discharge temperature

This temperature is within the limit.

Intercooler outlet pressure is 50.2 psia. Calculate the second section pressure ratio.

$$r_p = 105/50.2$$

$$r_p = 2.092$$

Evaluate the Section 2 discharge temperature.

$$T_2 = 565(2.092)^{.338}$$

$$T_2 = 725°R$$

$$t_2 = 725 - 460$$

$t_2 = 265°F$ discharge temperature

Because the temperature just calculated is right on the temperature limit and there is margin in the Section 1 temperature, the pressure may be arbitrarily adjusted to the first section to better balance the temperatures. A Section 1 discharge pressure of 54.5 psia is selected, which results in a new pressure ratio.

$$r_p = 54.5/24$$

$$r_p = 2.271$$

Now calculate a new Section 1 discharge temperature for the pressure just assumed.

$$T_2 = 540(2.271)^{.338}$$

$$T_2 = 712.5°R$$

$$t_2 = 712.5 - 460$$

$$t_2 = 252.5\ °F$$

The temperature is still within the required limit. Correct the cooler outlet pressure and evaluate a new ratio for Section 2. The corrected cooler outlet pressure is 52.5 psia.

$$r_p = 105/52.5$$

$$r_p = 2.0$$

Recalculate the discharge temperature for Section 2, using the previous cooler outlet temperature.

$$T_2 = 565(2.0)^{.338}$$

$$T_2 = 714.2°R$$

$$t_2 = 714.2 - 460$$

$$t_2 = 254.2°F$$

The temperature is now below the 265°F limit and consistent with the Section 1 temperature. At this point, the initial assumption for 2 sections can be considered a firm value.

Step 7. Calculate the polytropic head for each section, using the overall average compressibility of $Z_{2avg} = .97$.

Section 1

$$H_p = .97 \times 22.39 \times 540 \times 2.956(2.271)^{.338}$$

$$H_p = 11{,}074 \text{ ft-lb/lb}$$

Section 2

$$H_p = .97 \times 22.39 \times 565 \times 2.956(2.0)^{.338}$$

$$H_p = 9{,}576 \text{ ft-lb/lb}$$

Step 8. Develop the allowable head per stage by the use of one of the rules of thumb.

$H_{stg} = 10{,}000 - ((69 - 29)(100))$

$H_{stg} = 6{,}000$ ft-lb/lb

Divide the total head per section by the allowable head per stage to develop the number of stages required in each section.

Section 1

$z = 11{,}074/6{,}000$
$z = 1.84$ stages, round off to 2

Section 2

$z = 9{,}576/6{,}000$
$z = 1.6$ stages, round off to 2

Step 9. Calculate a head per stage for each section based on two stages each.

Section 1

$H_p = 11{,}074/2$
$H_p = 5{,}537$ ft-lb/lb head per stage, Section 1

Section 2

$H_p = 9{,}576/2$

$H_p = 4{,}788$ ft-lb/lb head per stage, Section 2

Use the geometric portion of Equation 5.12 to calculate the tip speed. Assume $\mu_p = .48$ for the pressure coefficient.

Section 1

$u_2 = (5{,}537 \times 32.2/.48)^{.5}$

$u_2 = 609.5$ fps tip speed first two stages

Section 2

$u_2 = (4{,}788 \times 32.2/.48)^{.5}$

$u_2 = 566.7$ fps tip speed last two stages

Step 10. From Figure 5-26 and the inlet volume to the first section, select an initial impeller diameter.

$d_2 = 25$ in.

Because the second section shares a common shaft with the first section, it is not necessary to look up a new impeller size. Apply the Section 1 impeller diameter, Equation 5.15, and the conversion constants of 12 in./ft and 60 sec/min. to calculate a shaft speed.

$$N = \frac{12 \times 60 \times 566.7}{\pi \times 5{,}588}$$

$N = 5{,}588$ rpm

With the shaft speed and the tip speed calculated in Step 9 for the Section 2 stages, calculate an impeller diameter using Equation 5.15.

$$d_2 = \frac{12 \times 60 \times 566.7}{\pi \times 25}$$

$d_2 = 23.24$ in. Section 2 impeller diameter

Step 11. Calculate the inlet volume into Section 2. Use $Z_{avg} = 97$, $P_1 = 52.5$ psia, and $t_1 = 105°F$. Substitute into Equation 2.10 as was done in Step 2.

$$Q_1 = \frac{.97 \times 22.39 \times 565}{52.5 \times 144} \times 3{,}200$$

$Q_1 = 5{,}194$ cfm inlet volume into Section 2

Calculate the last impeller volume for each section using Equation 5.18.

Section 1

$$Q_{ls} = \frac{10{,}971}{(2.271^{1/2})^{1/1.511}}$$

$Q_{ls} = 8{,}363.4$ cfm last stage volume, Section 1

Section 2

$$Q_{ls} = \frac{5{,}194}{(2.0^{1/2})^{1/1.511}}$$

$Q_{ls} = 4{,}129.4$ cfm last stage volume, Section 2

Use Equation 5.19 to evaluate the flow coefficient for the first and last impeller of each section.

Section 1

$\delta = (700 \times 10{,}971)/(5{,}588 \times 25^3)$

$\delta = .088$ flow coefficient, first stage

$\delta = (700 \times 8{,}364.4)/(5{,}588 \times 25^3)$

$\delta = .067$ flow coefficient, last stage

Section 2

$\delta = (700 \times 5{,}194)/(5{,}588 \times 23.24^3)$

$\delta = .052$ flow coefficient, first stage

$\delta = (700 \times 4{,}129.4)/(5{,}588 \times 23.24^3)$

$\delta = .041$ flow coefficient, last stage

Step 12. Use Figure 5-27 and the flow coefficients to determine the efficiencies for the stages.

Section 1

$\delta = .088, \eta_p = .79$

$\delta = .067, \eta_p = .80$

The average is

$\eta_p = .795$

Section 2

$\delta = .052, \eta_p = .793$

$\delta = .041, \eta_p = .78$

The average is

$\eta_p = .787$

Step 13. Recalculate the polytropic exponent.

Section 1

Use $k_{avg} = 1.345$

$$\frac{n-1}{n} = \frac{1.345-1}{1.345} \times \frac{1}{.795}$$

$$\frac{n-1}{n} = .323$$

$$\frac{n}{n-1} = 3.1$$

Section 2

Use $k_{avg} = 1.335$

$$\frac{n-1}{n} = \frac{1.335-1}{1.335} \times \frac{1}{.787}$$

$$\frac{n-1}{n} = .319$$

$$\frac{n}{n-1} = 3.136$$

Step 14. Use the polytropic exponents calculated in the previous step and recalculate the discharge temperature of each section to correct for the average stage efficiency.

Section 1

$T_2 = 540(2.271)^{.323}$

$T_2 = 703.8°R$

$t_2 = 703.8 - 460$

$t_2 = 243.8°F$ final Section 1 discharge temperature.

Section 2

$T_2 = 565(2.0)^{.319}$

$T_2 = 704.8°R$

$t_2 = 704.8 - 460$

$t_2 = 244.8°F$ final Section 2 discharge temperature.

The temperature is approximately 20°F below the 265°F temperature limit. The sections differ by less than 1°F. This is probably just luck because that good a balance is not really necessary. Also, it should be noted that to maintain simplicity the additional factors were ignored, such as the 10°F temperature pickup in the return stream due to internal wall heat transfer. Also, nozzle pressure drops for the exit and return were not used. Balance piston leakage was not used as it was in Example 5-3. When all the factors are used, the pressures for each section would undoubtedly need additional adjustment as would the efficiency. However, for the actual compression process, the values are quite realistic, and for doing an estimate, this simpler approach may be quite adequate.

Step 15. To complete the estimate, calculate the shaft power, using the conversion of 33,000 ft-lb/min/hp.

Section 1

$$W_p = \frac{3{,}200 \times 11{,}074}{33{,}000 \times .795}$$

$W_p = 1{,}350.8$ hp gas horsepower, Section 1

Section 2

$$W_p = \frac{3{,}200 \times 9{,}576}{33{,}000 \times .787}$$

$W_p = 1{,}179.9$ hp gas horsepower, Section 2

Combine the two gas horsepower values and add 1% for the mechanical losses.

$W_p = 1{,}350.8 + 1{,}179.9 + 25.3$

$W_p = 2{,}556.0$ hp compressor shaft power

Fan Laws

These relationships were actually developed for pumps instead of compressors, but they are very useful in rating compressors that are being considered for reapplication. The equations used to this point are adequate to perform any rerate calculation; however, looking at the fan laws may help establish another perspective. The following relationships are a statement of the fan laws.

$$Q_i \alpha N \tag{5.21}$$

$$H_p \alpha N^2 \tag{5.22}$$

$$W_p \alpha N^3 \tag{5.23}$$

The equations have been expressed as proportionals; however, they can be used by simply "ratioing" an old to a new value. To add credibility to fan law adaptation, recall the flow coefficient, Equation 5.19. The term Q_i/N is used which shows a direct proportion between volume Q_i and speed N. Equation 5.12 indicates the head, H_p, to be a function of the tip speed, u_2 squared. The tip speed is, in turn, a direct function of speed making head proportional to speed. Finally, the power, W_p, is a function of head multiplied by flow, from which the deduction of power, proportional to the speed cubed, may be made.

Curve Shape

Figure 1-3 presented a general form performance curve for each of the compressors. The centrifugal compressor exhibited a relatively flat curve compared to the other machines. Flat is defined as a relatively low head rise for a volume change. Translated to pressure terms, it means a relatively low pressure change for a given volume change. It is important to understand some of the basics that contribute to the curve shape.

Figure 5-24 shows that if the flow is reduced for the radial wheel, a reduction occurs in the vector, V_{r2}, but there is no influence on the tangential component of the absolute velocity, V_{u2}. In fact, the ratio of $V_{u2}/u_2 = 1$. In this case, the ideal curve would be flat, something that really does not happen due to the effect of slip and efficiency. Looking at the 60° curve, the V_{u2} vector will increase with a decrease in flow. This is shown as decrease in the length of the V_{r2} vector, raising the work input coefficient and putting a slope into the curve. Then, if the 45° vector triangle is examined, the same thing will happen: V_{u2} will increase for a decrease in the flow. Because the angle β_2 is less, the V_{u2} increases faster for 45° than for 60°, making for a steeper curve. This is consistent with the earlier statement about the higher reaction wheel having a steeper curve.

Flow passing through an impeller is constantly changing in volume because of the compressible nature of the gas. If an impeller is operated first with a light molecular weight gas and then a heavy gas, the curve will be steeper with the light gas because the volume ratio is higher for the heavy gas. An examination of Equation 5.12 shows that head for a given geometry is fixed, within reasonable limits. Therefore, substituting different molecular weights in the head equation will indicate a higher pressure ratio directly proportional to the molecular weight. The volume ratio, then, is directly proportional to the pressure ratio making it also directly proportional to the molecular weight. Since the geometry was not modified to match the different volume ratio, the vectors, V_{r2}, are shorter for the lower outlet volume. As such, the change to the vector V_{u2} is not as great and the curve is not as steep.

The compressibility of the gas going through the impeller causes some problems. The assumption in the use of the fan law, when speeding up an impeller, is that the inlet volume follows the speed in a proportional manner. At the same time, the head is increased as a function of the speed squared. Just as the head increases with a given gas, so does the pressure ratio and therefore the volume ratio. It wasn't pointed out, but the alert reader may have noticed that the outlet triangle, not the inlet triangle, was used to discuss the curve shape. The problem is that the outlet volume is not exactly proportional to the inlet volume. For a 10% speed change, the compressor does not truly respond with a 21% head change. For small speed changes the problem is not serious; however, the basics should be remembered if a compressor is being rerated.

One last item should be noted regarding the shape of the curve. As stages are put together, the overall flow range of the combined stages is never larger and, in most cases, is less than the smallest flow range of the individual stages. Because of the compounding effect, as the volume is changed, the combined curve is always steeper.

Surge

Notice that the left end of a centrifugal compressor pressure volume curve does not reach zero flow. The minimum flow point is labeled as the *surge limit* and is the lowest flow at which stable operation can be achieved. Attempted operation to the left of that point moves the compressor into surge. In full surge the compressor exhibits an extreme instability; it backflows to a point and then temporarily exhibits forward flow. This oscillating flow is accompanied by a large variety of noises, depending on the geometry and nature of the installation. Sometimes it is a deep low frequency booming sound and for other machines it is a squeal. The pressure is highly unsteady and the temperature at the inlet rises relatively fast. The latter is caused by the same gas backing up in the machine and then recompressing until the next backflow. Each pass through the compressor adds additional heat of compression. Mechanically, the thrust bearing takes the brunt of the action and, if not left in surge indefinitely, most compressors do survive. In fact, most compressors that have operated for any period have experienced surge at one time or another. If left unchecked, and assuming the thrust bearing is well-designed, the compressor will more than likely destroy itself from the temperature rise.

Surge is due to a stalling of the gas somewhere in the flow path, although opinions seem to differ as to exactly where. For the process plant type low head compressor, it would appear to start in the diffuser. It can also take place at one of several points in an impeller depending on the geometry. For compressors designed for higher heads, the primary stall point appears to move into the impeller. Compressors exhibit a phenomenon referred to as *incipient surge* or *stall*. This is where one element stalls but not severely enough to take the stage into a complete stall. An experienced listener can readily hear and identify the stall. If the flow is not further reduced, it can remain in this condition without further stalling. It is very close to the limit, however, and only a minor flow disturbance can trigger a full-stage surge, which may then spread through the whole compressor. Stall is a flow separation. It may be compared to an airplane wing that produces lift until the angle of attack exceeds a limiting value at which point separation becomes great and the ability to continue producing lift is lost.

Choke

The right side of the curve tends to slope in an orderly manner and then falls off quite rapidly. If taken far enough, the compressor begins to choke or experience the effect of "stonewall." If the internal Mach numbers are near 1 and/or the incidence angle on the inlet vane becomes high enough to reduce the entrance flow area and force the Mach number high enough, the compressor will *choke*. At this point, no more flow will pass through the compressor. The effect is much greater on high molecular weight gas, particularly at a low temperature and with the k value on the low side. The problem is that the compressor reaches the "stonewall" limit in flow before the designer had intended. If compressors are rerated, this effect must be kept in mind, particularly when the new conditions are for a lower molecular weight gas. It is possible to choke the front-end stages and starve the downstream stages, causing these stages to be in surge.

Normally, operation of a compressor in choke flow is relatively benign, particularly for compressors operating at nominal pressures of less than 2,000 psig. As the application pressurs are raised, with the higher resulting density, there is the possibility that the off-design differential pressures could become high enough to increase the stresses to a level of concern. It would be wise if in the application of very high density compressor, due to the nature of the operation, the supplier be advised if prolonged operation in choke flow is anticipated. The supplier should review past experience with similar installations and critique the design to avoid potential problems.

Application Notes

As with the reciprocating compressor, care must be exercised when liquids are present in the gas stream. Unlike the reciprocator, the centrifugal is somewhat more forgiving if the liquid is in the form of mist. Small droplets can pass through the machine without problem if the duration is short. The problems with liquid have both short- and long-term characteristics. Short term, the biggest problem with liquid is the ingesting of slugs of water. The compressor is in danger of severe mechanical damage if suddenly deluged with a great quantity of liquid. For this reason, compressors taking suction from vessels containing either liquid and vapor or vapors near saturation should have suction drums to trap any potential liquids. The suction drum is also a good idea where the possibility exists that condensation could take place in the suction lines, forming a slug on its way to the compressor. The long-term problem is with mist or small droplets. With time, any liquid will start an erosion of the moving parts, particularly the impeller vanes. As the tips of the vanes erode, the effective diameter of the impeller is reduced. The foregoing equations showed that the head-producing capability of the impeller is a function of the tip speed squared.

Interestingly enough, centrifugals can be washed "on stream" to counteract the effects of fouling. In some cases, where fouling is continuous and severe, a liquid wash may be used continuously. Care must be used and a certain amount of trial-and-error steps taken to ensure the proper quantity: enough to do the job and not enough to cause significant erosion. In the same manner, when a compatible liquid is available, liquid can be injected into the machine to provide auxiliary cooling. In all liquid injection applications, it is important not to inject the liquid so that it impinges on any of the surfaces. It is better to use tangential sprays to the degree practical to have the liquid flash in the gas stream.

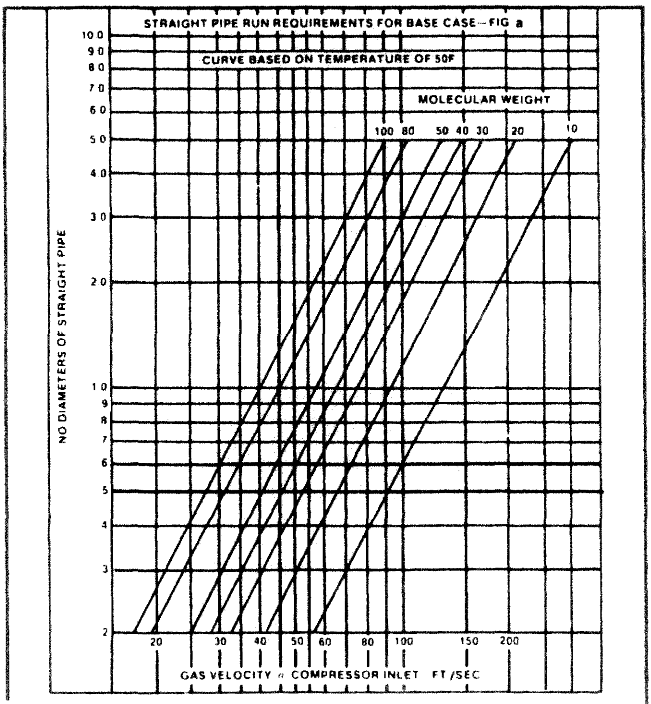
One question that arises quite often is the orientation of inlet piping and its influence on compressor performance. The flow into the impeller has been assumed axial or radial, depending on the impeller geometry, which means there is no pre- or antirotation and it is free from random flow distortions. While centrifugals are somewhat more forgiving than other machines like axials, there are limits. If the flow has rotation or distortion as it enters the impeller, the compressor performance will be influenced in a negative manner. Correct piping practices at the compressor inlet will help ensure the proper performance of the compressor. Figure 5-28 includes a set of curves that may be used as guidelines to establish a minimum length of straight pipe to use ahead of the inlet. The base case is shown in Figure 5-29 and consists of an elbow turned in the plane of the rotor. While the sketches are shown as multistage compressors, they may be used for axial entry single-stage compressors by obtaining the multiplier for the base case and taking the final result and multiplying by 1.25. The higher multiplier accounts for the more sensitive nature of the axial inlet. These sketches and pipe lengths are conservative, but should a vendor recommend a longer length, the vendor's recommendation should receive the first consideration. When there are problems achieving some of the minimum lengths, vaned elbows and straighteners can be used. Figures 5-30 and 5-31 offer suggestions for those not experienced in these areas. Again, these are methods that have been used, but are not the only, or necessarily the best, solutions for any and all applications.

Mechanical Design

Introduction

The centrifugal compressor is composed of a casing containing a rotating element, *rotor,* which is supported by a set of bearings. For most multistage compressors, shaft end seals are located in-board of the bearings.

(*text continued on page 192*)



COMPRESSOR INLET CORRECTION FACTORS FOR VARIOUS PIPING ARRANGEMENTS

	Factor	Figure
1. One long radius elbow (plane parallel to rotor)	1.0	a
2. One long radius elbow (plane normal to rotor)	1.50	b
3. Two elbows at 90° to each other with second elbow plane parallel to rotor	1.75	c*
4. Two elbows at 90° to each other with second elbow plane normal to rotor	2.0	d
5. Butterfly valve before an elbow		
a. valve axis normal to compressor inlet	1.5	e*
b. valve axis parallel to compressor inlet	2.0	—
6. Butterfly valve in straight run entering compressor inlet		
a. valve axis normal to rotor	1.5	—*
b. valve axis parallel to rotor	2.0	—
7. Two elbows in same plane (parallel to rotor)	1.15	f*
8. Two elbows in same plane (normal to rotor)	1.75	—
9. Gate valve (wide open)	1.0	—*
10. Swing check valve (balanced)	1.25	—*

Factors also apply to single stage, axial inlet compressors.

Note:

1. Factors are applied to the base straight run requirements from the chart.
2. Factors for butterfly valves assume minimum throttling at design conditions. If heavy throttling is required, factors should be doubled.
3. *For axial inlets, use 1.25 with appropriate figure.*

Figure 5-28. Chart for minimum straight inlet piping. Use this chart and the given factors in conjunction with Figure 5-29. (*Courtesy of Elliott Company*)

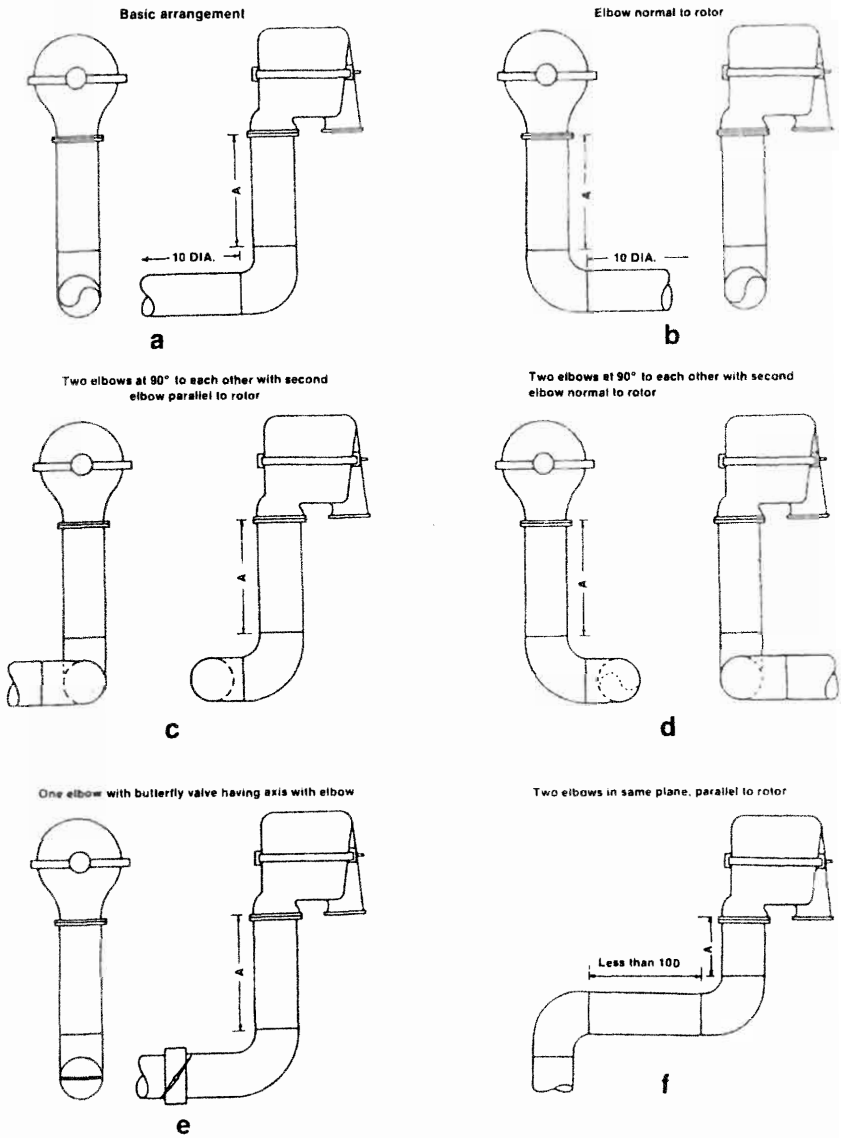


Figure 5-29. Methods of piping. To find "A," multiply the number of diameters of straight pipe from the chart in Figure 5-28 with the appropriate correction factor. (*Courtesy of Elliott Company*)

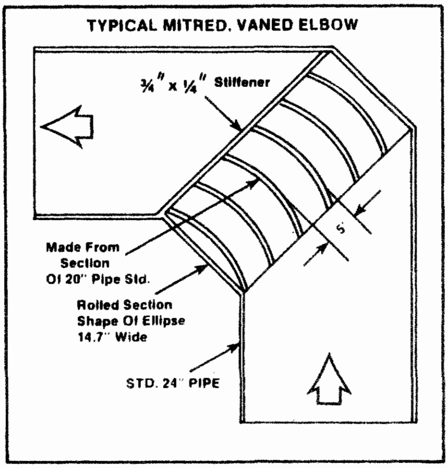


Figure 5-30. Elbow straightening vanes. (*Courtesy of Elliott Company*)

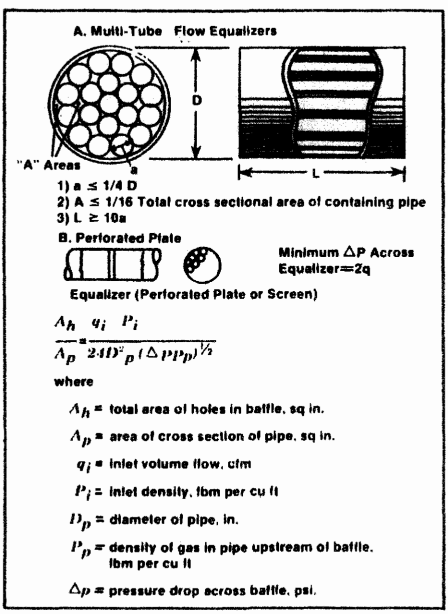


Figure 5-31. Straightening vanes. (*Courtesy of Elliott Company*)

(text continued from page 188)

The internal passages are formed by a set of diaphragms. Figure 5-32 depicts a typical multistage barrel compressor. Refer to this figure to locate the relative position of the various parts that are described in the following section.

