

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 sub-atmospheric 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

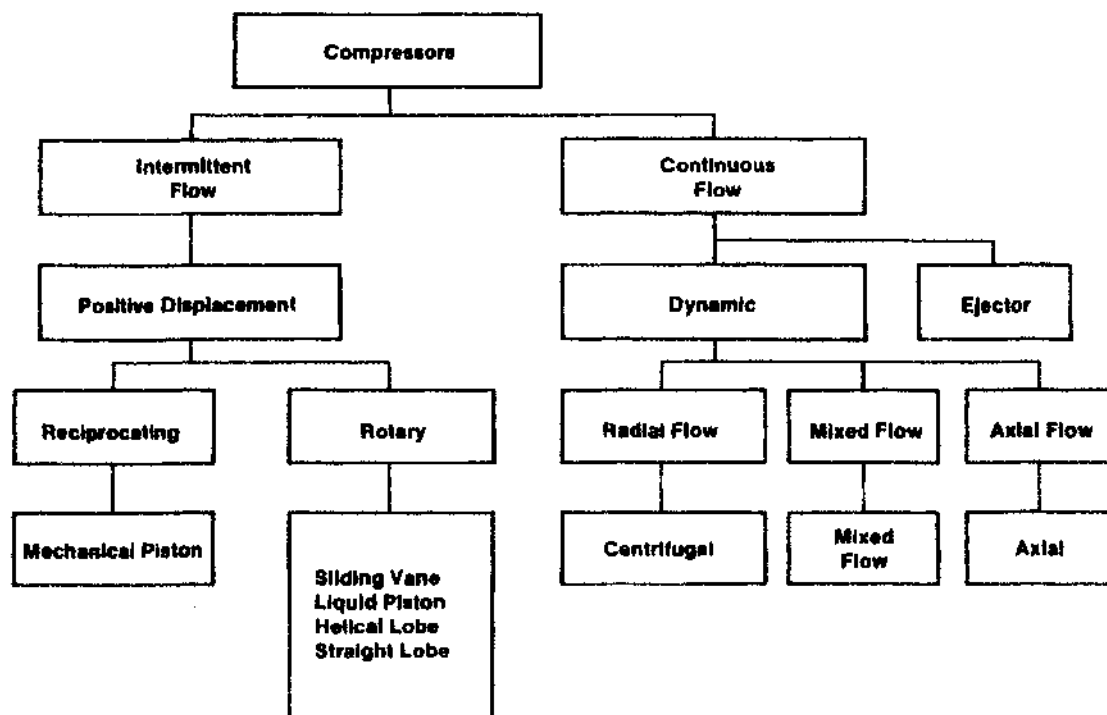


Figure 1-1. Chart of compressor types.

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.

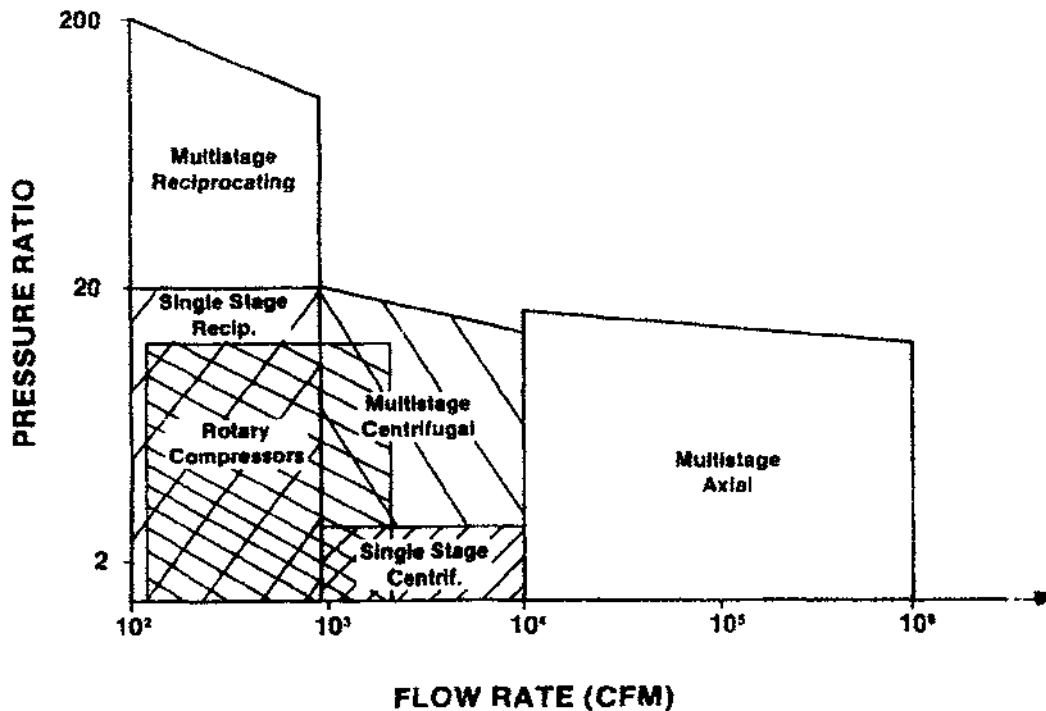


Figure 1-2. Typical application ranges of compressor types.

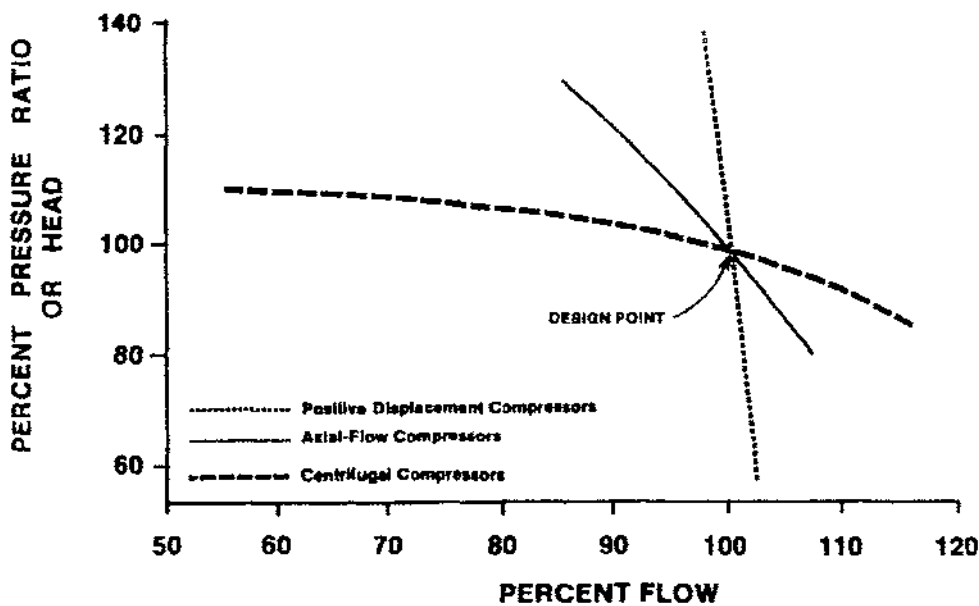


Figure 1-3. General performance curve for axial flow, centrifugal, and positive displacement.

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.

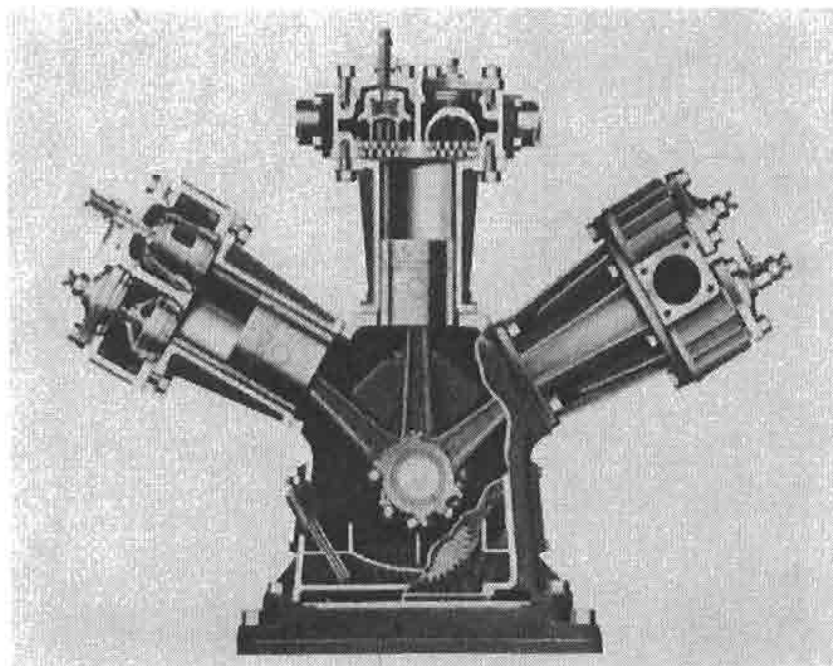


Figure 1-4. A three-stage single-acting reciprocating compressor. (*Courtesy of Ingersoll Rand*)

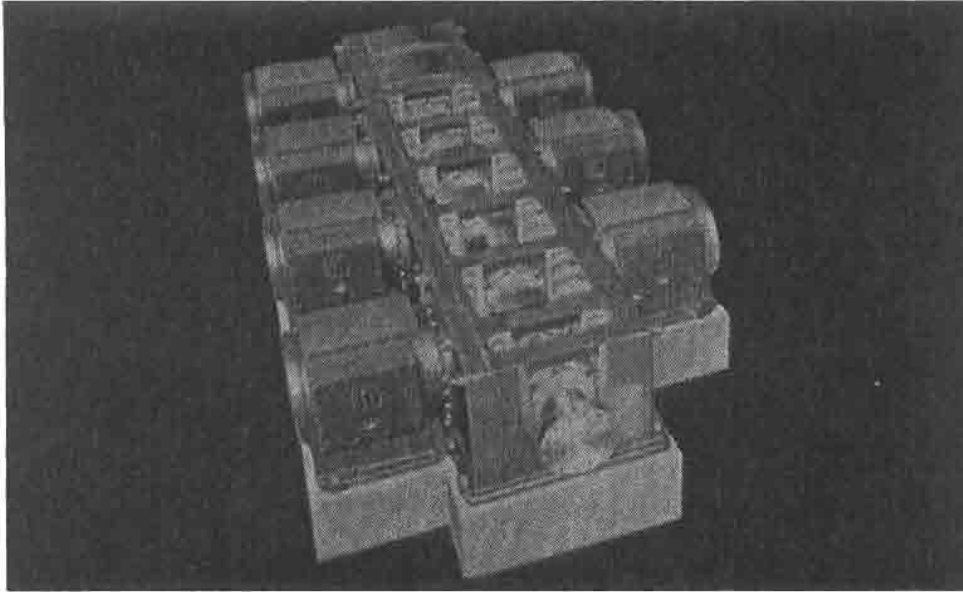


Figure 1-5. Cutaway of the frame end of a large multistage reciprocating compressor. (Courtesy of Dresser-Rand)

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

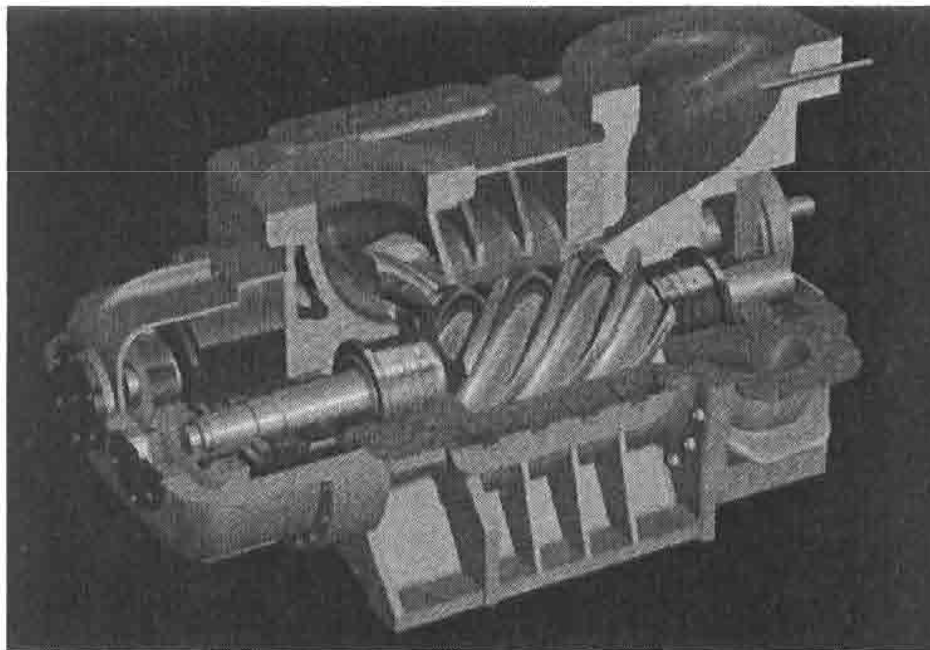


Figure 1-6. Cutaway of an oil-free helical-lobe rotary compressor. (*Courtesy of A-C Compressor Corporation*)

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.

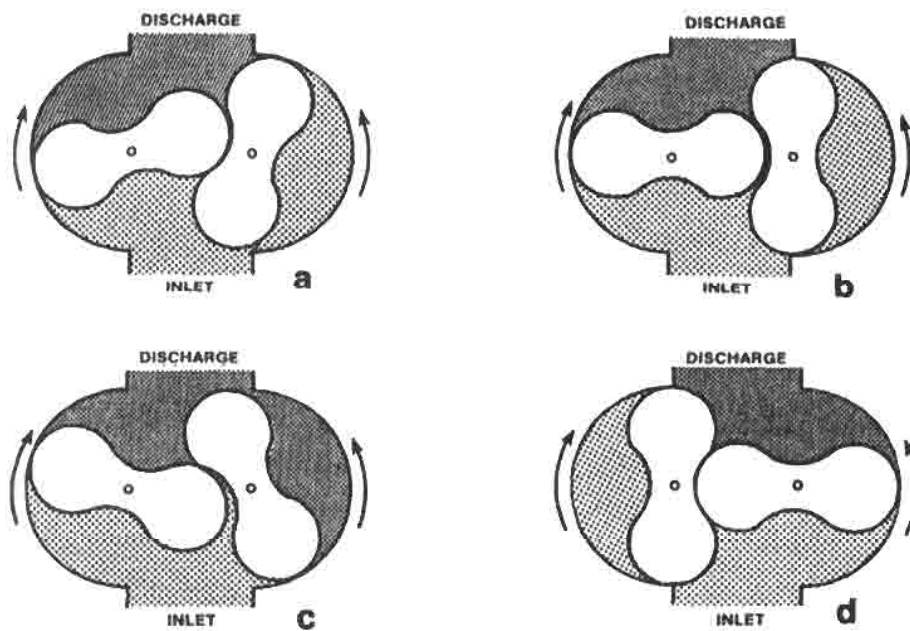


Figure 1-7. Operating cycle of a straight-lobe rotary compressor. (Modified, courtesy of Ingersoll-Rand)

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

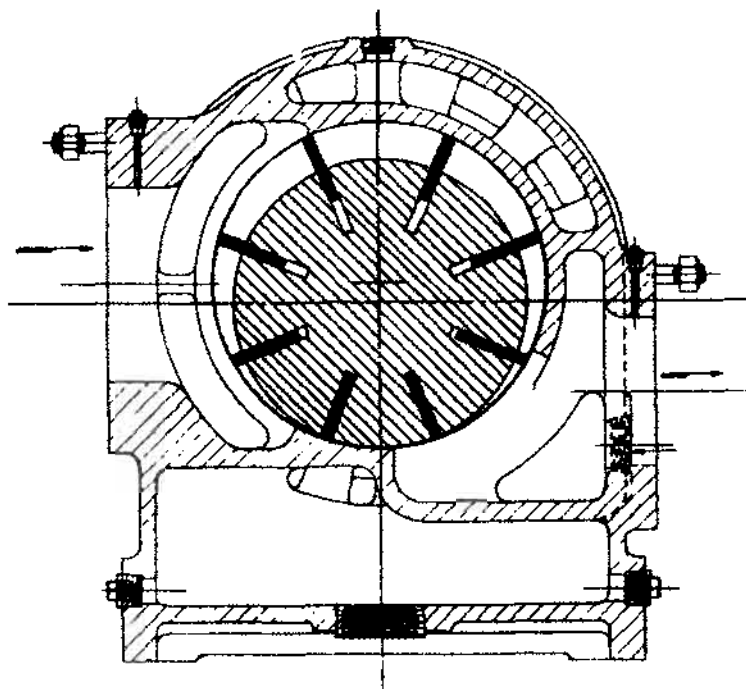


Figure 1-8. Cross section of a sliding vane compressor. (Courtesy of A-C Compressor Corporation)

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-

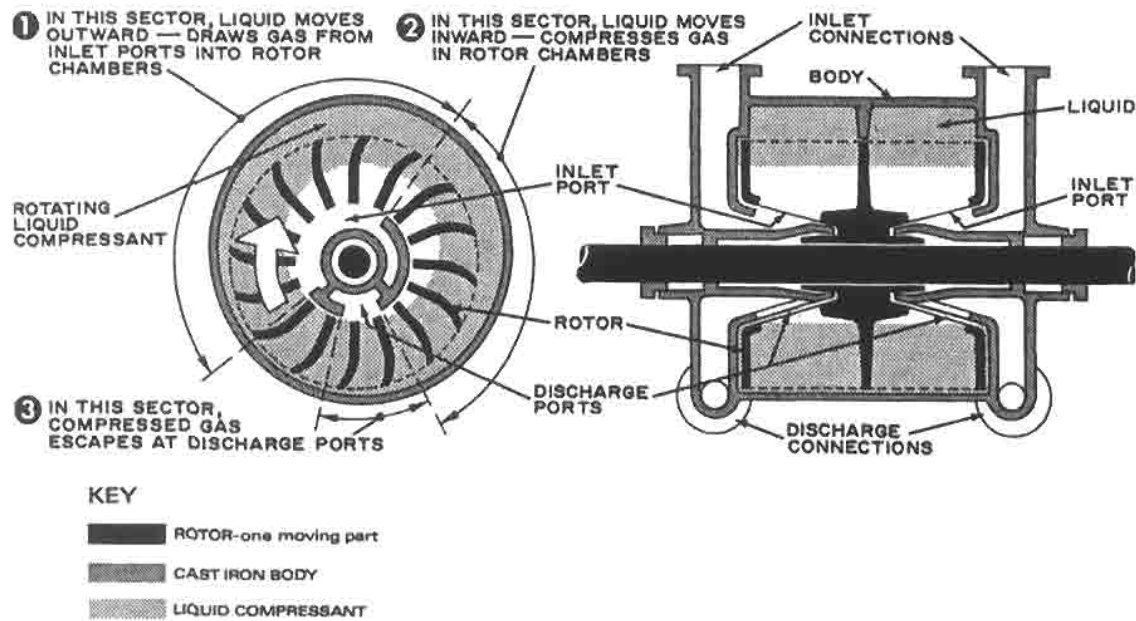


Figure 1-9. A sectional and end view of a liquid piston compressor. (Courtesy of Nash Engineering Co.)

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 multi-stage 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.

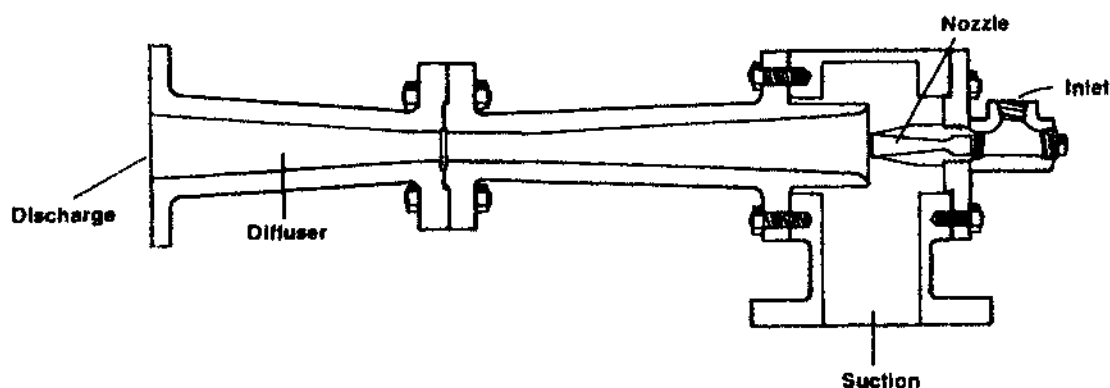


Figure 1-10. Cross section of an ejector. (Courtesy of Graham Manufacturing Co., Inc.)

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

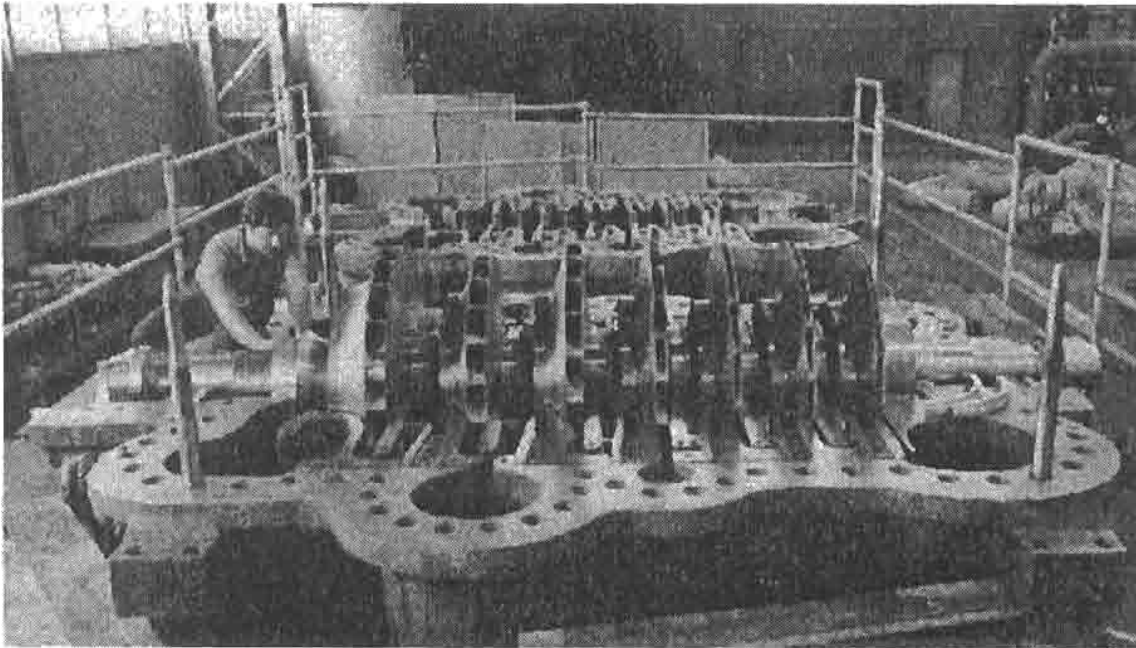


Figure 1-11. Radial-flow horizontally split multistage centrifugal compressor.
(Courtesy of Nuovo Pignone)

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 side-

streams. 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.

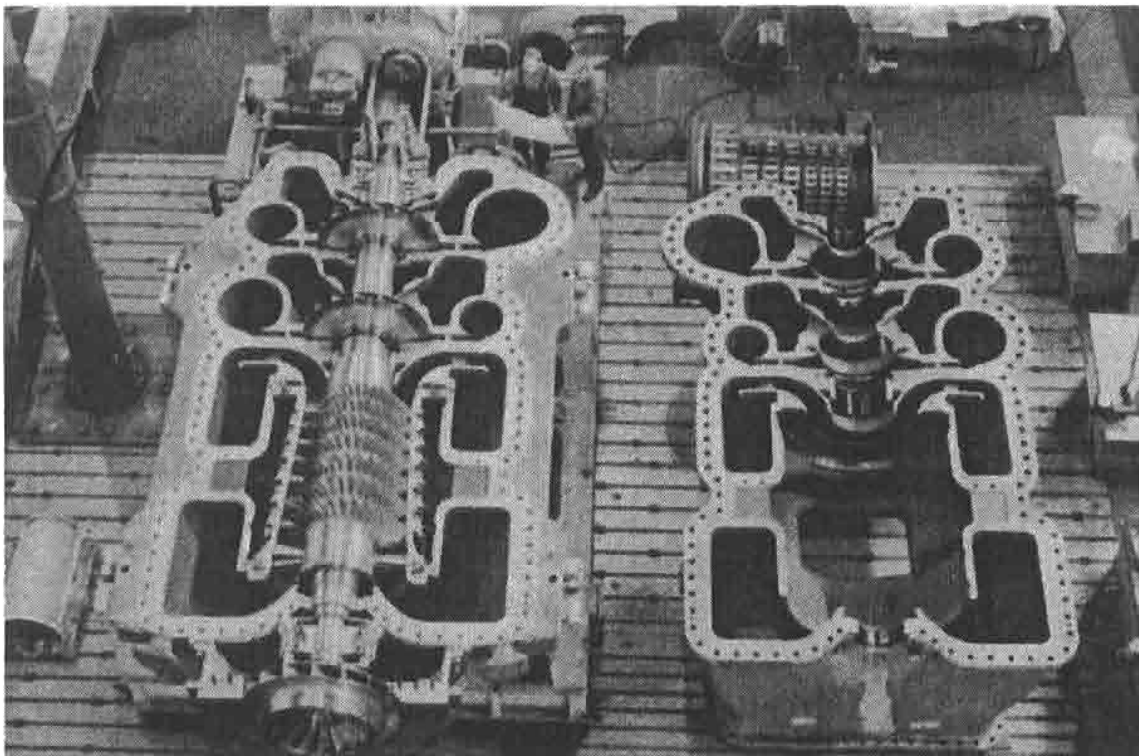


Figure 1-12. Axial-flow compressor. (Courtesy of Demag Delaval Turbomachinery Corp.)

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.

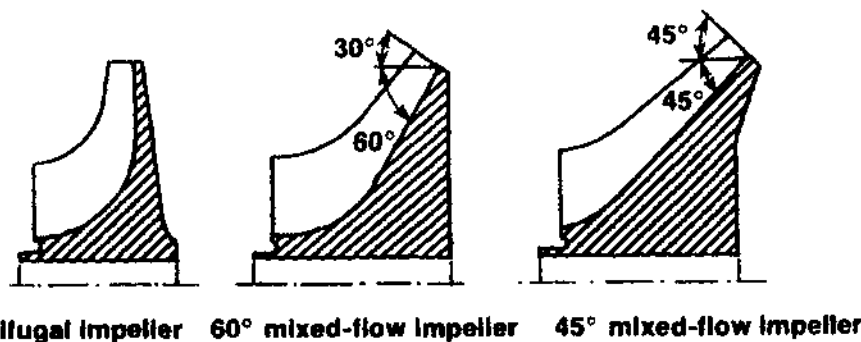


Figure 1-13. Comparison of radial- and mixed-flow compressor impellers.

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.

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

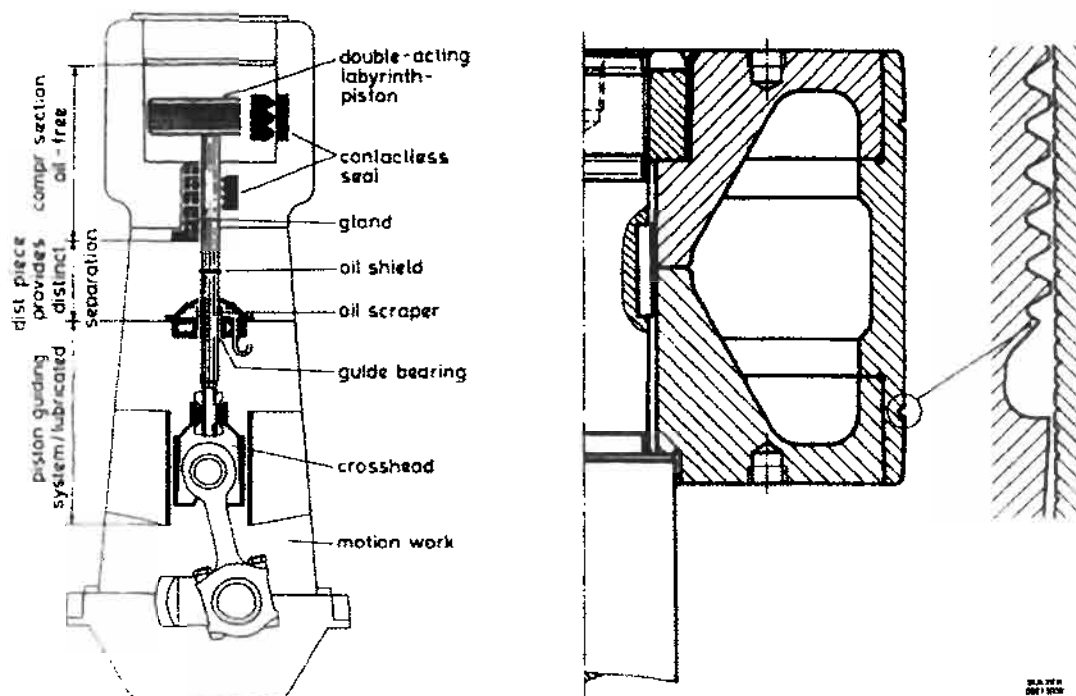


Figure 3-1. Labyrinth piston compressor. This non-lubed piston's circumferential labyrinths operate with a close clearance to the cylinder wall instead of rubbing. (Courtesy of Sulzer)

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.