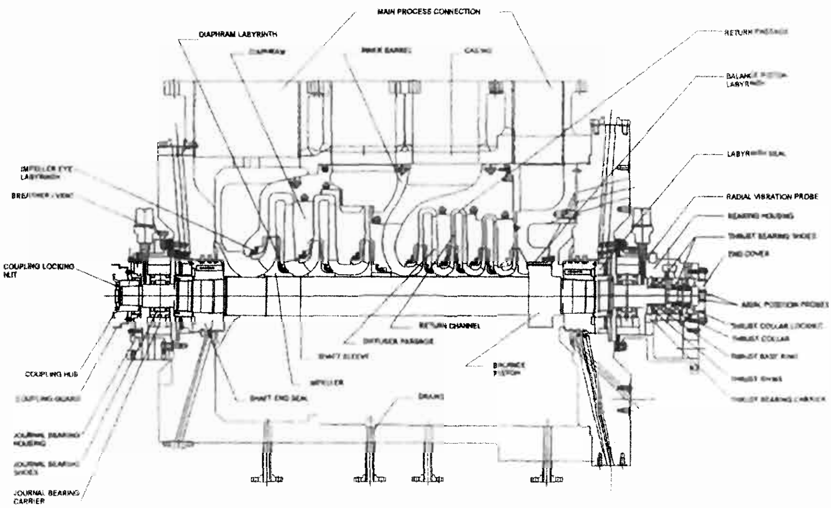


Figure 5-32. Centrifugal compressor nomenclature. (*Courtesy of A-C Compressor Corporation*)

Casings

All centrifugal compressor casings were initially of cast construction, and this method is still used on many casings today, particularly in the smaller sizes. In the past 15 years, some manufacturers have fabricated steel casings, generally converting their line of steel casings beginning with the larger frame sizes. The reason for this was economics; while the fabricated casings cost more to make on a per-unit weight basis, the net cost was less. Two factors were responsible. Quality of large steel castings was hard to control, with much time spent repairing the casing after inspection. Secondly, fabrication techniques and costs have improved significantly. There also came a side benefit of flexibility, once the manufacturer discovered he was no longer tied to a set of patterns or was bound by the time consumption and cost of pattern changes.

Casing materials are, in most cases, cast iron, nodular iron, or cast steel. Fabricated casings are generally made of carbon or alloy steel. Casings are, on occasion, made of austenitic stainless steel or one of the high nickel alloys. For low temperature inlet conditions, a low nickel alloy may be used. API Standard 617 [12] includes material guidelines in its appendix. The standard also mandates steel for all flammable and toxic gases, for air or nonflammable gas at pressures in excess of 400 psig, and for air or nonflammable gas with operating temperatures anywhere in the operating range in excess of 500°F.

The casing construction and materials covered to this point have generally applied to all kinds of compressors, including both horizontally split and vertically split. The vertically split, multistage barrel compressor is somewhat different. It is generally constructed of steel or steel alloy. It may be cast, fabricated, or, for very high pressure service, it may be forged. It should always be used when the gas contains hydrogen at or above a partial pressure of 200 psig. It may also be required in those services where the overall pressure is too high for the horizontally split compressor. This occurs when the horizontally split joint deforms too much at the operating pressure to maintain a gas-tight seal.

Diaphragms

The stationary members located inside a multistage casing are referred to as diaphragms. The function of the diaphragm is to act as a diffuser for the impeller and a channel to redirect the gas into the following stage. The diaphragm also acts as the carrier for the impeller eye seal and the interstage shaft seal. Diaphragms are either cast or fabricated. Most cast diaphragms are made of iron. Fabricated diaphragms are steel or composite steel and cast iron, with straightener or guide vanes of cast iron. Diaphragms are normally not highly stressed, with some exceptions. On compressors with out-in streams, if the differential pressure is relatively high from the outlet to the return nozzle, then the differential is taken across the diaphragm at the two nozzles. This diaphragm should be made of steel. The diaphragms are split, located with matching grooves in the upper and lower half casing and pinned to the upper half for maintenance ease. The diaphragms are hand-fitted to center them to the rotating element. It is important for the horizontal joint to match well, to keep the joint leakage to a minimum. On barrel compressors, the diaphragm assembly makes up an inner barrel (see Figure 5-33). The assembly and rotor are removed from the barrel casing as a unit using a special fixture. The diaphragm assembly is split to permit the removal of the rotor, and the diaphragms are generally constructed in the same way as those of the horizontally split compressor.

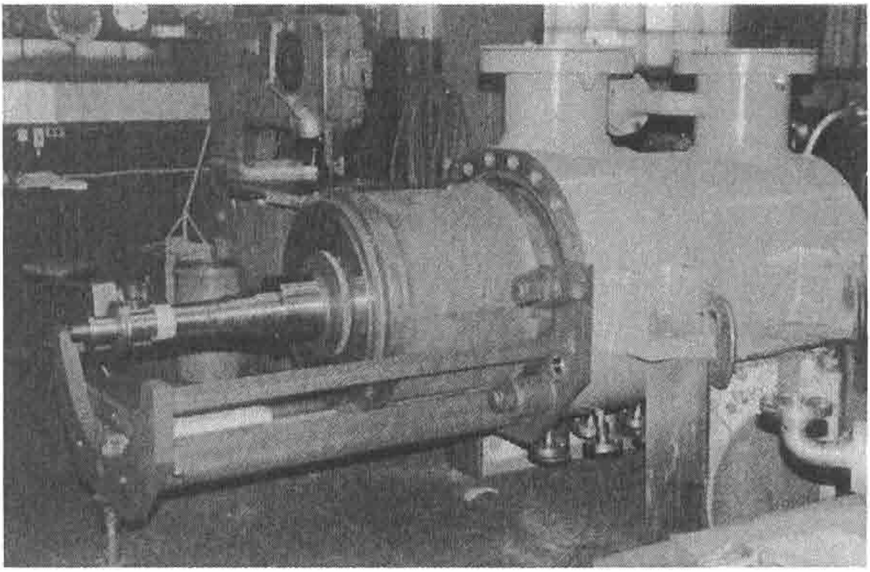


Figure 5-33. Inner barrel assembly. *(Courtesy of A-C Compressor Corporation)*

Casing Connections

Casing inlet and outlet nozzles are normally flanged. General preference, in process service, is for all casing connections to be flanged or machined and studded. On steel-cased machines, this normally is not a problem. On the smaller, refrigeration compressors that are highly standardized, constructed of cast iron, and originally designed for other than process service, connections will generally have flanged inlet and outlet nozzles. However, most of the auxiliary connections on these machines will be screwed. It is desirable to use standard flanges throughout the connections on the casing. However, for space reasons, on rare occasions, a nonstandard flange arrangement may become necessary. It is quite important to have the equipment vendor furnish all nonstandard mating flanges and associated hardware.

Forces and moments which the compressor can accept without causing misalignment to the machine are to be specified by the vendor. Many factors go into this determination, and as one may guess, the limits are determined quite arbitrarily in most cases. With all the many configurations a compressor can take, a single set of rules cannot fit all. Despite this, NEMA SM-23[13] for mechanical drive steam turbines is used as a basis. API 617 has adapted the NEMA nozzle criteria to centrifugal compressors. This works on larger steel-cased multistage compressors, but is not good for the overhung style. Moreover, the user or piping designers want a higher number to simplify piping design, while the manufacturer wants a small number to assure good alignment and fewer customer complaints. From a user's point of view, where long-term reliability is a must, the vote must go to the manufacturer. Experience shows that the lower the piping loads on the nozzles, the easier coupling alignment can be maintained. This seems reasonable since most compressors are equipped with plates called wobble feet to provide flexibility for thermal growth. The feet will flex from pipe loads as well as from the temperature. The piping loads tend not to align themselves as well with the shaft as the temperature gradients. Even when guides and keys are used, as is customary on the larger machines, they may bind despite the fact that they are stout enough to carry the load.

Impellers

Impeller construction was covered in the performance section and need not be repeated here. The impeller is the most highly stressed compressor component, and generally becomes the limiting item when it comes to establishing the rotating element performance limit. Impellers are made of low alloy steel for most compressors in process service, either chrome-moly or chrome-moly-nickel. Because of the high strength-to-weight ratio, many of the high head, integrally geared units use aluminum. Austenitic stainless, monel, and titanium are some of the other materials used for impellers in certain special applications, generally with corrosive gases involved. Stress levels must be adjusted for the materials involved. Some of the precipitation hardening steels in the 12 chrome alloy have been used and found to provide a good alternate material with moderately good corrosion resistance and very good physical properties.

Impeller construction for the cover-disk style impeller historically has been by built-up construction and welding. The traditional method uses die formed blades (see Figure 5-34). More recently, with the increased use of 5-axis milling, blades have been milled integrally with the hub disk. This alternate construction method is somewhat more costly because of the machining but produces a more accurate and repeatable gas path, which offsets the added expense (see Figure 5-35). Cover disks are welded to the blades to complete the milled impeller. Physical prop-

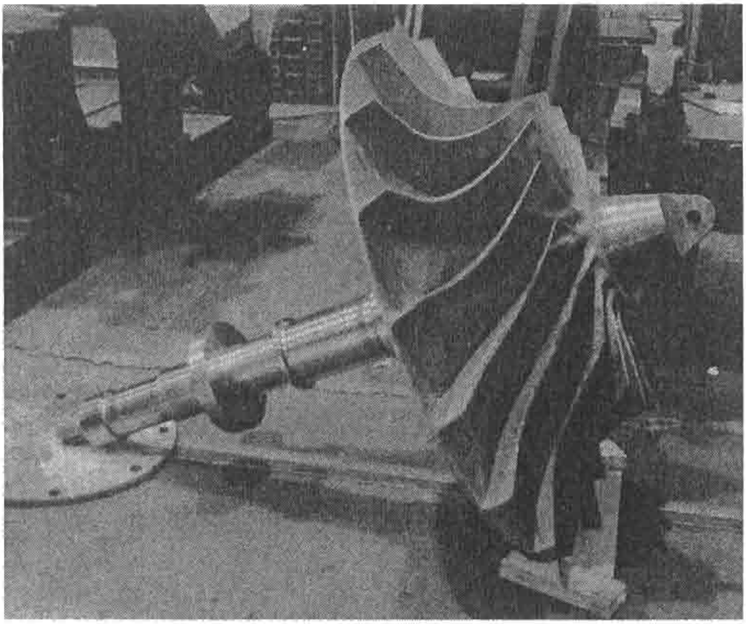


Figure 5-34. A fabricated centrifugal compressor impeller. (*Courtesy of A-C Compressor Corporation*)

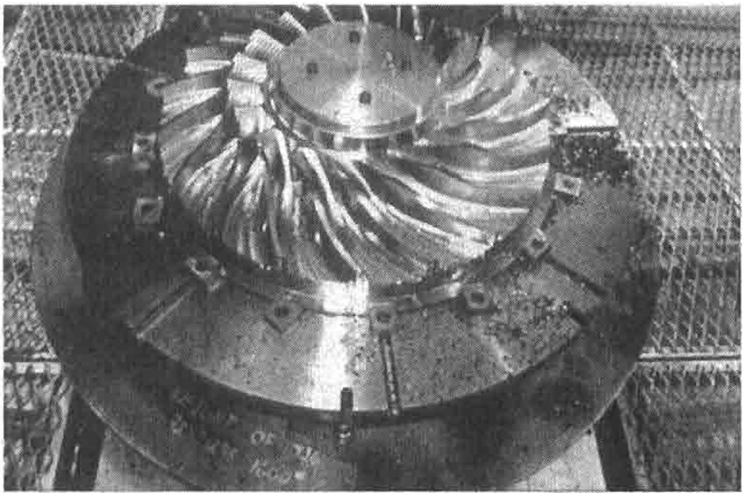


Figure 5-35. A centrifugal compressor impeller during manufacture. The blading was milled with a five-axis milling machine. The blading is integral with the back plate. (*Courtesy of Dresser-Rand*)

erties are derived by heat treating and stress relieving. Some small sizes are cast. In the semi-open construction, casting is quite common, though fabricated impellers are used. Fully open impellers, which are not as common, can be either fabricated or cast. Impeller shaft attachment for multistage applications is by shrinking the hub to the shaft either with or without a key, depending on the vendor philosophy. There are numerous other methods used, each peculiar to the individual vendor.

Although not universal, on most multistage compressors, the impellers are axially located by shaft sleeves. The sleeves form a part of the interstage seal and are shrunk onto the shaft with a shrink level less than the impeller.

Shafts

Shafts are made of material ranging from medium carbon to low alloy steel and are usually heat treated. Shafts were originally made of forgings for the compressors in process service. But because of the availability of high quality material, hot rolled bar stock has been used for shafts up to 8 inches in diameter. Bar stock shafts are given the same heat treatment and quality control as forgings. Many of the process users prefer a low alloy, chrome-moly-nickel material for shafting, particularly for compressors in critical service.

Shafts require a good finish that can be achieved by machining. Honing, or sometimes grinding, is used to improve the finish in selected areas. Since proximity probes are used with most process compressors, the probe area must receive extra attention to minimize mechanical and electrical runout. On the whole, the shaft is the foundation for good mechanical performance to keep the rotor dynamics in control and maintain good balance. The requirements are that the shaft must be round and all turns must be concentric to the journals. As simple as it sounds, it is not easy to accomplish. The tighter the tolerance, the closer to perfection, the more expensive that particular manufacturing step. However, some added expense at this point will save time in subsequent rotor balancing providing the user with a rotor that can be more easily maintained. By using CNC machine tools for manufacturing shafting, the cost should come down, quality improve, and the product should become more consistent.

Radial Bearings

Radial bearings or journal bearings are usually pressure-lubricated. Most compressors use two bearings on opposite ends of the rotor assembly or on the overhung design, located adjacent to each other between the drive coupling and the impeller. It is highly desirable for ease of maintenance to have the bearings horizontally split. On centrifugal compressors, the bearings size is not a function of the load but rather it is dictated by critical speed considerations. Rotors in centrifugal compressors are by nature not very heavy; therefore, the bearings are lightly loaded. Because of the light loading, there are potential bearing-induced rotor dynamics problems.

Straight cylindrical bearings, as shown in Figure 5-36, are the most simple in concept. Because of low resistance to bearing-induced prob-

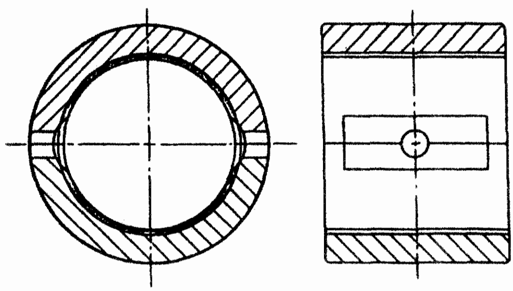


Figure 5-36. Straight cylindrical bearing. (*Courtesy of Turbocare, a Division of Demag Delavel Turbomachinery Corp., Houston facility*)

lems, application of this bearing is limited in centrifugal compressors. It is found normally in very large compressors with relatively heavy rotors and low compressor operating speeds. They are also used in fluorocarbon refrigerant compressors, where speeds are low because of high molecular weight and where relatively short rotors are used. As a minimum, most compressors with sleeve bearings use a modified sleeve bearing, such as the dam type shown in Figure 5-37. A relief groove is cut in the upper half of the bearing. The groove is stopped near the center of the upper portion of the bearing in a square, sharp-edged dam. As the shaft rotates, oil is carried through the groove to the end where the oil velocity is suddenly brought to a halt thereby converting it to pressure. A stabilizing force is formed on the top of the journal by the pressure. The maintenance of the sharp edge at the end of the bearing is very important. In service, if the groove ends become rounded, the bearing will cease to function as intended and can become unstable.

To facilitate maintenance and avoid the tedious scraping and other fitting steps required in early forms of plain journal bearings, replaceable

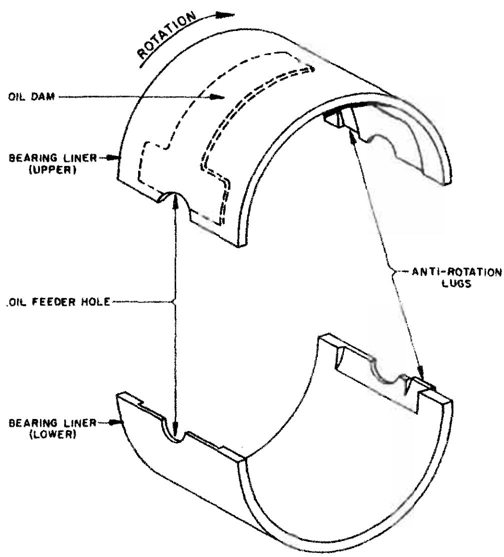


Figure 5-37. Dam type sleeve bearing. (*Courtesy of Elliott Company*)

inserts are used. The inserts are lined with a thin layer of babbitt on a steel backing. Precise manufacturing assures interchangeability. Babbitt thickness is a compromise, balancing enough depth for particle imbedability against keeping the strength up by staying close to the steel liner. This form of journal bearing is also referred to as a *liner bearing.*

The bearing most often found in centrifugal compressors is the *tilting pad bearing,* shown in Figure 5-38, which is inherently stable. The individual pads break up the rotating oil film and discourage the tendency for the oil to whirl. Each pad also acts as a separate force to keep the bearing loaded and thereby stabilized. The bearing, also known as the tilting shoe bearing, has grown in popularity in recent years and is found in most process compressors in critical service. The bearing can be furnished with various numbers of pads, with five being the most common. Bearing dynamics can be altered by a variety of configuration changes, such as load on or between pads. The number of pads can be changed for alternative dynamic parameters, with the four-pad bearing the most common alternative. Bearing clearance for a journal bearing is on the order of 1 to 1.5 mils per inch of journal diameter and is generally the same value for both the liner and the tilting pad bearings.

The pads are fabricated of steel with a babbitt coating, the thickness determined by the same argument as stated for the liner bearing. The

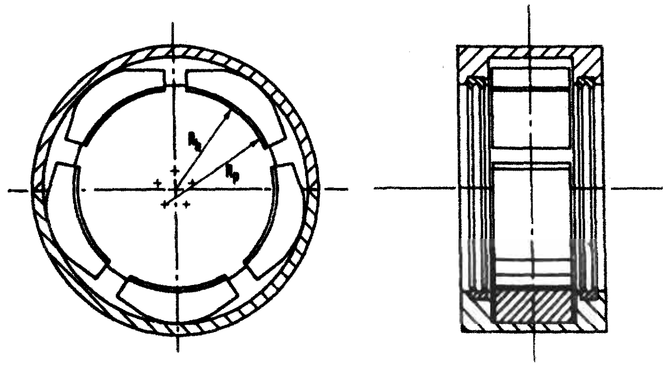


Figure 5-38. Five-pad tilting pad bearing. ***(Courtesy of Turbocare, a Division of Demag Delavel Turbomachinery Corp., Houston facility)***

backside of the pad is fitted with some form of rocker, the exact shape varying from one maker to the next. The pads are contained by a horizontally split base ring assembly.

Thrust Bearings

Centrifugal compressor impellers, with the exception of the open impeller, are thrust unbalanced. The machine also has a requirement for a location device to maintain axial clearances. For these reasons, all centrifugal compressors use some form of thrust bearing.

API 617 recognizes the need for the compressor thrust design to take into account peripheral factors such as the coupling. Gear couplings can transmit thrust to the compressor because of tooth friction. The standard uses an arbitrary friction coefficient of .25, which can be a design basis. Flexible element couplings transmit less thrust because of the lower flexing element axial stiffness.

The basic type of thrust bearing consists of a thrust collar attached to the shaft running against a *flat land* (see Figure 5-39). The land is normally a steel ring with a babbitted surface. The load-carrying capacity of this bearing is quite limited, making the bearing suitable only for locating purposes. This bearing is commonly used with double helical gear units and is not normally found in centrifugal compressors.