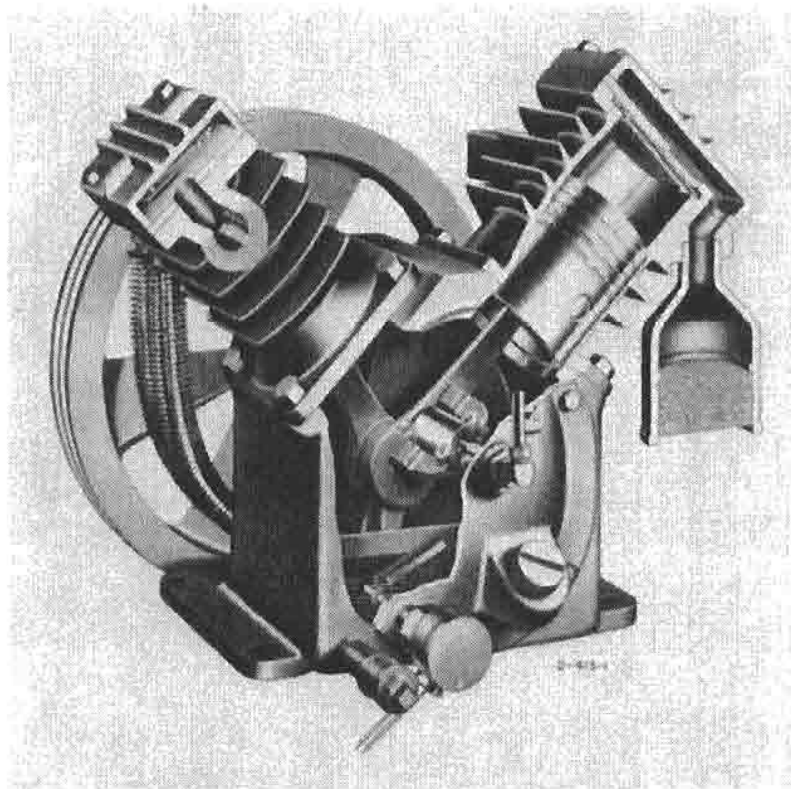


Figure 3-2. Trunk-piston type two-stage compressor with fins for air cooling.
(Courtesy of Ingersoll-Rand)

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.

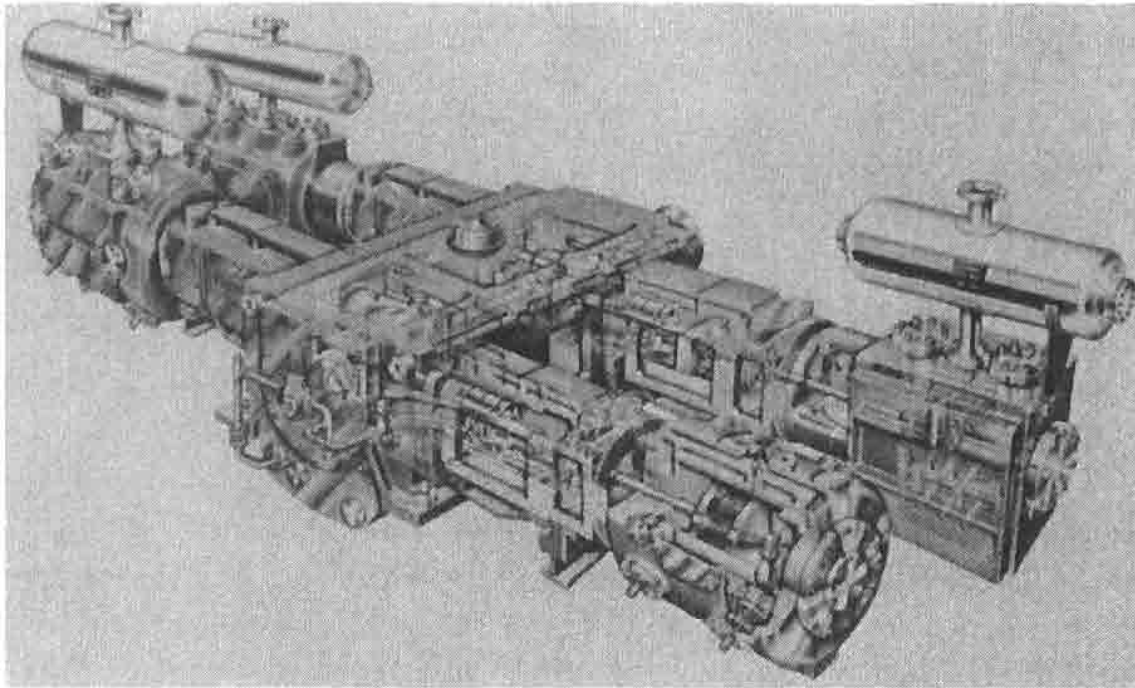


Figure 3-3. Typical multistage crosshead type compressor. (Courtesy of Nuovo Pignone)

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 latter 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.

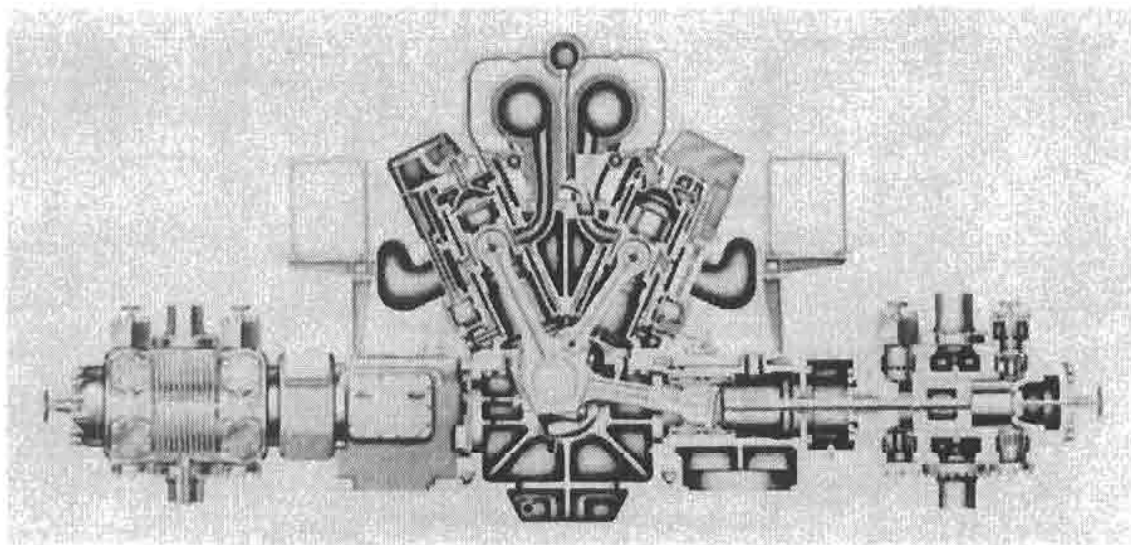


Figure 3-4. Cutaway of a two-stage piston engine driven compressor. (Courtesy of Dresser-Rand)

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

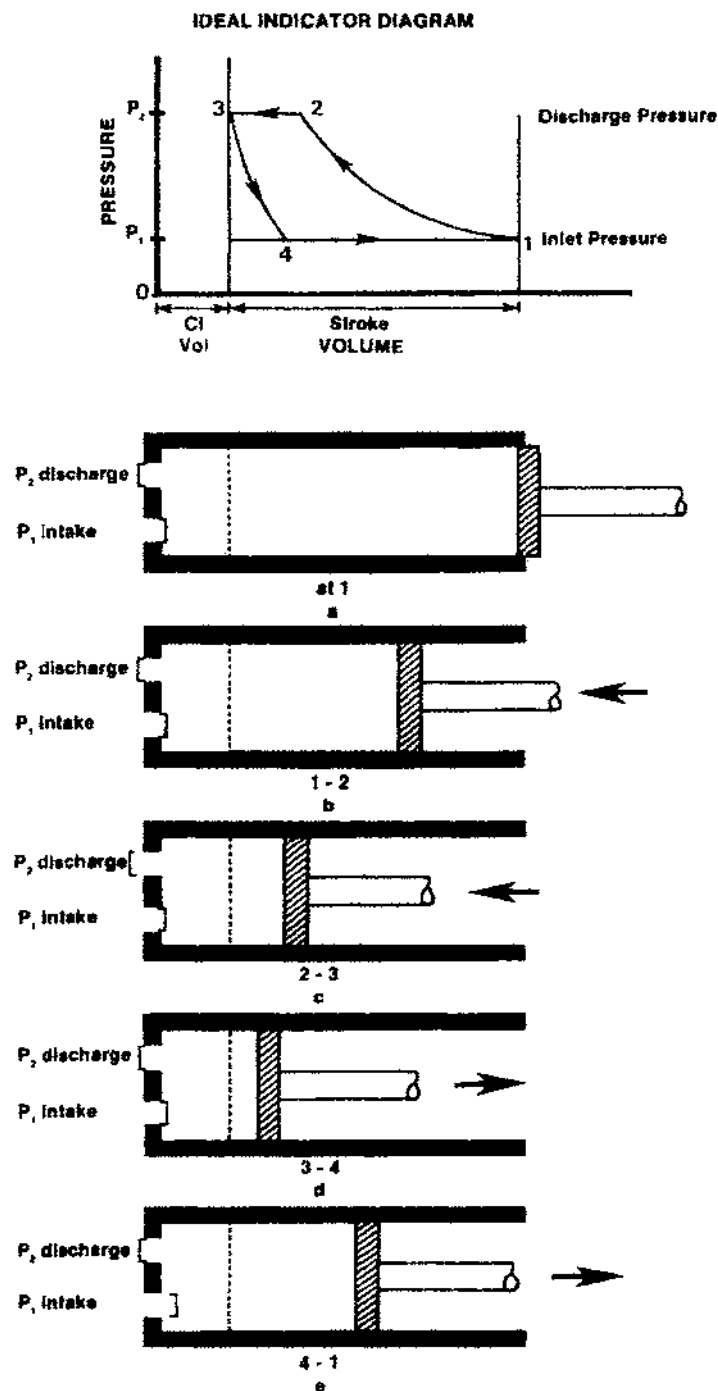


Figure 3-5. Steps in the cycle of reciprocating compressor.

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_1 . 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,

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

where

Pd = piston displacement

S_t = stroke

N = speed of the compressor

D = cylinder diameter

For a double-acting cylinder without a tail rod,

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

where

d = piston rod diameter

For a double-acting cylinder with a tail rod,

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

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} .

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

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.

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

$$f = Z_2/Z_1 \quad (3.6)$$

The inlet capacity of the cylinder is calculated by

$$Q_i = E_v \times Pd \quad (3.7)$$

Piston Speed

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

$$PS = 2 \times S_t \times N \quad (3.8)$$

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_1 (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

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

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

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

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.

$$\text{Work} = H_a \times w \quad (3.10)$$

then,

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

Substituting $P_1 Q_1$ for wRT_1 from Equation 2.7,

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

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

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

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

$$P_i = \sqrt{P_1 \times P_2} \quad (3.14)$$

This expression can be changed to

$$\frac{P_1}{P_i} = \frac{P_2}{P_i} \quad (3.15)$$

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

$$r_{pi} = r_{pi} \quad (3.16)$$

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

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

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} .

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

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

RECIPROCATING COMPRESSOR EFFICIENCIES

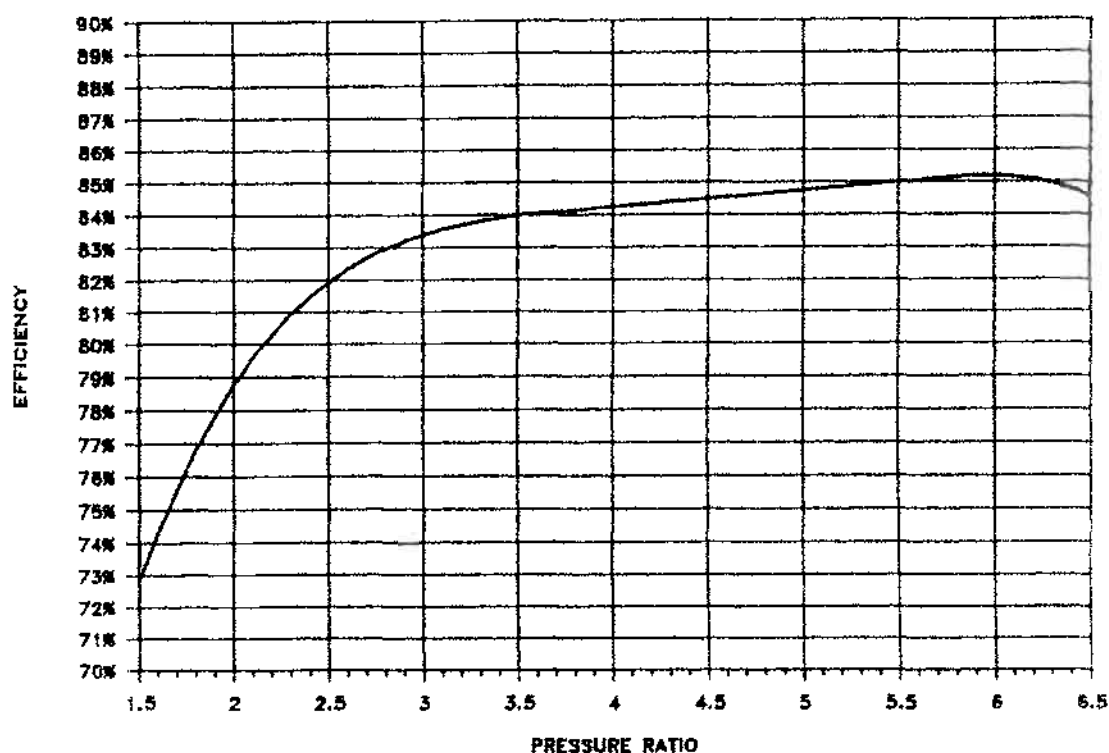


Figure 3-6. Reciprocating compressor efficiencies plotted against pressure ratio with a valve velocity of 3,000 fpm and a mechanical efficiency of 95%.

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.

$$v = 144 \frac{Pd}{A} \quad (3.19)$$

where

v = average gas velocity, fpm

Pd = piston displacement per cylinder, ft³/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: CO₂
 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 ft³ to convert the output to cfm.

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

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

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,

$$T_1 = 460 + 115$$

$$T_1 = 575^\circ\text{R}$$

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

$$T_r = 1.05$$

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

$$P_r = 1.6$$

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.

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

$$r_p = 2.0$$

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

$$T_2 = 674.7^\circ\text{R}$$

$$t_2 = 674.7 - 460$$

$$t_2 = 214.7^\circ\text{F discharge temperature}$$

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 in²ft² 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

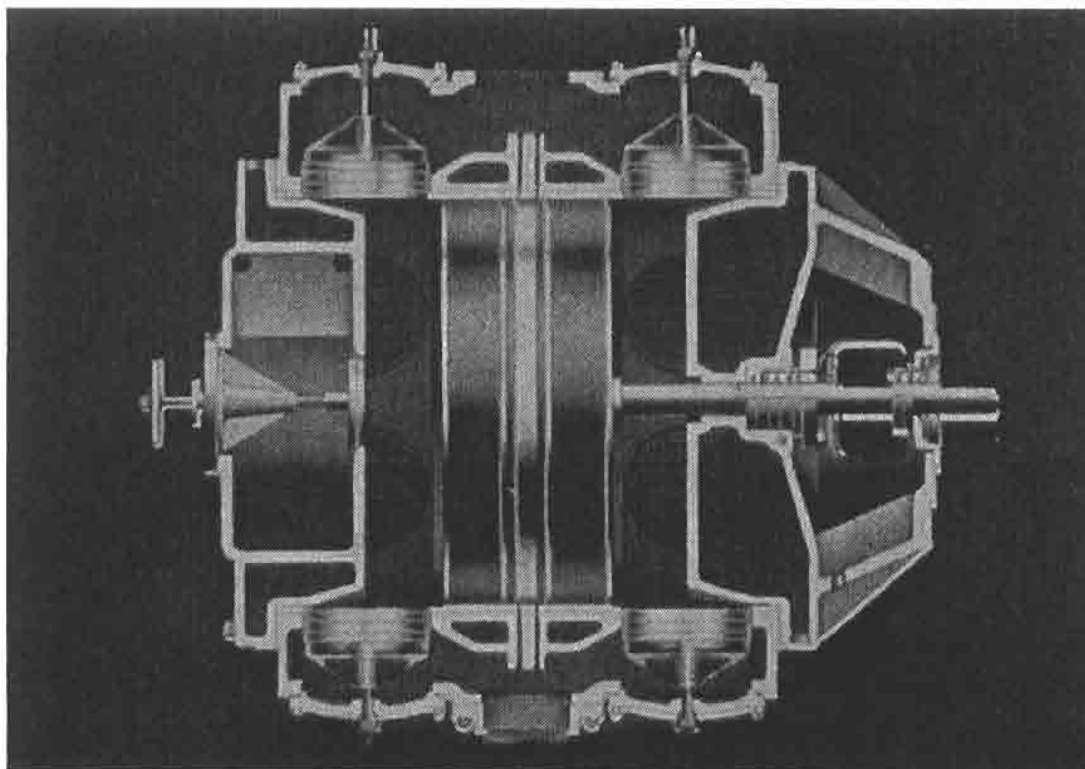


Figure 3-7. Cylinder with clearance pocket. (*Courtesy of Dresser-Rand*)

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.

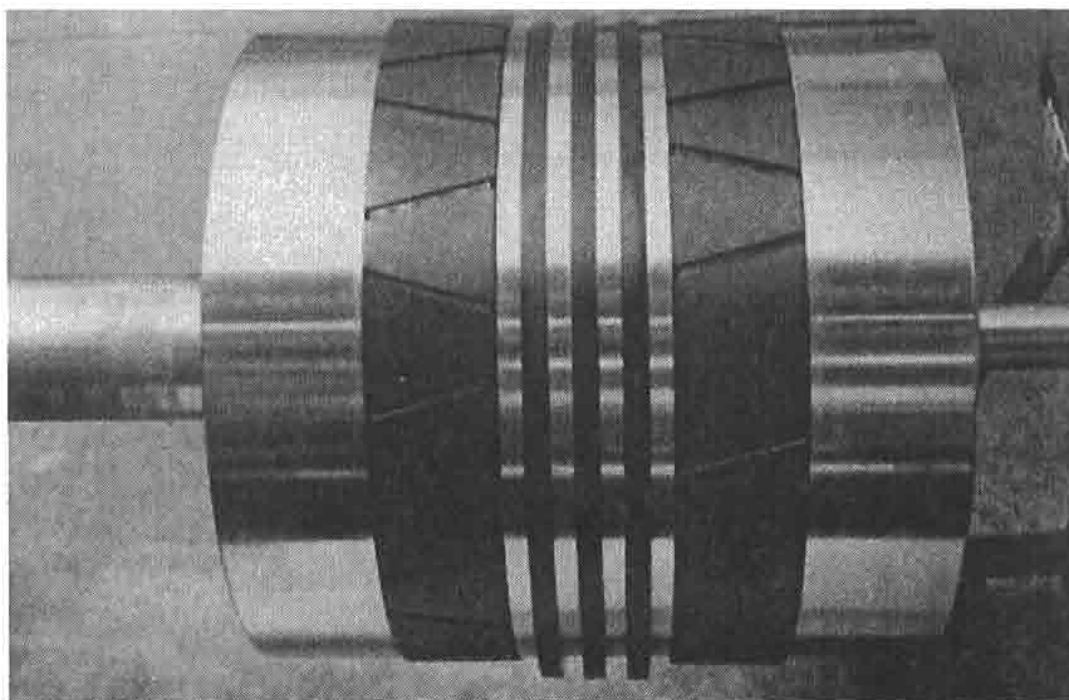


Figure 3-8. Piston rings and wear band. (*Courtesy of Nuovo Pignone*)

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

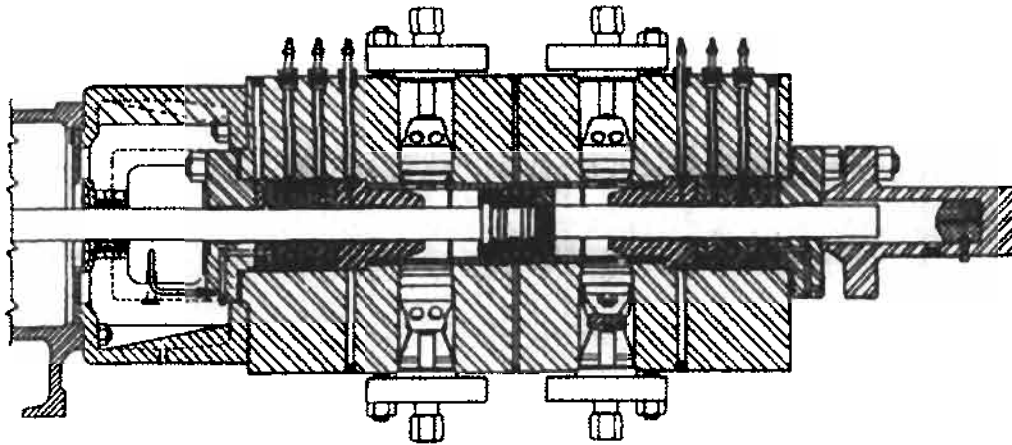


Figure 3-9. Diagram of cylinder with piston tail rod. (*Courtesy of Dresser-Rand*)

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.

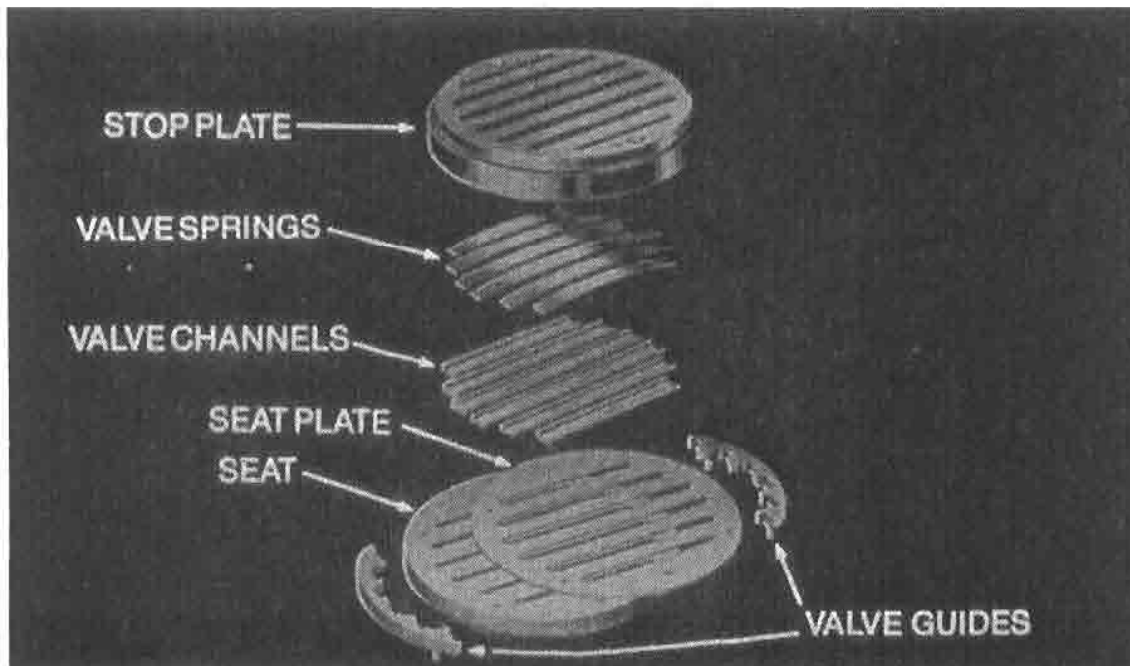


Figure 3-10. An exploded view of a cushioned channel valve. (Courtesy of Dresser-Rand)

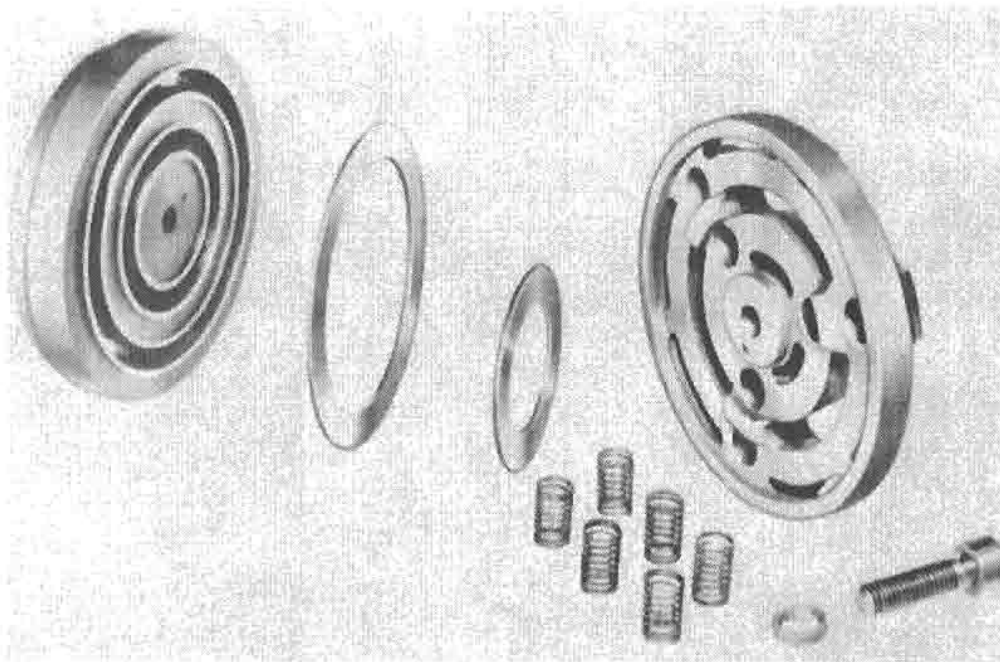


Figure 3-11. Exploded view of a concentric plate valve. (Courtesy of Dresser-Rand)

The poppet valve (see Figure 3-13) consists of multiple, same-size ports and sealing elements. The advantage of the valve is that it 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

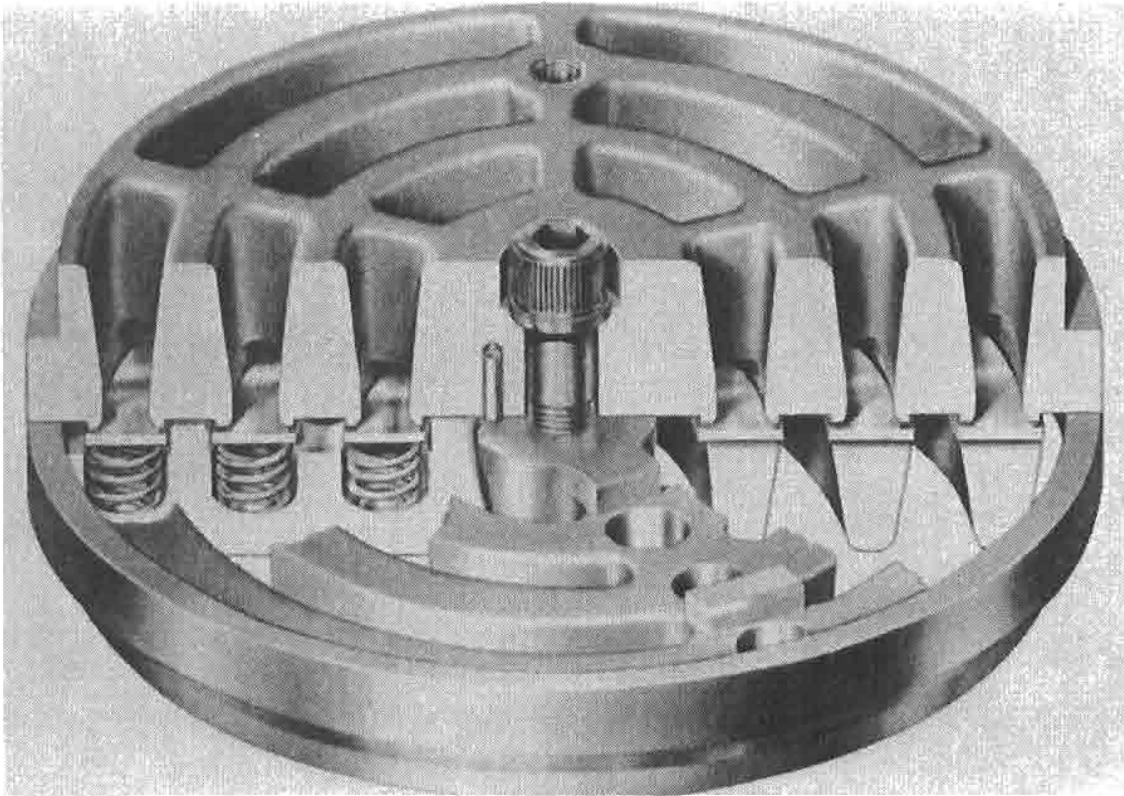


Figure 3-12. Cutaway of a ported plate valve. (*Courtesy of Dresser-Rand*)

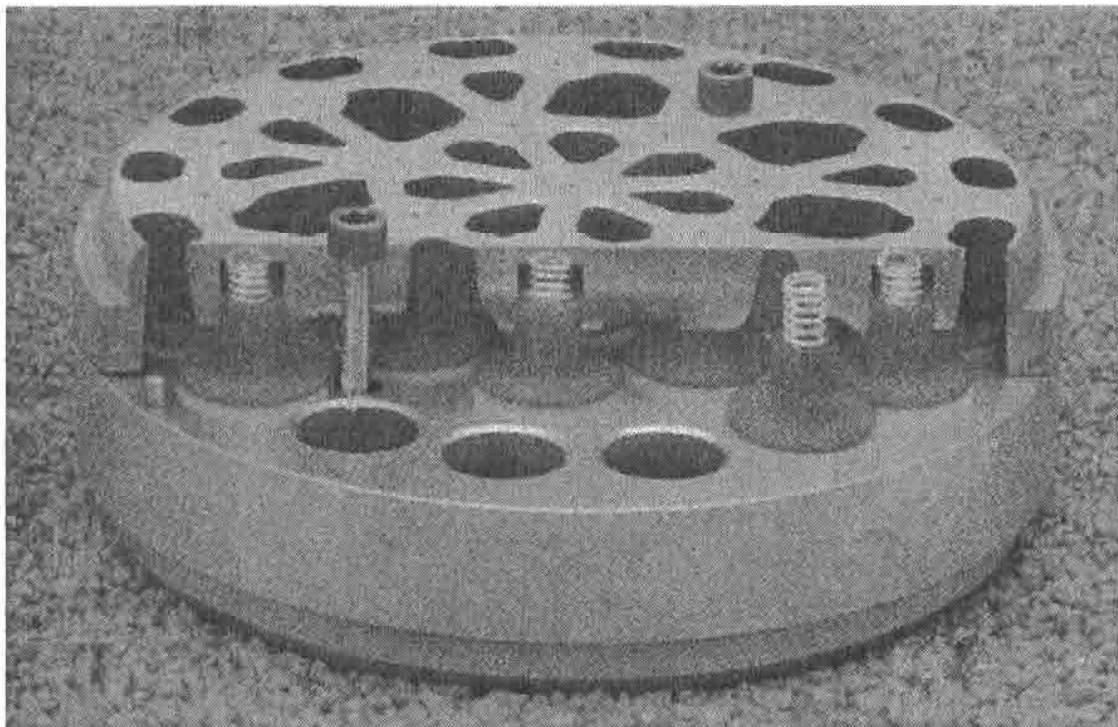


Figure 3-13. Cutaway of a poppet valve. (*Courtesy of Dresser-Rand*)

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 tan-

dem 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.

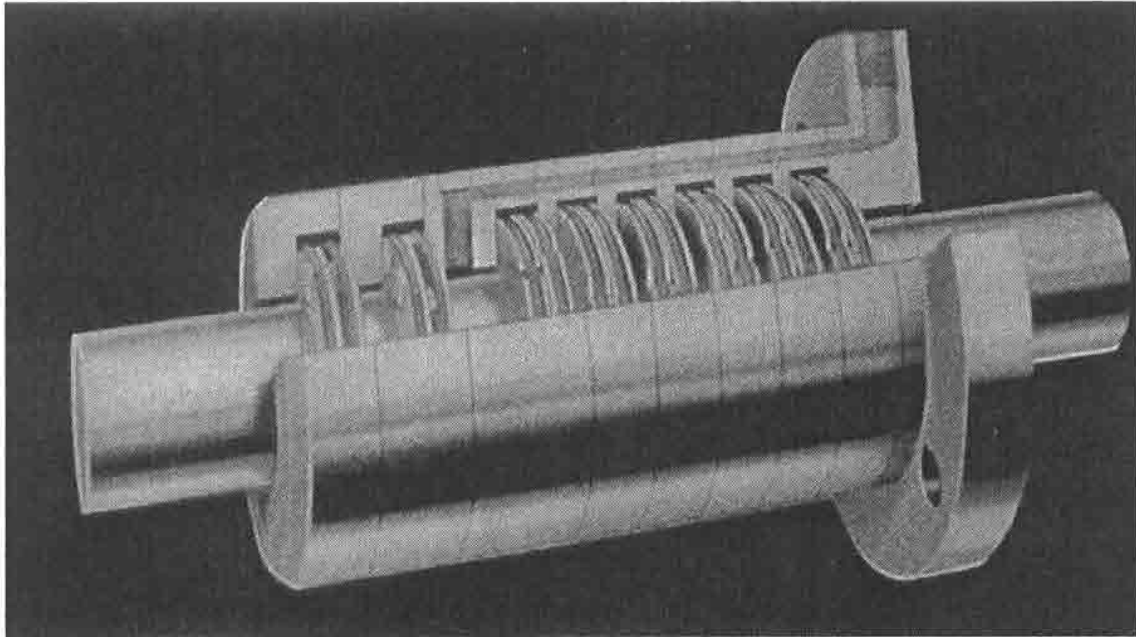


Figure 3-14. Rod packing box. Lantern rings in packing provide space into which a buffer may be injected to aid in sealing. (*Courtesy of Dresser-Rand*)

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

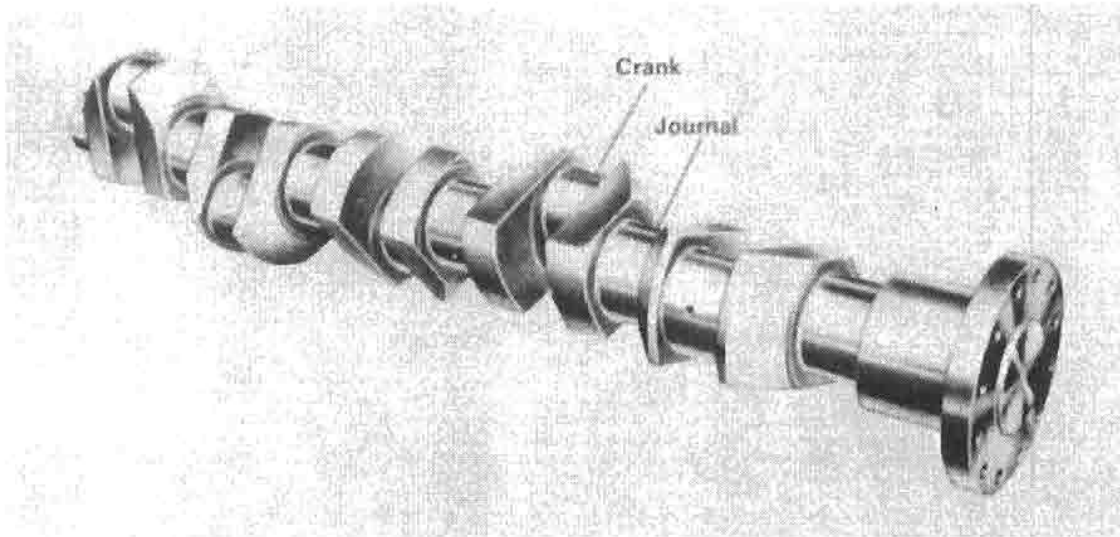


Figure 3-15. A five throw crankshaft with drilled oil passages. (Courtesy of Dresser-Rand)

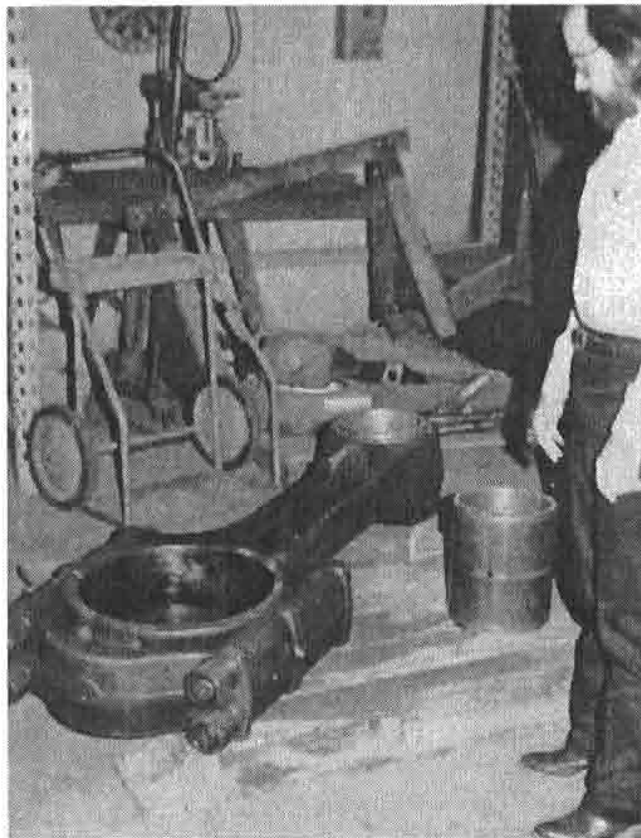


Figure 3-16. A large connecting rod.

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-

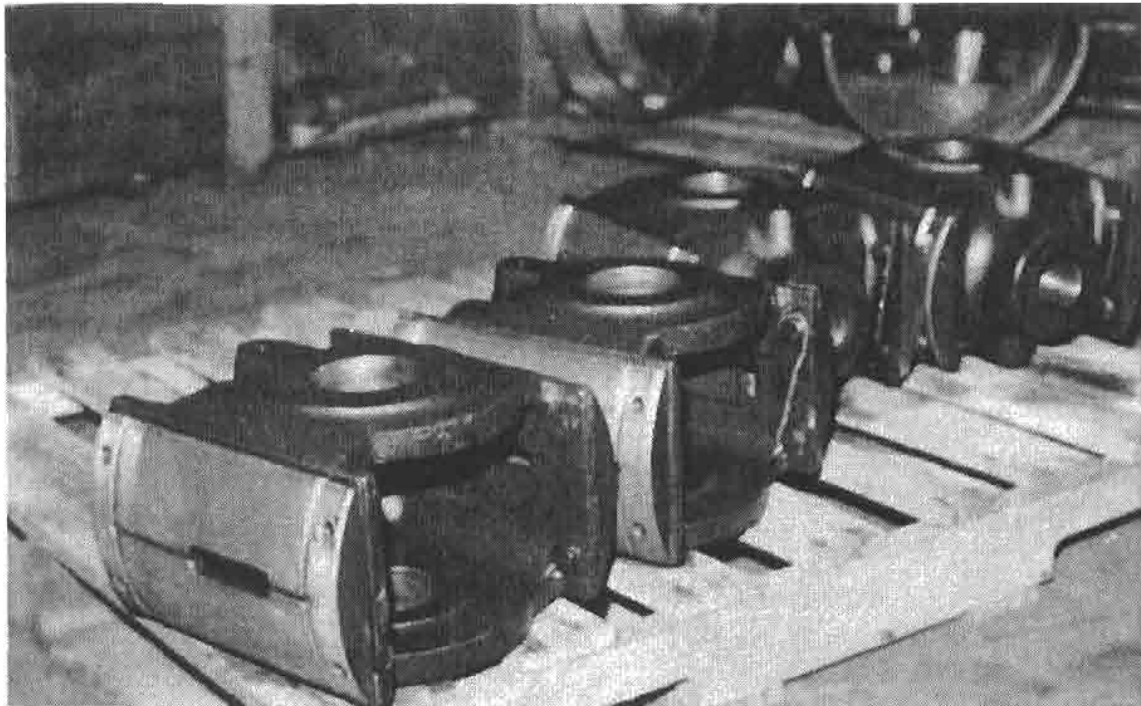


Figure 3-17. Crossheads.

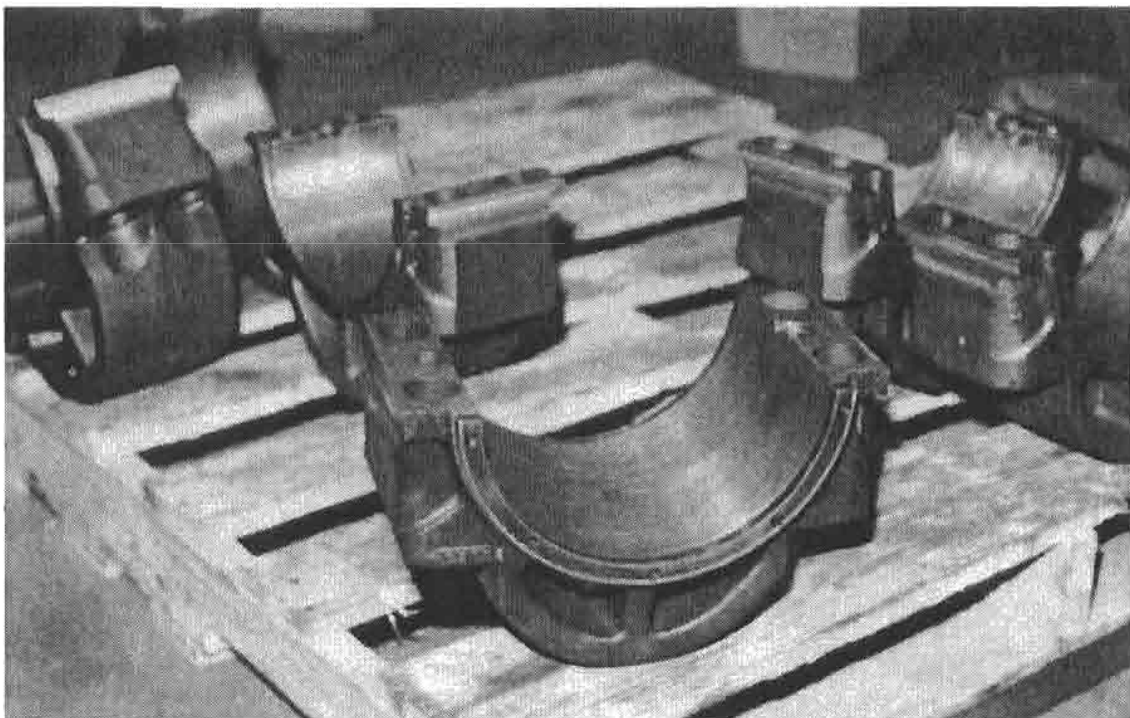


Figure 3-18. Split sleeve bearing caps.

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

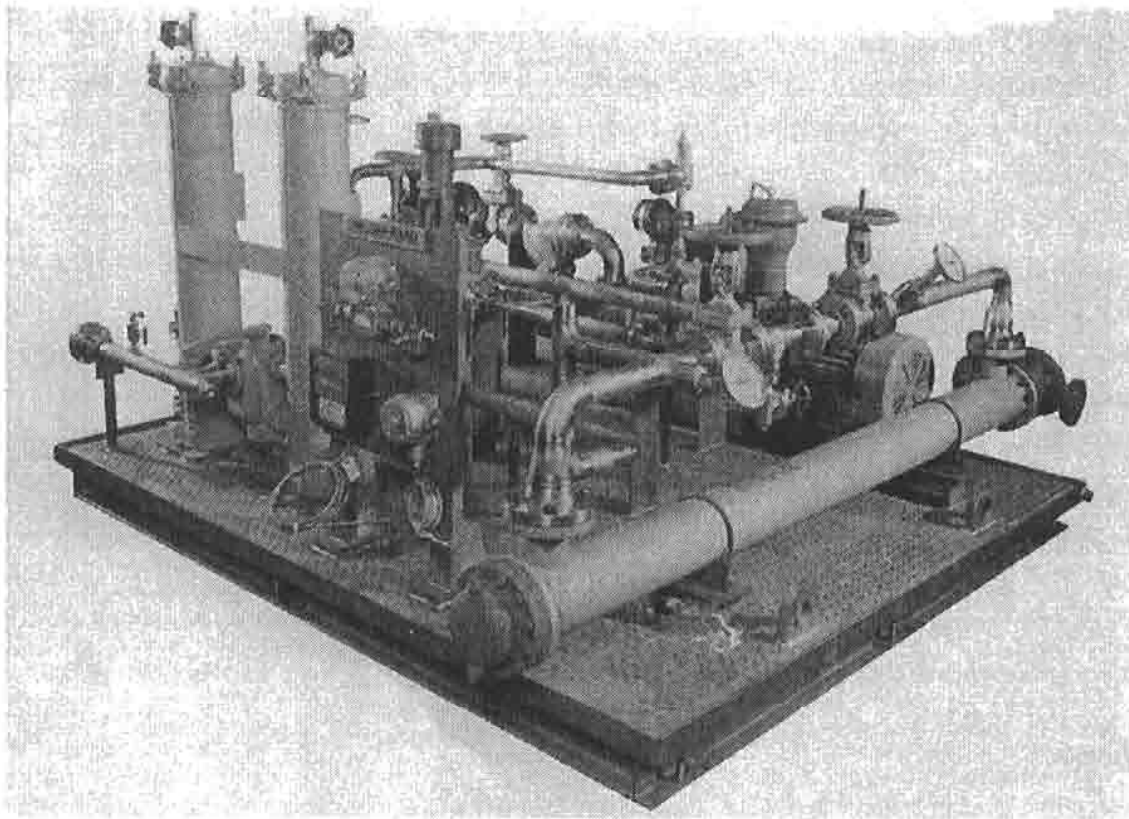


Figure 3-19. Pressurized lubrication system for a multistage reciprocating compressor. (Courtesy of Dresser-Rand)

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 CO₂ or H₂S 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-

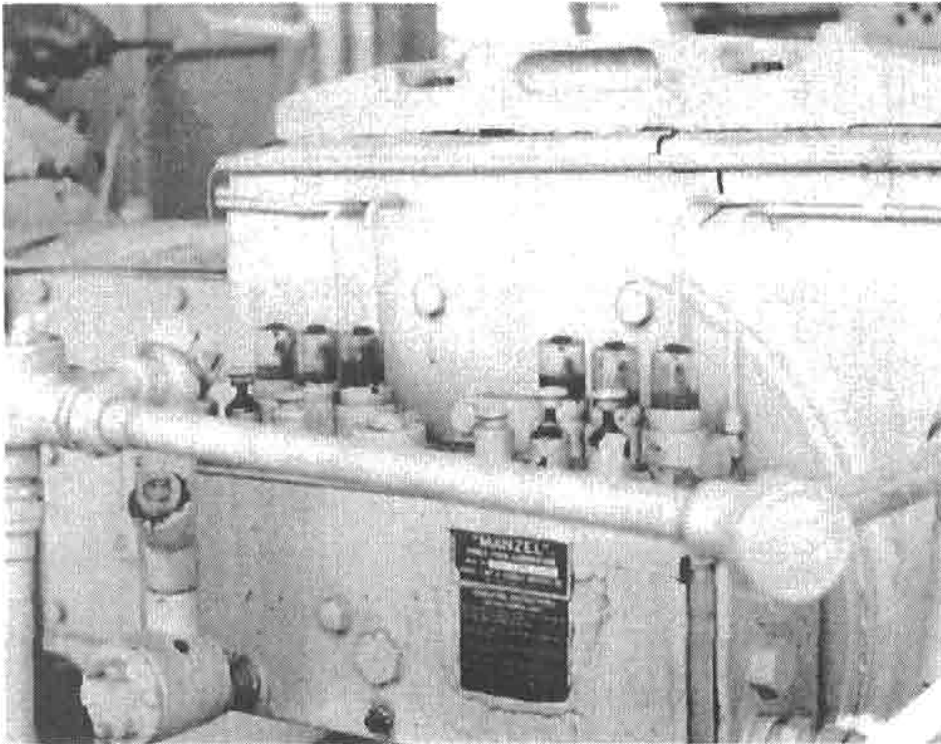


Figure 3-20. A multiplunger lubrication pump.

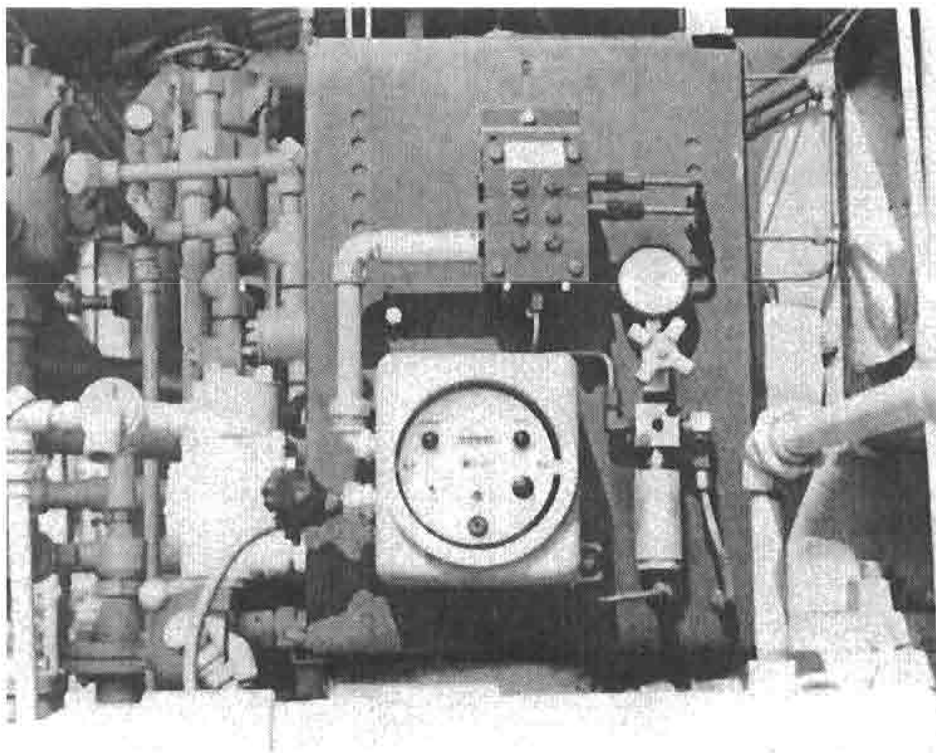


Figure 3-21. A single lubrication pump with a divider block.

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

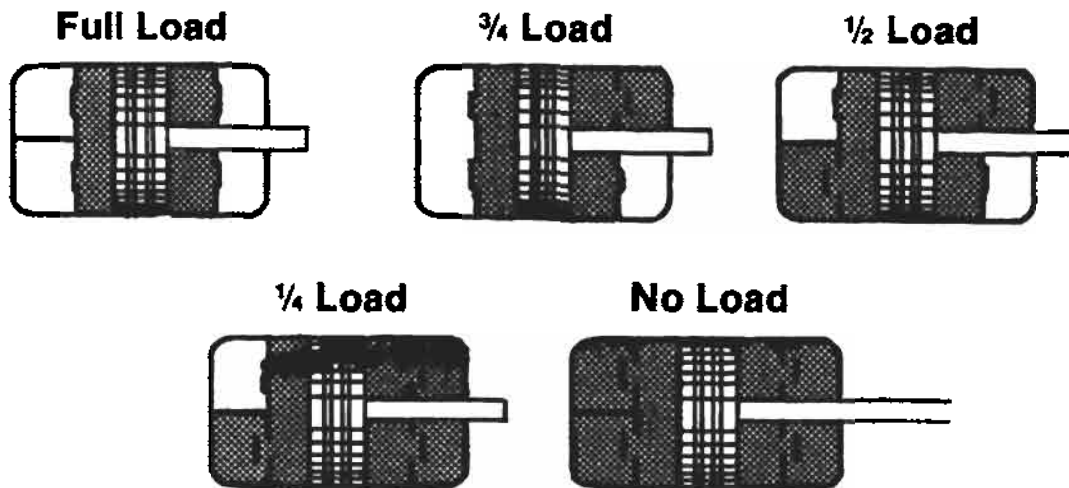


Figure 3-22. A 5-step clearance pocket unloading scheme. (Courtesy of Dresser-Rand)

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

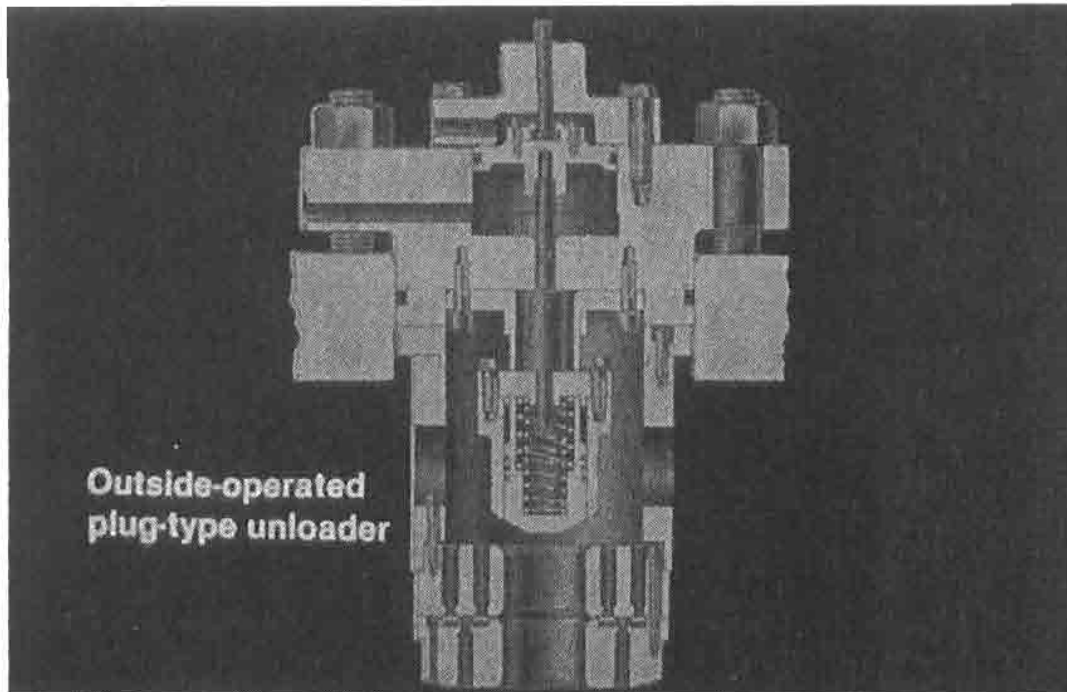


Figure 3-23. Plug type unloader. (*Courtesy of Dresser-Rand*)

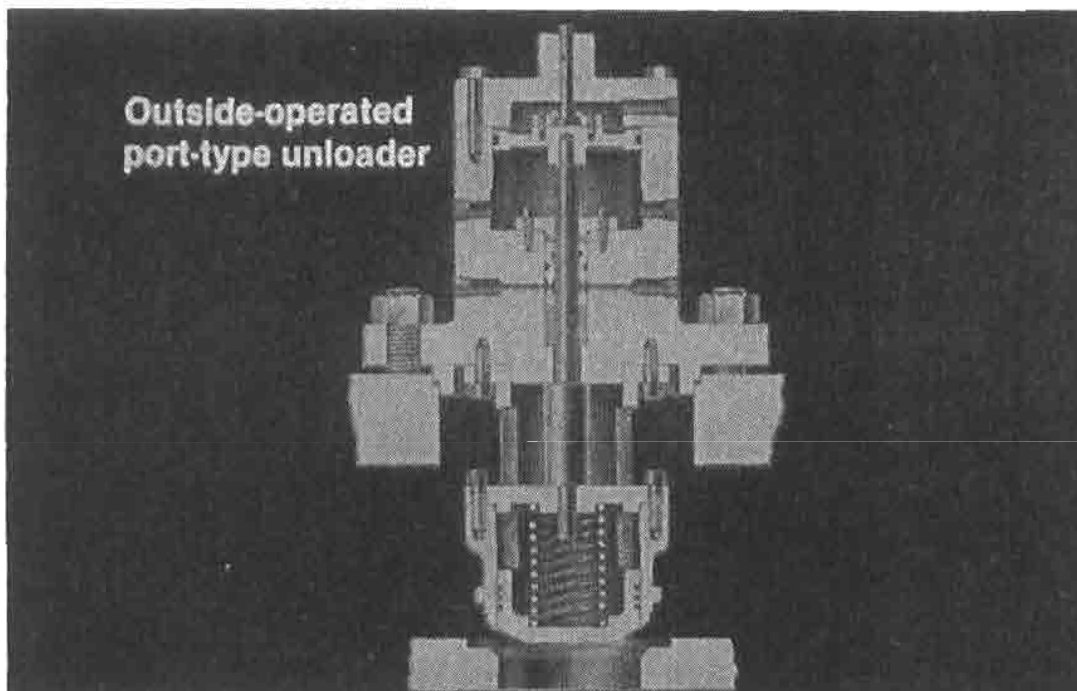


Figure 3-24. Port type unloader. (*Courtesy of Dresser-Rand*)

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.

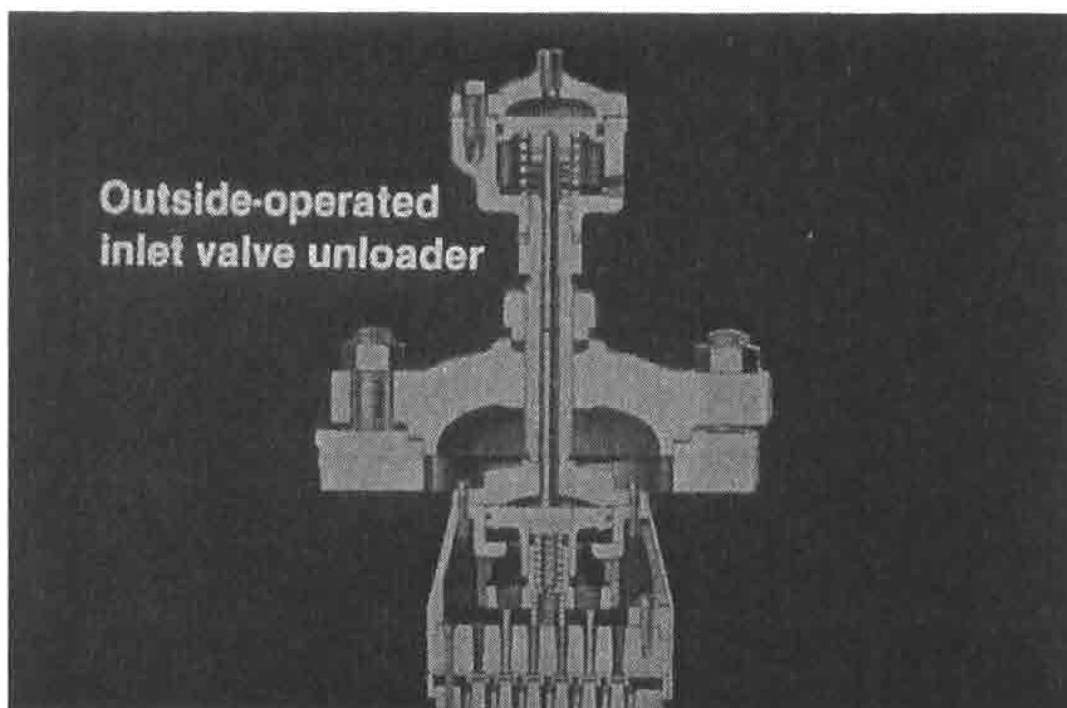


Figure 3-25. Plunger type unloader. Note the plunger finger used to hold the valve open when energized. (Courtesy of Dresser-Rand)

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.

$$P\% = \frac{10}{P_{\text{line}}^{1/3}} \quad (3.20)$$

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

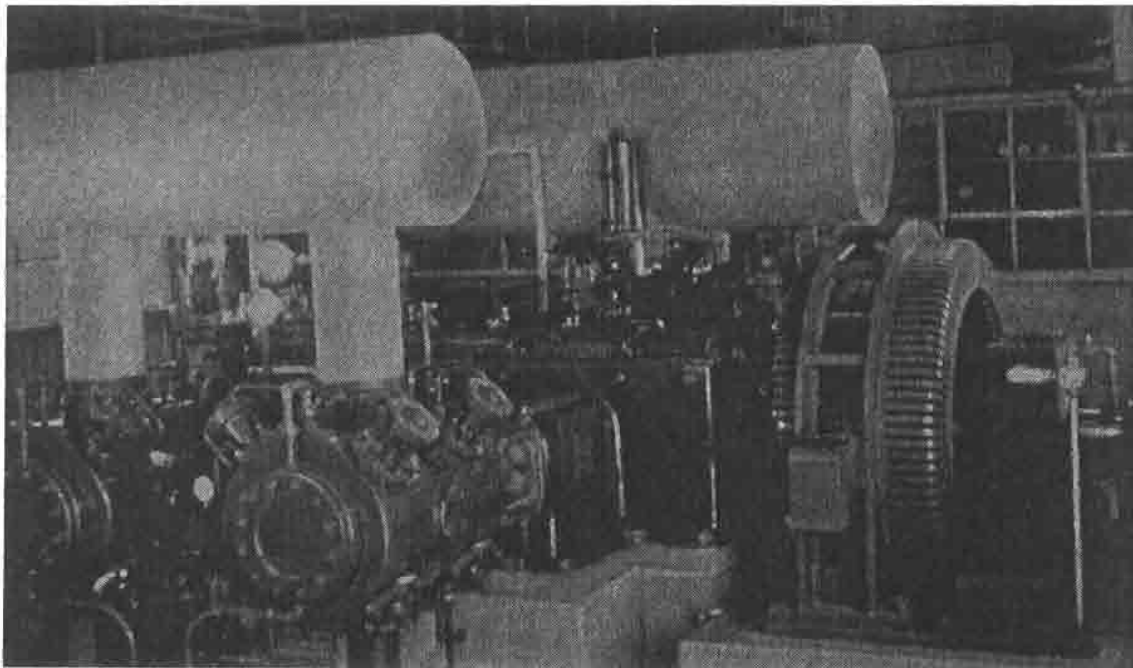


Figure 3-26. Manifold-type volume bottles are used where cylinders are operated in parallel, as on this two-stage, motor driven compressor. (Courtesy of Dresser-Rand)

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

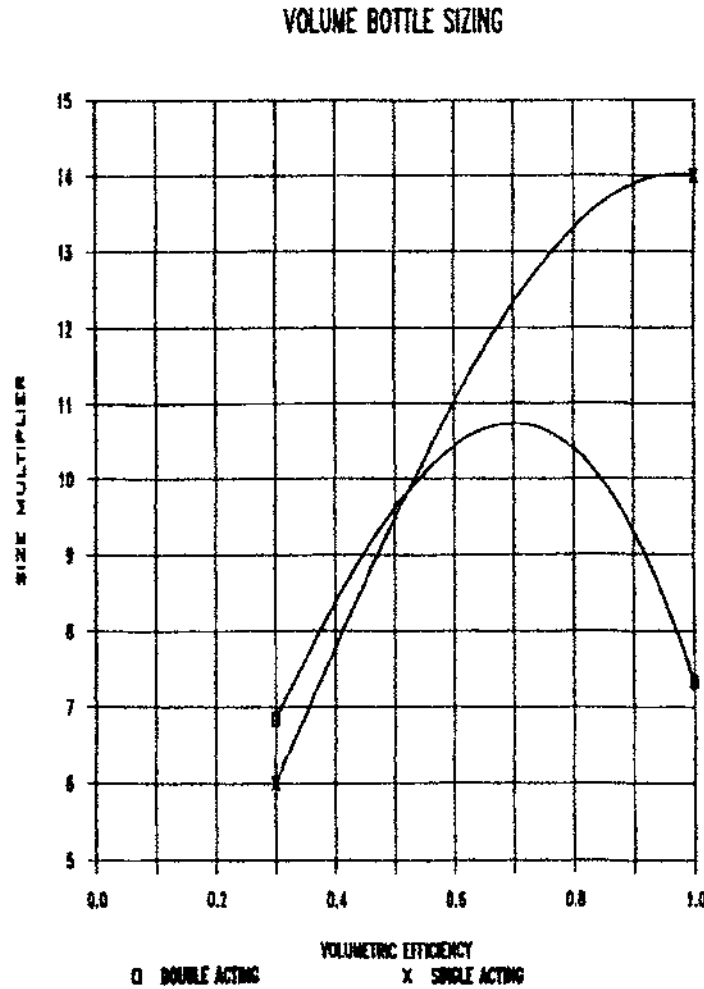


Figure 3-27. Volume bottle sizing graph.