A thrust bearing that physically resembles the flat land bearing is the *tapered land bearing.* The modification is the construction of the land. The land is grooved radially, dividing the land into segments that are individu-

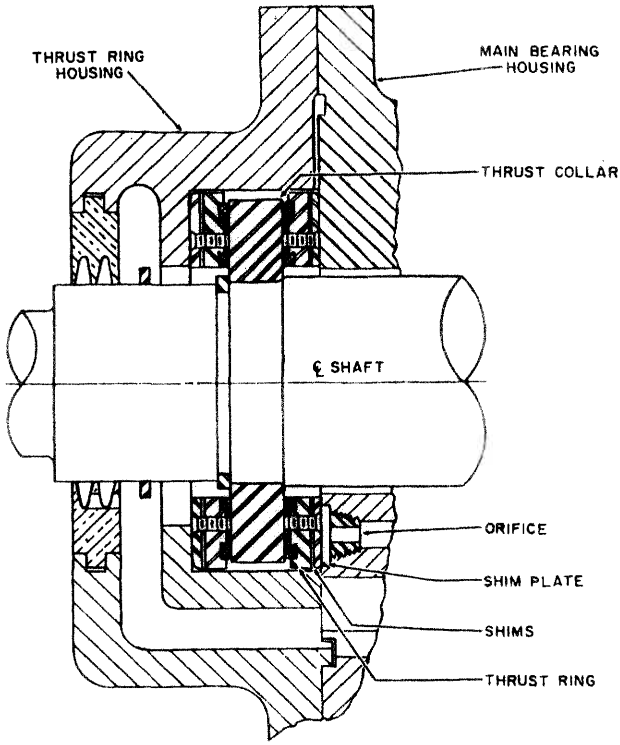


Figure 5-39. Basic thrust bearing. (*Courtesy of Elliott Company*)

ally tapered to form a wedge. As the collar rotates, relative to the land, oil is carried past the tapered wedges, developing pressure in the oil film resulting in an outward force. This force generates a load-carrying capacity. Theoretically, the tapered land bearing is capable of handling large axial loads, but it can only do so at a limited speed range. The bearing also requires good perpendicular alignment between the shaft and the land to maintain a uniform face gap. This bearing is used only for limited applications in the centrifugal compressor and when used is highly derated.

The tilting pad thrust bearing is available in two forms. The first form is named alternately for one or the other of the inventors, as it was developed by Albert Kingsbury in the United States and A.G.M. Michell in Australia working independently [14]. The bearing consists of a collar or thrust runner, attached to the shaft with the collar either integral or separable, and the stationary carrier in which the pads reside. Various numbers of pads are used, with six or eight being the most common. The pad consists of a babbitted segment, normally made of steel. The load is transmitted to the carrier by way of a button at the back of the pad, which also acts as the pivot point. The button may be centered or offset. The bearing is suitable for variable speeds because the pivoting feature allows the pad to adjust to the differing velocity of the oil film. The basic tilting pad thrust bearing is shown in Figure 5-40.

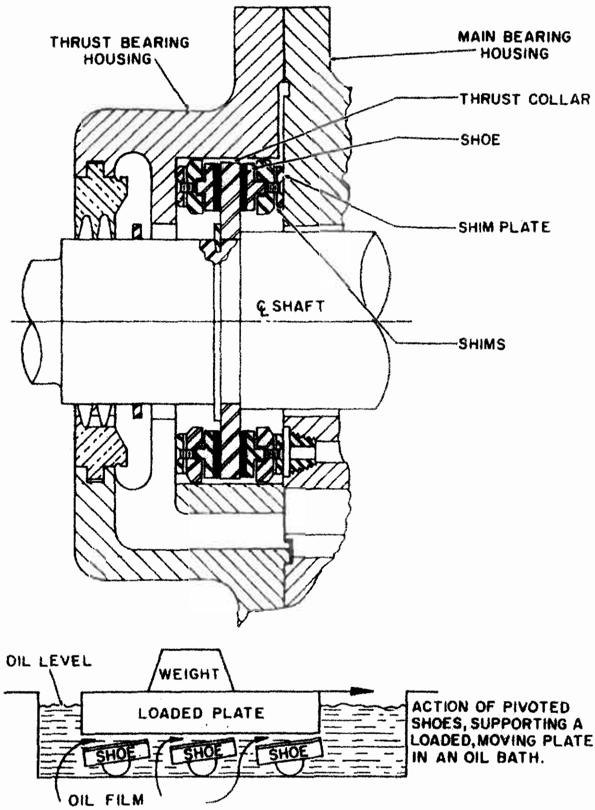


Figure 5-40. Tilting pad thrust bearing. (*Courtesy of Elliott Company*)

The second form of the multiple segment or tilting pad thrust bearing retains all the features of the first type, but includes a further refinement. This bearing is referred to as a *self-equalizing bearing*. Instead of a simple carrier to house the pads, each pad rides against an equalizing bar. Between each pad's equalizing bar is a secondary bar that carries the ends of two adjacent pad bars and transmits the load to the carrier ring. All bars are free to rock and thereby adjust themselves until all pads carry an equal share of the load (see Figure 5-41). The advantage of the self-

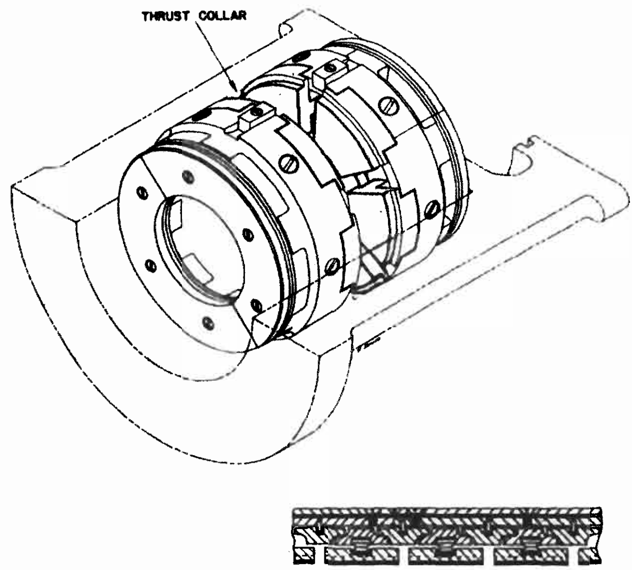


Figure 5-41. Self-equalizing tilting pad type thrust bearing. (*Courtesy of Elliott Company*)

equalizing bearing is obvious—it can adjust for minor irregularities of the rotor-to-bearing position.

Each bearing described requires a certain amount of axial space, with the simple thrust ring using the least and the self-equalizing bearing the greater amount. The thrust carrying capability of the latter two bearings is theoretically the same, and proponents of the Michell bearing cite deflections in the carrier ring as providing sufficient adjustment to achieve full potential load within the practical limits of bearing misalignment.

API 617 mandates the self-equalizing feature. Steady loads as high as 500 psi can be accommodated with transient loads going higher. However, conservative design practices and some encouragement from API tend to keep the loads on the thrust bearing in the range of 150 psi to 350 psi.

Pads for the multiple pad bearings may be made of a higher heat conducting material such as copper. Load-carrying capability can be increased by use of the copper pads. This option is a good alternate for difficult applications where size limits the use of a standard bearing. Many users do not permit the use of the alternate materials in a new compressor, using the argument that the option should be available for the solution of field problems. It should be mentioned that while the reference to copper is the common usage, in reality the material is a chromium copper alloy.

Compressors built to the API standard are required to have equal thrust capability in both directions; that is, the bearing is to be symmetrically constructed. However, the thrust bearings can be combined using various numbers of pads on each side, or a tapered land can be combined with the multiple-segment design. Other combinations of the four thrust bearings discussed are found in certain isolated applications.

The thrust bearing is responsible for large portions of the mechanical horsepower losses; however, it is the sophisticated multiple-segment bearing that has the highest losses of all the thrust bearing types. The power consumption is due to the churning of the oil in the essentially flooded bearing; therefore, care must be used in sizing the bearing to maintain margins for reliability. Alternative lubrication methods, such as the directed lubrication that uses a spray or jet to apply oil to the pads and eliminate the need for flooding, are available on some designs.

Bearing Housings

The bearing support system is normally separable from the casing, as mandated by API 617 and should be made of steel, particularly when used with a steel case. Provision should be made to maintain alignment of the rotor to the casing. The housing should be horizontally split and nonpressurized with provision for circulation of bearing lubrication. Care should be taken to prevent foaming of the lubricant. The housing is the desirable place to locate radial vibration probes as required by API 617.

The preceding paragraph assumes the bearings are located outboard of the seals on a multistage compressor, and also applies to most of the overhung types with the exception of the integrally geared machine. For the multistage, which has the seals outboard of the bearings, it is recognized that the housing will assume some pressure level used in the compressor. Therefore, provision should be made to minimize the amount of lubricant entering the gas stream. Also, for maintenance purposes, a port access to the bearing should be furnished. This type of compressor is generally limited to fluorocarbon refrigerant service and is not recommended for general gas service.

Magnetic Bearings

With the advent of magnetic bearings, the dream of an all-dry compressor can now be realized. This is to say that no external lube system is needed. Not all compressor applications at this point can qualify, because control oil is generally required for steam and gas turbine drivers. Gear bearing loads at present are higher than can be carried by current magnetic bearing designs.

The magnetic bearing is made up of a series of electromagnets located circumferentially around the shaft to form the radial bearing. The electromagnets (Figure 5-42) are laminated to limit the eddy current losses. The shaft must be fitted with a laminated sleeve (see Figure 5-43) for the

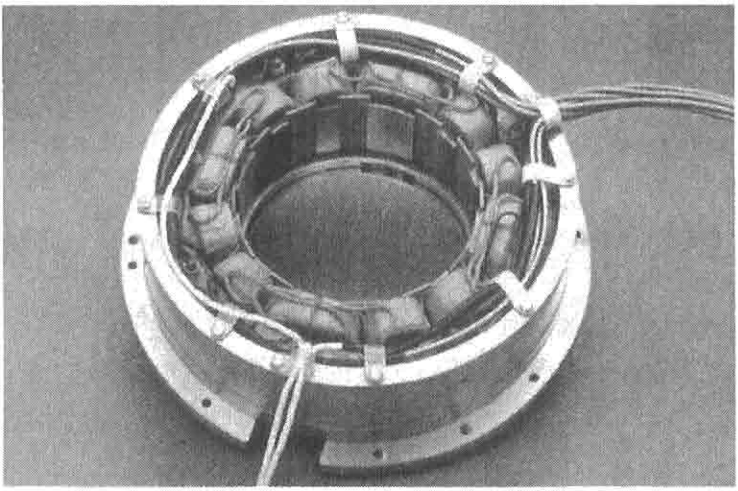


Figure 5-42. Radial magnetic bearing with a view of the circumferential electromagnets. (*Courtesy of Mafi-Trench Corp.*)

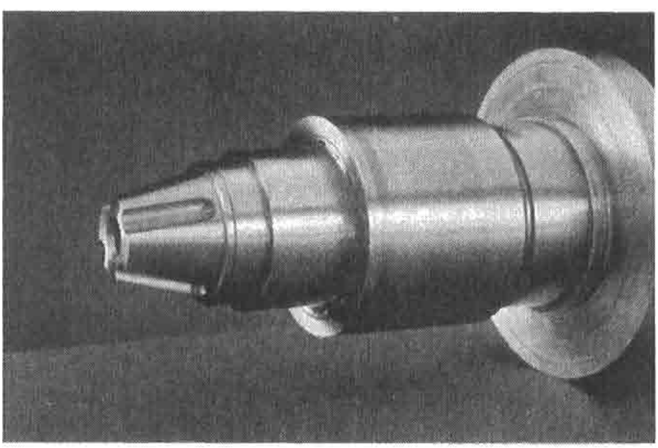


Figure 5-43. Radial magnetic bearing rotor sleeve. (*Courtesy of Mafi-Trench Corp.*)

same reason. The thrust is carried by a single-acting or dual-acting set of electromagnets (see Figure 5-44) depending on the need for a unidirectional or bidirectional thrust load. The magnetic bearing operates with a fixed air gap so there is no contact under operating conditions.

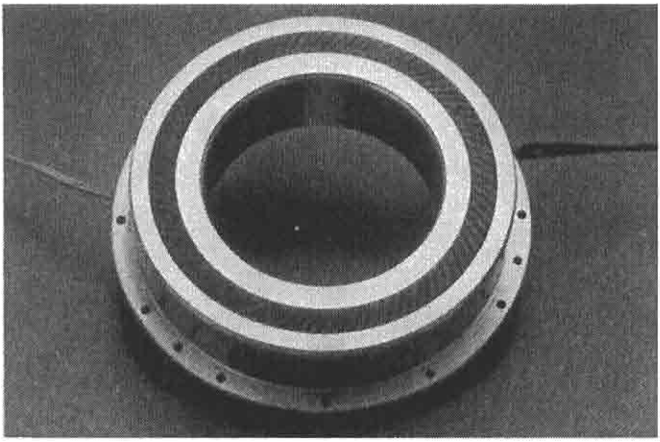


Figure 5-44. Magnetic bearing thrust electro-magnets. (*Courtesy of Mafi-Trench Corp.*)

Sensors are incorporated in the bearing assemblies to sense position of the rotor relative to the bearing. A servo control system uses the position information provided by the sensors to increase or decrease the bearing force on the rotor as needed to keep the rotor properly positioned. Magnetic bearings react differently than hydrodynamic or rolling element bearings in that the mechanical bearings react immediately to a load change, while the electromagnet in line with the load change must increase its force at the same rate to maintain rotor position. The actual rate at which the servo amplifier can increase the force is a function of volt ampere product of the amplifier. If the rate at which the load is applied exceeds the capability of the servo control, a temporary perturbation will be experienced before the shaft is brought back to its normal position.

The magnetic bearing load capacity on a per-unit basis is less than that available from hydrodynamic bearings. The specific load limit is at approximately 80 psi with typical design values of 60 psi. Higher values can be achieved with special magnetic materials, but these are not normally used in compressor applications. The load carried by bearing may be compensated by increasing the physical size of the bearing. The heaviest compressor rotor weight has been approximately 4,000 pounds.

Speed is not limited by a surface speed as in the hydrodynamic bearing. Unfortunately, however, the stresses in the thrust collar pose one limit and the other is caused by need for an auxiliary bearing, which does have a limitation based on the type of bearing being used.

The auxiliary bearing may be of the rolling element type, which is currently most common, or the dry lubricated bushing. The auxiliary bearing, which normally does not contact the shaft, is used to protect the rotating components from loss of the servo amplifiers (see Figure 5-45). The aux-

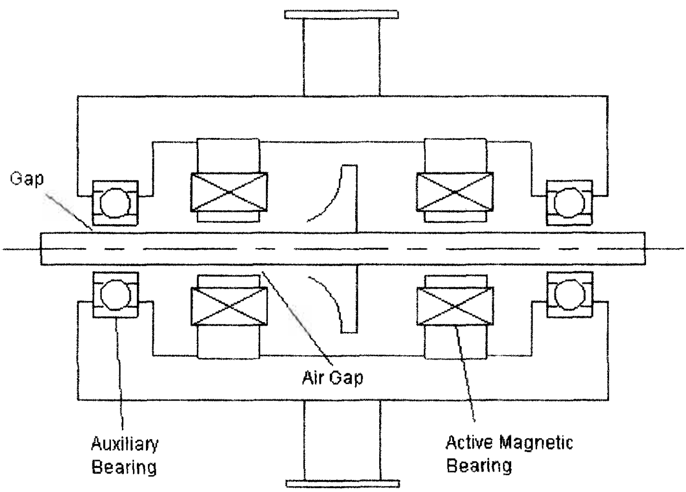


Figure 5-45. Schematic illustration of magnetic bearings and auxiliary rolling element bearings.

iliary bearing gap is approximately one half the air gap. Displacement due to momentary overload would also cause the auxiliary bearings to be pressed into service on a transient basis. It is critical that the compressor be tripped off line should the power to the magnetic bearings fail, because the auxiliary bearings have a limited life and are primarily intended for coastdown use. The life of the auxiliary bearings in general is considered to be five coastdowns from full speed. In some cases, the life has proven to be somewhat longer, particularly with the dry bushing design. Continued development in this area will no doubt increase this value in time.

An interesting aspect of the magnetic bearing is that in pre-startup the bearing servos are energized and the rotor levitated. It remains suspended as the startup begins. There is no minimum oil film type phenomena to pass through. On shutdown, the rotor is allowed to cease rotating and to remain in the levitated position until the power is removed should a full shutdown be required.

Balance Piston

It is desirable to have additional axial-load control on the multistage compressor. A *balance piston,* also referred to as the *balance drum,* can be located at the discharge end (see Figure 5-46). The balance piston consists of a rotating element that has a specified diameter and an extended rim for sealing. The area adjacent to the balance piston (opposite the last stage location) is vented, normally to suction pressure. The differential pressure across the balance piston acts on the balance piston area to develop a thrust force opposite that generated by the impellers. The pressure on the

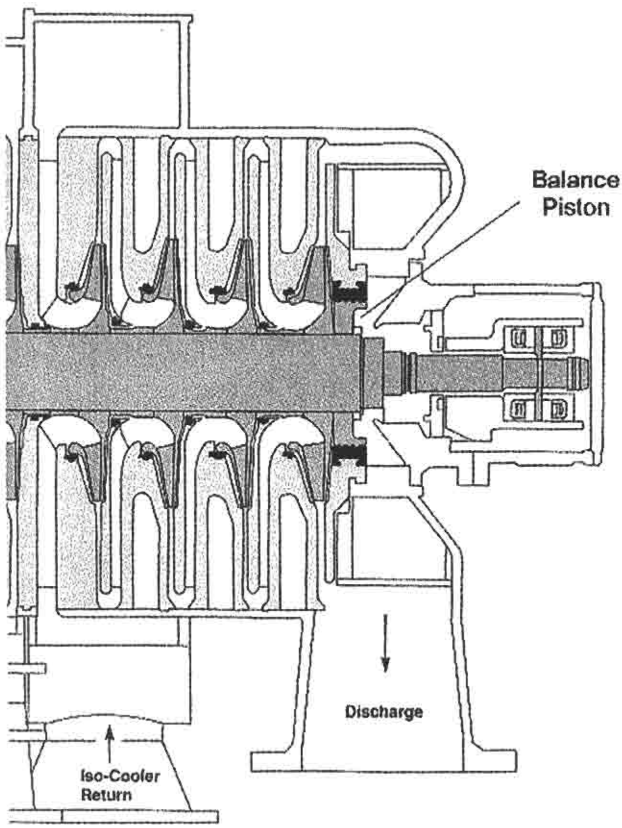
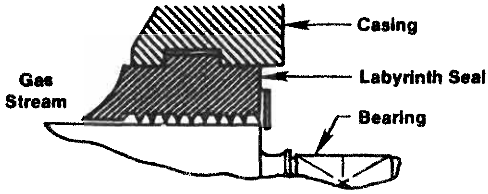


Figure 5-46. Balance piston. (*Courtesy of Elliott Company*)

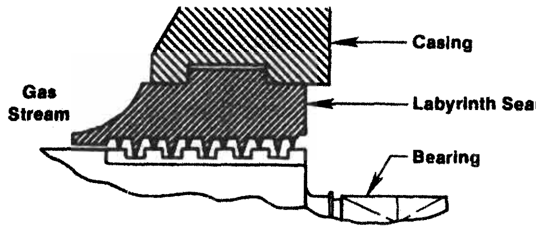
low pressure side of the balance piston is higher than the reference pressure by an amount equal to the resistance of the balance line, the line taking the flow from the low pressure cavity to the reference point. Line resistance, of course, is a function of the flow in the line. To permit efficient balance, an effective seal must be used at the rim of the balance piston because the leakage also represents parasitic power loss. In the earlier paragraph on sizing, a target value of 1% was used as a base value, recognizing that for higher pressure applications this value would tend to be greater. While full control of the thrust can be developed by controlling the diameter, limits are in order. Generally, the balance force is kept less than that developed by the impellers, with the thrust bearing taking the remainder of the load. This keeps the rotor on one face of the thrust bearing for all load conditions and is the recommended practice. An alternative philosophy overbalances the thrust with the balance piston, arguing that balance piston seal deterioration will unload the thrust bearing for more conservative design. The problem with this approach is that the rotor will tend to shift its operating position from one side of the thrust bearing to the other for varying loads and conditions. Because the thrust bearing has .012 to .015 inches of float, the rotor will not be in a fixed position, making instrumentation for rotor position difficult to judge. Also, oversizing the balance piston means a larger seal diameter, making the potential seal leakage greater. Besides the ramifications of the higher leakage, the method tends to be somewhat self-fulfilling in that the deterioration will tend to increase at a higher rate.

Interstage Seals

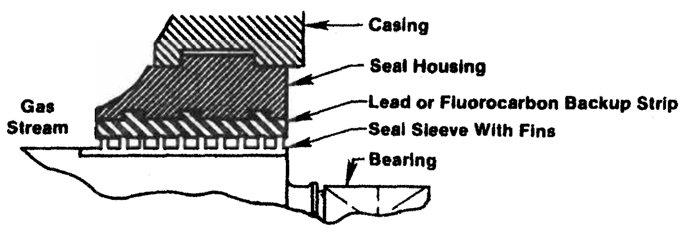
Interstage and balance piston seals of the labyrinth type are universally used in centrifugal compressor service. Multistage compressors are equipped with impeller eye seal and interstage shaft seals to isolate the stages. Figure 5-47 shows various labyrinth configurations. Labyrinth seals consist of a tooth-like form with spaces in between. Leakage is a function of both the tooth or fin clearance and the spacing. As shown in the figures, the fins can be stationary or rotating. The basic labyrinth design is the straight seal, where the teeth are at the same height. Another, rarely used for the interstage seal but frequently used for the balance piston, is the staggered or stepped form. When rotating seals are used, they can be machined integral on a sleeve or into the rim of the balance piston. Another type of rotating seal is constructed of a strip material with one edge rolled. The rolled edge is then caulked into a groove in the rotating



Straight Labyrinth Seal



Stepped Labyrinth Seal



Interference Labyrinth Seal

Figure 5-47. Three types of labyrinth seals. (*Courtesy of Elliott Company*)

element. The shape of the rolled edge gives the name *J-Strip*, which is sometimes used by the manufacturers who use the seal (see Figure 5-48). For use with the rotating seal, a soft backing surface is provided on the stationary surface opposite the seal. The backing can be lead, babbitt, or a stabilized fluorocarbon material. This arrangement allows the seal clearance to be set to a smaller value compared to the stationary finned seal with the objective of running the seal fins into the soft material to cut run-

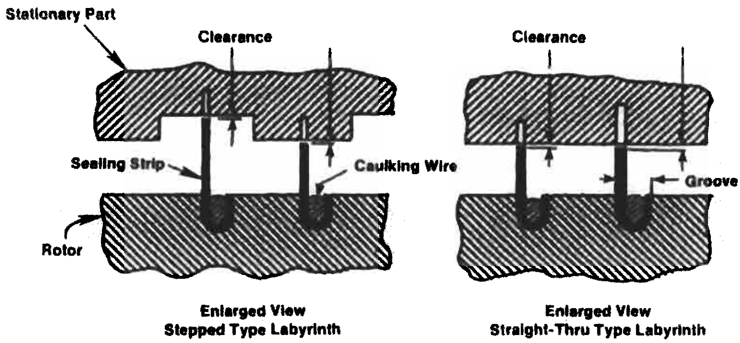


Figure 5-48. J-strip type labyrinth seal. *(Courtesy of A-C Compressor Corporation)*

ning grooves. While somewhat more expensive, the method does tend to keep leakage down and is particularly well-suited to fouling service; whereas, the stationary teeth would tend to fill and lose effectiveness.

When teeth are stationary, the material chosen for the labyrinth must be a relatively soft nongalling material, because the teeth tend to touch the shaft during upsets, startup, or shutdown. The clearance chosen must be large enough to avoid excessive rubbing yet close enough to control the leakage. If set too tight, the extra rubbing may cause the edges to roll and affect the performance. Overall, the stationary seal is simple and relatively easy to replace.

Shaft End Seals

Restrictive Seals

In controlling gas leakage, shaft end seals are either restrictive or positive in nature. The labyrinth seal is one form of restrictive seal. The reasoning for the labyrinth end seal is generally the same as discussed in the interstage seal section. A procedure for the calculation of restrictive seal leakage is given in Appendix D.