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.

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

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.

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

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

$$L_b = 2d_b \quad (3.23)$$

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,

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

$$E_v = .823 \text{ suction volumetric efficiency}$$

For the discharge volumetric efficiency, use Equation 3.21.

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

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

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

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

$$Pd/Rev = 318.1 \text{ in.}^3$$

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

$$\text{Suction multiplier} = 13.5$$

$$\text{Discharge multiplier} = 10.4$$

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

$$Vol = 13.5 \times 318.1$$

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

$$Vol_d = 10.4 \times 318.1$$

$\text{Vol}_d = 3,308.1 \text{ in.}^3$ 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.

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

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

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

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

Use Equation 3.23 to calculate the length.

$$L_{bs} = 2 \times 16.3$$

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

$$L_{bd} = 2 \times 14.9$$

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

References

1. *Compressed Air and Gas Handbook*, Third Edition, New York, NY: Compressed Air and Gas Institute, 1961.
2. Joergensen, S. H., *Transient Valve Plate Vibration*, Proceedings of the 1980 Purdue Compressor Technology Conference, Purdue University, West Lafayette, IN, 1978, pp. 73–79.
3. Woollatt, D., *Increased Life for Feather Valves of Failure Caused by Impact*, Proceedings of the 1980 Purdue Compressor Technology Conference, Purdue University, West Lafayette, IN, 1980, pp. 293–299.
4. Davis, H., *Effects of Reciprocating Compressor Valve Design on Performance and Reliability*, Presented at Mechanical Engineers, London, England, October 13, 1970 (Reprint, Worthington Corp., Buffalo, NY).
5. Tuymmer, W. J., "Maintaining Compressor Valves," *Power*, April 1978, pp. 41–43.

6. White, K. H., "Prediction and Measurement of Compressor Valve Loss," *ASME 72-PET-4*, New York, NY: American Society of Mechanical Engineers, 1972.
7. Szenasi, F. R. and Wachel, J. C., "Analytical Techniques of Evaluation of Compressor-Manifold Response," *ASME 69-PET-31*, New York, NY: American Society of Mechanical Engineers, 1969.
8. Damewood, Glen and Nimitz, Walter, "Compressor Installation Design Utilizing an Electro-Acoustical System Analog," *ASME 61-WA-290*, New York, NY: American Society of Mechanical Engineers, 1961.
9. Nimitz, Walter, "Pulsation Effects on Reciprocating Compressors," *ASME 69-PET-2*, New York, NY: American Society of Mechanical Engineers, 1969, 1.
10. Wachel, J. C., "Consideration of Mechanical System Dynamics in Plant Design," *ASME 67-DGP-5*, New York, NY: American Society of Mechanical Engineers, 1967.
11. Nimitz, Walter W., *Pulsation and Vibration*, Part I. Causes and Effects, Part II. Analysis and Control. *Pipe Line Industry*, Part I, August 1968, pp. 36–39. Part II, September 1968, pp. 39–42.
12. Mowery, J. D., *Rod Loading of Reciprocating Compressors*, Proceedings of the 1978 Purdue Compressor Technology Conference, Purdue University, West Lafayette, IN, 1978, pp. 73–89.
13. Von Nimitz, Walter W., *Reliability and Performance Assurances in the Design of Reciprocating Compressor Installation—Part I Design Criteria, Part II Design Technology*, Proceedings of the 1974 Purdue Compressor Technology Conference, Purdue University, West Lafayette, IN, 1974, pp. 329–346.
14. Safriet, B. E., "Analysis of Pressure Pulsation in Reciprocating Piping Systems by Analog and Digital Simulation," *ASME 76-WA/DGP-3*, New York, NY: American Society of Mechanical Engineers, 1976.
15. API Standard 618, *Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services*, Fourth Edition, Washington, DC: American Petroleum Institute, 1995.
16. Scheel, Lyman F., *Gas Machinery*, Houston, TX: Gulf Publishing Company, 1972.
17. Evans, Frank L. Jr., *Equipment Design Handbook for Refineries and Chemical Plants*, Vol. 1, Second Edition, Houston, TX: Gulf Publishing Company, 1979.
18. Loomis, A. W., Editor, *Compressed Air and Gas Data*, Third Edition, Woodcliff Lake, NJ: Ingersoll-Rand, 1980.
19. Cohen, R., "Valve Stress Analysis for Fatigue Problems," *ASHRAE Journal*, January 1973, pp. 57–61.

20. Hartwick, W., "Power Requirement and Associated Effects of Reciprocating Compressor Cylinder Ends, Deactivated by Internal By-Passing," *ASME 75-DGP-9*, New York, NY: American Society of Mechanical Engineers, 1975.
21. Hartwick, W., "Efficiency Characteristics of Reciprocating Compressors," *ASME 68-WA /DGP-3*, New York, NY: American Society of Mechanical Engineers, 1968.
22. Carpenter, A. B., "Pulsation Problems in Plant Spotted by Analog Simulator," *The Oil and Gas Journal*, October 30, 1967, pp. 151–152.
23. *Engineering Data Book*, Ninth Edition, 1972, 4th Revision 1979, Tulsa, OK: Gas Processors Suppliers Association, 1972, 1979, pp. 4–12, 4–13.
24. Wachel, J. C. and Tison, J. D., *Vibrations in Reciprocating Machinery and Piping Systems*, Proceedings of the 23rd Turbomachinery Symposium, Texas A&M University, College Station, TX, 1994, pp. 243–272.

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 entrée 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

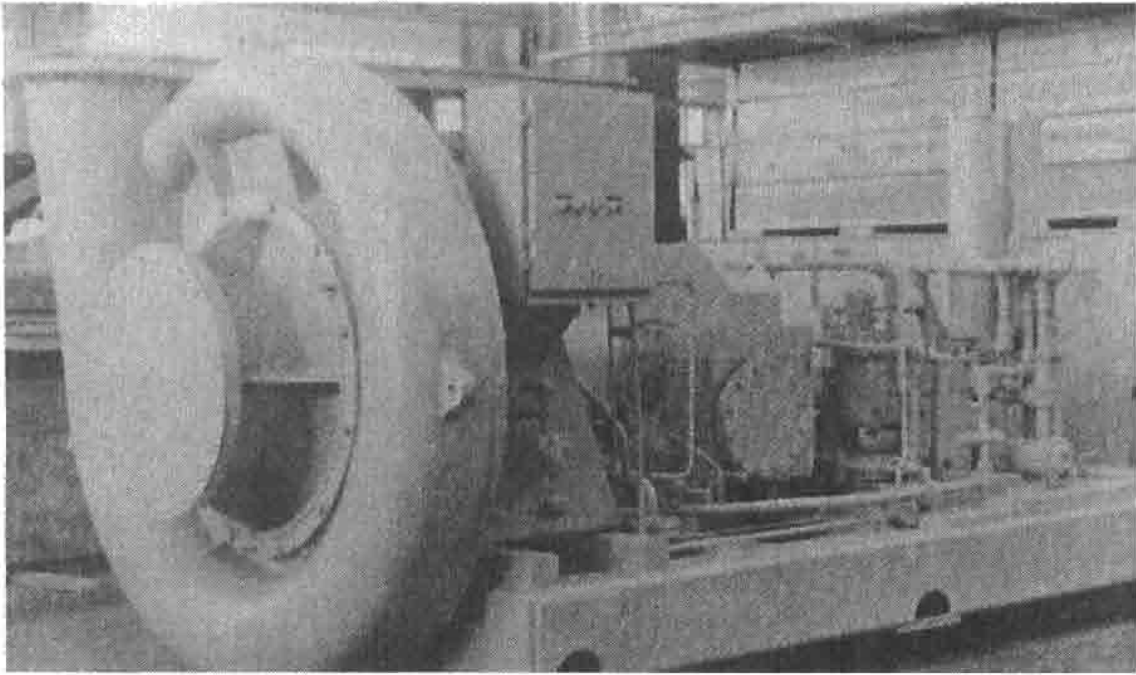


Figure 5-1. Single-stage, vertically split, overhung style centrifugal compressor. (Courtesy of Elliott Company)

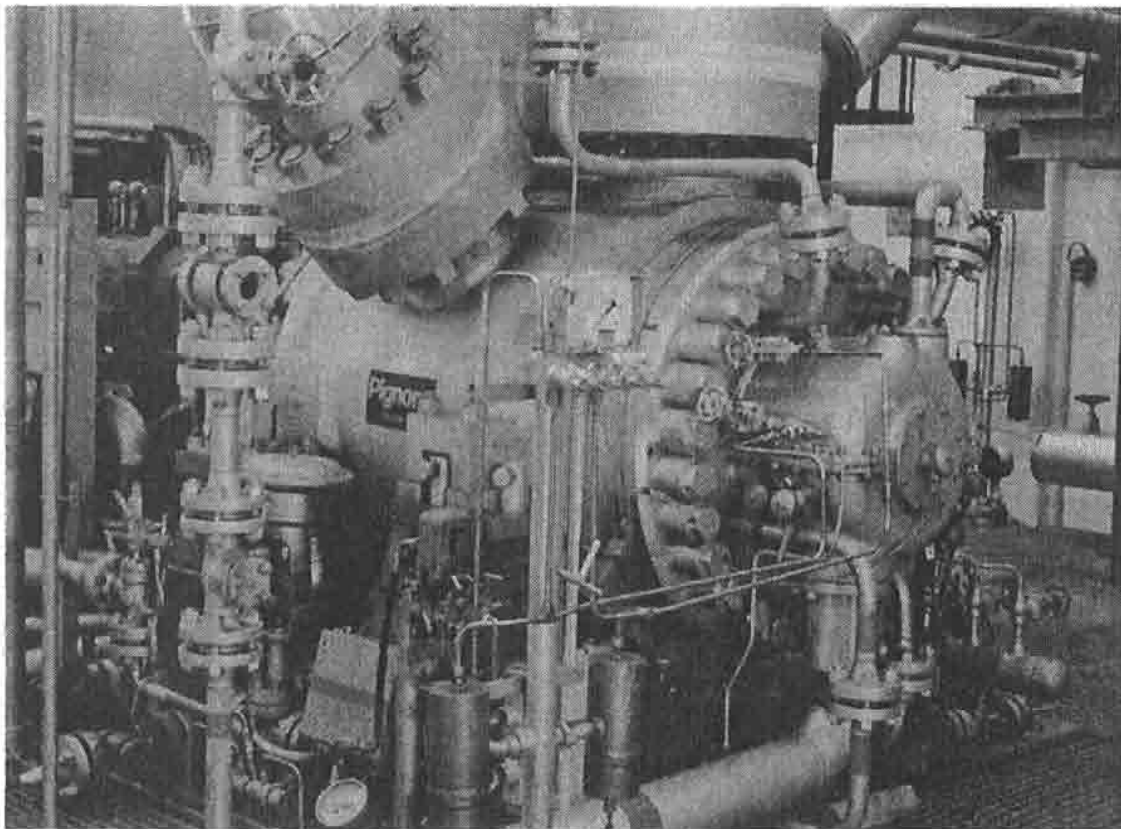


Figure 5-2. Multistage barrel type compressor. (Courtesy of Nuovo Pignone)

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.

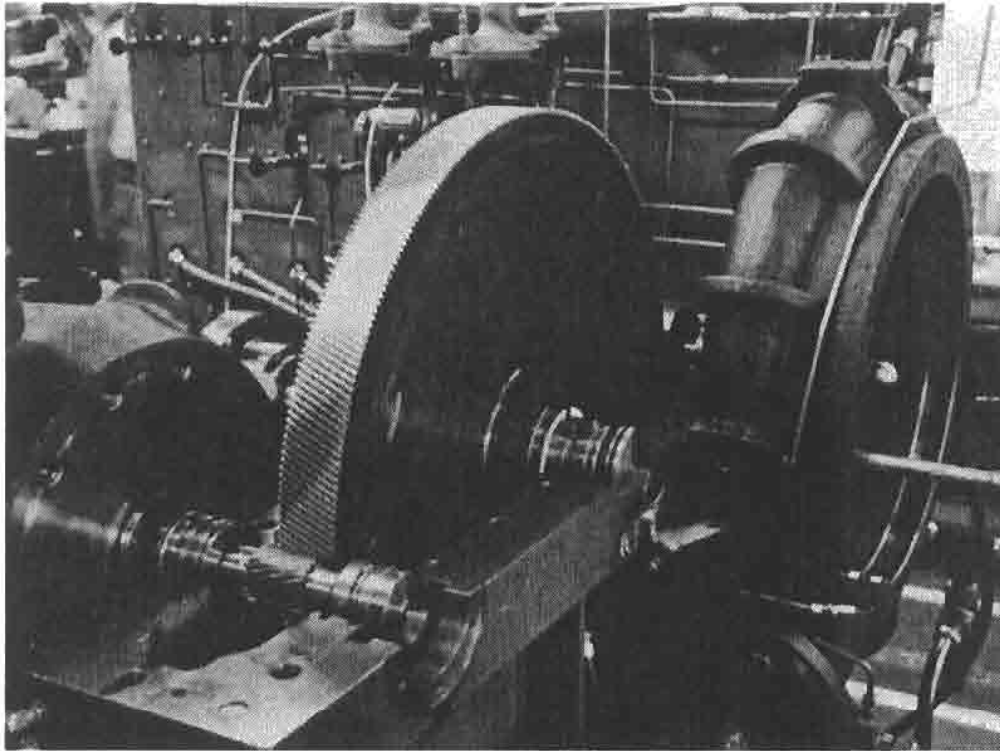


Figure 5-3. Overhung, gearbox mounted centrifugal compressor. (*Courtesy of Atlas Copco Comptec Inc.*)

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.

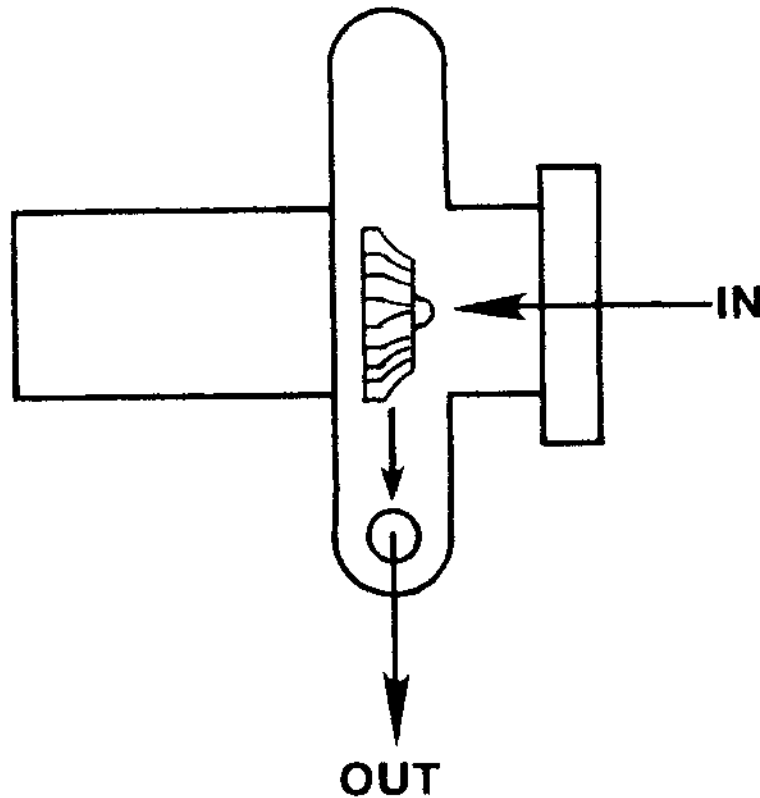


Figure 5-4. Diagram of a single-stage overhung type centrifugal compressor.

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.

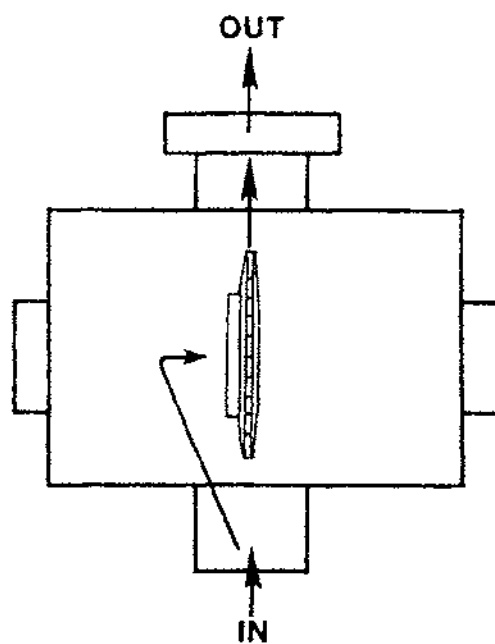


Figure 5-5. Diagram of a beam type single-stage compressor.

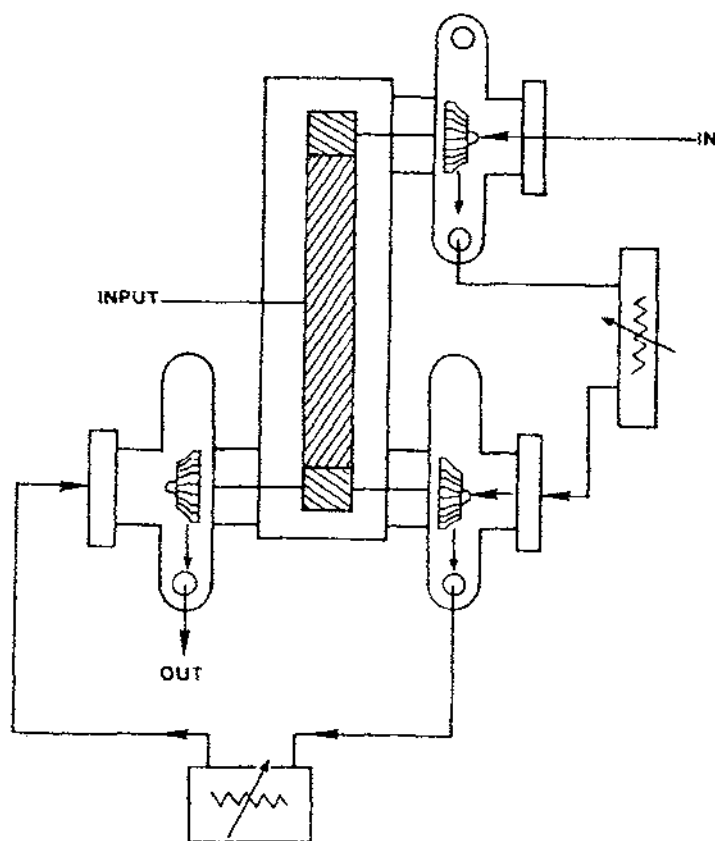


Figure 5-6. Flow diagram and schematic of an integrally geared compressor.

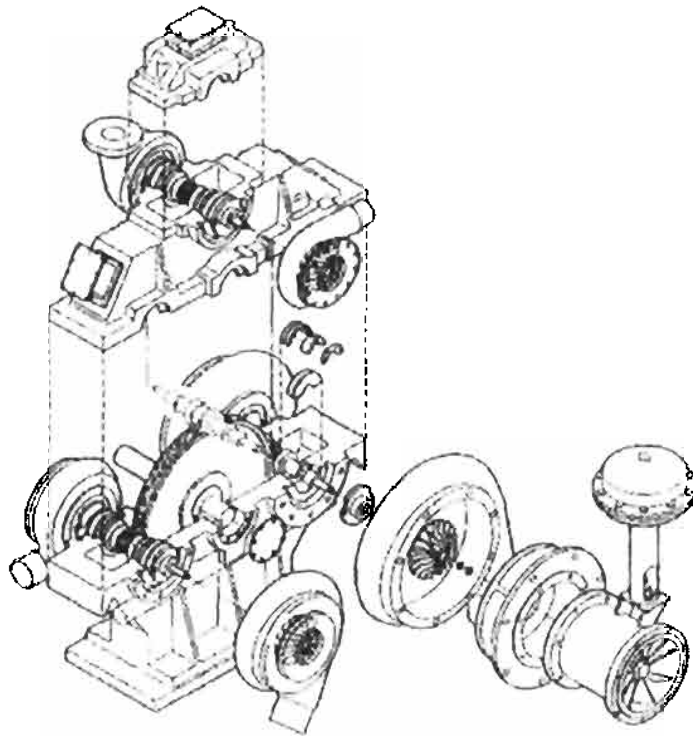


Figure 5-7. An exploded view of an integrally geared compressor. (Courtesy of Cooper Turbocompressor)

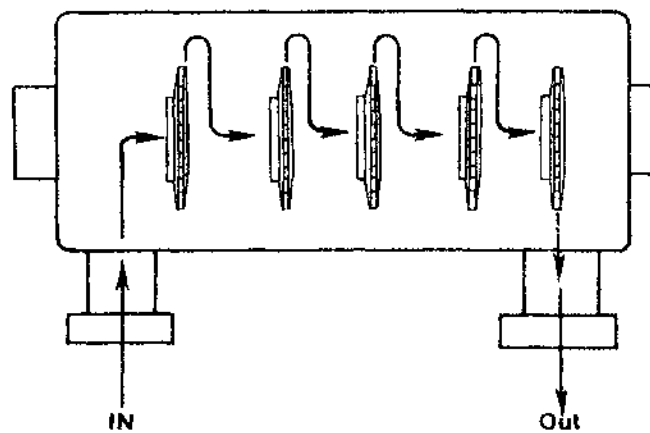


Figure 5-8. Diagram of a multistage centrifugal compressor with a straight-through flow path.

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

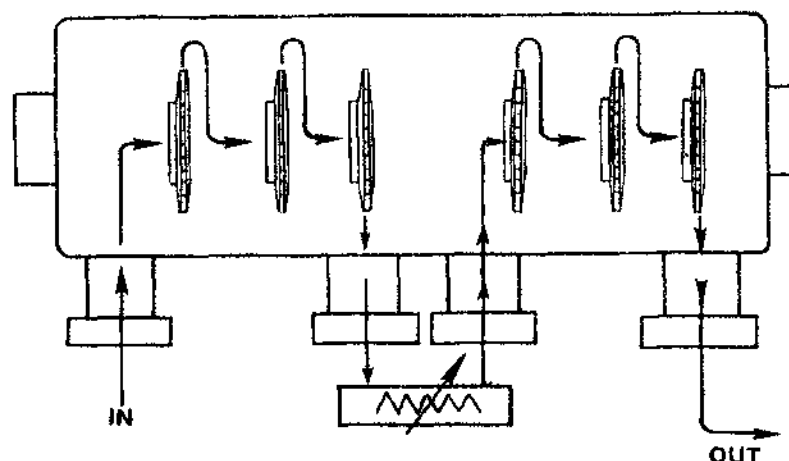


Figure 5-9. Diagram of an in-out arrangement with intercooling.

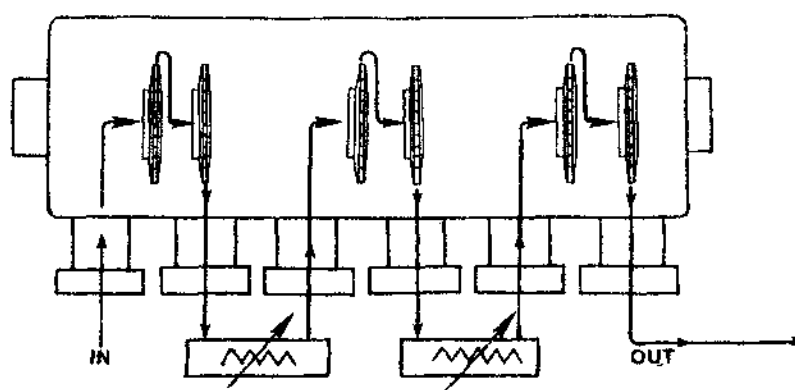


Figure 5-10. Diagram of a double-cooled centrifugal compressor.

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

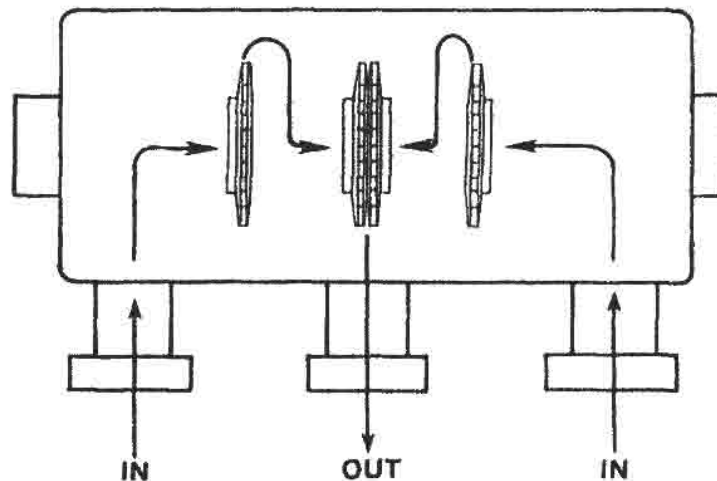


Figure 5-11. Diagram of a double-flow compressor with two inlets.

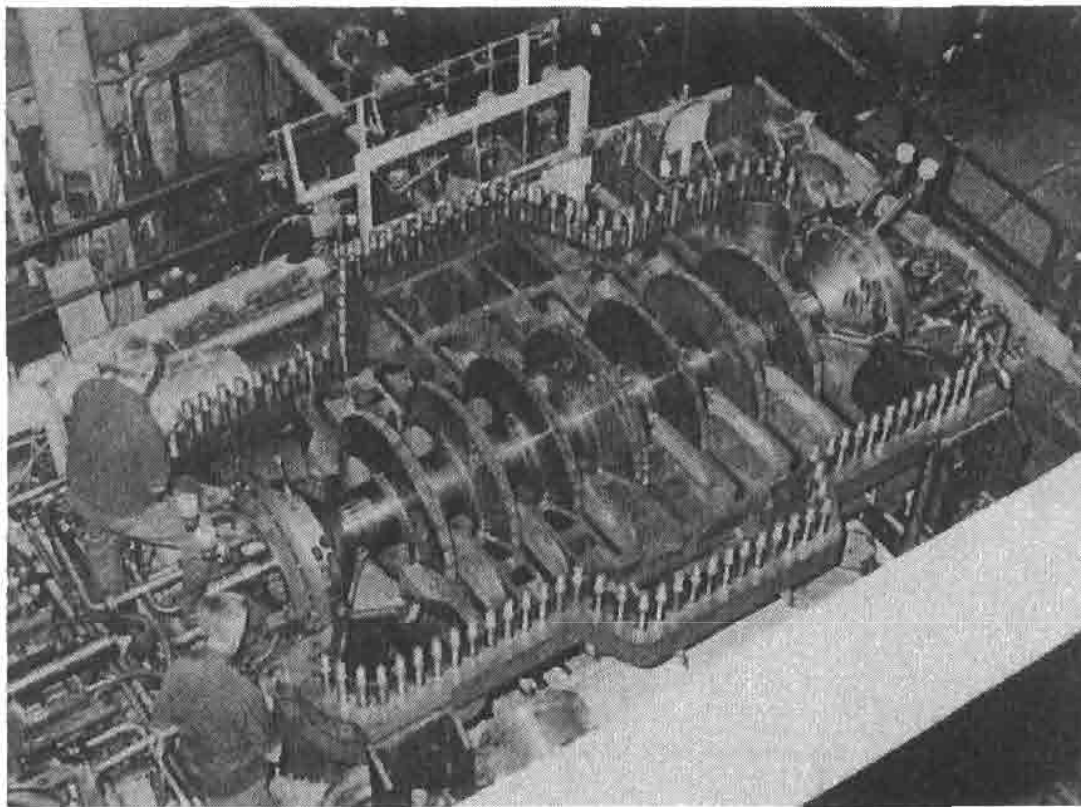


Figure 5-12. A double-flow compressor with inlets on each end and a common center discharge. (Courtesy of Elliott Company)

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-

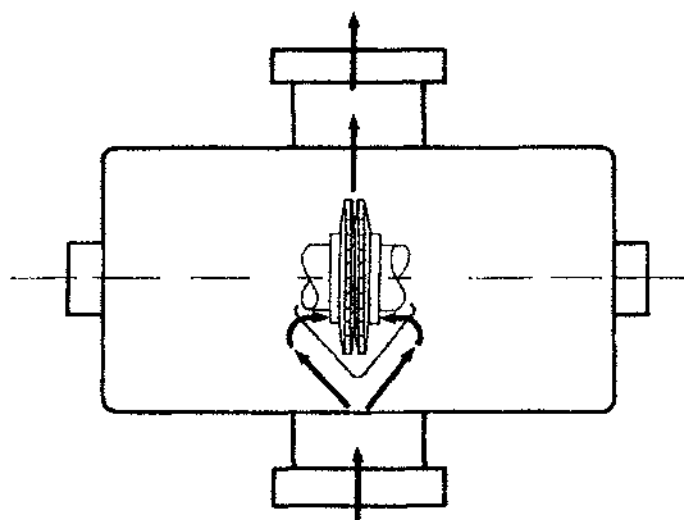


Figure 5-13. Diagram of a double-flow compressor with flow split internally.

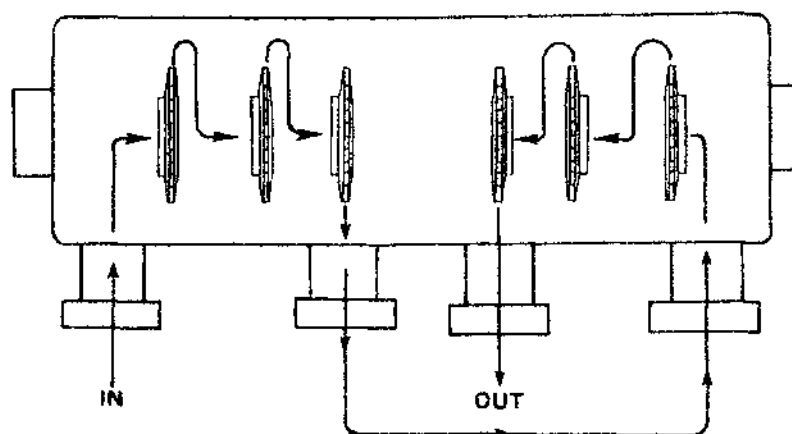


Figure 5-14. Diagram of an arrangement used to overcome a thrust balance problem.

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

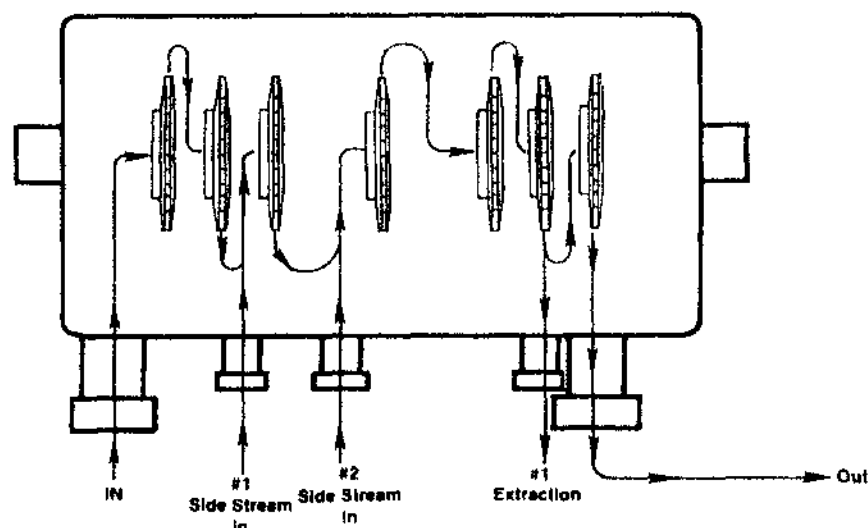


Figure 5-15. Diagram of flow path through a sidestream compressor.

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

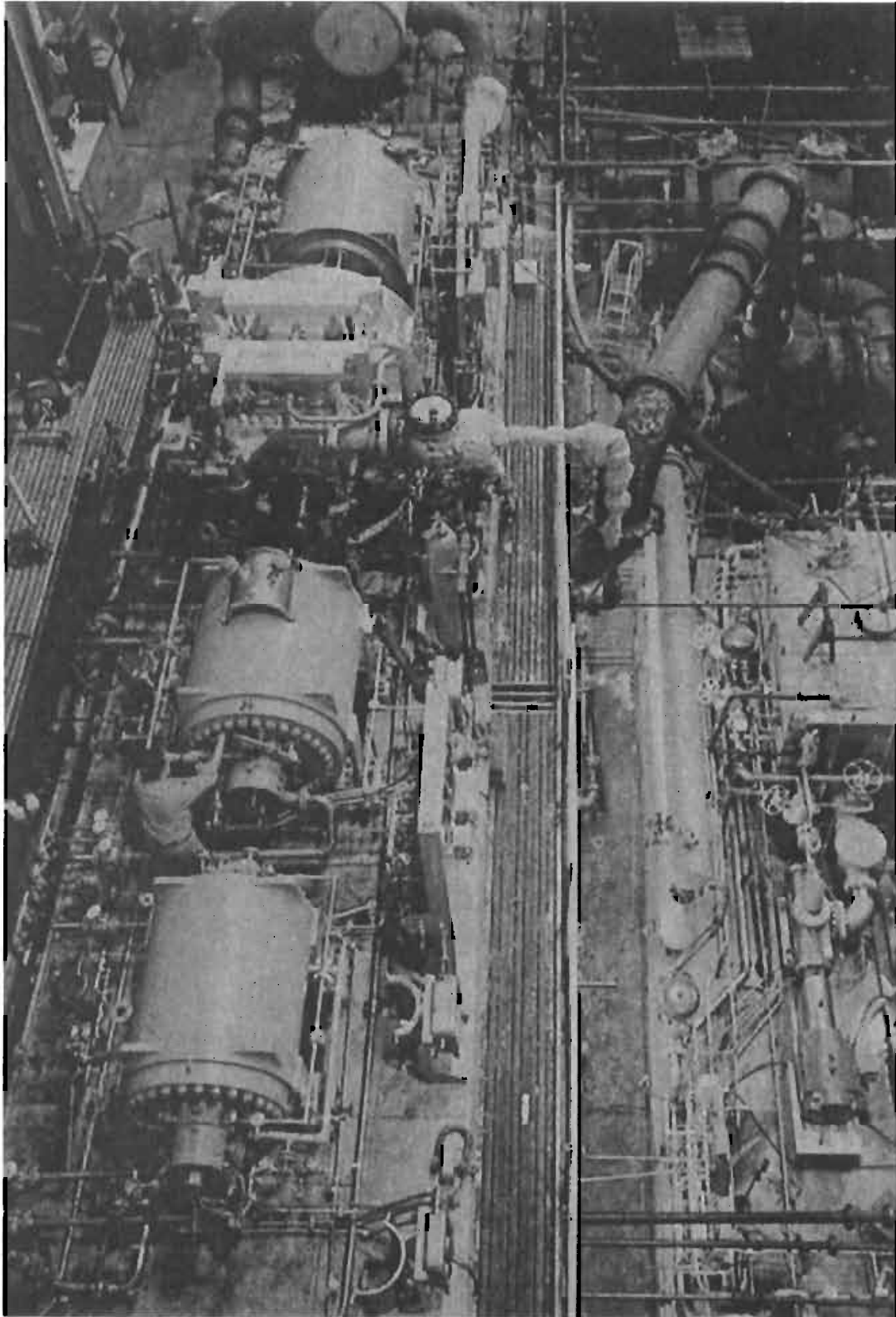


Figure 5-16. A tandem driven multi-body centrifugal compressor train with a steam turbine driver. (*Courtesy of Demag Delaval Turbomachinery Corp.*)

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.

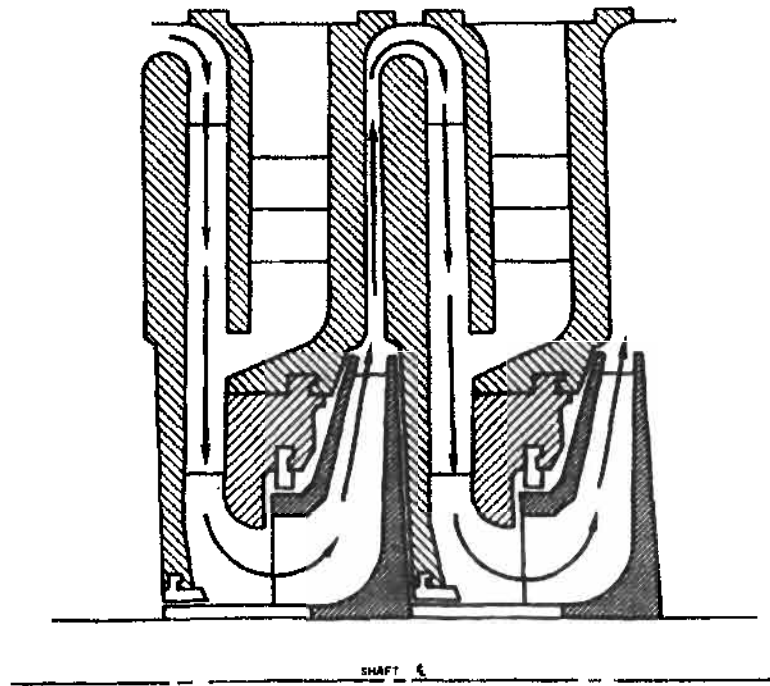


Figure 5-17. Flow path through typical stages on a multistage unit. (Courtesy of Elliott Company)

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

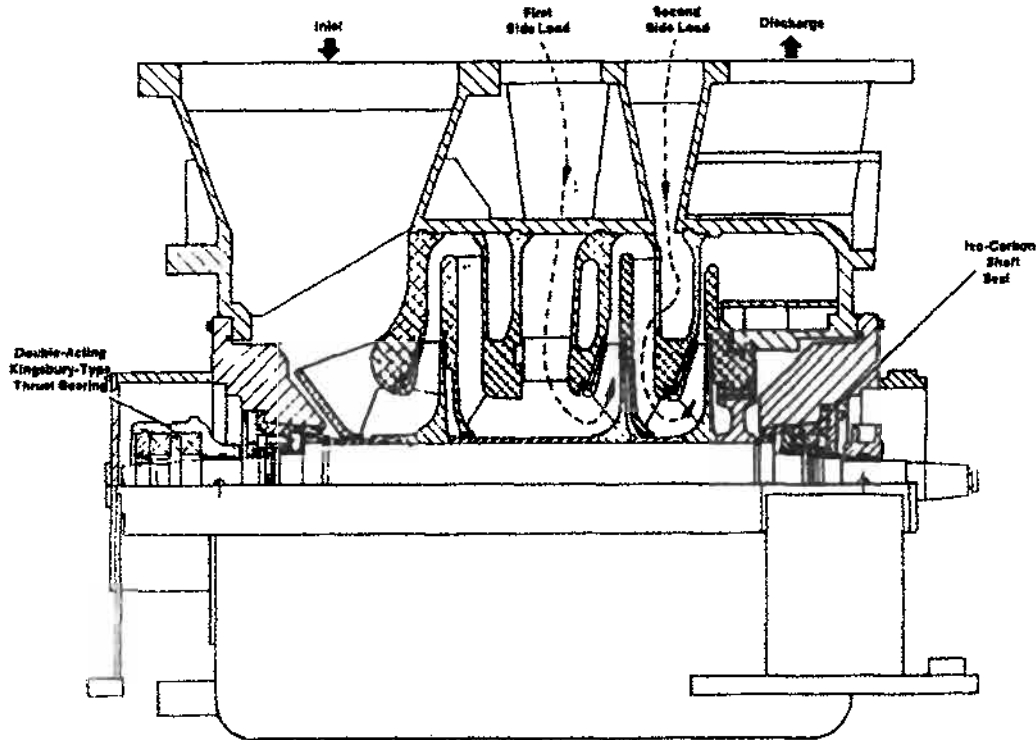


Figure 5-18. Two methods of directing sidestream flows into through flows.
(Courtesy of Elliott Company)

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-

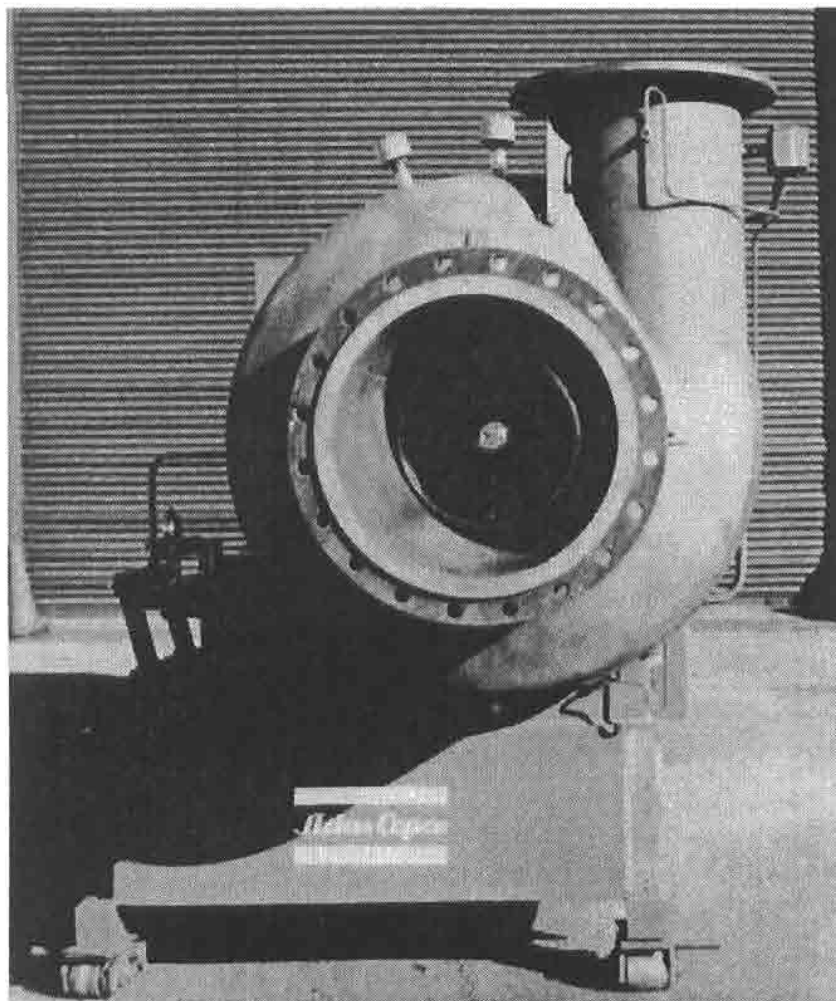


Figure 5-19. The impeller blades can be seen in this view through the inlet of a single-stage compressor. (Courtesy of Atlas Copco Comptec, Inc.)

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

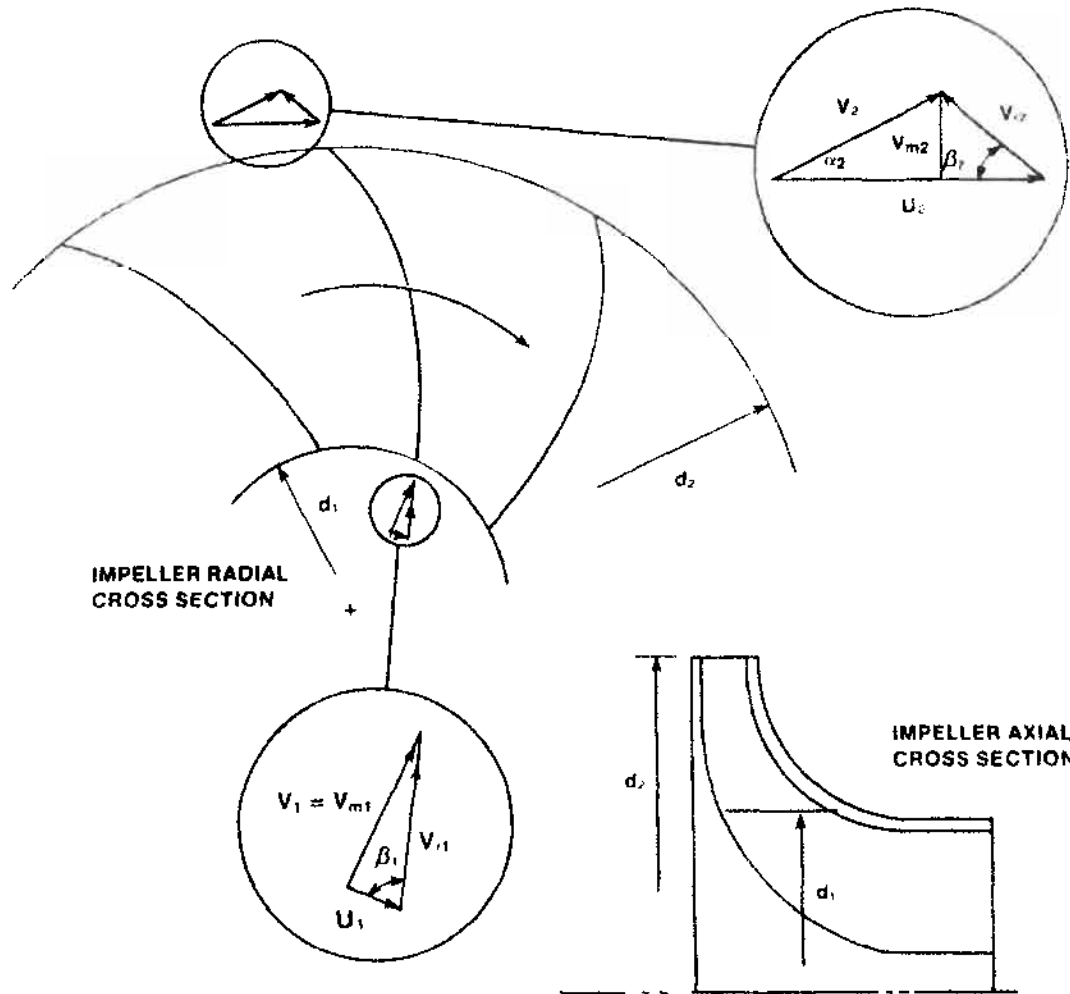


Figure 5-20. Impeller inlet and outlet flow vector triangles.

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 diagrammatically 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.

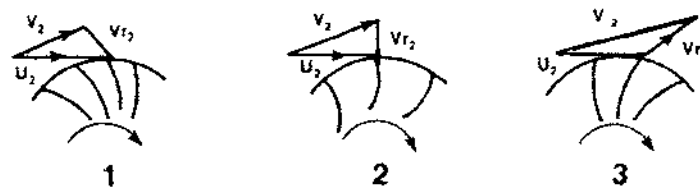
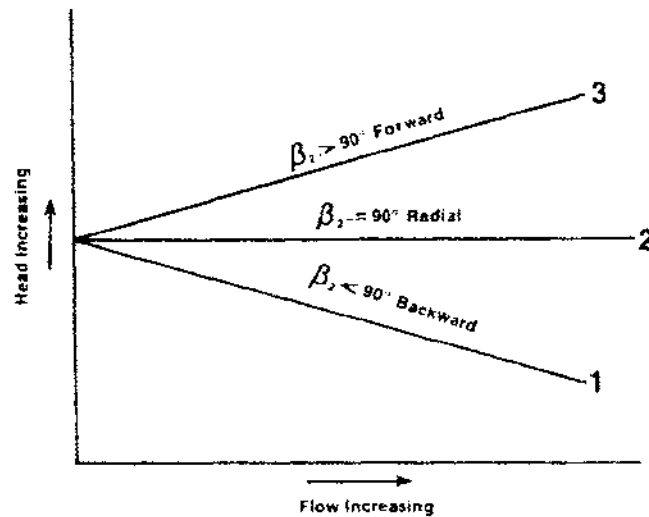


Figure 5-21. Diagram depicting backwards, radial and forward curved blades.

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, $\beta_2 = 90^\circ$, 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

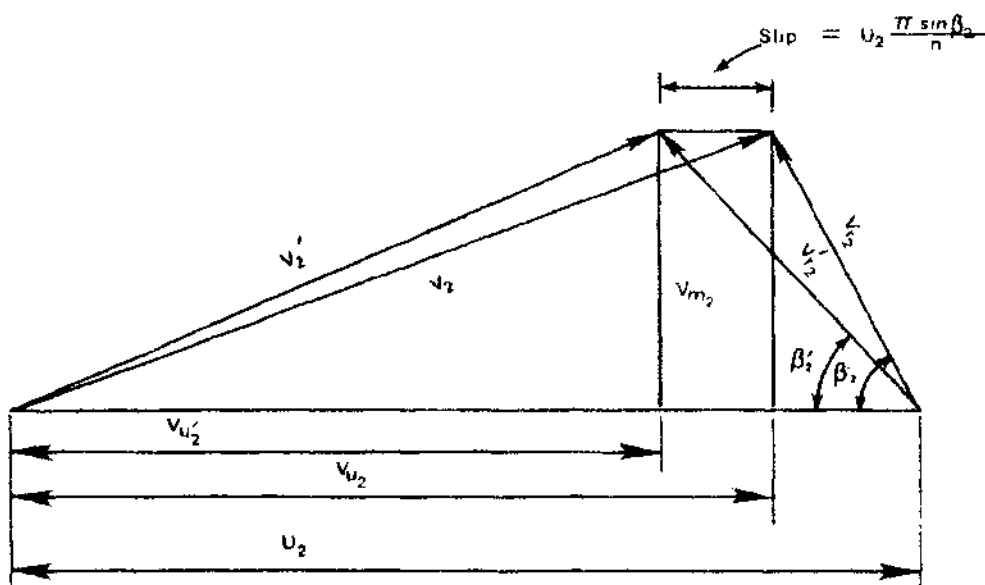


Figure 5-22. Discharge velocity vector triangle showing the effect of slip.

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:

$$\zeta_i = \frac{V_{u2}}{u_2} \quad (5.1)$$

where

V_{u2} = tangential component of the absolute velocity
 u_2 = impeller tip velocity

The ideal head input to the stage is given by

$$H_{in\ ideal} = (1/g)\zeta_i u_2^2 \quad (5.2)$$

The Stodola slip factor is defined as

$$\text{Slip} = u_2 \frac{\pi \sin \beta_2}{n_v} \quad (5.3)$$

where

β_2 = geometric vane exit angle
 n_v = number of vanes in the impeller

The slip factor SF follows.

$$\text{SF} = \frac{V'_{u2}}{V_{u2}} \quad (5.4)$$

Reference is made to Figure 5-22, where

$$V'_{u2} = V_{u2} - \text{slip} \quad (5.5)$$

Substituting into Equation 5.4 yields the following slip factor equation:

$$SF = 1 - \frac{u_2}{V_{u2}} \left(\frac{\pi \sin \beta_2}{n_v} \right) \quad (5.6)$$

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.

$$\zeta = \frac{V_{u2}}{u_2} (SF) \quad (5.7)$$

By replacing the ideal work input coefficient with actual work input coefficient, the actual head input can be written as

$$H_{in} = (1/g)\zeta u_2^2 \quad (5.8)$$

If the head coefficient is written as

$$\mu = \eta \zeta \quad (5.9)$$

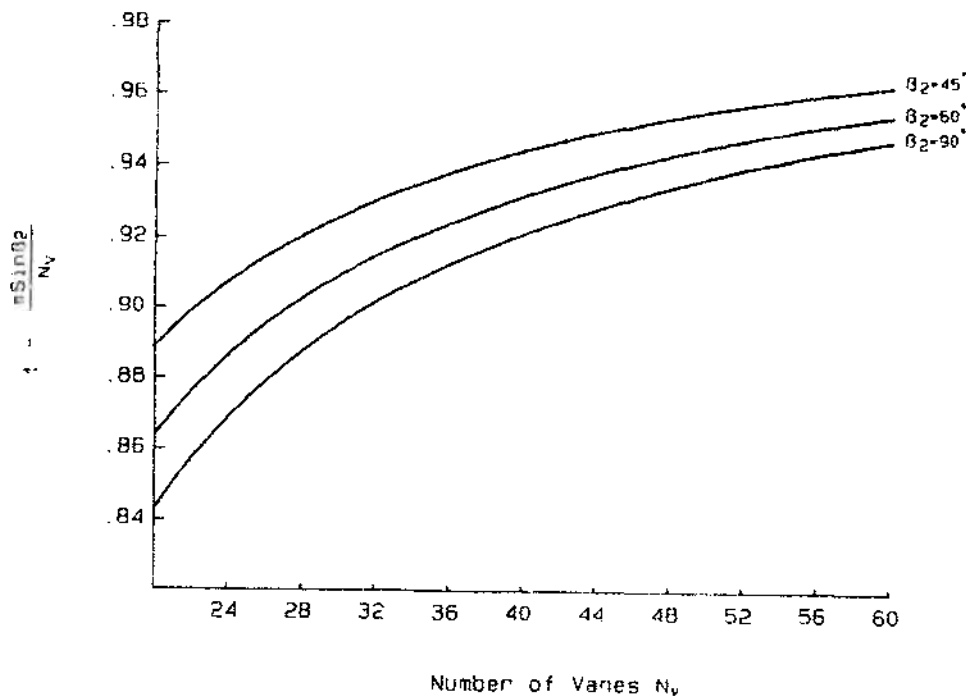


Figure 5-23. Geometric relationship of Stodola slip function.

where

η = stage efficiency, then

$$H_{\text{out}} = (1/g)\mu u_2^2 \quad (5.10)$$

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.

$$H_a = \frac{\mu_a u_2^2}{g} = Z_{\text{avg}} RT_1 \frac{k}{k-1} (r_p^{\frac{k-1}{k}} - 1) \quad (5.11)$$

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.

$$H_p = \frac{\mu_p u_2^2}{g} = Z_{\text{avg}} RT_1 \frac{n}{n-1} (r_p^{\frac{n-1}{n}} - 1) \quad (5.12)$$

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

$$N_s = \frac{NQ_1^{1/2}}{H_a^{3/4}} \quad (5.13)$$

$$D_s = \frac{DH_a^{1/4}}{Q_1^{1/2}} \quad (5.14)$$

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

$$R_i = \frac{2 + \cot \beta_2}{4} \quad (5.15)$$

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-

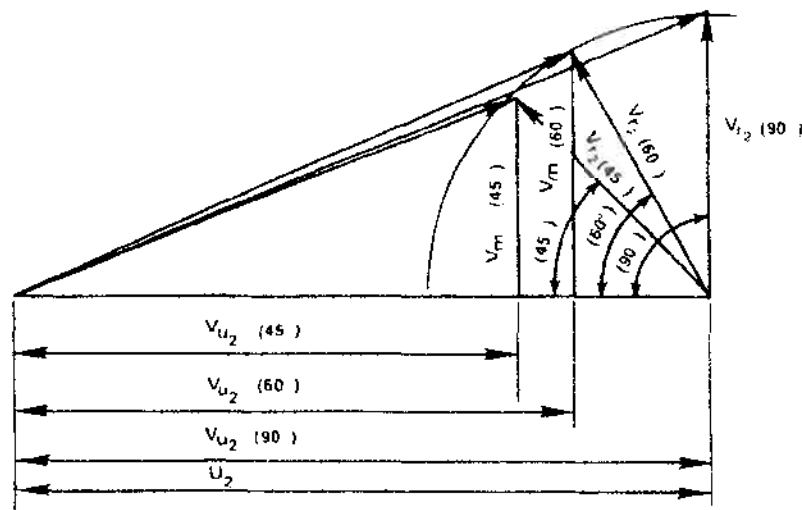


Figure 5-24. Vector tip triangle without slip, showing the effect of different exit vane angles.

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,

$$R_i = .50$$

$$.50 \times .90 = .45$$

$$.50 \times .60 = .30$$

$$.45 + .30 = .75 \text{ net stage efficiency}$$

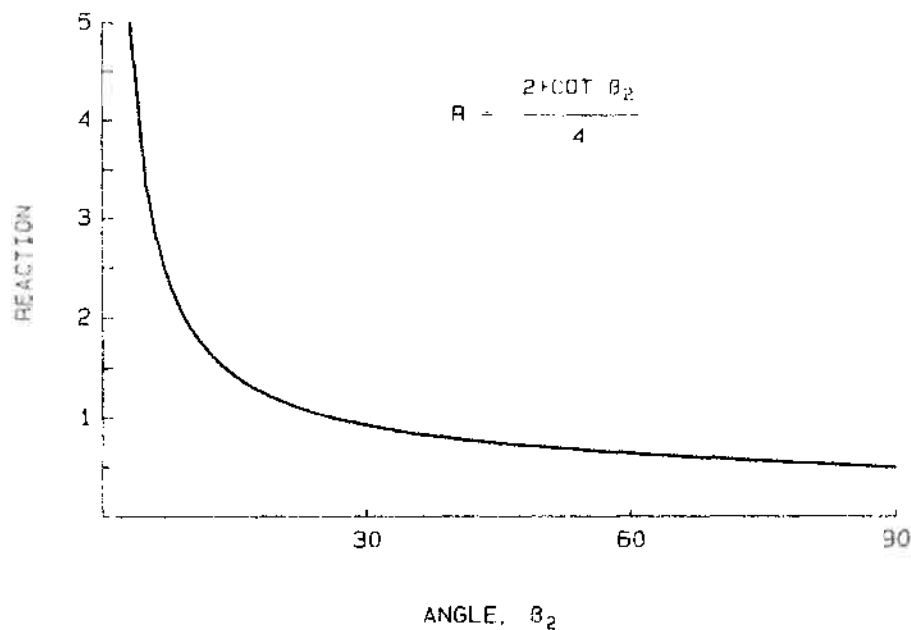


Figure 5-25. Theoretical reaction without slip.