Another common form of restrictive seal is the *carbon ring seal* (see Figure 5-49). This seal consists of a series of carbon rings, using either solid or segmented rings. The segmented rings are enclosed with a retaining spring, called a garter spring. This seal, while somewhat more complex, is easier to replace than its solid counterpart. The carbon ring seal is able to operate with a close clearance, closer than bearing clearances, because the rings can move radially and the carbon acts to self-lubricate

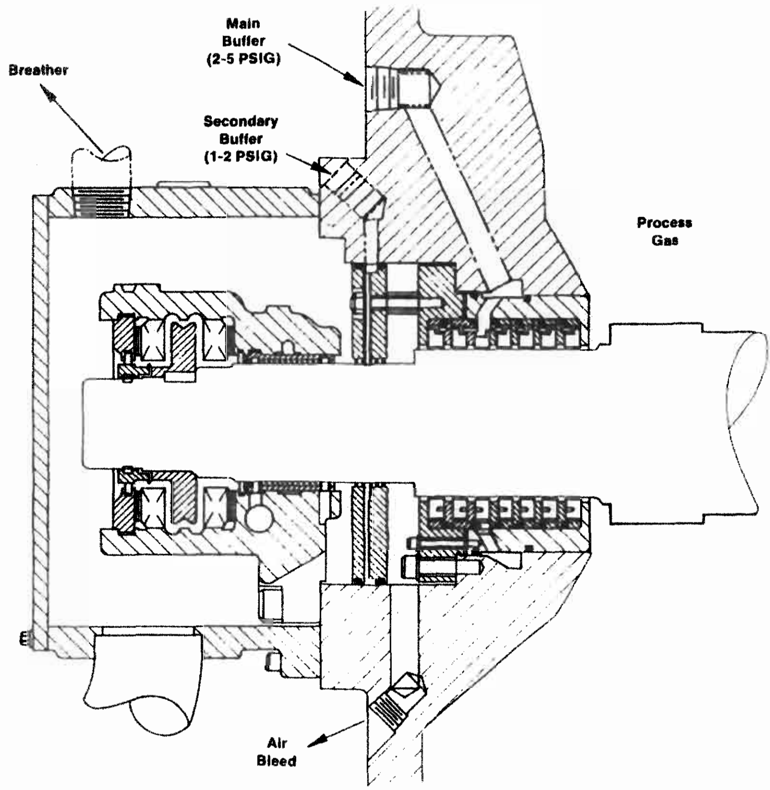


Figure 5-49. Dry carbon ring seal. The carbon rings on this seal are buffered by dry air. (*Courtesy of Elliott Company*)

when the seal rubs. Because rubbing does take place from time to time, the carbon ring tends to need more frequent replacement than the labyrinth. But for equal axial length, the carbon ring seal can be designed for leakage an order of magnitude less.

Liquid Buffered Seals

The positive seals are positive in the sense that the process gas is completely controlled, and in most applications, can be designed to avoid the loss of any gas, if the process gas and the sealing fluid are compatible to permit safe separation. In any event, the gas taken from the process is orders of magnitude lower than is the case for the restrictive seal. The positive seals take on the form of a *liquid film seal* or a *contact seal,* also known as the *mechanical seal.* The buffer fluid aids in the sealing process in the liquid film type and acts as coolant in both types. Each manufacturer generally has a proprietary form for one or both types of seal. Figures 5-50, 5-51, and 5-52 show the various seals available. The liquid film type operates with a close clearance and is used for high pressure applications. One modification of the liquid film seal uses a pumping bushing to control gas side leakage and, therefore, operates at bearing clearances (see Figure 5-52).

The contact seal can be used under 1,000 psig. It is more complex, but has the advantage of not leaking while shut down. The contact seal is used extensively in refrigeration service where the compressor is part of a closed loop, and the shutdown feature is desirable. As mentioned, the seals must have a source of cooling and buffer fluid. In many cases, this fluid is lubricating oil. If contamination is not a problem, a combined lube and seal system can be used.

Positive seals have been used in flammable and some toxic services. In toxic applications, an isolating seal must be included in the seal configu-

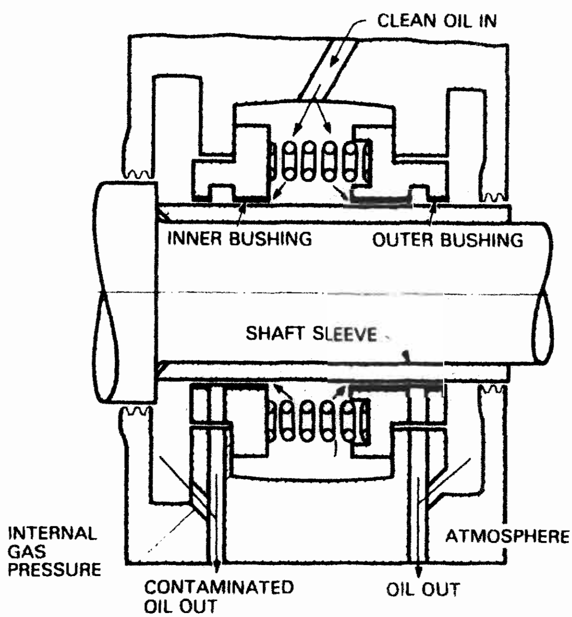


Figure 5-50. Liquid film shaft seal [12].

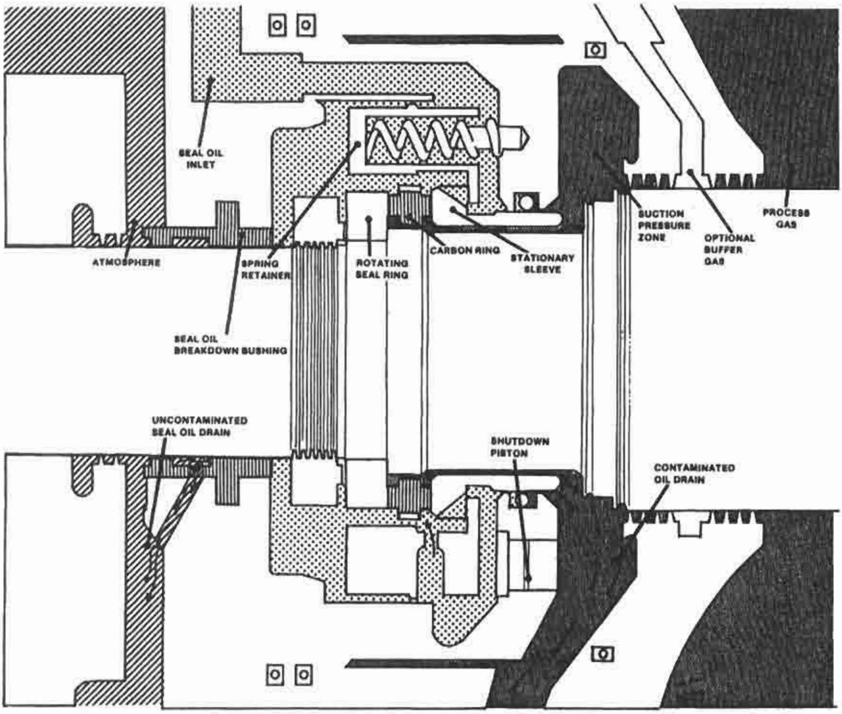


Figure 5-51. Mechanical shaft seal. (*Courtesy of Elliott Company*)

ration. By careful application, the isolating seal can also act as a backup to the primary seal.

In all situations, seals must function over the entire operating range, including startup and shutdown. If a compressor shuts down and is to be restarted hot after being down only a short time, the possibility exists of differential growth of the various components, closing the clearances to the point of seizure of the parts. The seal should be selected well inside its operating pressure range. With the liquid buffered seals, a value for the allowable leakage toward the gas side must be determined. This liquid is removed from the compressor by traps, referred to as *sour oil pots,* even when the fluid can be recycled. On small to intermediate compressors, the leakage flow should not be more than three to five gallons per day (gpd). Large compressors can have larger leakages, but should not average more than ten gpd per seal.

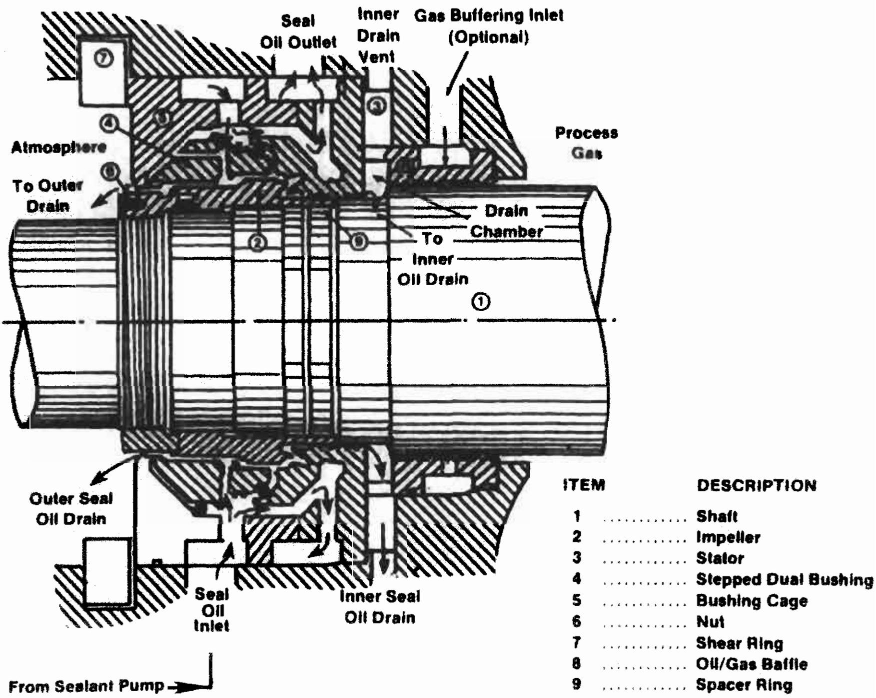


Figure 5-52. Liquid film type seal with pumping bushing. (*Courtesy of A-C Compressor Corporation*)

Dry Gas Seals

Dry gas seals are in the positive seal class and have the same basic design features as mechanical face seals with one significant difference. The dry gas seal has shallow grooves cut in the rotating seal face located part way across the face. The grooves may be in a spiral pattern; the exact location and pattern vary from one manufacturer to another. Lubrication and separation is effected by a microscopically thin film of gas. This implies some finite amount of leakage, which is quite small but must be accounted for in the design.

The seal unit located at each end of the multistage compressor rotor is installed as a cartridge. The cartridge has positive locating features to permit proper placement on installation. It normally includes a provision to ensure that cartridges are not interchanged from the intended end. This is to prevent reverse rotation on the unidirectional configurations.

Gas leakages range from less than 1 Scfm to 1 Scfm. The maximum rubbing speed is considered to be 590 fps. Operating pressures may range up to 3,000 psi. The temperature range using elastomers range from –40°F to 450°F. By using non-elastomers in the seal design, the temperature range is widened to –250°F to 650°F. From these values, it can be seen that the dry gas seal has a wide application range potential.

The dry gas seal has numerous advantages, but, as with most things in life, it also comes with some disadvantages. It is fair to state that for most of the applications, the good outweighs the bad and as such these seals are used extensively in the industry. However, each application should be evaluated on its own merits.

Probably the biggest single advantage of dry gas seals is getting rid of the seal oil. The seal oil system, even when part of a combined lube and seal system, is a complex assembly. With the dry gas seal, the lubrication oil system is all that is needed to service the compressor train bearings and, on turbine driven units, to also supply turbine control oil. As an aside, it makes feasible the dream long held by the compressor vendors of having a standardized lube system line.

Eliminating the oil gets rid of the disposal problem of the contaminated oil, which must be properly disposed of or cleaned up and recycled. It also eliminates the fouling problems in components downstream of the compressor. Despite all efforts to the contrary, oil from liquid buffered seals finds its way into the gas stream.

In most applications, the net loss of gas is less. The oil buffered seal loses gas both with the contaminated oil due to gas in solution and through the gas leakoff required to keep the various differential pressures in the proper orientation.

In those application where the cross-coupling effects from the oil seal were detrimental to the rotor dynamics, the use of the gas seal is a distinct advantage. However, the down side is that should the oil seal have provided a good measure of damping, the impact on the rotor dynamics is reversed. None of this is irreversible, but certainly must be kept in mind at the time of design.

As stated, the dry gas seal does come with its own set of disadvantages. The biggest of these is that the buffer gas must be reliable. Loss of buffer gas in some cases will reverse the differential pressure across the seal faces, which will damage the seal in short order. The seals will operate at a zero differential pressure level, but when possible, even a small differential in the proper direction is recommended by the manufacturers. Another disadvantage is the requirement for clean and dry gas at the seal faces. The issue of providing a dry gas supply to the seal is covered in Chapter 8. For dirty gas applications, a sidestream from the compressor discharge will have to be filtered and injected on the process side of the seal. Of course, all buffer gas must be filtered. A 2-micron nominal level is considered sufficient. While the requirement for cleanliness of the gas is a disadvantage, it is not unique to the gas seal as the liquid buffered seal, particularly the mechanical type, also has a relatively stringent cleanliness requirement.

One final negative comment: some of the dry gas seals are unidirectional. This is a problem for compressors that are subject to reverse rotation. It is a problem for using a common spare seal for a compressor, because the rotation makes a seal rotor end specific. For compressors prone to reverse rotation and for the spare parts concern, seals that are bidirectional are available. There may be a small leakage penalty. Other considerations are that the compressor bearings may not tolerate reverse rotation, making the seal limitation not the only factor. Also, though definitely not recommended, unidirectional seals have rotated in the reverse direction for short periods of time without any major problem. The best solution is to address the reverse direction problem itself. The negatives were pointed out only as a caution to the user. The dry gas seal advantages definitely outweigh the negatives and are a significant addition to compressor shaft sealing.

Seal configurations are single, tandem, and double opposed (shown in Figures 5-53 A, B, and C, respectively). The single configuration, as the name implies, is a single set of sealing faces with the leakage either flared or vented. The tandem seal, which is probably the most common, consists of two single seals oriented in the same direction. The first seal is considered a primary seal and handles full pressure, while the second seal, which is referred to as secondary, operates at near zero differential and acts as a backup to the primary. Figure 5-54 shows a tandem seal. The leakage is removed from between the seals, and either flared, vented, or recovered if the recovery system can maintain a relatively low pressure. In applications where it is undesirable to permit the primary gas to leak through the secondary seal, such as with hazardous gas, a baffle can be installed between the primary and secondary seal. An additional port is added to permit the injection of a secondary gas with inert properties. This secondary gas then flows through the secondary seal. A variation of the tandem seal is referred to as the triple seal, which uses a two-seal arrangement to break down the pressure. By design, the two seals divide the pressure drop approximately in half and use the third seal as a backup.

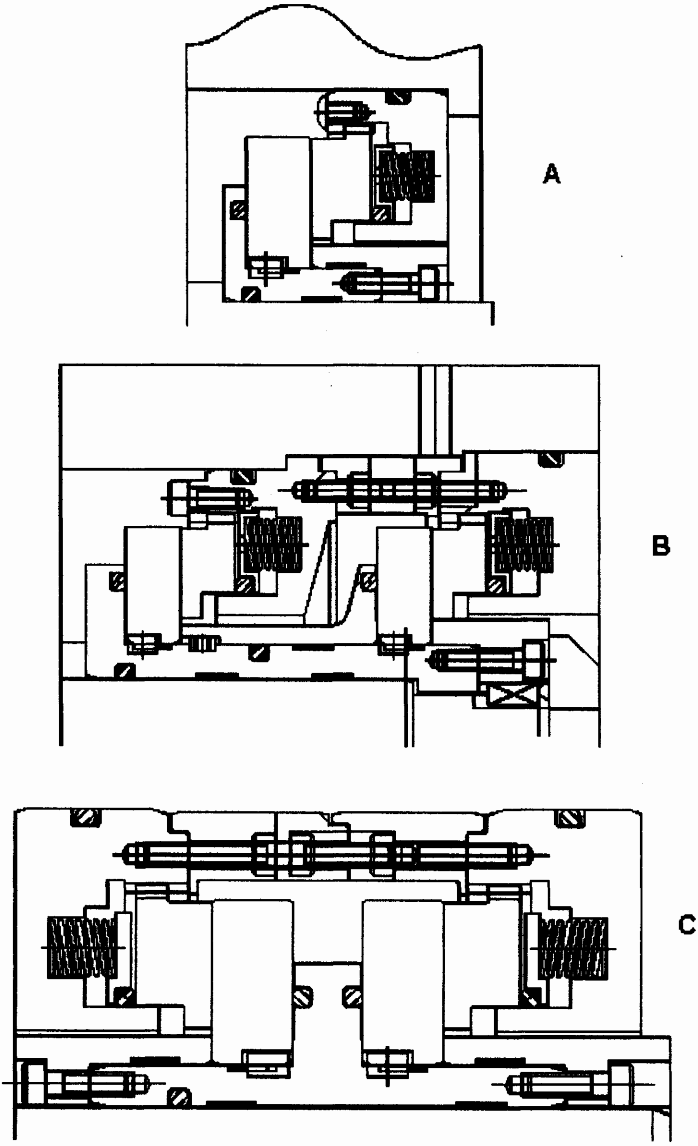


Figure 5-53. Section drawings of non-contacting dry gas seals: A. single seal arrangement, B. tandem seal arrangement, C. double opposed seal arrangement. (*Courtesy of John Crane International*)

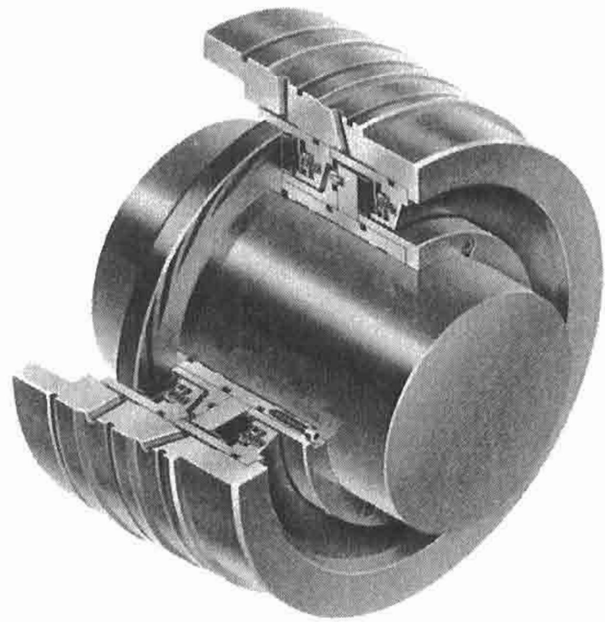


Figure 5-54. Cutaway of a tandem arrangement non-contacting dry gas seal. **(*Courtesy of John Crane International*)**

The double opposed seal is used in applications where a zero process leakage is mandated. The seal consists of two seal faces, with the process side seal reversed. An inert gas is injected between the two seals at a positive differential over the process gas pressure. A small amount of the inert gas leaks into the process. The process must be able to accept the contamination of the buffer gas for this seal to be used.

Dry gas seals use a separation seal on the bearing side of the seal as a barrier. The purpose of the barrier seal is to prevent lubricating oil from migrating along the shaft and into the dry gas seal. This seal also serves the purpose of preventing any gas leakage from the dry gas seal from leaking into the bearing cavity.

The barrier seals come in two basic forms. One is a labyrinth design, which is probably the most common. It has the features of a conventional labyrinth discussed earlier. The alternative is the carbon ring seal. The carbon ring is used either as a single ring or, in some cases, it is of a multiple ring configuration. In the latter, it is normally a double ring. The carbon ring may be split and use a garter spring around the outside segments or it may be one piece. The carbon ring has the advantage of being a lower leakage seal and uses less barrier gas. It also provides a more effective seal against oil migration. The carbon seal features in general were covered in the earlier discussions on carbon ring seals.

Capacity Control

Probably the most widely used capacity control for the centrifugal compressor is speed control. The capacity curve when used with speed control covers a wide range. While electric variable speed motors offer a continuation to the speed control practice, there are some other alternatives available. Suction throttling has been widely used and offers a reasonable control range for a relatively low cost.

A more efficient control available on some centrifugals is the *movable inlet guide vane*. The movable inlet guide vane adds pre-whirl to the gas stream entering the impeller, which, in turn, reduces the axial component of the absolute velocity, which controls the capacity to the impeller as discussed earlier. By modifying the inlet whirl component, the capacity is reduced with little loss in efficiency. It is obvious that this method would be most effective on the single-stage compressor (see Figure 5-55). It can be used on the multistage compressor, however, it can only be installed in front of the first impeller. If the compressor has more than a few stages or is more complex in arrangement the idea is not practical.

The guide vanes in a single-stage are generally pie shaped and center pivoted. They are located directly in the flow path immediately in front of the impeller. The shanks of the vanes extend through the inlet housing and connect to an external linkage. The linkage is connected to a power operator to supply the motive power to position the vanes. Control for the vanes can be by a remote manual station or connected to an automatic control as the final element.

In the multistage compressor, the vanes are rectangular and located in a radial position ahead of the first impeller, with a linkage connecting the vanes to a power positioner. From that point, the control is affected in the same manner as the single-stage.

The largest problem with the use of movable inlet guide vanes is the danger of the vanes or the mechanism sticking. Obviously, the vanes are not suitable for dirty or fouling gas service. The vane bearings and linkage should be buffered with clean, dry gas and exercised regularly. While this does take extra effort, the vanes will work well and give efficient control.

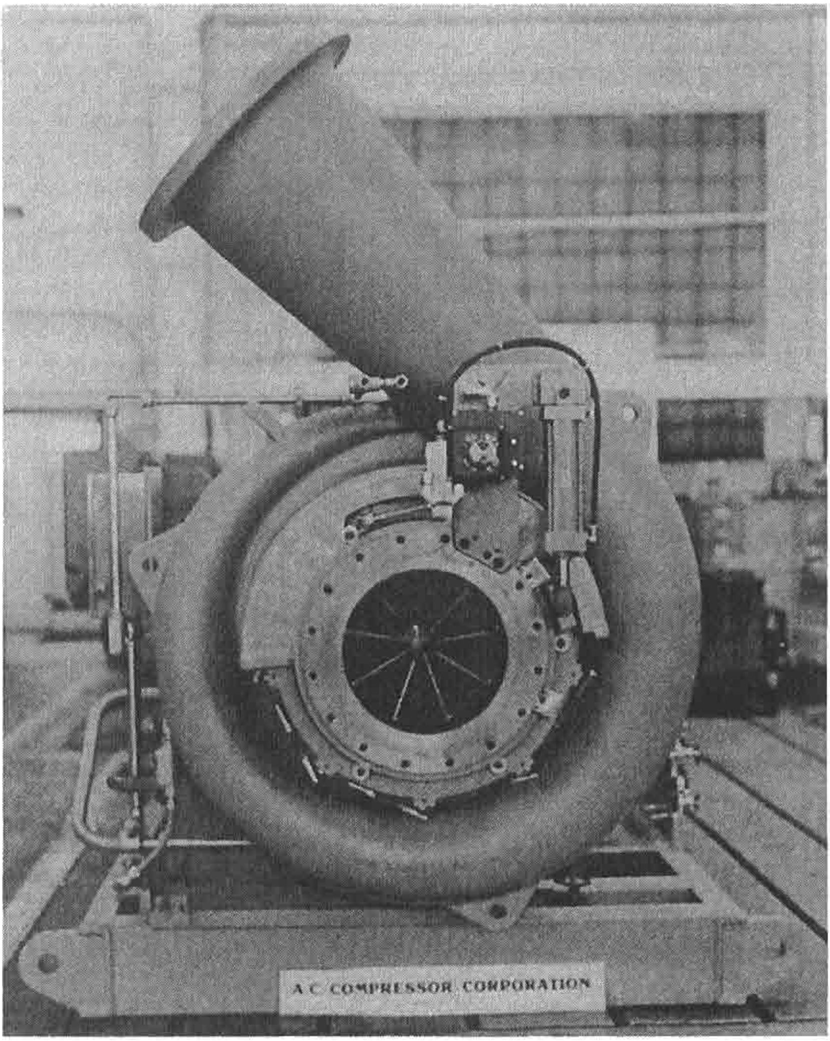


Figure 5-55. Single-stage centrifugal with movable inlet guide vanes. (*Courtesy of A-C Compressor Corporation*)

Maintenance

At the risk of misleading the reader, this section will just touch a few points concerning maintenance. One frequently asked question is, what is critical and what needs special consideration when performing maintenance on the centrifugal compressor? While there are many areas that must be carefully reviewed, the clearances should be restored as close as practical to the new machine values. Probably the single most important consideration, therefore, concerns concentricity. Interstage seal clearances should be concentric to the rotating element. It is better to allow larger than desired clearances in the machine than to leave seals in an eccentric condition. Leakages approach 2½ times the concentric values for eccentric seals, with the same average clearance.