For a 45° backward leaning impeller,

$$R_i = .75$$

The diffuser then converts $1 - R_i$ or .25

$$.75 \times .90 = .68$$

$$.25 \times .60 = .15$$

$$.68 + .15 = .83 \text{ net stage efficiency}$$

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.

$$T_2 = T_1 r_p^{\frac{n-1}{n}} \quad (5.16)$$

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

$$N = \frac{u_2}{\pi d_2} \quad (5.17)$$

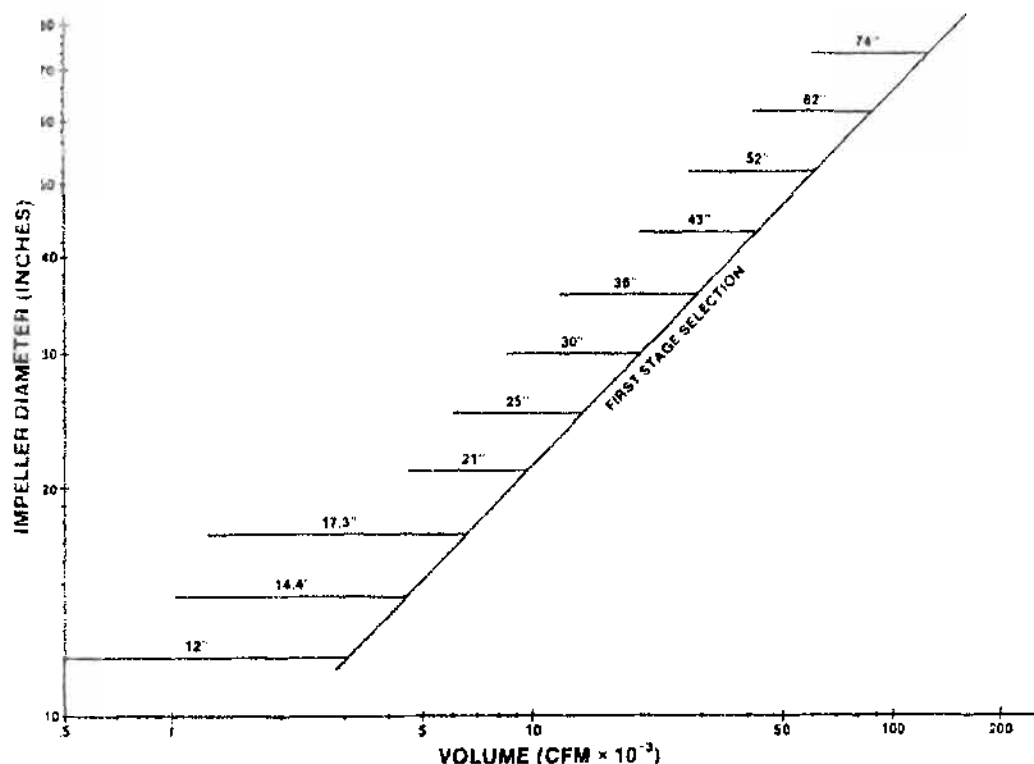


Figure 5-26. Estimation of impeller diameter using inlet volume.

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

$$Q_{ls} = \frac{Q_{in}}{\left(r_p^{1-\frac{1}{z}}\right)^n} \quad (5.18)$$

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 .

$$\delta = 700 \frac{Q_i}{Nd_2^3} \quad (5.19)$$

where

Q_i = volumetric flow, ft³/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)

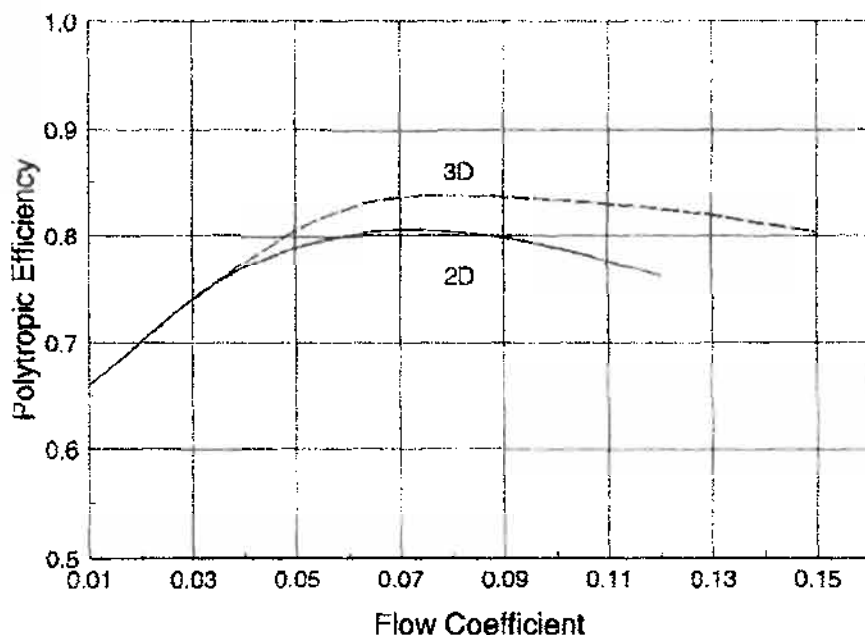


Figure 5-27. Centrifugal stage efficiency vs. flow coefficient for 2D and 3D blading.

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

$$t_w = \frac{t_i + t_d BP}{1 + BP} \quad (5.20)$$

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:

$$Q_1 = 6,171 \text{ cfm inlet volume}$$

$$w_m = 437.5 \text{ lbs/min}$$

$$mw = 28.46 \text{ molecular weight}$$

$$P_1 = 14.7 \text{ psia inlet pressure}$$

$$t_1 = 90.0^\circ\text{F inlet temperature}$$

$$T_1 = 550^\circ\text{R absolute inlet temperature}$$

$$R_m = 54.29 \text{ specific gas constant}$$

Add the following conditions to complete the application:

$$k = 1.395 \text{ isentropic exponent for air}$$

$$P_2 = 40 \text{ psia discharge pressure}$$

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,

$$r_p = 40/14.7$$

$$r_0 = 2.721 \text{ pressure ratio}$$

Step 2. Calculate the total required polytropic head using Equation 2.73, assuming a value for $Z_{\text{avg}} = 1$:

$$H_p = 1 \times 54.29 \times 550 \times 2.646 (2.721^{.378} - 1)$$

$$H_p = 36,338.4 \text{ ft-lb/lb overall polytropic head}$$

Step 3. Determine the number of stages, z , required using the recommended 10,000 ft-lb/lb head per stage.

$$z = 36,338.4/10,000$$

$$z = 3.63 \text{ stages, round off to 4}$$

Calculate a new head per stage using four stages:

$$H_p = 36,338.4/4$$

$$H_p = 9,085 \text{ ft-lb/lb}$$

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.

$$u_2 = (9,085 \times 32.2/.48)^.5$$

$$u_2 = 780.7 \text{ fps impeller tip speed}$$

From Figure 5-26 and the inlet volume, select an initial impeller diameter.

$$d_2 = 17.3 \text{ inches impeller diameter}$$

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.

$$T_2 = 550 \times 2.721^{.359}$$

$$T_2 = 787.8^\circ\text{R}$$

$$t_2 = 787.8 - 460$$

$$t_2 = 327.8^\circ\text{F discharge temperature}$$

Calculate the power required using Equation 2.78 and the recommended 1% for mechanical losses.

$$W_p = \frac{437.5 \times 36,338.4}{33,000 \times .79} + .01W_p$$

$$W_p = 609.8 + 6.1$$

$$W_p = 615.9 \text{ hp total for the compressor}$$

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:

$$mw = 53.0$$

$$k_1 = 1.23$$

$$Z_1 = 0.97$$

$$t_1 = 85^\circ\text{F}$$

$$P_1 = 40 \text{ psia}$$

$$P_2 = 120 \text{ psia}$$

$$w = 2,050 \text{ lb/min}$$

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.

$$T_1 = 85 + 460$$

$$T_1 = 545^\circ\text{R}$$

Step 3. Calculate the polytropic exponent using Equation 2.71. Assume an efficiency of $\eta_p = .75$. Use as $k_{\text{avg}} = k_1 = 1.23$.

$$\frac{n-1}{n} = \frac{1.23-1}{1.23} \times \frac{1}{.75}$$

$$\frac{n-1}{n} = .249$$

$$\frac{n}{n-1} = 4.011$$

$$n = 1.332$$

Step 4. From Equation 2.64,

$$r_p = 120/40$$

$$r_p = 3.0 \text{ pressure ratio}$$

Step 5. Calculate the estimated discharge temperature using Equation 5.14.

$$T_2 = 545 \times 3^{.249}$$

$$T_2 = 716.7^\circ\text{R absolute discharge temperature estimate}$$

Convert to $^\circ\text{F}$:

$$t_2 = 716.7 - 460$$

$$t_2 = 256.7^\circ\text{F}$$

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.

$$w = 1.01 \times 2,050$$

$$w = 2,070.5 \text{ lb/min net flow to the impeller.}$$

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.

$$t_w = \frac{85 + 256.7(.01)}{1.01}$$

$$t_w = 86.7^\circ\text{F corrected impeller inlet temperature}$$

Convert to absolute:

$$T_w = 86.7 + 460$$

$$T_w = 546.7^\circ\text{R}$$

Step 6. Substitute into Equation 2.10 and using $144 \text{ in}^2/\text{ft}^2$.

$$Q_1 = \frac{.97 \times 29.15 \times 546.7}{40 \times 144} \times 2,070.5$$

$$Q_1 = 5,557 \text{ cfm inlet flow to the impeller}$$

Step 7. Calculate the total required polytropic head using Equation 2.73, assuming the average value of $Z_{\text{avg}} = .97$.

$$H_p = 0.97 \times 29.15 \times 546.7 \times 4.011(3^{2.49} - 1)$$

$$H_p = 19,508 \text{ ft-lb/lb total polytropic head required}$$

Step 8. Determine the number of stages required using the modified rule of thumb on head per stage, H_{stg} .

$$H_{\text{stg}} = 10,000 - ((53 - 29)100)$$

$$H_{\text{stg}} = 7,600 \text{ ft-lb/lb}$$

$$z = \frac{19,508}{7,600}$$

$$z = 2.57 \text{ stages, round off to 3}$$

Calculate a new head per stage using three stages.

$$H_p = 19,508/3$$

$$H_p = 6,502.7 \text{ ft-lb/lb head per stage}$$

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.

$$u_2 = (6,502.7 \times 32.2/.48)^{1/2}$$

$$u_2 = 660.5 \text{ fps impeller tip speed}$$

Step 10. From Figure 5-26 and the inlet volume, select an initial impeller diameter.

$$d_2 = 17.3 \text{ in. initial impeller diameter}$$

Use Equation 5.17 to calculate the initial speed, N .

$$N = \frac{60 \times 12 \times 660.5}{\pi \times 17.3}$$

$$N = 8,750 \text{ rpm compressor shaft speed}$$

Step 11. The volume into the last impeller is calculated with the use of Equation 5.18.

$$Q_{1s} = \frac{5,557}{(3^{1-1/3})^{1/1.332}}$$

$Q_{1s} = 3,206$ 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.

$$T_2 = 546.7 \times 3.236$$

$T_2 = 708.5^\circ\text{R}$ absolute discharge temperature

$$t_2 = 708.5 - 460$$

$t_2 = 248.5^\circ\text{F}$ 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.

$$mw = 69$$

$$k_1 = 1.35$$

$$k_2 = 1.33$$

$$Z_1 = .98$$

$$Z_2 = .96$$

$$t_1 = 80^\circ\text{F}$$

$$P_1 = 24 \text{ psia}$$

$$P_2 = 105 \text{ psia}$$

$$w = 3,200 \text{ lbs/min}$$

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.

$$T_1 = 80 + 460$$

$$T_1 = 540^\circ\text{R}$$

Substitute into Equation 2.10 and using the conversion constant of 144 in.²/ft², 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 $\eta_p = .75$.

$$k_{\text{avg}} = (1.35 + 1.33)/2$$

$$k_{\text{avg}} = 1.34$$

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,

$$r_p = 105/24$$

$$r_p = 4.375 \text{ overall pressure ratio}$$

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.

$$T_2 = 540 \times 4.375^{.338}$$

$$T_2 = 889.3^\circ\text{R}$$

$$t_2 = 889.3 - 460$$

$$t_2 = 429.3^\circ\text{F discharge temperature}$$

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.

$$r_p = 4.375^{1/2} \text{ pressure ratio per section}$$

$$r_p = 2.092$$

$$P_2 = 2.092 \times 24$$

$$P_2 = 50.2 \text{ psia first section discharge pressure}$$

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.

$$T_2 = 540(2.175)^{.338}$$

$$T_2 = 702.2^\circ\text{R}$$

$$t_2 = 702.2 - 460$$

$$t_2 = 242.2^\circ\text{F first section discharge temperature}$$

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.

$$T_2 = 565(2.092)^{.338}$$

$$T_2 = 725^\circ\text{R}$$

$$t_2 = 725 - 460$$

$$t_2 = 265^\circ\text{F discharge temperature}$$

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.

$$T_2 = 540(2.271)^{.338}$$

$$T_2 = 712.5^\circ\text{R}$$

$$t_2 = 712.5 - 460$$

$$t_2 = 252.5^\circ\text{F}$$

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.

$$T_2 = 565(2.0)^{.338}$$

$$T_2 = 714.2^\circ\text{R}$$

$$t_2 = 714.2 - 460$$

$$t_2 = 254.2^\circ\text{F}$$

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 $Z_{2\text{avg}} = .97$.

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.

$$H_{\text{stg}} = 10,000 - ((69 - 29)(100))$$

$$H_{\text{stg}} = 6,000 \text{ ft-lb/lb}$$

Divide the total head per section by the allowable head per stage to develop the number of stages required in each section.

Section 1

$$z = 11,074/6,000$$

$$z = 1.84 \text{ stages, round off to 2}$$

Section 2

$$z = 9,576/6,000$$

$$z = 1.6 \text{ stages, round off to 2}$$

Step 9. Calculate a head per stage for each section based on two stages each.

Section 1

$$H_p = 11,074/2$$

$$H_p = 5,537 \text{ ft-lb/lb head per stage, Section 1}$$

Section 2

$$H_p = 9,576/2$$

$$H_p = 4,788 \text{ ft-lb/lb head per stage, Section 2}$$

Use the geometric portion of Equation 5.12 to calculate the tip speed. Assume $\mu_p = .48$ for the pressure coefficient.

Section 1

$$u_2 = (5,537 \times 32.2/.48)^{.5}$$

$$u_2 = 609.5 \text{ fps tip speed first two stages}$$

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 $t_1 = 105^\circ\text{F}$. 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

$$Q_{1s} = \frac{10,971}{(2.271^{1/2})^{1/1.511}}$$

$Q_{1s} = 8,363.4$ cfm last stage volume, Section 1

Section 2

$$Q_{1s} = \frac{5,194}{(2.0^{1/2})^{1/1.511}}$$

$Q_{1s} = 4,129.4$ 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

$$T_2 = 540(2.271)^{.323}$$

$$T_2 = 703.8^\circ\text{R}$$

$$t_2 = 703.8 - 460$$

$$t_2 = 243.8^\circ\text{F final Section 1 discharge temperature.}$$

Section 2

$$T_2 = 565(2.0)^{.319}$$

$$T_2 = 704.8^\circ\text{R}$$

$$t_2 = 704.8 - 460$$

$$t_2 = 244.8^\circ\text{F final Section 2 discharge temperature.}$$

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

$$W_p = \frac{3,200 \times 11,074}{33,000 \times .795}$$

$$W_p = 1,350.8 \text{ hp gas horsepower, Section 1}$$

Section 2

$$W_p = \frac{3,200 \times 9,576}{33,000 \times .787}$$

$$W_p = 1,179.9 \text{ hp gas horsepower, Section 2}$$

Combine the two gas horsepower values and add 1% for the mechanical losses.

$$W_p = 1,350.8 + 1,179.9 + 25.3$$

$$W_p = 2,556.0 \text{ hp compressor shaft power}$$

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.

$$Q_i \propto N \quad (5.21)$$

$$H_p \propto N^2 \quad (5.22)$$

$$W_p \propto N^3 \quad (5.23)$$

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 pres-

sure 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 pressures 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.

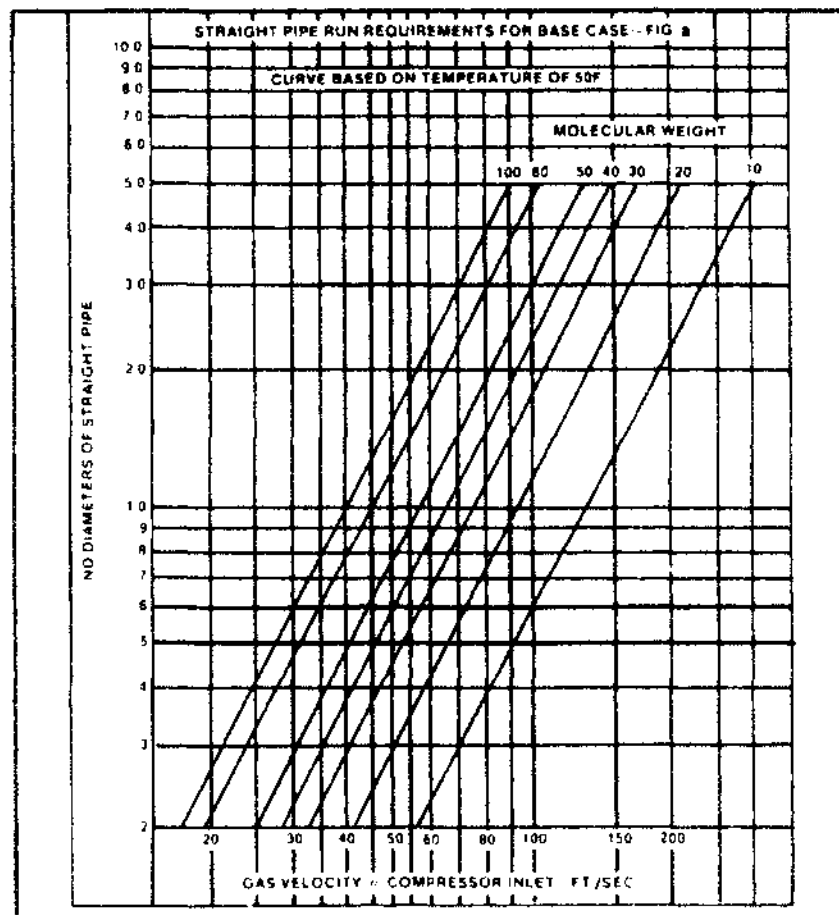
One question that arises quite often is the orientation of inlet piping and its influence on compressor performance. The flow into the impeller has been assumed axial or radial, depending on the impeller geometry, which means there is no pre- or antirotation and it is free from random flow distortions. 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	6. Butterfly valve in straight run entering compressor inlet		
2. One long radius elbow (plane normal to rotor)	1.50	b	a. valve axis normal to rotor	1.5	...
3. Two elbows at 90° to each other with second elbow plane parallel to rotor	1.75	c	b. valve axis parallel to rotor	2.0	...
4. Two elbows at 90° to each other with second elbow plane normal to rotor	2.0	d	7. Two elbows in same plane (parallel to rotor)	1.15	...
5. Butterfly valve before an elbow			8. Two elbows in same plane (normal to rotor)	1.75	...
a. valve axis normal to compressor inlet	1.5	e	9. Gate valve (wide open)	1.0	...
b. valve axis parallel to compressor inlet	2.0	f	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)

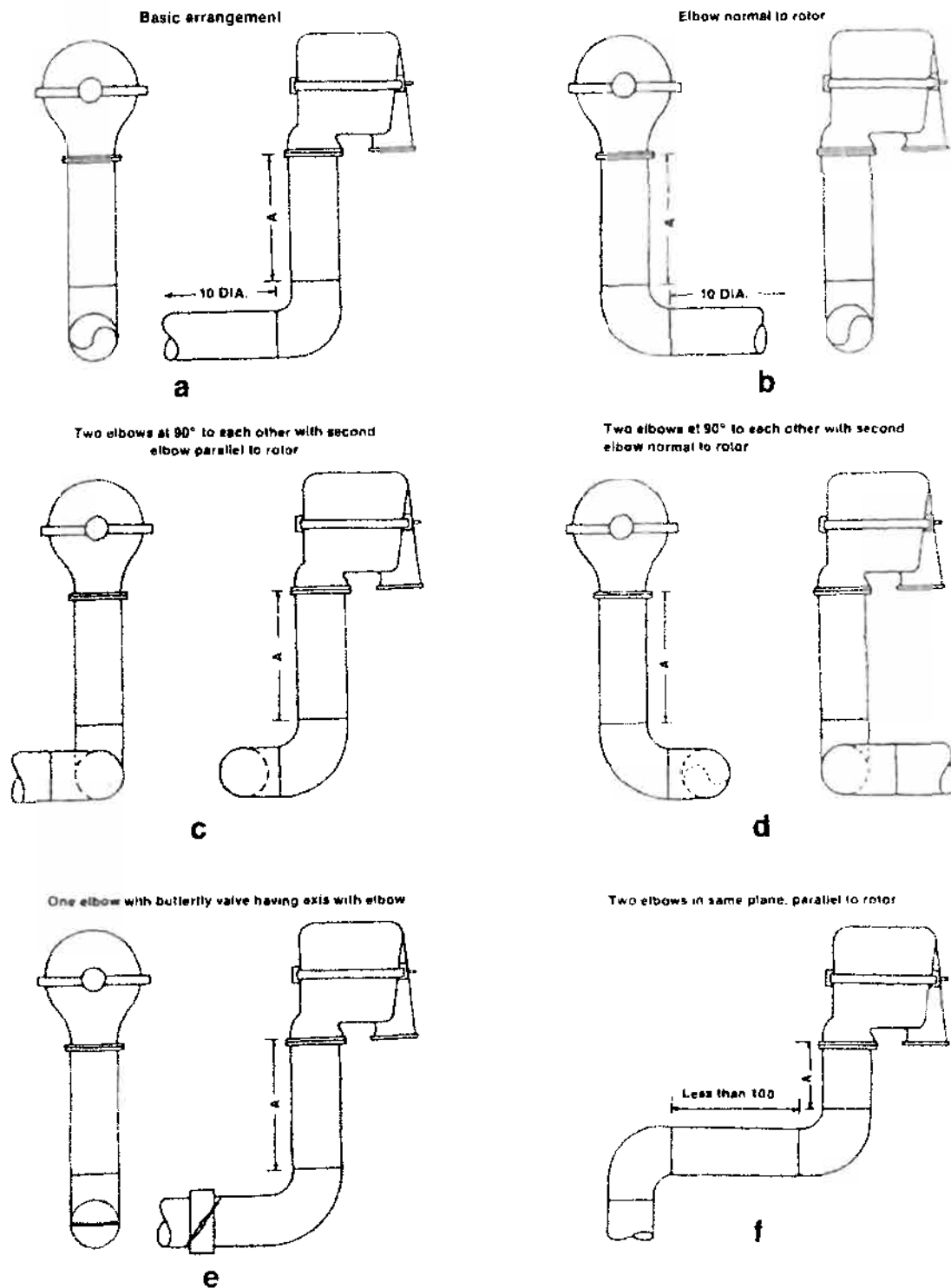


Figure 5-29. Methods of piping. To find "A," multiply the number of diameters of straight pipe from the chart in Figure 5-28 with the appropriate correction factor.
(Courtesy of Elliott Company)

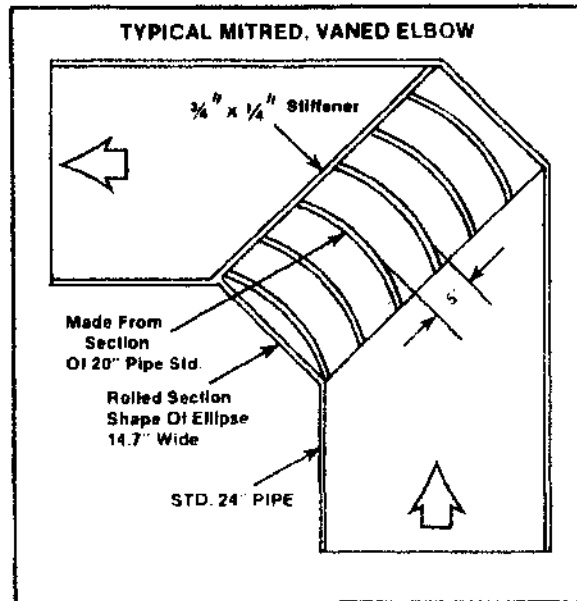


Figure 5-30. Elbow straightening vanes. (Courtesy of Elliott Company)

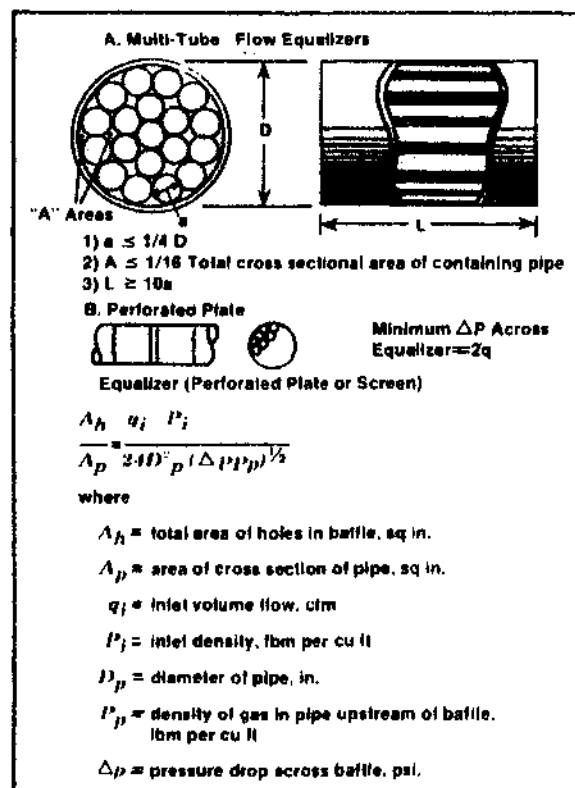


Figure 5-31. Straightening vanes. (Courtesy of Elliott Company)

(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.

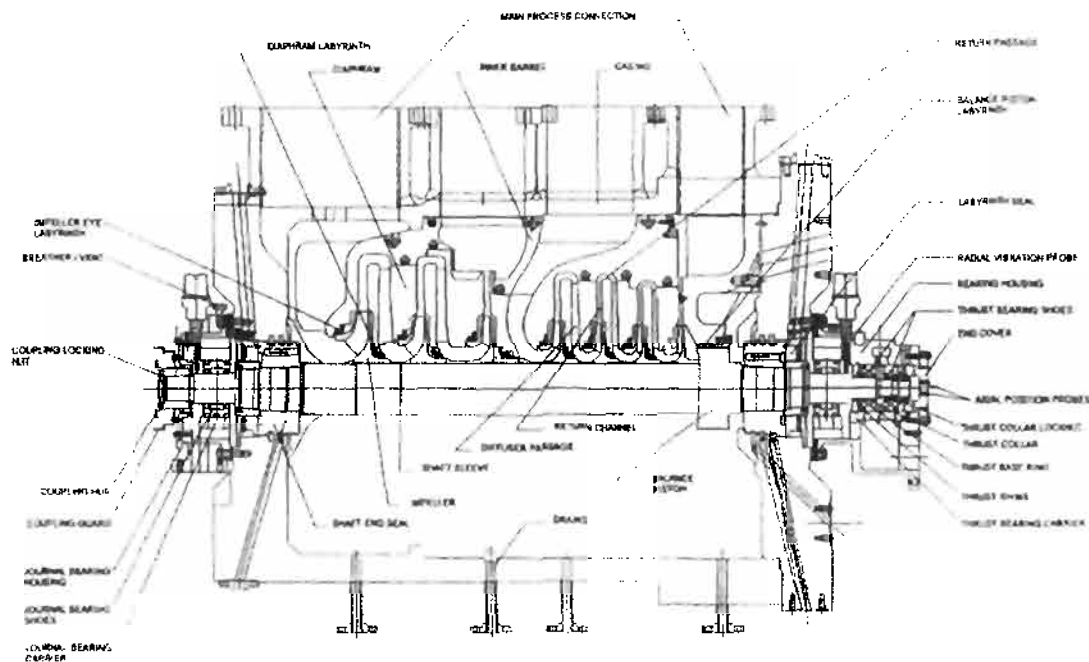


Figure 5-32. Centrifugal compressor nomenclature. (Courtesy of A-C Compressor Corporation)

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. Cas-

ings 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.