While the axial position of multistage impellers to their diffusers is not critical, they should line up reasonably well. Impellers are not extremely sensitive to leading-edge dings and minor damage, but anything, such as erosion on the exit tips, that tends to decrease the effective diameter of the impeller is more serious. Front shroud clearance on open impellers should be maintained close to the design values to minimize capacity loss.

Bearings normally have a specified clearance range. Allowing clearances to exceed the specified maximum clearance may encourage the onset of rotor dynamics problems. Dams in dam type bearings are very critical. The edge of the relief must be square and sharp, not rounded. The clearance of this bearing is also quite sensitive and must remain inside the specified limits for stability.

Care must be taken in the assembly of the buffered shaft end seals, particularly in the area of the secondary *o-ring* seals. A cut or damaged ring can allow more oil to be bypassed than from a damaged main seal.

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6

Axial Compressors

Historical Background

The basic concepts of multistage axial compressors have been known for approximately 130 years, being initially presented to the French Academie des Sciences in 1853 by Tournaire. One of the earliest experimental axial compressors was a multistage reaction type turbine operating in reverse. This work was performed by C. A. Parsons in 1885. Needless to say, the efficiency was not good, primarily because the blading was not designed for the condition of a pressure rise in the direction of flow. Around the turn of the century, a few axial compressors were built using blading based on propeller theory. The efficiency was better but still marginal, achieving levels of 50 to 60%. Further development of the axial compressor was retarded by ignorance of the underlying fluid mechanics principles.

World War I and the interest in aviation gave rise to a rapid development of fluid mechanics and aerodynamics. This, in turn, gave a renewed impetus to axial compressor research. The performance of the compressor was considerably improved by the isolated air foil theory. As long as the pressure ratio per stage was moderate, the axial compressors were capable of quite high efficiencies. The compressor began to see commercial service in ventilating fans, air-conditioning units, and steam-generator fans.

Beginning in the 1930s, interest was increased in axial compressors as a result of the quest for air superiority. Efficient superchargers were necessary for reciprocating engines in order to increase power output and improve aircraft high altitude performance. With the development of efficient compressor and turbine components, turbojet engines for aircraft also began receiving attention. In 1936, the Royal Aircraft Establishment in England began the development of axial compressors for jet propulsion. A series of high performance compressors was developed in 1941 [1]. In this same period, Germany was doing similar research that ultimately produced several jet aircraft. In the United States, research was directed by the National Advisory Committee for Aeronautics (NACA). This was the forerunner of the National Aeronautics and Space Administration (NASA).

In the development of all these units, increased stage pressures were sought by using high blade cambers and closer blade spacing. Under these conditions the blades began to affect each other, and it became apparent that the isolated airfoil approach was not adequate. Aerodynamic theory was, therefore, developed specifically for the case of cascaded airfoils. In addition to the theoretical studies, systematic experimental investigations of airfoils in cascade were conducted to provide the required empirical design information.

While the aircraft-oriented research was going on in the mid '30s, commercial axial compressors were being built and installed in various process plants. The technology from the aircraft industry did not penetrate the commercial compressor business until 1958 when many of the NACA reports were declassified. Today, much of the commercial compressor design worldwide is based on the published NACA reports. An interesting comparison is to take similar applications from different time periods and look at the number of stages required to perform a given pressure ratio on air. For example, the '30s vintage compressor sized to compress air from 14.7 to 45 psia required 21 stages. If the '50s technology is applied to the same application, only 11 stages are needed.

Description

Axial compressors are high speed, large volume compressors but are smaller and somewhat more efficient than comparable centrifugal compressors. The axial compressor's capital cost is higher than that of a centrifugal but may well be justified by energy cost in an overall evaluation. The pressure ratio per stage is less than that of the centrifugal. In a general comparison, it takes approximately twice as many stages to perform the same pressure ratio as would be required by a centrifugal. The characteristic feature of this compressor, as its name implies, is the axial direction of the flow through the machine.

The energy from the rotor is transferred to the gas by rotating blades—typically, rows of unshrouded blades. Before and after each rotor row is a stationary (stator) row. The first stator blade row is called the guide vane.

The volume range of the axial starts at approximately 30,000 cfm. One of the largest sizes built is 1,000,000 cfm, though this size is certainly not common. The common upper range is 300,000 cfm. The axial compressor, because of a low pressure rise per stage, is exclusively manufactured as a multistage machine.

By far, the largest application of the axial compressor is the aircraft jet engine. The second most common usage is the land-based gas turbine, either the aircraft derivative or generic designed type. In last place of the applications comes the process axial compressor. All principles of operation are exactly the same. About the most obvious difference is that the gas turbine compressor is a higher pressure ratio machine and therefore has more stages.

Performance

Blades

While the process engineer does not need to understand the aerodynamic theory that makes the axial compressor perform, a few basics may be helpful. Acquaintance with the blading nomenclature and the fundamental velocity triangle may also keep the terms used around these compressors from seeming like a foreign language. The airfoil in most axials is of the NACA type, generally a 65 series. A typical designation is 65-(18) 10. The 65 defines the profile shape. The 18 represents a lift coefficient of 1.8 and the 10 designates a thickness/chord ratio of 10%. Figure 6-1 is a sketch of a blade cascade with various angles and profiles defined. The blade angles are designated as β. The *chord,* b, is the length of the straight line connecting the leading and the trailing edge. The *pitch,* s, is the measure of the circumferential spacing. *Solidity,* σ, is the relative interference, obtained by the ratio of b/s. *Camber,* is the curved mean line of the blade as is defined as the angular difference between the

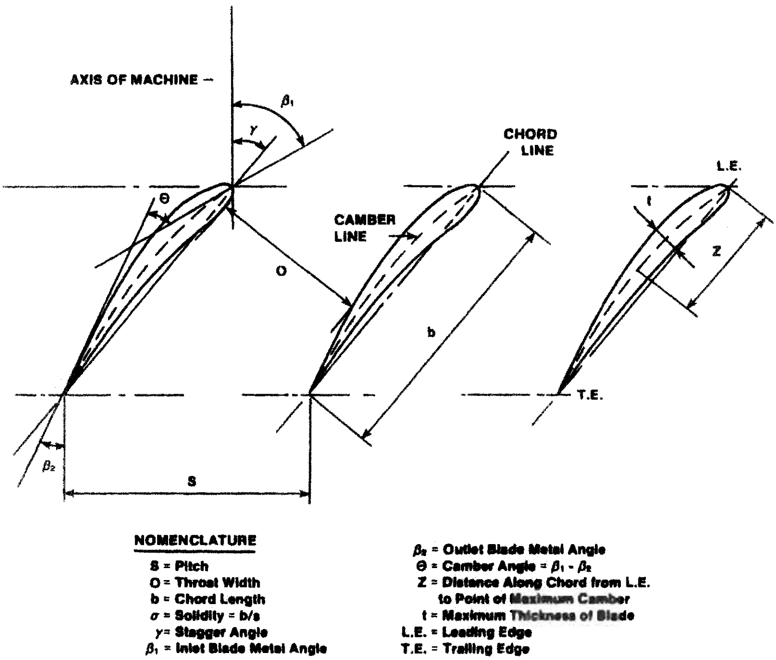

Figure 6-1. Airfoil nomenclature.

inlet blade angle β_1 and outlet blade angle β_2. *Aspect ratio,* AR, is the blade height, h, divided by the chord. *Stagger angle,* γ, is the angle between the chord line and machine axis. The gas angles are designated as α. Figure 6-2 gives a comparison of the various 65 series profiles. Note the increase in camber as the lift coefficient increases.

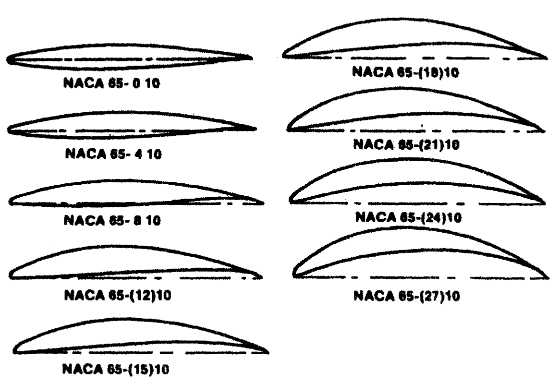

Figure 6-2. The NACA 65 series of cascade airfoils.

Compression Cycle

The gas enters the axial compressor through a guide vane row, then is acted on by the rotor, and moves on through the first stator row. From the stator, the gas again enters the rotor and exits to the next stator row. The gas continues its axial path until it has been acted on by all the rotor-stator pairs (stages). After the last stator, the gas may encounter an additional stator row or two, referred to as straightener rows. These stator blades take the whirl component out of the gas, so entry into the final diffuser or collector may be accomplished with a minimum of loss. Figure 6-3 shows a diagram of the blade path and the resulting vector triangles. Because of the finite length of the blades, the blade velocity, u, for the

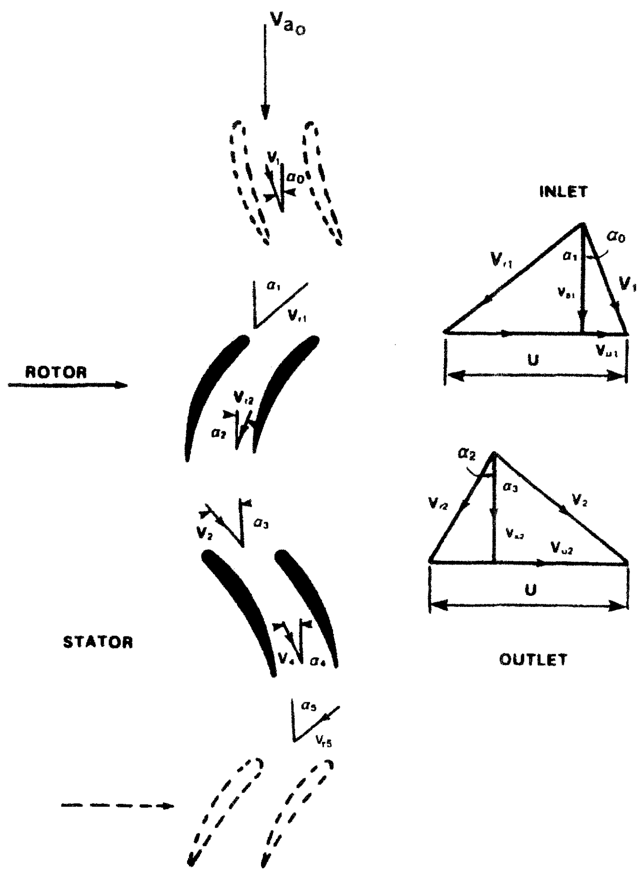

Figure 6-3. Velocity diagrams for an axial-flow compressor stage.

diagrams will be assumed to apply to the mean height of the blade. This simplifying assumption will be carried through most of the discussion because the three-dimensional flow path becomes quite complex. The absolute velocity is designated as V_1 and V_2. The relative velocity is given by V_{r1} and V_{r2}. The axial velocity is V_a.

Figure 6-4 is a diagram of the gain in energy of the gas shown by the enthalpy plot and the increase and decrease in velocity as the pressure is increased. While energy can only be added at the rotor blades, shown by

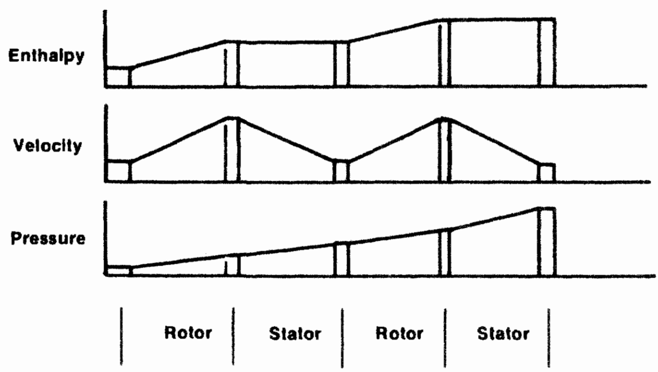

Figure 6-4. Variation of enthalpy, velocity, and pressure through an axial-flow compressor. (Modified from [4])

a horizontal line at the stator, the gain in pressure can be divided between the rotor and stator, as indicated by the steady increase in pressure as the gas moves from rotor through the stator.

Reaction

The degree of *reaction,* R, in the axial compressor is defined as the ratio of the static differential pressure in the rotor to the static differential pressure developed across the stage.

$$R = \frac{\Delta P_{\text{static rotor}}}{\Delta P_{\text{static stage}}} \tag{6.1}$$

The reaction can also be derived in terms of velocity components as given in the following relationship:

$$R = \frac{1}{u}\left(\frac{V_{ru1} + V_{ru2}}{2}\right) \tag{6.2}$$

$$R = \frac{1}{u}\left(V_{rum}\right) \tag{6.3}$$

Equation 6.3 shows that the degree of reaction can be seen directly on the velocity diagram as proportional to the whirl component of the mean relative velocity. These diagrams are helpful in comparing blade configurations, and with a further knowledge of the actual blade angles, a considerable amount of information can be obtained. Figure 6-5 shows a series of blade arrangements. These are all for the same blade speed, axial velocity, and change of whirl velocity. This, in effect, says that they have the same flow coefficient, pressure coefficient, and energy transfer.

Figure 6-5(a) shows the special case of axial outflow, associated with a single-stage fan with a stator row preceding the rotor. This case has no residual whirl velocity at the exit. As a multistage design, it offers the advantage of acceleration in the stator, since $R > 1$, which has the effect of smoothing out the flow and providing the best possible conditions for the rotor. However, it has the disadvantage of having a very high relative velocity, V_{r1}, and possibly a high Mach number. It is, therefore, unsuited for the first stage of the compressor, where V_a and u are high and the temperature is at its lowest, but may be more suited for the later stages where the Mach number may be lower.

Figure 6-5(b) is a case for a reaction of unity, that is, all the pressure rise is in the rotor, with the stator blades acting only as guide vanes to deflect the gas. A reaction of unity is aerodynamically the equivalent of $R = 0$ or impulse, as shown at Figure 6-5(f) since it corresponds to an interchange of the moving and fixed blade row.

Figure 6-5(c) shows the diagram for R between 0.5 and 1.0 for the special case of axial inflow associated with a single-stage fan with a stator following the rotor to remove the residual whirl velocity. This is necessary when the gas is drawn straight into the rotor. As a multistage design, this arrangement offers no special advantages or disadvantages. It will be seen again that from the aerodynamic viewpoint, it is the equivalent of R between 0 and 0.5, Figure 6-5(e).

Figure 6-5(d) shows $R = 0.5$ or the symmetrical case, in which rotor and stator are similar. $V_1 = V_{r2}$ and $V_2 = V_{r1}$. The degree of reaction is defined as

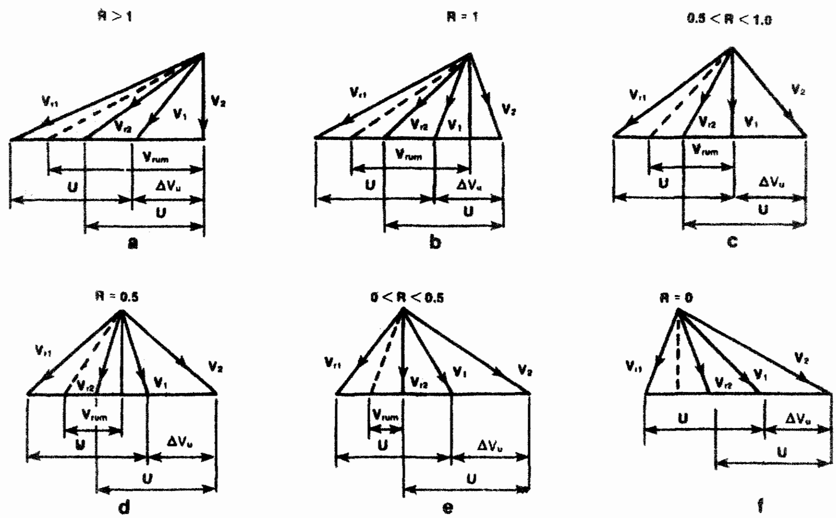

Figure 6-5. Velocity diagrams for various degrees of reaction in an axial-flow compressor stage. (***Reprinted with permission of Macmillan Publishing Company from*** **Principles of Turbomachinery** ***by D. G. Shepherd, Macmillan Publishing Company, 1956)***

$$R = 1 - \frac{(V_{u1} + V_{u2})}{2u} \tag{6.4}$$

From the symmetry $V_{u1} = V_{ru2}$, hence

$$R = 1 - \frac{V_{ru2} + V_{u2}}{2u} \tag{6.5}$$

substituting $u = V_{ru2} + V_{u2}$

$$R = 1 - \frac{u}{2u}$$

$$R = .5$$

Any effect of Mach number is experienced by rotor and stator equally and thus neither (or both) are limiting, and this Mach number will be lower than for other degrees of reaction under the conditions stated. If equal lift and drag are assumed in both rotor and stator, then optimum efficiency is obtained with $R = 0.5$ and $V_a/u = 0.5$. Although the latter is not always true, it does provide a useful criterion. Furthermore, the blade angles are similar in rotor and stator, which may be an advantage in the manufacturing phases in relation to tooling and inspection facilities. These factors help make the 50% reaction design a common choice by the compressor manufacturers when using the NACA 65 series blading.

It would seem that a high-reaction design would be desirable for the highest stage pressure rise and, therefore, the fewest number of stages for a given overall pressure ratio. Another factor must be considered however, and that is Mach number. The diagram for R = 1.0 shows that the relative inlet velocity to the rotor is considerably higher than for R = 0.5. The cascade data are for low speed (exclusive of Mach number). Therefore, the high reaction stage might be penalized by poor efficiency due to high Mach number, or alternatively, u and V_a would have to be reduced and thus the gain in energy transfer might be nullified. It is not possible to generalize further on degree of reaction, because the choice rests on the selection of velocities, Mach numbers, and effect of Mach number on efficiency. It may be reasonable, however, to state that for a high performance compressor, one in which high velocities and high efficiencies are required, a controlled diffusion airfoil may be used (see Figure 6-6). This type of airfoil has been made possible with the development of computational fluid dynamics (CFD). By using the controlled diffusion airfoil, the optimum reaction can be raised to a range of .6 to .7.

Stagger

For a given degree of reaction and value of V_a or flow rate, a choice can be made of the stagger angle or setting of the blades. From the consideration of two defining equations:

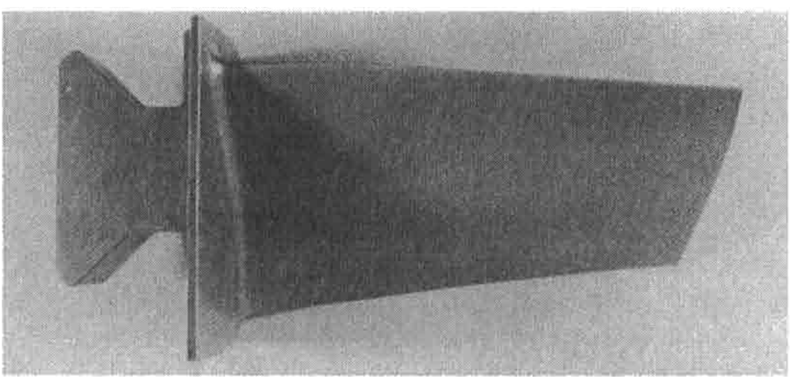

Figure 6-6. An axial compressor controlled diffusion airfoil. (*Courtesy of Elliott Company*)

$$R = \frac{V_a}{2u}(\tan \alpha_1 + \tan \alpha_2) \tag{6.6}$$

$$E = \frac{uV_2}{g}(\tan \alpha_1 - \tan \alpha_2) \tag{6.7}$$

where

E = energy transfer

For fixed values of R and V_a, a selection of a value of α_2 (corresponding to a value of stagger angle) fixes a limiting value of α_1 from cascade data. Thus the blade speed, u, is determined and then the energy transfer E is found.

To show the general effect of outlet angle or stagger, an approximate relation is used,

$$\tan \alpha_1 - \tan \alpha_2 = \frac{1.55}{1 + 1.5\,s/b} \tag{6.8}$$

where

s = blade pitch
b = blade chord

For the conditions of s/b = 1 and R = 0.5, the result is

$$\tan \alpha_1 - \tan \alpha_2 = 0.62$$

and from Equation 6.6,

$$\frac{V_a}{u} = \frac{1}{\tan \alpha_1 + \tan \alpha_2} \tag{6.9}$$

substituting the previous calculated value for tan α,

$$\frac{V_a}{u} = \frac{1}{0.62 + 2\tan \alpha_2} \tag{6.10}$$

As the stagger angle increases, for higher values of α_2, the optimum V_a/u decreases and thus for a given value of V_a, the blade speed, u, must increase. Also as α_2 increases, $(\alpha_1 - \alpha_2)$ decreases, hence high stagger implies high rpm and blades of low camber.

Another important factor in design is the steepness of the characteristic curve, that is, the variation of pressure ratio with mass flow (see Figure 1-3). From consideration of the velocity diagram for 50% reaction, such as (d) of Figure 6-5, it can be shown that the symmetrical arrangement gives

$$\tan \alpha_1 = \frac{u - V_a \tan \alpha_2}{V_a} \tag{6.11}$$

and, substituting in Equation 6.7, this produces

$$E = \frac{u}{g}(u - 2V_a \tan \alpha_2) \tag{6.12}$$

Differentiating with respect to V_a, and selecting the proportional variables

$$\left(\frac{\partial E}{\partial V_a}\right) u \propto (-\tan \alpha_2) \tag{6.13}$$

that show the energy transfer, E, which is head or pressure ratio change with respect to axial velocity, V_a, which is mass flow at a greater rate as α_2 increases. Therefore, high stagger blades tend to have a steeper characteristic curve. However, another feature of increased stagger that cannot be demonstrated simply, but requires use of cascade data, is that the design point or point of maximum efficiency on the characteristic curve is at a value of mass flow somewhat greater than that for maximum pressure ratio. As a result, the design point is further away from the surge mass flow and allows more flexibility on that part of the characteristic curve. Low stagger, on the other hand, tends to place the design point closer to the surge point. While this approach offers more flexibility for increased mass flow, some efficiency may be sacrificed to avoid operating too near surge.

Since for a given value of V_a the blade speed must be greater for high stagger, the energy transfer is increased because

$$E \alpha uV_a \tag{6.14}$$

and thus fewer stages may be required. Because the relative velocity is higher, the Mach number may be prohibitive, or alternatively thinner blades may have to be used in order to obtain a higher critical Mach number for the drag value. Thinner blades would require larger chords in order to maintain reasonable levels of bending stress. As a net result, the overall length would not decrease in proportion to the decrease in number of stages.

Earlier, it was stated, on the basis of simplifying assumptions, that the maximum efficiency for 50% reaction blading was obtained at a value of $V_a/u = 0.5$, requiring mean gas angles of 45°. The assumption for this result was that the drag-lift ratio was constant. In actual practice, cascade data indicate that drag-lift is not constant but increases as α_2 increases. It would appear that the maximum efficiency may be close to $\alpha_2 = 30°$. However, the reduction in efficiency is not severe because for values of α_2 of 15° and 45°, the drop is only about 1%.

Compressor performance can be changed by alteration of blade stagger. This is usually done by changes of stator stagger in preference of changes to the rotor. This can be accomplished on both process compressors and gas turbines, including the aircraft engine by use of variable stator vane control. While the mechanism is somewhat complex, it gives the axial compressor, with its inherently steep pressure-volume curve, the ability to be matched to a changing load without changing the speed. Figure 6-7 is a pressure-volume chart for an axial compressor with a partial stator vane control (only a portion of the stator blades is movable).

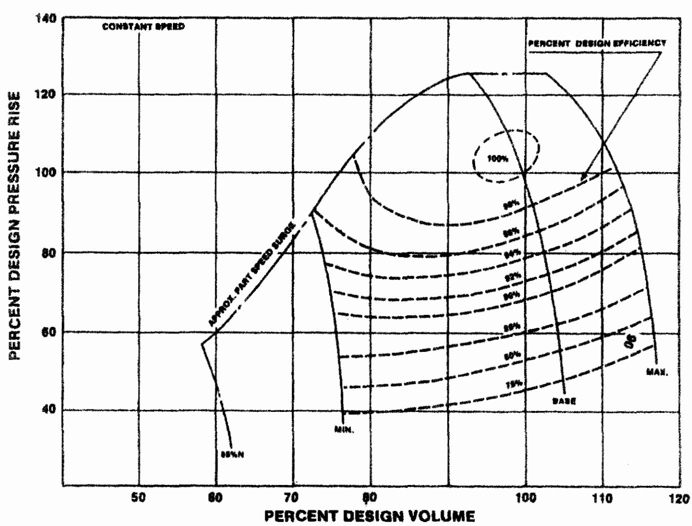

Figure 6-7. Pressure-volume chart for an axial-flow compressor with partial stator vane control. (*Courtesy of A-C Compressor Corporation*)

Curve Shape

The role of stagger, in developing the curve shape, was just covered; however, the shape deserves a few comments on comparison with other compressors. Even though it is a dynamic compressor, the axial's inherently steep pressure-volume curve makes it more akin to the positive displacement compressor shown in Figure 1-3. The steep curve contrasts with the flat curve of the centrifugal compressor. The horsepower characteristic also contrasts the shape of the centrifugal horsepower curve. Note that while the centrifugal's required horsepower increases with volume, the axial compressor's required horsepower does just the opposite (see Figure 6-8). This unique characteristic should be kept in mind when starting an axial. The axial is unloaded by opening a discharge bypass or otherwise removing the downstream load restriction during startup. This means that the lowest gas load is away from surge, compared to the centrifugal which may be unloaded by discharge throttling, which tends to bring the machine up to speed in surge.

The steep pressure-volume curve permits the axial compressor to operate very well in parallel with other axial compressors. The pressures do not have to match precisely to permit load sharing, as the steepness of the curve allows for adjustment without danger of going into surge or taking wild load swings as sometimes happens when attempts are made to operate centrifugal compressors in parallel.

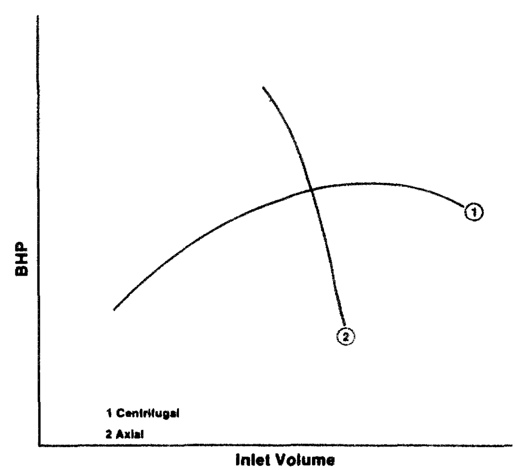

Figure 6-8. Typical brake horsepower vs. inlet volume curves for a centrifugal and an axial-flow compressor.

The axial compressor matches up well with a centrifugal compressor in a tandem-driven series connected arrangement, when higher process pressures are needed. The interstage matching is quite easy as the steep curve will provide sufficient changes in discharge volume to easily accommodate the requirement of the centrifugal. It can be thought of as stacking a constant ratio compressor on top of the axial's more vertical curve. With the axial compressor's high volume and speed attribute, the two machines will match in speed and volumes very neatly without the use of interstage gears.

Surge

In Chapter 5, this characteristic was applied to centrifugal compressors. The airplane wing analogy of stall was used, which is very directly applicable to the axial's airfoil-shaped blades. The incidence angle, described earlier, defines the onset of surge by stating that when the incidence exceeds the stall point, as developed from the cascade data, the foil ceases to produce a forward motion to the gas. When the gas cannot move forward, it moves in reverse, opposing the incoming flow. When the two collide, there is a noise, sometimes very loud. Recompression of the gas causes the temperature to rise very high very quickly. There have been cases where, when the blades were sufficiently strong not to break from the unsteady forces, they melted. It is more normal with prolonged surge to experience catastrophic blade breakage. The axial can also exhibit a phenomenon referred to as *rotating stall.* Rotating stall (propagating stall) is generally encountered when the axial compressor is started or operated too near the surge limit. This is especially true for compressor with adjustable vanes with the vanes in their extreme open or closed position. Vane movement is limited in some cases to minimize this problem. A flow perturbation causes one blade to reach a stalled condition before the other blades. This stalled blade does not produce a sufficient pressure rise to maintain the flow around it, and an effective flow blockage or a zone of reduced flow develops. This retarded flow diverts the flow around it so that the angle of attack increases or decreases on adjacent blades. These blades, with the increased angle of attack, stall and stay in a cell-like form. The cell then propagates around the stage or possibly two in which it occurs and at some fraction (40–75%) of rotor speed. Once begun, the cells continue to generate, causing inefficient performance, and if not terminated, may continue until a blade failure occurs. This is especially true if the cell's rotating speed coincides with the blade's natural frequency. Rotating stall is sometimes accompanied by audible pressure pulsation. Momentarily venting the compressor can inhibit cell formation. Besides the fact that the compressor tends to unload when taken to the maximum flow condition when starting, the additional problems are avoided.

Sizing

A short procedure will enable the reader to size an axial compressor. The complete and rigorous sizing of the axial is quite tedious and requires a comprehensive computer program. By making a few simplifying assumptions, a reasonable first approximation can be derived. While axials have been used rather extensively for gas service in their history, the bulk of the applications has been in air service. This is the first limitation to the sizing method, as it is only good for gases in the air molecular weight range eliminating the Mach number considerations. The head is assumed to be reasonably well divided over the various stages, and axial velocity is assumed constant throughout the machine. The numerical values are tabulated as follows:

Hub/tip ratio, $d_h/d_t = 0.7$ minimum, 0.9 maximum

Adiabatic efficiency, $\eta_a = 0.85$

Pressure coefficient, $\mu = 0.29$

Mean blade velocity, $u_m = 720$ fps

Because the constants and frame data include units, the relations presented here will depart from the primitive form used elsewhere in the book and will incorporate the necessary units.

Calculate an inlet volume, correct for moisture, if necessary, as outlined in Chapter 2. Select the frame size using Figure 6-9. It's probably a good idea to select the smallest frame that seems to accommodate the volume at the start. The frame size has been made easy as it is the hub diameter in inches. There have been five sizes presented, which should cover most ranges encountered, although the commercially available size range extends over a somewhat broader volume range. Select the number of stages, z.

Calculate the overall required head from the pressure ratio and the inlet temperature using Equation 2.70 from Chapter 2. It is repeated here for convenience.

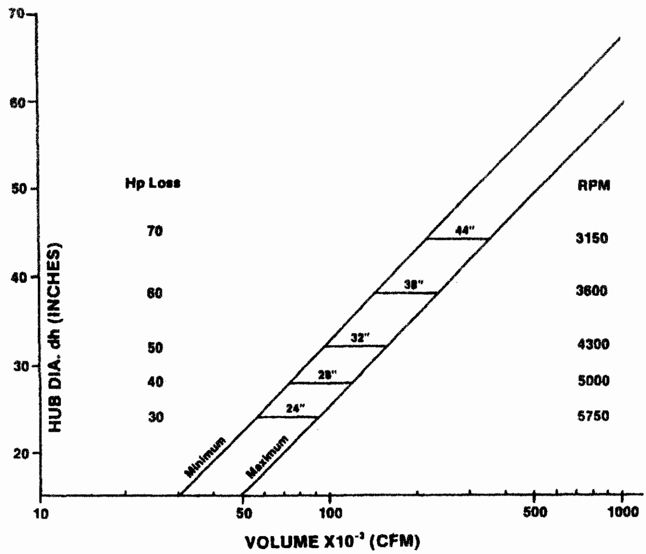

Figure 6-9. Axial compressor.

$$H_a = Z_{avg} \, RT_1 \, \frac{k}{k-1} \left(r_p^{\frac{k-1}{k}} - 1\right) \tag{2.70}$$

$$z = \frac{gH_a}{\mu \, (u_m)^2} \tag{6.15}$$

Round off the number of stages to the nearest whole number, then recalculate the pressure coefficient, μ, using

$$\mu = \frac{gH_a}{z \, (u_m)^2} \tag{6.16}$$

If the pressure coefficient just calculated is within 5 % of the original value of 0.29, then proceed using the calculated pressure coefficient and the assumed value of mean blade velocity as the final value. Continue to the speed calculation. If the pressure coefficient is higher than 5 %, add an additional stage to the compressor and again calculate the pressure coefficient using Equation 6.16.

If the pressure coefficient is now, or was in an earlier step, 5% under the 0.29 value, calculate a new mean blade velocity using the rounded-off number of stages and the original pressure coefficient, 0.29. Use the calculated blade velocity in the subsequent step for compressor speed. Calculate the speed.

In order to calculate the speed, a mean blade diameter must be established, and to calculate the mean diameter, a tip diameter is needed. The first step is to calculate the tip diameter.

$$d_t = [(6.05\ Q_1/u_m) + d_h^2]^{1/2} \tag{6.17}$$

where

d_t = tip diameter, inches
d_h = hub diameter, inches
Q_1 = inlet flow, cfm
u_m = mean blade velocity, fps

then calculate a mean diameter, d_m in inches:

$$d_m = \frac{d_t + d_h}{2} \tag{6.18}$$

Before proceeding, make sure the hub tip ratio is within the minimum limit of 0.67. If satisfactory, continue with the speed calculation. If the value is unsatisfactory, repeat the previous steps with an alternate frame choice:

$$N = \frac{229\ u_m}{d_m} \tag{6.19}$$

where

N = shaft speed, rpm

The speed must not exceed the speed given in Figure 6-9 for the selected hub diameter. Calculate the last stage volume using the following:

$$Q_{ls} = \frac{Q_1}{(r_p^{z-1/z})^{1/k}} \tag{6.20}$$

where

Q_{ls} = last stage inlet volume, cfm
r_p = pressure ratio across the compressor
k = isentropic compression exponent

Using Equation 6.16, calculate a stage tip diameter. Then check the hub tip ratio against the maximum value, 0.9. If the value is greater, the last stage blading is getting too short and probably the only solution is to use a smaller frame.

The guidelines presented are simplified and may not be sufficient for all applications. This does not mean that an axial cannot be used, because the vendors can perform a much more complex analysis and change factors that this simplified method chose to hold constant. Undoing some of these values is probably beyond the scope of most of the users. The best way to interpret a potential application is that an extra measure of care might be exercised when going out for bid. This can generate additional questions concerning the vendor's proposal.

To complete the sizing, calculate the discharge temperature using the Equation 4.6 from Chapter 4.

$$t_2 = t_1 + \frac{T_1 \left(r_p^{\frac{k-1}{k}} - 1\right)}{\eta_a} \times \eta_t \tag{4.6}$$

where

t_2 = discharge temperature, °F
T_1 = inlet absolute temperature, °R
t_1 = inlet temperature, °F
η_a = adiabatic efficiency

Calculate the shaft horsepower, using Equation 4.7 from Chapter 4. Read the mechanical losses from Figure 6-9.

$$W_s = \frac{w \times H_a}{33{,}000\,\eta_a} + \text{Mech losses} \tag{4.7}$$

where

w = weight flow of the gas in the compressor, lb/min.
H_a = total adiabatic head, ft-lb/lb

Example 6-1

Size an axial compressor for air service using the procedures outlined in the chapter. The following conditions are given:

Molecular weight:	28.65
Isentropic exponent:	1.395
Compressibility:	1.0
Inlet temperature:	80.0°F
Inlet pressure:	23.0 psia
Discharge pressure:	60.0 psia
Weight flow:	28,433.7 lb/min.

Step 1. Use Equation 2.5 to calculate the specific gas constant.

$R = 1{,}545/28.65$

$r = 53.93$ specific gas constant

Convert temperature to absolute.

$T_1 = 460 + 80$

$T_1 = 540°R$

$k/(k - 1) = 1.395/.395$

$k/(k - 1) = 3.53$

$(k - 1)/k = .395/1.395$

$(k - 1)/k = .283$

Step 2. Substitute into Equation 2.10 and, using the conversion constant of 144 in^2/ft^2, calculate the inlet volume.

$$Q_1 = \frac{1.0 \times 53.93 \times 540}{23 \times 144} \times 28,433.7$$

$$Q_1 = 250,000 \text{ cfm inlet flow}$$

Select the frame size (hub diameter, d_h) using Figure 6-9. At the inlet volume value, a 44 frame is selected with a maximum speed of 3,150 rpm. This frame has a 44-inch hub diameter.

Step 3. To calculate the number of stages, the overall head is required. The head is calculated using Equation 2.70 and $r_p = 60/23 = 2.61$ for the pressure ratio.

$$H_a = 1 \times 53.93 \times 540 \times 3.53\,(2.61^{.283} - 1)$$

$$H_a = 32,080.2 \text{ ft-lb/lb total adiabatic head}$$

Then using Equation 6.12 and the pressure coefficient, $\mu = .29$ and $u_m = 720$ fps given in the chapter.

$$z = 32.2 \times 32,080.2/(.29 \times 720^2)$$

$$z = 6.87$$

This value is rounded to the next whole number, 7. Recalculate μ using Equation 6.13 and the number of stages.

$$\mu = 32.2 \times 32,080.2/(7 \times 720^2)$$

$$\mu = .285$$

Since the μ is within 5% of the target value of .29, then use the mean blade velocity, $u_m = 720$ fps, as a final value and proceed with the sizing.

Step 4. Calculate the tip diameter using Equation 6.17.

$$d_t = [(6.05 \times 250,000/720) + 44^2]^{1/2}$$

$$d_t = 63.53 \text{ in. first stage tip diameter}$$

Check the hub-to-tip ratio, d_h/d_t. If greater than .67, continue. If not, go back to Step 2 and try another frame size.

$$d_h/d_t = 44/63.53$$

$$d_h/d_t = .69$$

This is greater than the stated limit, proceed to the next step.

Step 5. Calculate a mean diameter in preparation for calculating the compressor speed. Use Equation 6.18.

$$d_m = (63.53 + 44)/2$$

$$d_m = 53.77 \text{ in. mean blade diameter}$$

Calculate the speed using Equation 6.18.

$$N = 229 \times 720/53.77$$

$$N = 3066 \text{ rpm compressor speed}$$

The speed just calculated is less than the maximum speed of 3150 given for the frame and is therefore acceptable.

Step 6. Calculate the last stage volume using Equation 6.20.

$$Q_{ls} = \frac{250{,}000}{(2.61^{7-1/7})^{1/1.395}}$$

$$Q_{ls} = 138{,}698.9 \text{ cfm last stage volume}$$

Using Equation 6.14 calculate the last stage tip diameter.

$$d_t = [(605 \times 138{,}698.9/720) + 44^2]^{1/2}$$

$$d_t = 50.9 \text{ in. last stage tip diameter}$$

Check the last stage hub-to-tip ratio. It should be less than 0.9. If a problem is encountered with meeting this ratio, select another frame. If one frame exceeds the lower limit and the alternative choice exceeds the higher limit, multiple cases may be needed. The casing passing the lower hub-to-tip limit of .67 should be selected, except with the pressure ratio varied until the high limit of .9 can be met. The balance of the compression could be completed with a centrifugal compressor. In sizing a centrifugal compressor by the procedure outlined in Chapter 5, the speed of the axial can be assumed to be the centrifugal speed. This would permit a tandem drive arrangement. Cooling can be added, depending on the discharge temperature of the axial.

Calculate the last stage hub-to-tip ratio.

$$d_h/d_t = 44/50.9$$

$$d_h/d_t = .864$$

This value is less than the limit of 0.9. Proceed to the next step.

Step 7. Calculate the discharge temperature using the efficiency stated of .85 and Equation 4.6.

$$t_2 = 80 + \frac{540\,(2.61^{.283} - 1)}{.85}$$

$$t_2 = 278.2°\text{F discharge temperature}$$

Step 8. Calculate the shaft horsepower using Equation 4.7 and the mechanical losses from Figure 6-9 at the frame selection. Use the efficiency $\eta_a = .85$ as recommended.

$$W_s = \frac{28{,}433.7 \times 32{,}080.2}{33{,}000 \times .85} + 70$$

$$W_s = 32{,}589\text{hp shaftpower}$$

Application Notes

The axial compressor is a highly refined, sophisticated compressor. It is capable of very high efficiency, to the point that some of the designers feel there is no area of improvement left. As efficiency gets higher, the margin left between the ideal and current design makes each point much more difficult to achieve. Most of the development activity has centered on higher velocities, and the development of cascade data at the higher Mach numbers. The developments are more significant in aircraft engines where power-to-weight (size) ratio has a greater impact. The technology, however, is being applied to the land-based axial compressor. While the cost of the machine will somewhat follow the number of stages, and cost is probably one of the more significant factors retarding the application growth. The higher Mach number stages are more expensive to manufacture, somewhat offsetting the savings of having fewer stages. Gas turbines seem to be using the newer technologies as their size capabilities are increased.

Because the axial is a sophisticated compressor, it tends to show its "blueblood" at times, in lack of ability to cope with common plant problems such as fouling. The sophisticated airfoils, while capable of such nice high efficiency performance, have a real problem with dirt. It does not have to be polymers or other chemical reactions of the kind that cause problems with the centrifugal, but rather it can be ordinary atmospheric air. Some of the tendency to foul can be averted by changing the reaction at the expense of efficiency. This has not been completely successful, however, due to the complex modes in which fouling takes place. The best solution is filtration, which is attended by an increase in inlet pressure drop. The filtration should be of the dry type. Moisture, even a high humidity, can make whatever dirt does pass through the filter stick to the blading. On-stream washing has been successful in some cases, but must be carefully done and is somewhat of a trial-and-error method, until an operable mode is established. An alternative to washing is the use of organic abrasives. These have been reported as an effective and low-cost method of cleaning up this type of build-up [2].

Larger axial compressors have a physical space problem with the inlet nozzle, requiring a departure from the conventional round flanged nozzle customarily used in centrifugal application. This means either custom engineered rectangular duct work supplied by the user or an off-machine transition piece. For atmospheric suction compressors, where the inlet is connected to a nearby filter housing, this is not a serious problem.

Inlet startup screens have been recommended for other compressors covered in the earlier chapters. If the point has not been made yet, it should be with the axial compressors. Considering that most of the cost of the compressor is in the hundreds of vulnerable blades just waiting to be hit by some foreign object, it should be obvious that some protection is needed until the piping has been proven to be clear and clean.

Mechanical Design

Casings

The casings on axial compressors are somewhat unusual, because of the disproportionately large inlet and outlet nozzles. This makes the compressors appear to be only nozzles connected by a long tube. Casings can be fabricated or cast, with the fabricated obviously being steel, while the castings can be cast iron or cast steel. In some designs, the casing is an outer shell containing an inner shell, which acts as the stator vane carrier. In other designs, the stators are directly carried on the casing, which are of a one-part construction. With this latter design, the casing is made up of three distinct parts, bolted at two vertical joints. The parts are the inlet section, the center body with the stators, and the discharge section. The three sections are also split horizontally for maintenance. With the three piece-bolted construction, a mixture of fabrications and castings may be used. The mounting feet are attached to the outside casing and so located as to provide a more or less centerline support. As mentioned earlier, some designs use a rectangular inlet section to provide more axial clearance. The reason for using an entire separate casing is that it forms a separate pressure casing that can readily be hydrotested. The disadvantage is that there is more material involved making the cost higher. The compressors that use the integral stator section or single case approach have somewhat of a cost advantage, and in general, may have a slight advantage in being able to keep the stator carriers round, because of the end-bolting to the other casing components. The disadvantage is that there are more joints to seal and maintain. Checking out the entire casing for strength and leakage in a hydro and gas test becomes somewhat more complex.

Stators

As already described, the stators may be carried in a separate inner casing or may be carried by the outer, center section of the main casing. When movable stator vanes are used, the vanes pass through the wall of the carrier. The outer side of the casings exposes the shank ends, which are used as shafts to connect to the linkage that will control the movement. Because of leakage at the mounting bushings in the stator liner or carrier, the single-case unit has some sealing problems, which are inherently taken care of in the double-case construction. The single-case construction uses a lagging over the linkage, which can act as a collector.

For non-movable construction, some vendors use a conical shank and bolt arrangement. The stator vanes are set to a gauge at the factory and locked in place by tightening the hold-down nut and then staking. This construction permits flexibility for capacity adjustment at relatively low cost because the vane stagger can be reset without the manufacture of new parts or a complete machine disassembly. Other bolting arrangements are manufactured to give the same flexibility as the conical shanks. Some designs use a more permanent fixture for the stator vanes, setting the vanes in a diaphragm similar to the steam turbine or fixing the vane to the stator casing by dovetailing.

The stator vanes are usually not shrouded. This is not a hard and fast rule. In practice, often vanes are mixed with some shrouded and some unshrouded. When a separate stator inner case is used, it is normally of cast iron. If the temperature is expected to exceed 500°F, the liner will be multipart with the discharge end being made of steel.