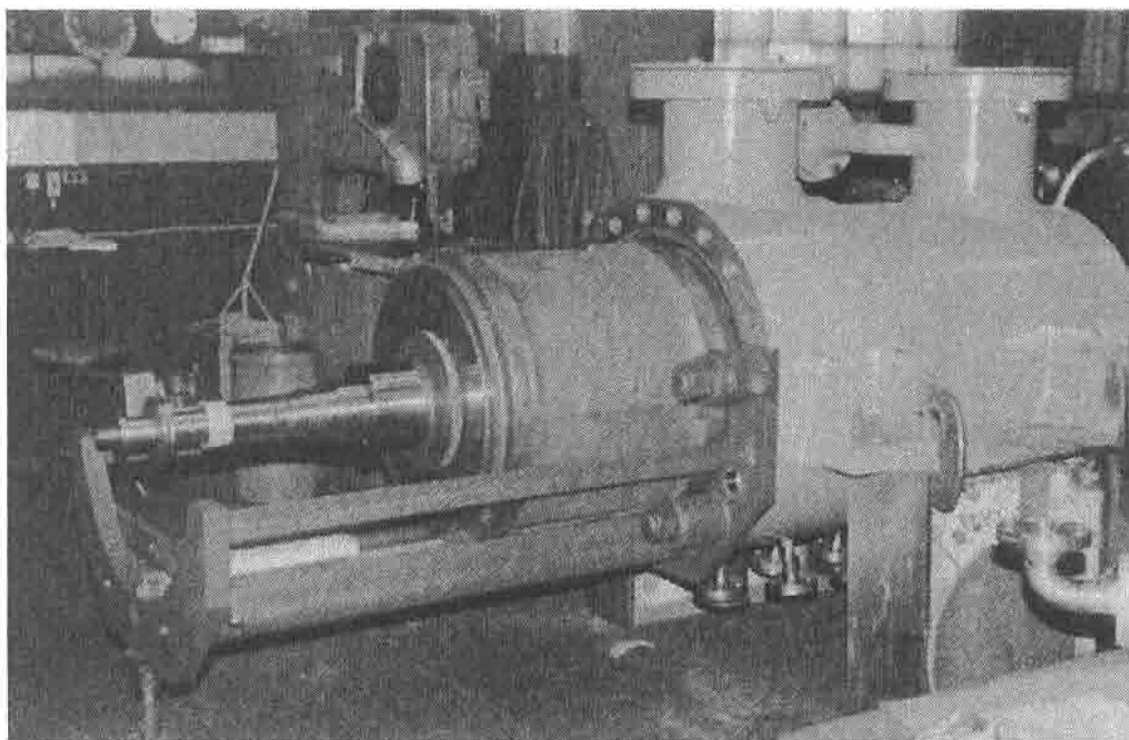


Figure 5-33. Inner barrel assembly. (*Courtesy of A-C Compressor Corporation*)

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-

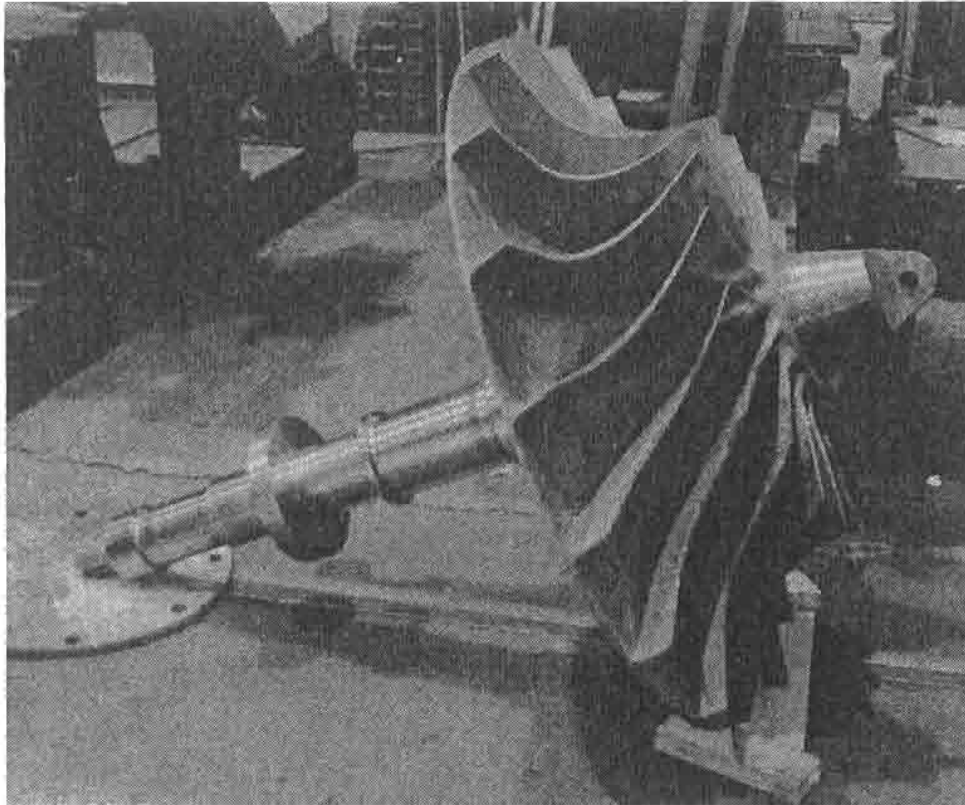


Figure 5-34. A fabricated centrifugal compressor impeller. (*Courtesy of A-C Compressor Corporation*)

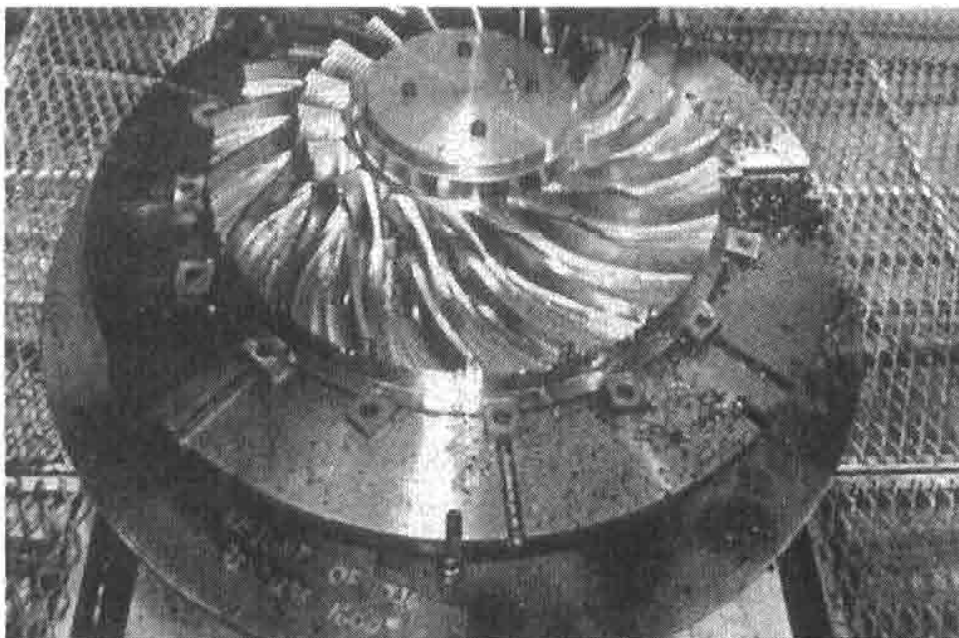


Figure 5-35. A centrifugal compressor impeller during manufacture. The blading was milled with a five-axis milling machine. The blading is integral with the back plate. (*Courtesy of Dresser-Rand*)

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 inter-stage 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 bearing 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-

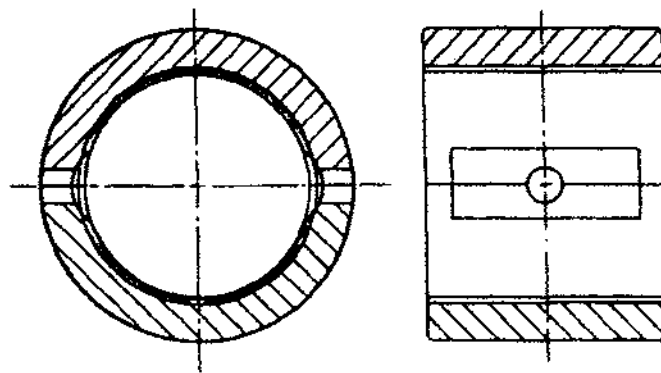


Figure 5-36. Straight cylindrical bearing. (*Courtesy of Turbocare, a Division of Demag Delavel Turbomachinery Corp., Houston facility*)

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

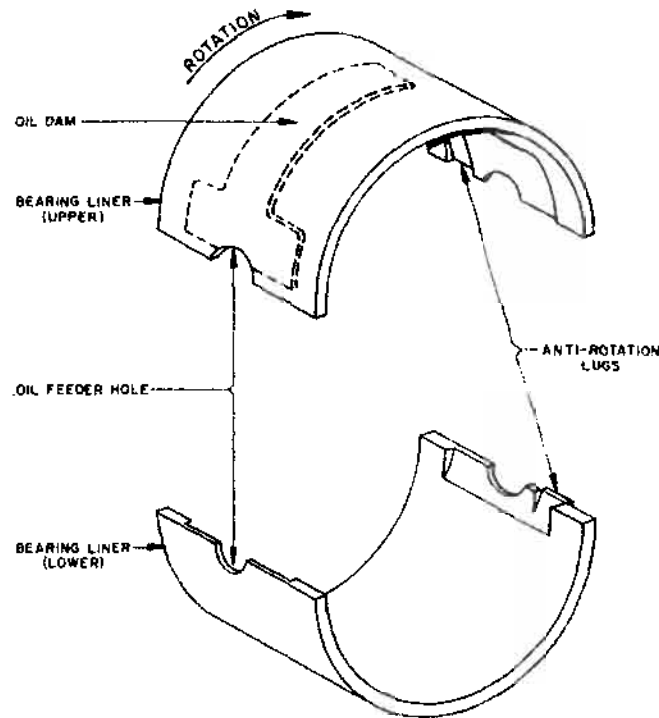


Figure 5-37. Dam type sleeve bearing. (Courtesy of Elliott Company)

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

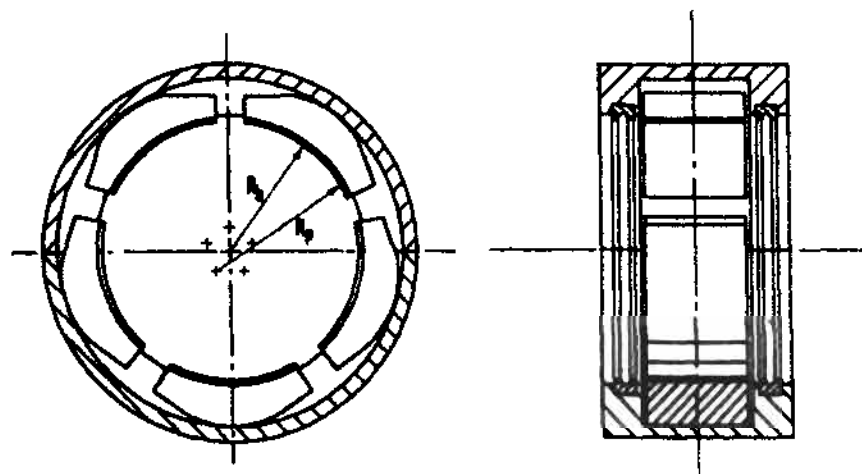


Figure 5-38. Five-pad tilting pad bearing. (Courtesy of Turbocare, a Division of Demag Delavel Turbomachinery Corp., Houston facility)

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-

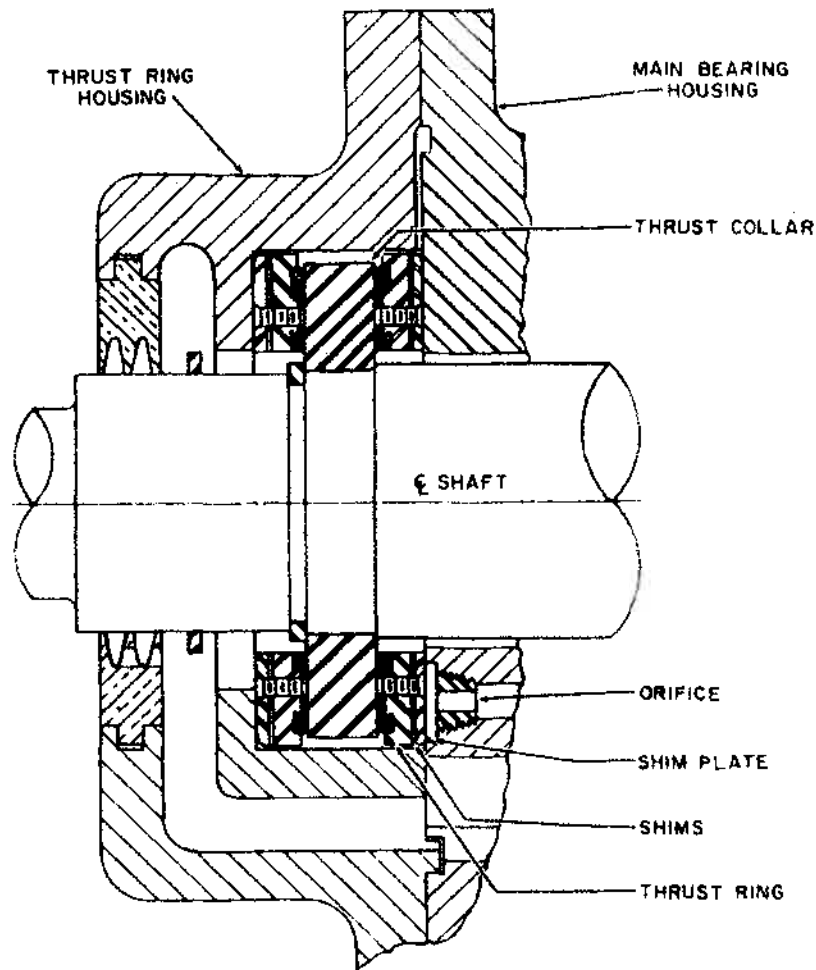


Figure 5-39. Basic thrust bearing. (Courtesy of Elliott Company)

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.

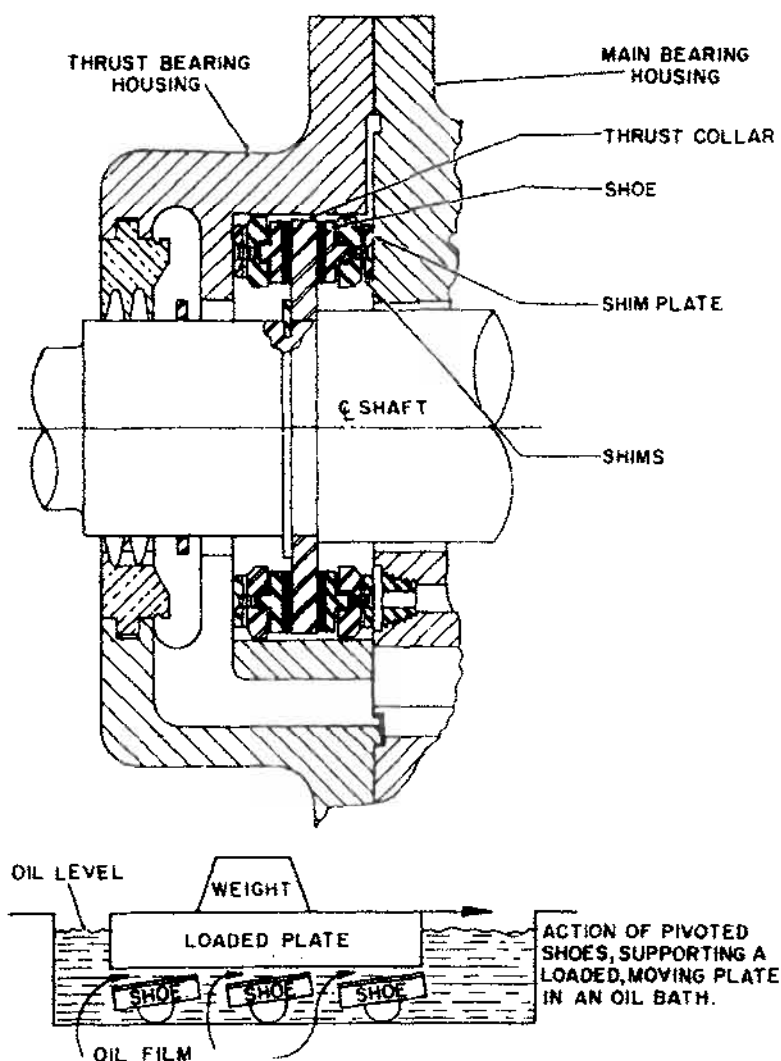


Figure 5-40. Tilting pad thrust bearing. (Courtesy of Elliott Company)

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-

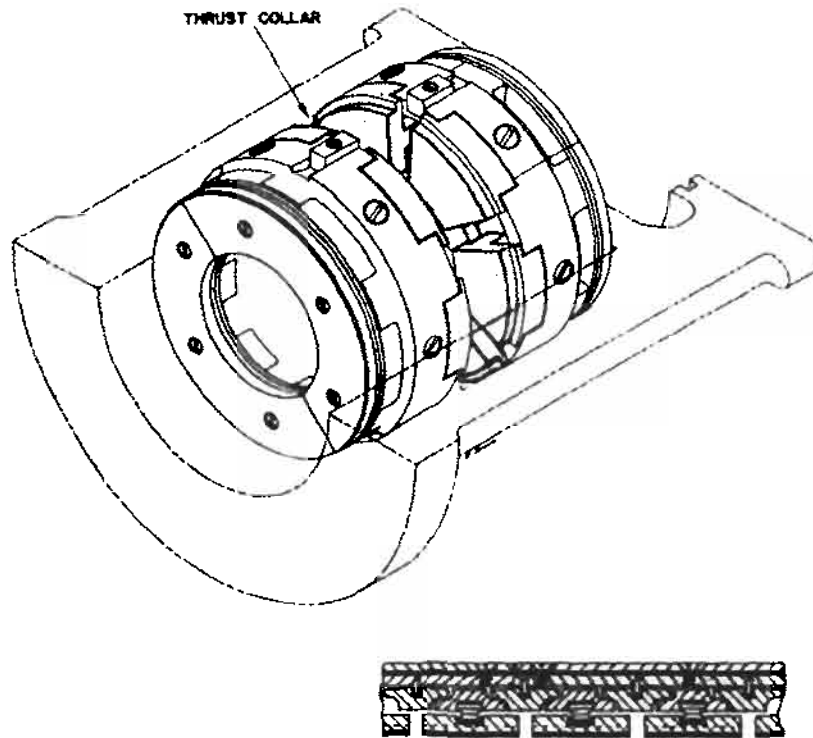


Figure 5-41. Self-equalizing tilting pad type thrust bearing. (*Courtesy of Elliott Company*)

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

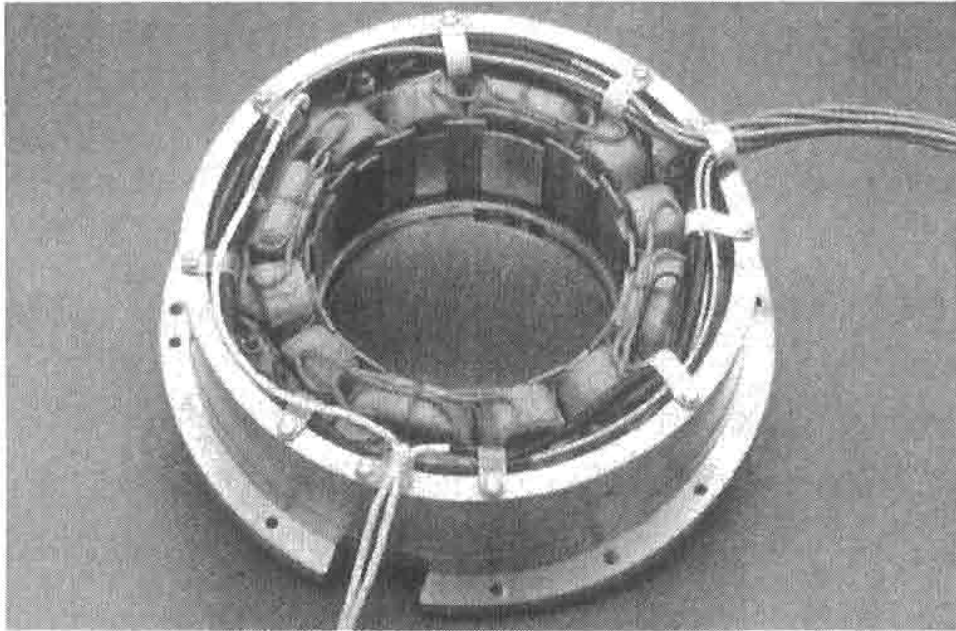


Figure 5-42. Radial magnetic bearing with a view of the circumferential electromagnets. (Courtesy of Mafi-Trench Corp.)

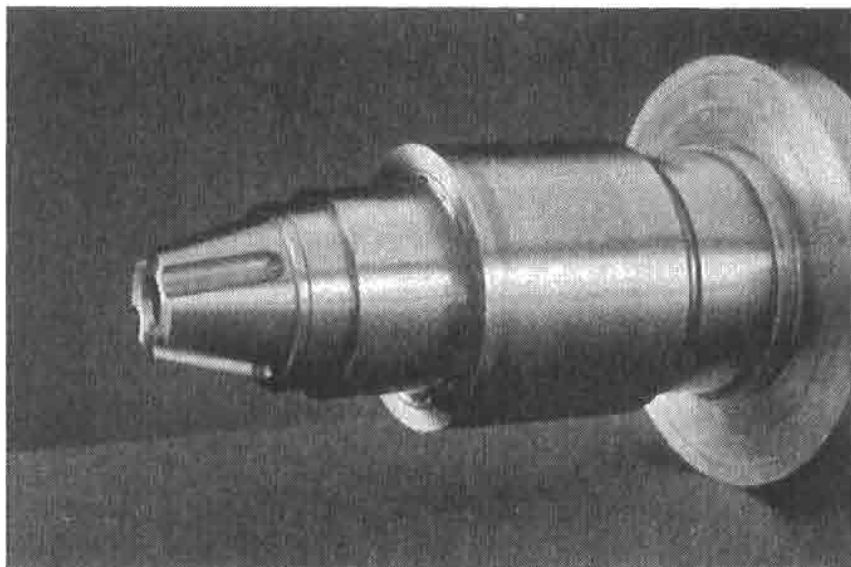


Figure 5-43. Radial magnetic bearing rotor sleeve. (Courtesy of Mafi-Trench Corp.)

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.

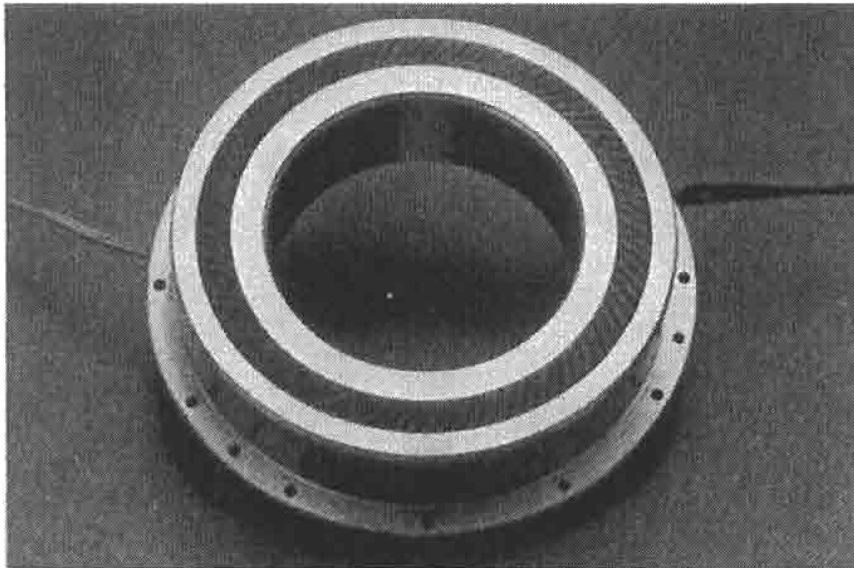


Figure 5-44. Magnetic bearing thrust electro-magnets. (*Courtesy of Mafi-Trench Corp.*)

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-

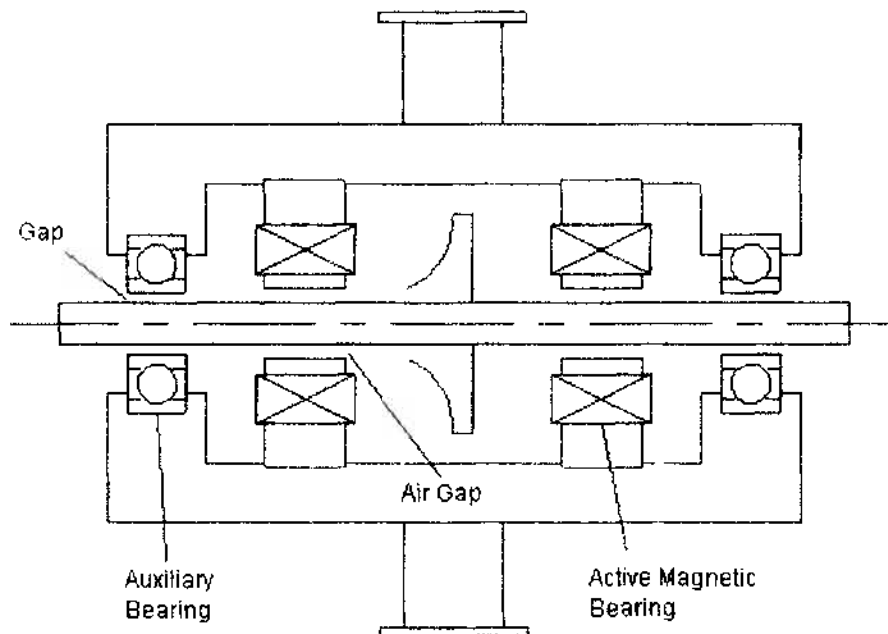


Figure 5-45. Schematic illustration of magnetic bearings and auxiliary rolling element bearings.

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

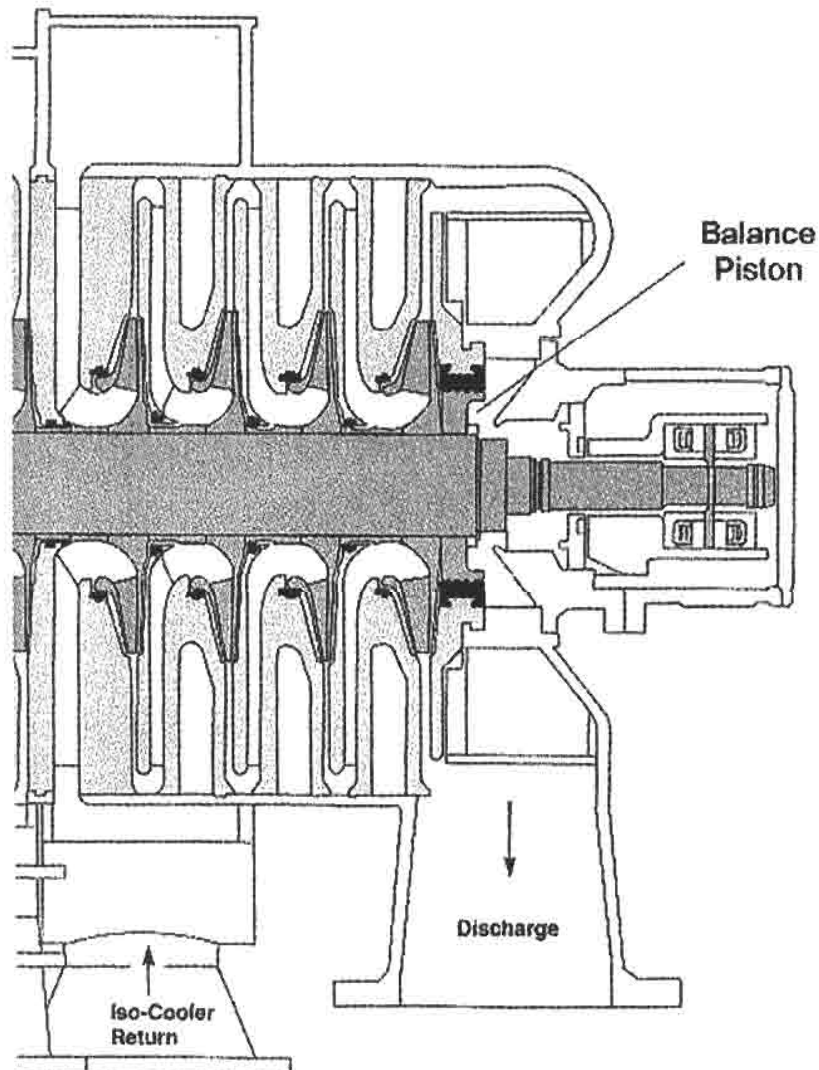


Figure 5-46. Balance piston. (Courtesy of Elliott Company)

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

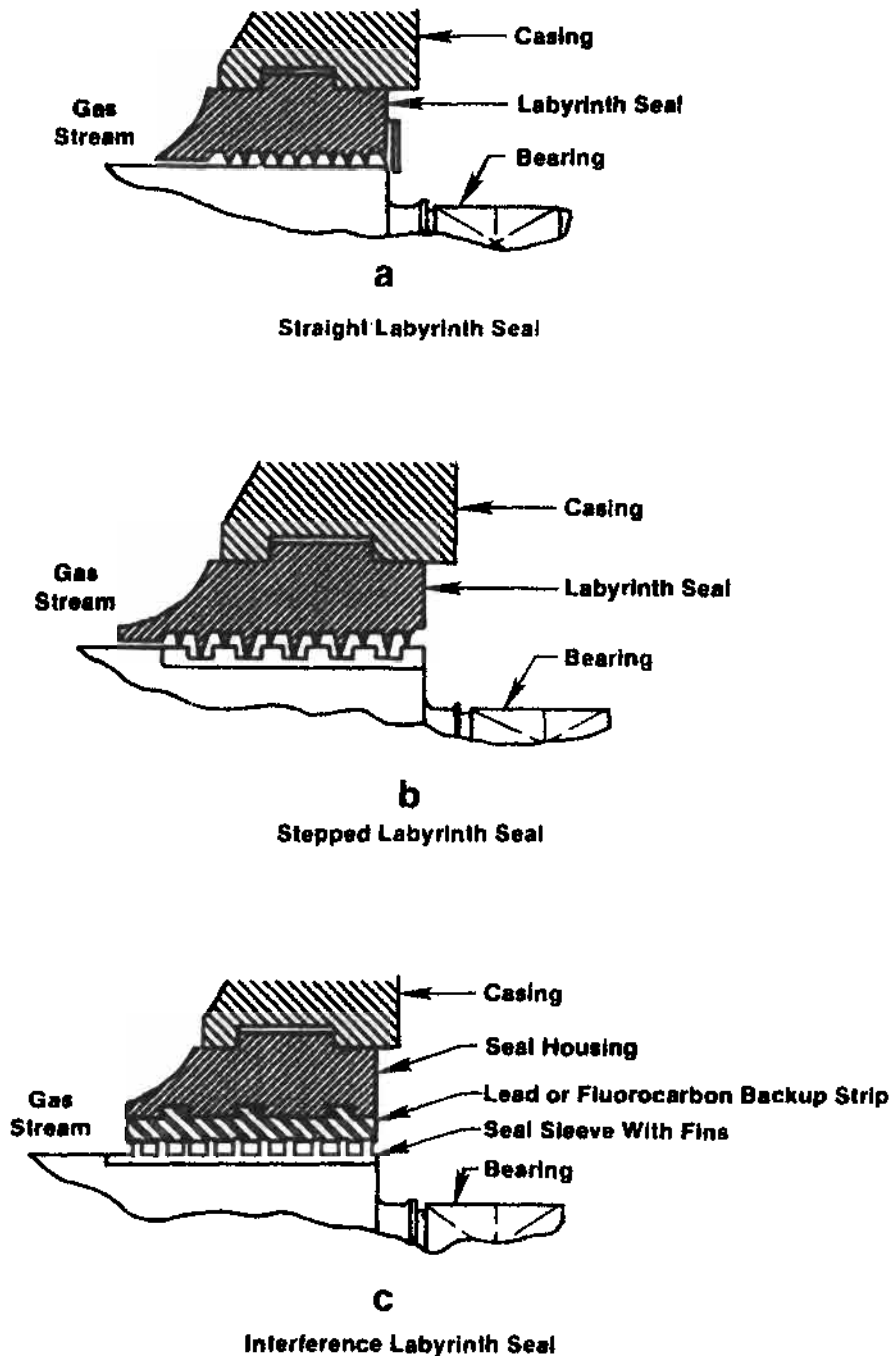


Figure 5-47. Three types of labyrinth seals. (*Courtesy of Elliott Company*)

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-

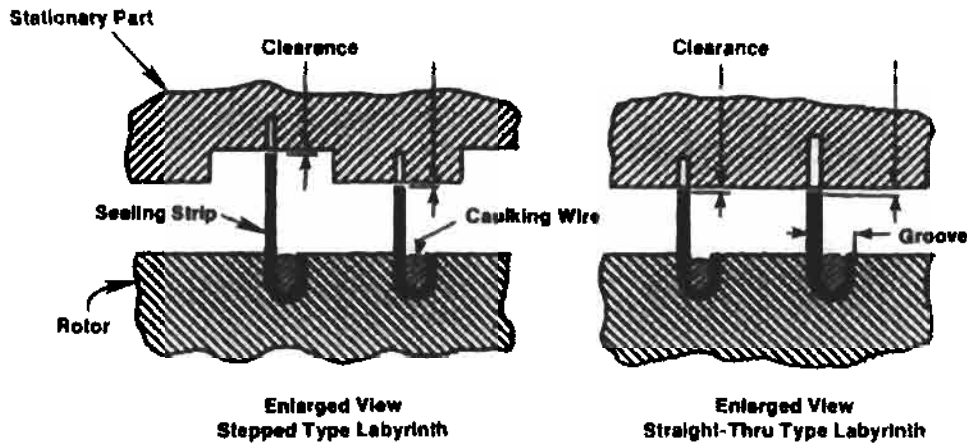


Figure 5-48. J-strip type labyrinth seal. (Courtesy of A-C Compressor Corporation)

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.