Casing Connections

The casing inlet and outlet are flanged, for connection to the user's piping system, with the exception of some rectangular inlet connections as already mentioned. Communication with the manufacturer must take place concerning allowable forces and moments. While this is recommended for all types of compressors, the axial is a relatively small machine with disproportionately large piping, which results in the inherent potential for large forces and moments.

The balance chamber leakoff line, while recommended to be held within the confines of the compressor casing may well turn out to require some user piping. There are some situations where the desire for keeping open space around the compressor for maintenance may require compromise on the part of the user. The balance of the connections on the axial are for lube oil and other auxiliary equipment not different from that found on other compressors.

Rotor

Rotor construction tends to vary from vendor to vendor. The blades are attached to the outer surface of the rotor. The rotor may be of basic disc or drum type construction, with the disc type having some variations. The two most common disc construction modes are shrunk discs on a shaft and stacked discs, normally through-bolted together. A final method is the solid rotor construction.

When the disc construction method is used, the blades are attached by a single-lobe dovetail root design. The slots are broached into the rotor and the blade roots fitted into the slots and keyed in place. Figure 6-10 shows a bladed disc type rotor.

When the discs are of the shrunk-on design, they are made up individually and stacked onto the shaft by first heating the disc to dilate the bore. They are then allowed to cool and thus attach themselves to the shaft. Keys are normally not used. When the discs are of the stacked design, the discs are equipped with rabbet fit to radially lock the discs to maintain concentricity of assembly. The through-bolts are usually tensioned by stretching hydraulically to a precise value to ensure the mechanical integrity of the assembly.

An alternate method not used too much at this time is the drum design. The drum construction is somewhat different from the disc, in that the rotor body is of cylindrical construction. By using the hollow drum, conical roots, of bolted construction, can be used for the rotor, again, allowing for stagger adjustments to fine tune the axial compressor to the application if the need arises. The setting is done to a gauge at the factory, as with the stators.

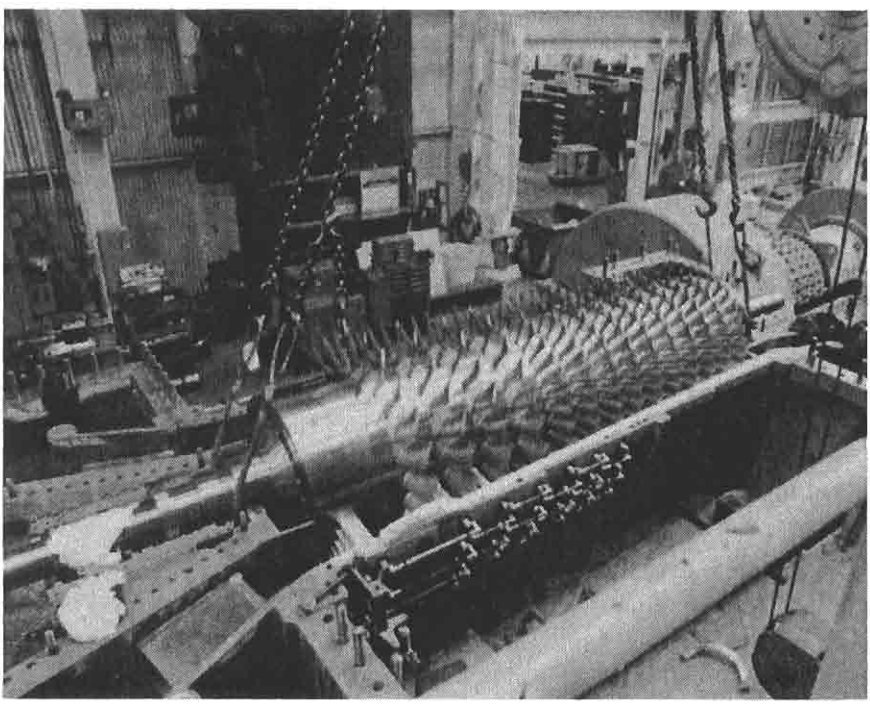

Figure 6-10. A 14-stage axial-flow rotor. (*Courtesy of Elliott Company*)

For smaller compressors, where the speed is relatively high and space is limited, a solid rotor construction is used. This is similar to the disc type of construction, except that the discs are an integral part of the rotor. Blade attachment slots are cut into the rotor, similar to the slots cut into the discs. Rotor blades are rarely, if ever, shrouded.

Rotor material in all cases is low alloy steel with an appropriate heat treatment to match the stresses imposed by the blades and rotor weight. The rotor is generally manufactured from a forging with the material being a chrome-molybdenum alloy such as AISI 4140 or AISI 4340.

Shaft

Shafting takes on several forms to match the various rotor construction methods. Obviously, for the solid rotor, the shaft is a part of the overall rotor. For the shrunk-on discs, the shaft is a continuous member, carrying the discs in the center section. Concentricity of all turns and good control on the roundness of the shaft are critical, if a balanced, smooth running compressor is to result.

The more unique form of shafting is used in the bolted disc and drum designs. Here, the *stub-shaft* design is used. A stub-shaft is fitted to each end of the center body, whether disc or drum. The design must anticipate all possible sources of stress, so the proper shrink can be applied to the interface. An interference fit is used at the interface to ensure concentricity for all operating conditions. It should include a reasonable allowance for momentary overspeed, particularly if a turbine driver is used. The design should consider the potential temperature transients that may be encountered at startup, shutdown, and hot restarts. Some arbitrary allowance should be made for torsional transients, even when not ordinarily anticipated. The shaft material, for the separate shafting, can be made of a different material than that used for the rotor body, although there is not much reason to do it that way. The heat treatment used could be different without compromising the overall rotor.

Blading

Axial compressor blades are usually forged and milled. Precision casting has been used on occasion. The most common material used is a 12 chrome steel, in the AISI 400 series, and is also known as 400 series stainless steel. While the stator blades are occasionally shrouded, the rotor blades are free-standing. Lashing wires have been used on rotor blades, but are generally used to solve a blade vibrational stress problem.

In a new compressor, the wires should not be used because the stress problems should be solved in a fundamental design manner without having to resort to "fixes." When the manufacturer designs the blades, the vibrational characteristics for both rotor and stator blades should be established. The basic bending resonances and the higher orders to which the blades may be excited should be established. Care should be taken to avoid any direct excitation sources, such as splitter vane or stator and guide vane passing frequencies. If possible, any fundamental and lower order resonances should be at least three to four times higher than any of the running speed. The manufacturer should supply a *Campbell diagram* to demonstrate that the compressor is free of direct excitations. When resonances do exist in the operating range, the vendor should demonstrate his understanding of the stress level and provide some assurance to the user that the compressor will not have premature blade failures. One method to convey the information is with a *Goodman diagram.* Some users ask for Goodman diagrams for all stages, regardless of any resonant interferences, to demonstrate that a conservative design concept was used throughout the blading design. While stress levels in the rotor blading are by far the most severe, resonances can occur in the stator blades that have been known to fail when excited by one of the compressor's operating frequencies. This is a rare occurrence, but must still be considered. Most reported failures are caused by rubs or foreign object damage. Regardless of the cause, if a stator blade should break and drop into the gas path where it can be struck by the rotor blades, the wreck is just as traumatic as a direct rotor blade failure.

While much has been said about the axial compressor that would give the impression that the machine is not durable, nothing could be farther from the truth. The compressor has logged hundreds of thousands of hours in trouble-free operation, and will do so if properly designed and operated with reasonable care. It may not be quite as abuse-resistant as the centrifugal. However, as the efficiency and performance of centrifugals is upgraded they tend to become less abuse-resistant. The object of any operating group should be a conscientious effort to properly operate and maintain the equipment so that everyone will benefit.

Bearings

The bearings used in axial compressors are the same journal and thrust type used in the centrifugal compressor. Refer to Chapter 5 for a complete description of these bearings.

For axial compressors, the journal bearings are of the plain sleeve type for the larger, slower speed compressors. They are of the tilting pad type for the smaller, higher speed machines. The sleeve bearing is normally housed in a spherically seated carrier. The bearings require pressure lubrication as do most of the other compressors.

The thrust bearing is generally the tilting-pad type bearing. Most vendors apply the recommendation that the thrust bearing be of the symmetrical design with leveling links. Axial compressors have a high inherent thrust load, so the thrust bearing is quite important in the overall reliability of the compressor. While this is true for the other compressors as well, it deserves an extra emphasis here.

Balance Piston

The axial compressor is inherently always a reaction type of machine. In regards to axial thrust, this means the rotor is subjected to a differential pressure across each rotating blade row. The differential pressures convert to an axial force at each rotor row that totals to a rather high value when taken over the normal number of stages. A thrust bearing would be prohibitive in size to carry the generated thrust. Fortunately, the geometry of the axial provides space for a large balance piston at the discharge end of the compressor. In fact, the construction of the axial compressor rotor is such that the placement of a labyrinth seal on the hub diameter and another labyrinth seal on the shaft forms a balance cavity. The balance piston cavity is normally vented to the suction end of the compressor or to the atmosphere on air machines. The balance piston seal leakage is charged to the compressor as a loss. As in the centrifugal, the return gas represents a head loss due to the heating effect of the return gas, and a direct loss in capacity due to the quantity of gas bypassed.

Seals

Because most axial compressors are in air service, most are equipped with labyrinth type end seals. There are no interstage seals in the machines with unshrouded stator blades. The balance piston seal, a labyrinth type, is the only internal seal. There is no reason that axials cannot use some of the other seals as described in Chapter 5, such as the controlled leakage or the mechanical contact type, if the gas being handled by the compressor needs a more positive seal. If there is any problem at all, it will be that the seal rubbing velocity will be higher due to the larger shaft and relatively high speed. In the past, where axials have been used in process gas service, these problems were overcome.

Capacity Control

Earlier in the chapter, the movement of the stator blades to change the capacity of the axial compressor was discussed. In conjunction with extended shanks and linkage, as mentioned, the axial offers a very good dynamically controlled capacity range. Depending on the range desired, linkage may be installed on the guide vanes only or on the guide vanes and stator vanes starting from the front of the compressor. Control has been used on up to 100% of all the stator vanes; however, the range gained after 50% stator control is generally marginal and not worth the extra expense. The linkage in Figure 6-11 is typical. The linkage is normally connected to one or more power cylinder operators for manual operation from a remote location, or for use as the final control element in an automatic system. Use in an automatic control system is good, particularly if there is a little noise on the control signal. This acts to keep the vanes *live* and the linkage and bearings from fouling and sticking. The sticking of the linkage is probably the largest single problem with the vane control. For manual systems, a regularly scheduled linkage exercise can help keep the movement free. Use of dry clean gas purges are also helpful. The range of the axial is enhanced considerably by the use of movable vanes.

Maintenance

Most of the maintenance ideas presented in Chapter 5 are also applicable to the axial. Concentricity of the rotor clearances in the stator is again stressed. Rotor tip clearances allow a reasonable tolerance. A 10% clearance change is not greatly significant.

When an axial is taken out of service, the blading should be carefully cleaned and inspected. If minor damage is noted, the blade should be repaired or, if the damage is significant, it should be replaced. In most cases, minor, foreign-object "dings" or cracks can be ground out and blended. If several blades in a row show cracking, a note of the accrued operating time of that row should be made and the manufacturer consulted to ensure that a more serious problem isn't beginning to show. With

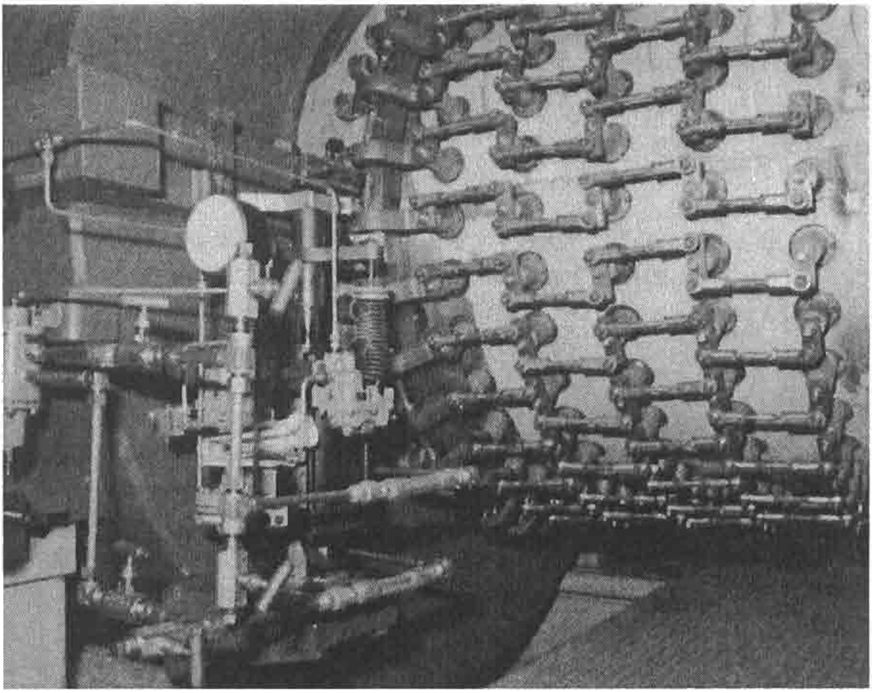

Figure 6-11. Adjustable stator blade linkage. (*Courtesy of Elliott Company*)

reasonable operation, not too many hours in surge, or no gross upsets, blading should last 50,000 hours or more. If no cracks are detected, blades should be randomly selected and examined by one of the NDT methods. Should any cracks be detected, very strong consideration should be given to replacing the entire blade row in which the cracked blade was located.

If movable stator blades were used, the linkage should be checked for binding and wear. If a clean gas purge was used to keep the linkage clean and dry, the source and supply lines should be inspected to make sure they all work as intended when the compressor is restarted.

Beyond these items, the balance of the maintenance procedure should include the customary bearing and seal checks.

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7

Drivers

Introduction

Drivers for compressors must supply torque of a specified value at a certain speed. Whenever the driver speed characteristics are not directly useable, they may be modified by a speed increasing or reducing gear. The only exceptions to the speed-torque criteria are a limited number of the reciprocating compressors. These are the integral engine and the direct steam cylinder driven machines. The balance of the reciprocators align themselves with the other compressors, receiving their input energy by way of shaft torque at a given speed.

The driver is a prime mover capable of developing the required torque at a constant speed or over a range of speeds. The driver's energy source can be either electrical or mechanical. Electrical energy is used by motors, either of the induction or synchronous type, while the mechanical covers a multitude of sources. It may be a fuel, as in internal or external combustion engines, or it may be a gas, such as steam or process gas used in a turbine or expander.

This chapter will describe all common compressor drivers, but as a practical consideration, details on selection or sizing, hazardous area applications, and installations will not be covered. For additional information refer to the National Electrical Code [1] and API publications RP 500 and RP 540 [2, 3] as well as some of the general references given.

Electric Motors

Modern technology has reduced the size of motors, increased their expected life and improved their resistance to dirt and corrosion. Other important developments of the last 30 years are brushless excitation for synchronous motors and new two-speed, single-winding, induction motors.

Long-term cost of ownership is the prime factor in the selection of any equipment. Selection should be based on the least expensive, most efficient motor that will meet the requirements. Improper selection increases operating costs. Oversized motors are commonly purchased either because the actual load requirements are not known or because of anticipated load growth. It is important that the motor be sized for all known design and offdesign operating conditions. Gross oversizing is wasteful because motors perform best and cost less to operate (maximum power factor and efficiency) at the manufacturer's rating. The best checks against improper size are careful review of drive requirements prior to purchase and periodic checks of the individual motors in operation.

Serious consideration should also be given to enclosure selection. Many improvements have been made in recent years in both enclosures and insulation. Therefore, it is important to review purchasing practices to make sure they are based on today's technology.

Review the motor requirements and specifications to make sure that all the unnecessary, nonstandard, special features have been eliminated. Each special requirement such as nonstandard mounting dimensions and nonstandard bearings should be eliminated unless it can be demonstrated the feature is cost effective. In actual practice, many special features are specified because of an isolated case of trouble that occurred years ago. Likewise, some special features may become obsolete through changes in refinery or chemical plant practice or through improved manufacturing techniques.

Synchronous and induction motors cannot always be compared on an equal speed basis. In geared applications such as high-speed centrifugal compressor drives (above 3,600 rpm), the most economical induction motor speed is usually 1,800 rpm. The most economical synchronous motor speed for the same application might be 900 or 1,200 rpm, depending on the horsepower required. For compressor drives above 3,600 rpm, motor prices must be studied within the total evaluation concept.

For 3,600-rpm compressor drives below 5,000 hp, simplicity of installation almost dictates using the two-pole induction motor. No gear is required, and the overall electrical and mechanical installation is the simplest possible.

Electric motors above 5,000 hp are custom-designed for the specific application, taking into consideration compressor characteristics and the power system parameter limitations.

While logistics favor the use of the two-pole motors, there is a break point in mechanical design between the four-pole and two-pole motors. Because opinions vary concerning the desirability of two-pole motors, those contemplating their use should first investigate the "track record" of the motor supplier in the required size range.

A 5,000 hp break point has been used rather arbitrarily, as larger motors are built, ranging in size to 30,000 hp. Generally as the size increases, the synchronous motor becomes more competitive. However, final selection is not only dictated by the driver economics above, but includes the power system as well.

Voltage

The least expensive motors, typically below 200 hp, are low voltage (less than 600 volts). Above 500 hp and up to 5,000 hp, the preferred voltage is 4,000 volts for use on a 4,160 volt system. It should be noted that motor nameplate voltages are commonly about 5% less than the nominal voltage of the system they are used on. However, synchronous motors that may operate at a leading power factor, in general, have a nameplate voltage equal to the nominal system voltage. As the motors become larger, coil capacity becomes more of an issue and increased space is available for insulation, so it becomes economically practical to use higher voltage motors and controls. It is apparent that, as the size of a refinery or chemical plant increases, the distribution system increases, and it is necessary to go to higher system voltages. For distribution system voltages above 5,000 volts, the usual practice is to transform down to 2,400 or 4,160 volts for large motors. Cable and transformer costs frequently make it economical to select motors with higher voltages such as 13.8 kilovolts. Numerous 11 to 14 kV motors are now in service, even outdoors.

Lightning and switching surges that damage motors are related to high-voltage motors. The insulation level of motors is below that of many other types of apparatus such as transformers, switchgear, cables, etc. Because of the low insulation level, the system lightning arresters will not protect the motors adequately. Special surge protection equipment is often needed, particularly for large motors with long cores. Surge protection consists of special low-discharge voltage arresters in parallel with surge capacitors. The capacitor slopes off the wave front to reduce the motor winding turn-to-turn voltage, and the arrester limits the voltage rise to a safe value. Surge capacitors are used without the arrester in some applications To give maximum protection, the arrester should be mounted at the motor terminals.

Enclosures

The enclosure selected affects cost substantially. Corrosive and hazardous atmospheres encountered dictate the protection needed.

Today's standard motor enclosure for indoor applications is the open, drip-proof enclosure for induction and high-speed synchronous motors. For large motors, open, drip-proof construction is available up to about 20,000 hp and is used for squirrel-cage, synchronous, and wound-rotor motors.

For larger motors, the next degree of protection is weather-protected, Type I (WPI), which is an open machine with ventilating passages constructed to minimize the entrance of rain, snow, and airborne particles to the electric parts.

Where a higher degree of protection and longer life is desired, weather-protected, Type II (WP II) motors are recommended. This type of enclosure is used on the motor-driven compressors shown in Chapter 8, Figure 8-15. They are equipped with extensive baffling of the ventilating system so that the air must turn at least three 90° corners before entering the active motor parts. Maximum air-intake velocity is limited to no more than 600 fpm, and blow-through provisions are also provided. In this way, rain, snow, and dirt carried by driving winds will be blown through the motor without entering the active parts. The WP II are the predominant choice for large outdoor motors in petrochemical applications in North America.

Totally Enclosed Motors

The choice for severe applications below 250 hp is the totally enclosed fan-cooled (TEFC) motor (see Figure 7-1). TEFC motors separate the internal and external ventilating air. Breathing is the only way external air ever gets inside. At some size above 250 HP, manufacturers presently switch from a rib-cooled to an exchanger-cooled motor for totally

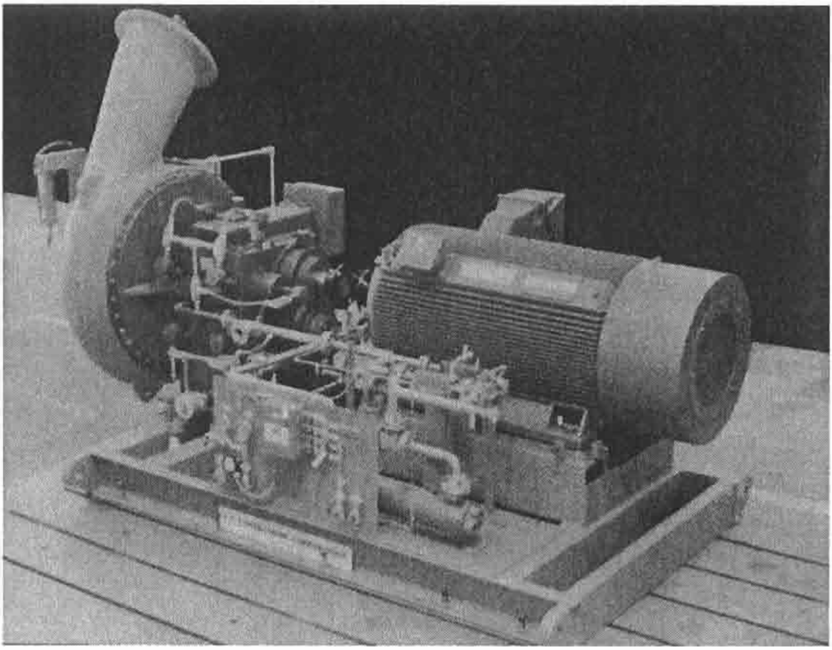

Figure 7-1. A direct-connected, TEFC two-pole motor driven centrifugal single-stage compressor. (*Courtesy of A-C Compressor Corporation*)

enclosed fan-cooled (TEFC) applications. These exchanger-cooled motors are often referred to as totally enclosed air-to-air cooled (TEAAC) motors. Fin-cooled TEFC motors are becoming available in ever increasing horsepower, and are presently available from some manufacturers at sizes up to 2000 hp. These appear to be the preference of many users for the most severe applications. Above 500 hp (approximately), totally enclosed motors with water-to-air coolers (TEWAC) cost much less than TEFC motors and, in large ratings of synchronous motors, may cost even less than WP II. Large synchronous machines are frequently enclosed and supplied with coolers for mounting in the motor foundation at lower cost than integral-mounted coolers.

Totally enclosed motors offer the highest degree of protection against moisture, corrosive vapors, dust, and dirt. TEWAC motors also have the advantage of reduced noise level.

Division 1 Enclosures

Explosion-proof motors can withstand an internal explosion without igniting a flammable mixture outside the motor. These motors are totally enclosed, fan-cooled, and specially machined to meet Underwriter's Laboratory Standards. Explosion-proof squirrel-cage motors are available up to 3,000 hp at 3,600 rpm, but the larger sizes are usually not practical or economical.

Force-ventilated (F-V) motors are suitable for hazardous locations. Safe air is brought in through a duct system, passed through the motor, and then discharged preferably through another duct system to the limits of the hazardous location. The ventilating ducts should be pressurized to prevent entrance of contaminated air. This indoor construction has been largely replaced by outdoor motors such as weather-protected or totally enclosed types. It might be noted that while the F-V is normally used indoors, there are exceptions where it has been put in outdoor service.

Inert Gas-filled

Inert gas-filled motors can also be used in refineries and chemical plants, but their applications are limited. They have tightly fitted covers and oil seals around the shaft to minimize gas leakage, are continually pressurized with an inert gas or instrument air, and are equipped with an internal air-to-water heat exchanger. Inert gas-filled motors are suitable for any hazardous location but require auxiliaries such as cooling water, gas pressurizing system, and control accessories.

Insulation

Electrical insulation is continually being improved. The motor manufacturers make use of this and other technological developments to put more power into smaller, lighter, more efficient packages. Modern insulating materials can withstand heat, moisture, and corrosive atmospheres, and new metals can withstand more mechanical punishment. Computer design techniques are also helpful.

Insulation systems were first classified according to the material used, and permissible temperatures were established based on the thermal aging characteristics of these materials. For example, Class B insulation was defined as inorganic materials such as mica and glass with organic binders; 130°C was the allowable maximum operating temperature. The present definition of insulation system Class B stipulates that the system be proven ". . . by experience or accepted tests . . . to have adequate life expectancy at its rated temperature, such life expectancy to equal or exceed that of a previously proven and accepted system." The definition is now functional rather than descriptive.

The newest catalogs show standard induction motors designed with Class B insulation for operation in a 40°C ambient with 80°C rise by resistance at 100% load for motors with 100% service factor. Class F insulation, with the capability of operating up to a 105°C rise by resistance, is today frequently offered as standard for machines with a Class B rise, particularly the larger sizes. Many users specify this as a standard. Previously, induction motor ratings were based on temperature rise by thermometer.

These changes require an explanation. National Electrical Manufacturers Association (NEMA) standards previously allowed three methods of temperature determination: (1) thermometer, (2) resistance, and (3) embedded detector. Motor engineers have long recognized that measuring temperature rises by placing a thermometer against the end windings does not give the best indication of insulation temperatures near the conductors in the slot. The average temperature rise of any motor can be measured by resistance. This will give a better indication of the temperature in the hottest part of the winding than will thermometer measurement. On machines equipped with temperature detectors, there will usually be a difference in the readings taken by an embedded detector and by winding resistance, with the detector reading usually slightly higher, because it is embedded near the hottest part of the winding. Adequate placement and selection of the embedded detectors can be a factor in determining the quality of this type of monitoring. For example, longer detectors may provide more of an average temperature, and detectors near the fan end of a long TEFC motor may not see the hottest winding temperatures.

The resistance method gives an average temperature of the whole winding. Some parts will be hotter than others; usually the end turns will be somewhat cooler than parts of the winding in the middle of the iron core. NEMA committee members have been collecting test data on many machines to determine the correlation between temperature measurements by detector and by resistance, and the standards are periodically updated to reflect any of the technology improvements.

NEMA standards do not give any fixed maximum operating temperature by any class of insulation. Briefly, NEMA states that insulation of a given class is a system that can be shown to have suitable thermal endurance when operated at the temperature rise shown in the standards for that type of machine. Standards for synchronous motors and induction motors with a 100% service factor specify 80°C rise by resistance for Class B insulation. Also, the total temperature for any insulation system is dependent upon the equipment to which it is applied. For example, railway motors with Class B insulation have rated standard rises by resistance of 120°C on the armature. Induction motors with service factor have 90°C rise at the service factor load.

Service Factor

For many years it was common practice to give standard open motors a 115% *service factor* rating; that is, the motor would operate at a safe temperature at 15% overload. This has changed for large motors, which are closely tailored to specific applications. Large motors, as used here, include synchronous motors and all induction motors with up to 16 poles (450 rpm at 60 Hz).

New catalogs for large induction motors are based on standard motors with Class B insulation of 80°C rise by resistance, 1.0 service factor. Previously, they were 70°C rise by thermometer, 1.15 service factor.

Service factor is mentioned nowhere in the NEMA standards for large machines. There is no standard for temperature rise or other characteristics at the service factor overload. In fact, the standards explicitly state that the temperature rise tables are for motors with 1.0 service factor. Neither standard synchronous nor enclosed large induction motors have included service factor for several years.

Today, almost all large motors are designed specifically for a particular application and for a specific driven machine. In sizing the motor for the load, the horsepower is usually selected so that additional overload capacity is not required. Therefore, customers should not be required to pay for capability they do not require. With the elimination of the service factor, standard motor base prices have been reduced 4–5% to reflect the savings. Users should specify standard horsepower ratings, without service factor for these reasons:

1. All of the larger standard horsepowers are within or close to 15% steps.
2. As stated in NEMA, using the next larger horsepower avoids exceeding standard temperature rise.
3. The larger horsepower ratings provide increased pull-out torque, starting torque, and pull-up torque.
4. The practice of using 1.0 service factor induction motors would be consistent with that generally followed in selecting horsepower requirements of synchronous motors.

The common practice of using Class F insulated motors with a Class B rise at 1.0 SF in effect provides some obtainable service factor above 1.0 if the user is willing to operate the motor up to the Class F limits in response to some contingency. In many cases this provides at least 15% margin.

In NEMA size ranges, motors with service factors are still available; however, for compressor drives, it would be better if they were not. Experience with operation into the service factor rating has not been satisfactory.

Synchronous Motors

Synchronous motors have definite advantages in some applications. They are the obvious choice to drive large, low-speed reciprocating compressors and similar equipment requiring motor speeds below 600 rpm. They are also useful on many large, high-speed drives. Typical applications of this type are geared, high-speed (above 3,600 rpm), centrifugal compressor drives of several thousand horsepower.

A rule of thumb that was used in the past for constant speed applications was to consider the selection of a synchronous motor where the application horsepower was larger than the speed. This, of course, was only an approximation and tended to favor the selection of a synchronous motor and would be considered too severe by current standards. However, the rule can aid in the selection of the motor type by giving some insight as to when the synchronous might be chosen. For example, applications of several hp per rpm often offer a distinct advantage of the synchronous over the induction motor. In fact, at the lowest speeds, larger sizes and highest hp/rpm ratios may be the only choice.

One interesting characteristic of synchronous motors is their ability to provide power factor correction for the electrical system. Standard synchronous motors are available rated either 100% or 80% leading power factor. At 80% power factor, 60% of the motor-rated kVA is available to be delivered to the system as reactive kVA for improving the system power factor. This leading reactive kVA increases as load decreases. At zero load, with rated field current, the available leading reactive kVA is approximately 80% of the motor rated kVA. The unity power factor machine does not provide any leading current at rated load. However, at reduced loads, with constant field current, the motor will operate at leading power factor. At zero load, the available leading kVA will be about 30% of the motor rated kVA.

Because of their larger size, 80% power factor motors cost 15–20% more than unity power factor motors, but the difference may be less costly than an equivalent bank of capacitors. An advantage of using synchronous motors for power factor correction is that the reactive kVA can be varied at will by field current adjustment. Synchronous motors, furthermore, generate more reactive kVA as voltage decreases (for moderate dips), and therefore tend to stabilize system voltage better than capacitors, since they supply less leading kVA when the voltage is decreased.

When higher than standard pull-out torque is required, 80% leading power factor motors should be considered. The easiest way to design for high pull-out is to provide additional flux, which effectively results in a larger machine and allows a leading power factor. The leading power factor motor may, therefore, be less expensive overall. However, leading power-factor motors are generally "stiffer" electrically, and may need to be evaluated against the higher current pulsations that will result from reciprocating compressors or other pulsating loads. Larger flywheels can normally be applied to compensate for this effect.

In addition to power factor considerations, synchronous motor efficiency is higher than similar induction motors. Efficiencies are shown in Table 7-1 for typical induction and unity power factor synchronous motors. Leading power factor synchronous motors have efficiencies approximately 0.5–1.0% lower.

Table 7-1
Full Load Efficiencies

Hp	600 RPM	1800 RPM	3600 RPM
250	91.0	93.5	94.5
	93.4*	—	—
1,000	93.5	95.4	9.52
	95.5*	—	—
5,000	—	97.2	97.0
	97.2*	97.4*	—
10,000	—	97.5	97.4
—	97.6*	—	
15,000	—	97.8	—
	—	98.1*	—

**Synchronous Motors, 1.0 PF*
Source: Modified from [7] & [15]

Direct-connected exciters were once common for general purpose and large, high-speed synchronous motors. At low speeds (514 rpm and below), the direct-connected exciter is large and expensive. Motor generator sets and static (rectifier) exciters have been widely used for low-speed synchronous motors and when a number of motors are supplied from a single excitation bus.

Brushless Excitation

One of the most significant developments in recent years is *brushless excitation* for synchronous machines. This development became possible with the availability of reliable, long-life, solid-state control and power devices (diodes, transistors, SCR's, etc.). As the name indicates, there are no brushes, collector rings, or commutators on the motor or exciter. This eliminates brush, collector ring and commutator maintenance and permits the use of synchronous motors in many hazardous (Class 1, Group D, Division 2) and corrosive areas where conventional motors could not be used without extensive additional protection. All the advantages of conventional synchronous motors are retained: constant speed, high efficiency, power factor correction, and varied performance capability. High precision, fast-acting, solid-state field application control is rotor-mounted and provides the same full complement of functions as a conventional synchronizing panel. Brushless excitation is presently almost universally used.

The *exciter* is an AC generator with a stator-mounted field. Direct current for the exciter field is provided from an external source, typically a small variable voltage rectifier mounted at the motor starter. Exciter output is converted to DC through a three-phase, full-wave, silicon-diode bridge rectifier. Thyristors (silicon-controlled rectifiers) switch the current to the motor field and the motor-starting, field-discharge resistors. These semiconductor elements are mounted on heat sinks and assembled on a drum bolted to the rotor or shaft.

Semiconductor control modules gate the thyristors, which switch current to the motor field at the optimum motor speed and precise phase angle. This assures synchronizing with minimum system disturbance. On pull-out, the discharge resistor is reapplied and excitation is removed to provide protection to the rotor winding, shaft, and external electrical system. The control resynchronizes the motor after the cause of pull-out is removed, if sufficient torque is available. The field is automatically applied if the motor synchronizes on reluctance torque. The control is calibrated at the factory and no field adjustment is required. The optimum slip frequency at pull-in is based on total motor and load inertia. All control parts are interchangeable and can be replaced without affecting starting or running operation.

Motor Equations

The following equations are useful in determining the current, voltage, horsepower, torque, and power factors for three phase AC motors:

$$I = .746\text{hp}/(1.73\ E\ \eta\ PF) \tag{7.1}$$

$$kVA = 1.73\ I\ E\ /\ 1{,}000 \tag{7.2}$$

$$kW_{input} = kVA \times PF \tag{7.3}$$

$$kW_{shaft} = kW_{input} \times \eta \tag{7.4}$$

$$hp = kW/.746 \tag{7.5}$$

$$T = 5250\ hp/N \tag{7.6}$$

$$PF = kW/kVA \tag{7.7}$$

where

E = volts (line-to-line)
I = current (amps)
PF = power factor
η = efficiency
hp = horsepower
kW = kilowatts
kVA = kilovoltamperes
N = speed, rpm
T = torque, ft-lb

A typical medium-size, squirrel-cage motor is designed to operate at 2–3% slip (97–98% of synchronous speed). Synchronous speed is determined by the power system frequency and the stator winding configuration. If the stator is wound to produce one north and one south magnetic pole, it is a two-pole motor. There is always an even number of poles (two, four, six, eight, etc.). The synchronous speed is

$$N = 2 \times 60f/P \tag{7.8}$$

where

N = speed, rpm
f = frequency, Hz
P = number of poles

The actual operating speed will be slightly less by the amount of slip. Slip varies with motor size, load, and application. Typically, the larger and more efficient the motor, the less full-load slip. A standard 10-hp motor may have 2½% slip; whereas, motors over 1,000 hp may have less than ½% slip. Operating slip can be approximated by multiplying % load by full-load slip.

Compressor and Motor

Coordinating a motor with driven equipment involves selecting the proper motor horsepower, comparing motor and driven equipment speed-torque curves, determining required starting time and effect of momentary voltage losses, providing for thrust conditions by using proper coupling and end-play limits, and coordinating bearing lubrication requirements.

Motor horsepower required to operate the compressor is dictated by process plant operating conditions. Usually there are minimum conditions, normal (design) conditions, and maximum conditions. Economic sizing of the motor depends on exercising good judgment when determining capacity to be provided in the motor over the usual maximum operating conditions. Two other factors should be considered: capability of the compressor casing and likelihood that plant requirements will dictate a future change in the rotor to take advantage of casing capability. For a guide to the actual size, API generally recommends that the motor be sized at 110% of the greatest power required by the compressor. This allows some margin for compressor deterioration.

Selecting Compressor Motors

The first step in selecting motors for large compressors (1,500 hp and over) is to determine the motor voltage, speed, and enclosure type.

Motor voltage selection is determined by economics and by the availability of adequate system capacity to permit motor starting without excessive voltage drop. Restrictions by the utility or by the size of the plant's generating capacity may limit the maximum drop to less than 20%. In some cases, system dips as small as 5% or less may be the maximum tolerable. For example, many utilities insist on limiting dips to this magnitude at the customer interface, and some motors will be installed at a system location that can influence large amounts of critical lighting or other similar loads. Applications of this type will often require some type of soft start. However, in the usual refinery or chemical plant process unit, large blowers and compressors are very seldom started once the plant has been on stream for some time. The undesirable effects normally associated with local drops as high as 20% or more have been tolerable when occurring infrequently.

The motor cost is but one facet of any cost study for selecting voltage level. The study must compare installed cost of motor, starting equipment, transformers, and power and control cables at the various levels under consideration, as well as plant standards.

Higher voltage levels such as 13,200 volts are sometimes more economical overall, although the cost of the motor itself is higher than it would be at 2,300 or 4,000 volts. For example, if the plant has a 13.8 kV distribution system, it can be more economical to install a 13,200-volt motor than to provide the primary switchgear required for lower voltage motors. Operating experience at levels as high as 13,200 volts has been good, but it is not extensive when compared to lower voltages. The number of manufacturers with experience at these levels is more limited. Also, there are motor design considerations that limit the minimum size at which these high voltage levels can be applied. The motor manufacturer probably would not recommend a 3,000-hp, 13,200-volt motor as a first choice.

When the plant distribution voltage is above utilization levels, for instance 23 kV or higher, economics will usually favor the 2,300- or 4,000-volt motor. However, each application has its own peculiarities. An examination of the relative cost of alternate schemes sometimes favors 4,160 volts. This is true particularly with motors in the 4,000 hp and larger sizes, and short circuit levels are above 150 MVA.

Where speed-increasing gears are a consideration, the 3,600-rpm motor is eliminated for all practical purposes because of higher cost, less favorable torque characteristics, and mechanical design considerations.

The motor and gear combination must provide the proper input speed to the compressor. Therefore, speed selection should consider whether the 1,800 rpm, or the 1,200 rpm motor and the corresponding gear provides the more economical combination. Before selecting motor speed, motor characteristics should be obtained for both speeds. The speed-torque and speed-current curves, power factor and efficiency values should be compared and evaluated. The most important considerations in matching the motor to a compressor are:

1. Motor speed-torque characteristics
2. Load accelerating torque requirements
3. Motor supply voltage during acceleration

For the slower speed compressors, an evaluation must be made concerning the use of direct-connected motors that are more expensive, or the use of a less expensive 1,800 rpm or a 1,200 rpm motor with a speed-reducing gear. As before, the evaluation must include all factors, such as installation and the extra space, as well as some factor for additional maintenance.

Starting Characteristics

The closer the motor and compressor speed-load curves are to each other, the longer the driver will take to reach full speed, and the hotter the motor will get (see Figure 7-2).

A motor speed-torque curve for a compressor (250 to 1,000 hp or more) does not look the same as a smaller machine (10 or 20 hp) (see Figure 7-3). NEMA Standard MG-1 gives minimum locked rotor-torque values as a function of motor size at 1,800 rpm (see Table 7-2).

Before final motor selection can be completed, a comparison of the selected compressor's required speed-torque must be made with the pro-

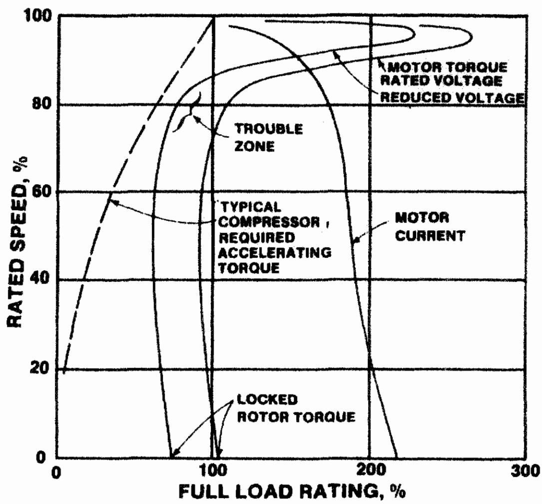

Figure 7-2. Speed-load curve for a centrifugal compressor and motor. Startup difficulty could occur at the "trouble zone" where the curves are very close [7].

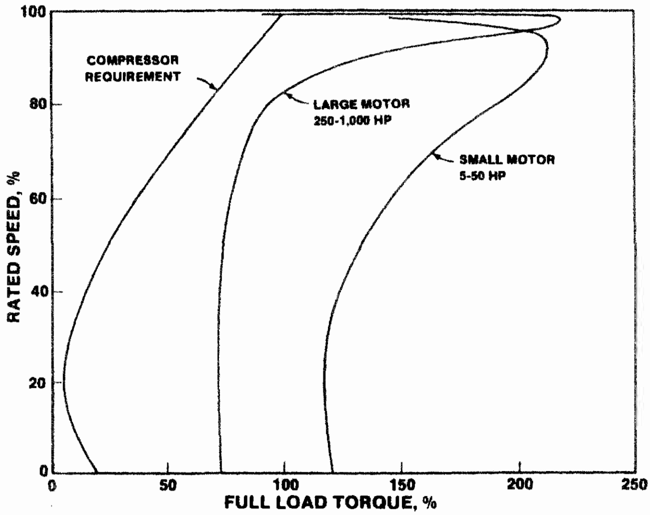

Figure 7-3. Typical speed-torque curves for large and small motors [7].

Table 7-2
Locked Rotor Torque vs. Motor Size

Rated hp	Minimum locked rotor torque percent of full load torque
5	185
10	165
100	125
250	80
2000	60

Source: [7]

posed motor curves. Each kind of compressor has its own unique curve, as illustrated by Figure 7-4. Whenever feasible, either from the compressor design or process considerations, unloading should be considered. Unfortunately, some compressors can not be readily unloaded. For the higher inertia loads, the torsional inertia must be taken into account to determine the starting time. For lower inertia loads, such as with some of the rotary compressors, the motor represents the principal inertia. Probably the worst assumption to make is that compressors follow the centrifugal square law curve, or that there is a typical compressor curve. While a little unusual, Figure 7-5 illustrates a standard motor for a typical compressor.

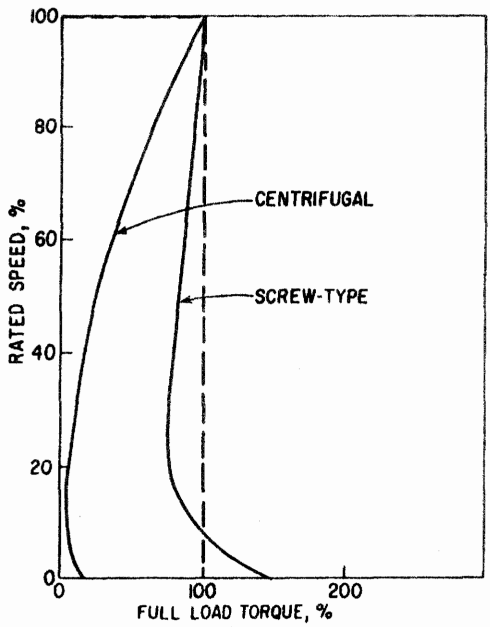

Figure 7-4. Speed-torque curve for two compressor types. Each type of compressor has its own curve [7].

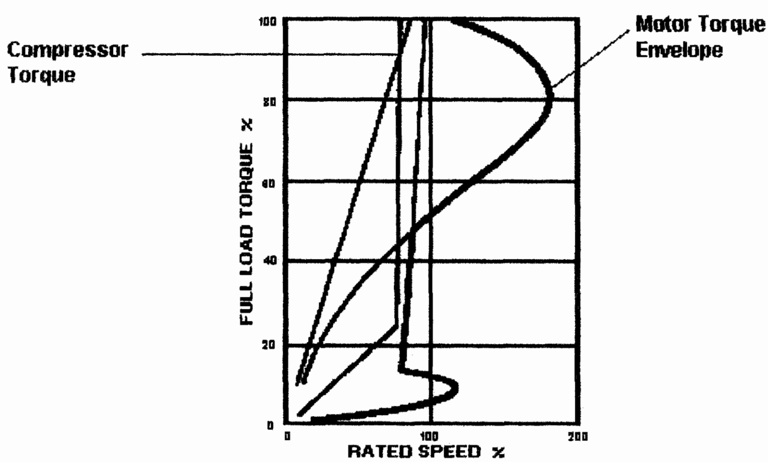

Figure 7-5. There is no standard speed-torque curve for accelerating compressors. Attempting to make a standard motor to meet all needs (heavy line) would impose unnecessary demands on motor design [7].

Some engineers specify motors for across-the-line starting. Motor output torque varies approximately as the square of applied voltage. A 10% voltage drop means a 20% drop in torque, which is enough to keep some drives from ever reaching full speed. This relationship should be borne in mind for operation as well as starting. A lower than required voltage while operating may appear as an overloaded compressor.

Starting Time

The voltage drop during starting should be calculated using the speed current data, locked rotor power factor, and the distribution system constants. Speed-torque values at this reduced voltage can be calculated, assuming that the torque varies as the voltage squared. These values are compared to driven equipment curves.

In comparing speed-torque characteristics, driven equipment torque requirement at any speed must not exceed 95% of motor torque at that speed. Also, the driven equipment speed-torque curve for the loaded condition is used. If less than 5% accelerating torque is available, it is considered doubtful that the motor can successfully accelerate within the allowable time. Theoretically, 1% difference should permit acceleration; however, 5% provides the margin for data and calculation inaccuracies. Because the time for the motor to accelerate through a given speed interval is inversely proportional to the accelerating torque, close approaches of the load-to-motor starting torque may result in excessively long starting times. Even if the motor does not stall at some sub-operating speed, as would be expected if the accelerating torque margin went to zero, the time to accelerate may result in the motor exceeding its allowable starting thermal limit and, at best, experiencing a protection relay trip prior to reaching the operating speed. The loaded curve is suggested for the basic evaluation because in many process applications, unloading may not be feasible. Also, if available, it presents extra margin. It should be pointed out that higher starting torque motors are more expensive. There is a possibility that they will require higher starting currents. To add insult to injury, induction motors with the higher starting torque design will probably be somewhat less efficient. Synchronous motors designed for higher starting torque may be difficult to find.

Starting times for large motors driving high-inertia loads, such as centrifugal compressors, can be 20 seconds or longer. The motor draws locked-rotor current for most of this period. These high currents maintained for such long periods cause winding and rotor temperature to rise rapidly.