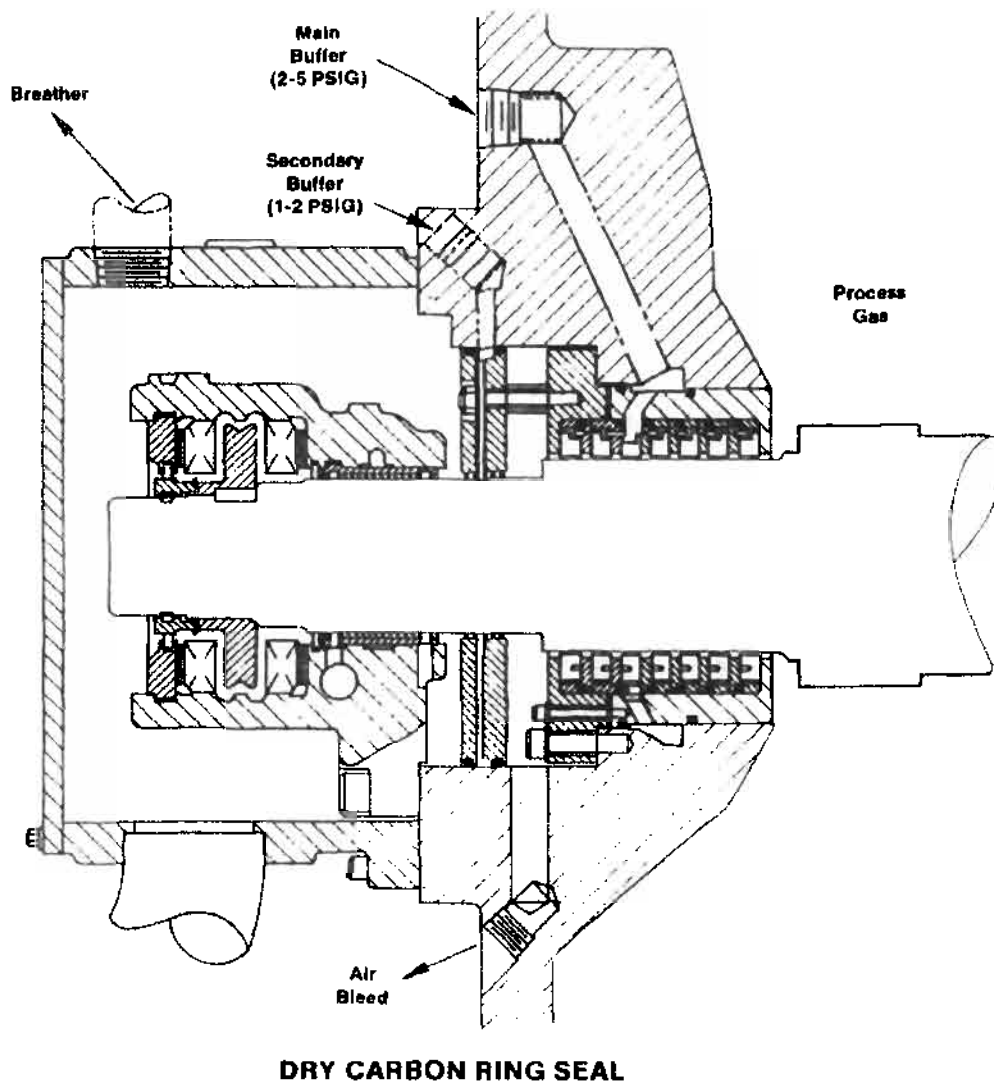
When teeth are stationary, the material chosen for the labyrinth must be a relatively soft nongalling material, because the teeth tend to touch the shaft during upsets, startup, or shutdown. The clearance chosen must be large enough to avoid excessive rubbing yet close enough to control the leakage. 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



DRY CARBON RING SEAL

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-

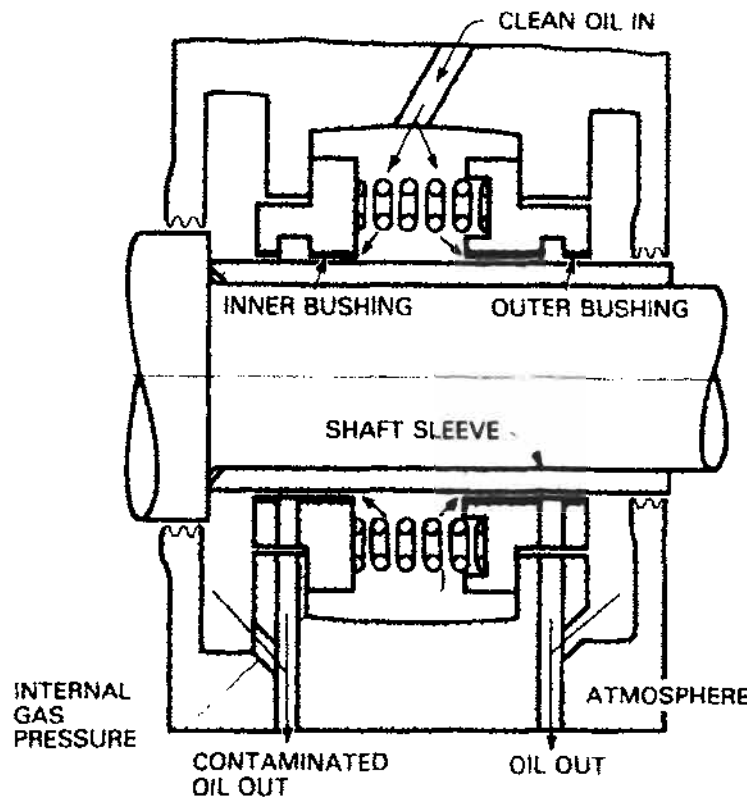


Figure 5-50. Liquid film shaft seal [12].

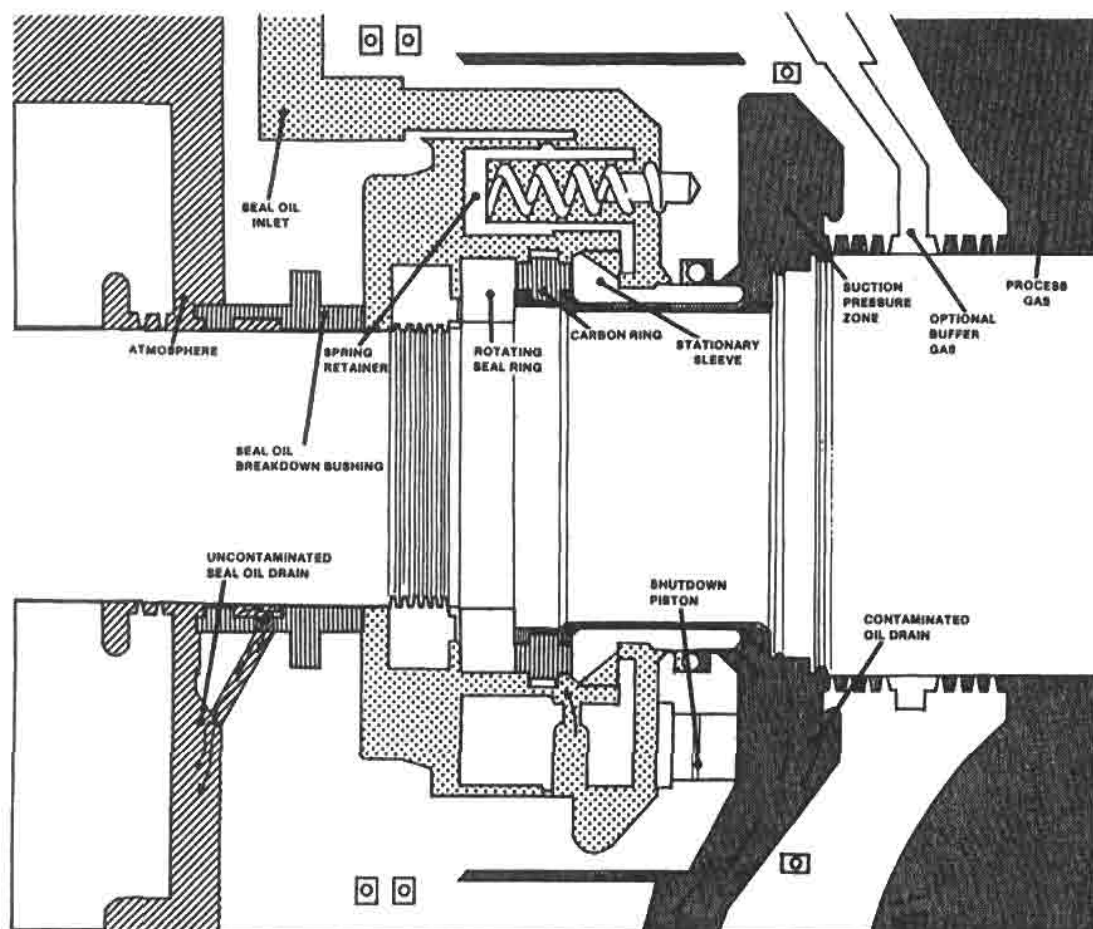


Figure 5-51. Mechanical shaft seal. (Courtesy of Elliott Company)

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.

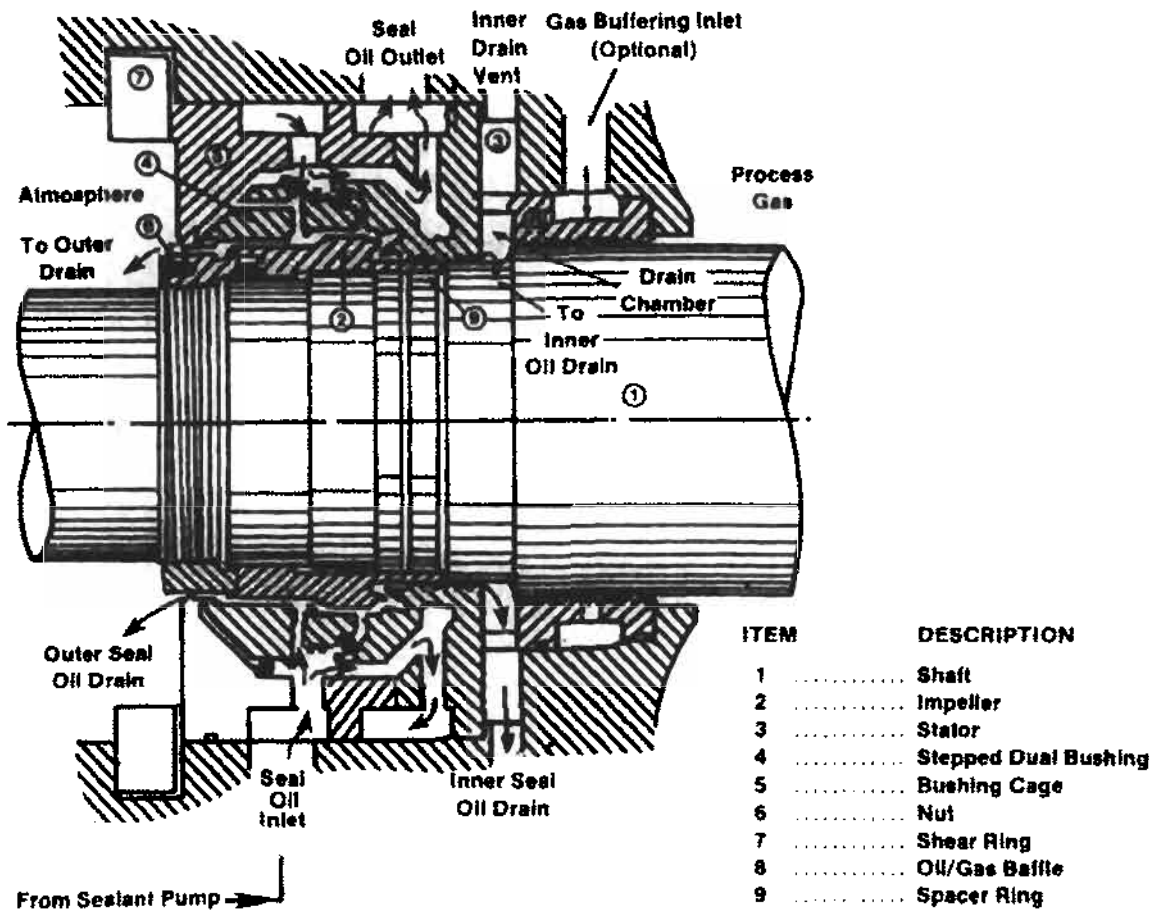


Figure 5-52. Liquid film type seal with pumping bushing. (Courtesy of A-C Compressor Corporation)

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.

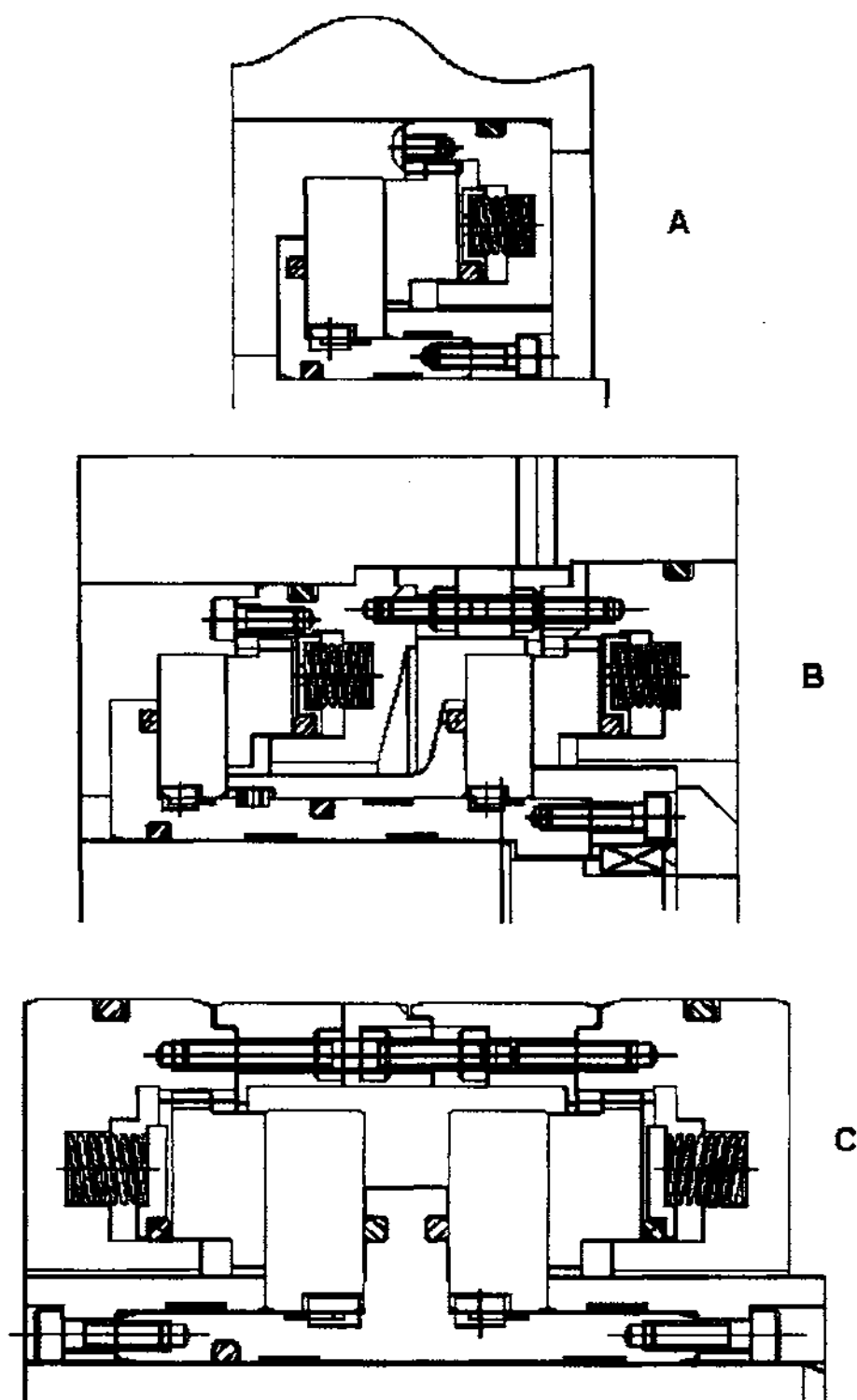


Figure 5-53. Section drawings of non-contacting dry gas seals: A. single seal arrangement, B. tandem seal arrangement, C. double opposed seal arrangement. (Courtesy of John Crane International)

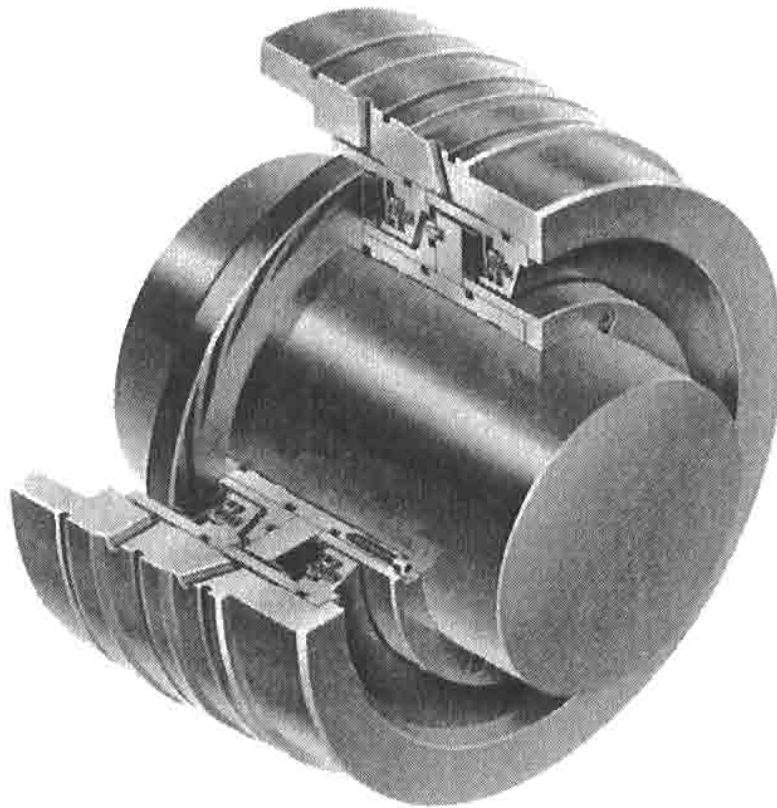


Figure 5-54. Cutaway of a tandem arrangement non-contacting dry gas seal.
(Courtesy of John Crane International)

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 mul-

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.

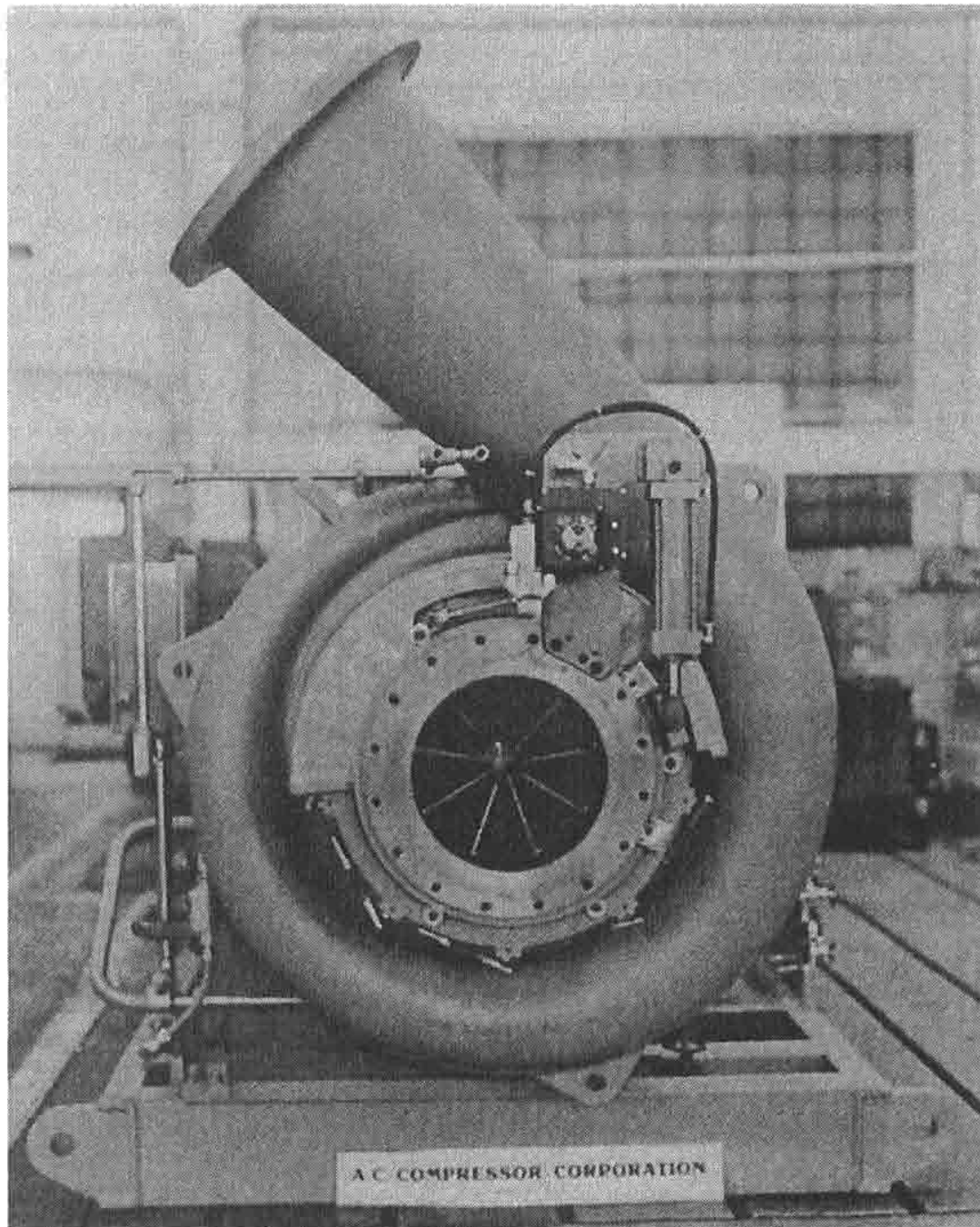


Figure 5-55. Single-stage centrifugal with movable inlet guide vanes. (Courtesy of A-C Compressor Corporation)

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 $2\frac{1}{2}$ 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.

References

1. Balje, O. E., "Study on Design Criteria and Matching of Turbomachines, Part B: Compressor and Pump Performance and Matching of Turbocomponents," *ASME Paper No. 60-WA-231*, ASME Transactions, Vol. 84, *Journal of Engineering for Power*, January 1962, p. 107.
2. Boyce, Meherwan P., *Gas Turbine Engineering Handbook*, Houston, TX: Gulf Publishing Company, 1982.
3. Durham, F. P., *Aircraft Jet Power Plants*, Englewood Cliffs, NJ: Prentice-Hall, Inc., 1951.
4. Lapina, R. P., *Escalating Centrifugal Compressor Performance, Process Compressor Technology*, Vol. I, Houston, TX: Gulf Publishing Company, 1982.
5. Scheel, Lyman F., *Gas Machinery*, Houston, TX: Gulf Publishing Company, 1972.
6. Sheppard, D. G., *Principles of Turbomachines*, The MacMillan Co., 1956, pp. 60, 67, 9th Printing 1969, pp. 238–244.

7. Wiesner, F. J., "A Review of Slip Factors for Centrifugal Impellers," *ASME 66-WA/FE-18*, American Society of Mechanical Engineers, New York, NY, 1966.
8. Hallock, D. C., "Centrifugal Compressor, the Cause of the Curve," *Air & Gas Engineering*, January 1968.
9. Boyce, Meherwan P., et al., *Practical Aspects of Centrifugal Compressor Surge and Surge Control*, Proceedings of the 12th Turbomachinery Symposium, Purdue University, West Lafayette, IN, 1983, pp. 147–173.
10. Hackel, R. A. and King, R. F., "Centrifugal Compressor Inlet Piping—A Practical Guide," *Compressed Air & Gas Institute*, Vol. 4, No. 2.
11. Brown, Royce N., "Design Considerations for Maintenance Clearance Change Affecting Machine Operation," Dow Chemical USA, Houston, TX, 1976.
12. API Standard 617, *Centrifugal Compressors for General Refinery Services*, Sixth Edition, 1995, Washington, DC: American Petroleum Institute, 1979.
13. NEMA Standards Publication No. SM 23-1979, *Steam Turbines for Mechanical Drive Service*, National Electrical Manufacturers Association, Washington, DC, 1979.
14. Elliott Company, *Compressor Refresher*, Elliott Company, Houston, TX., pp. 3–29 (other pages)
15. Dugas, J. R., Southcott, J. F. and Tran, B. X., *Adaptation of a Propylene Refrigeration Compressor With Dry Gas Seals*, Proceedings of the 20th Turbomachinery Symposium, Texas A&M University, College Station, TX, 1991, pp. 57–61.
16. Feltman, P. L., Southcott, J. F. and Sweeney, J. M., *Dry Gas Seal Retrofit*, Proceedings of the 24th Turbomachinery Symposium, Texas A&M University, College Station, TX, 1995, pp. 221–229.
17. Bornstein, K. R. et al., *Applications of Active Magnetic Bearings to High Speed Turbomachinery with Aerodynamic Rotor Disturbance*, Proceedings of MAG '95 Magnetic Bearings, Magnetic Drives and Dry Gas Seals Conference, The Center for Magnetic Bearings, A Technology Development Center of the Center for Innovative Technology and the University of Virginia, Charlottesville, VA, August 1995.

8

Accessories

Introduction

As with many consumer goods, compressors cannot be purchased without accessories. Some of the accessories are essential to the basic operation of the compressor, such as lubrication systems and couplings. Other accessories, such as the anti-surge control equipment, are optional in order to enhance operation but are not essential. The accessories may be purchased from the equipment vendor or they may be purchased from another party—a decision open to the purchaser. Generally, the accessories essential to the operation are purchased from the equipment vendor as part of the original purchase. The lubrication system and couplings are classic examples of this option. Intercoolers, while essential, are not normally purchased from the equipment vendor if the application is a process compressor.

This chapter will cover some of the more common accessory items for compressors such as the lubrication system, gears, coupling, instrumentation, vibration monitoring, and process control. The subject is broad and far-reaching. It is hoped that, for the first-time user, this discussion will be a good introduction and, for the veteran, it may offer another perspective on the subject.

Lubrication Systems

Lubrication is a fundamental requirement for all compressors with the exception of those equipped with an alternative form of bearing such as the magnetic bearing. If it is a tiny unit, the lubricant may be sealed into the rolling element bearings by the bearing manufacturer. In process service, lubrication of bearings takes on a more elaborate form. Some of the smaller units will probably use an attached oiler or an oil mist system. Because this affects only the smaller units, this section will deal primarily with those compressors using force-feed lubrication.

The ring-oiled bearing might be considered the most fundamental and basic of the lube systems (see Figure 8-1). The ring rides on top of the shaft and is dragged at part-shaft speed by friction. The lower portion of the ring resides in a reservoir of oil. In its most primitive state, the reservoir is the lubricant source and heat sink. The rotating ring moves oil from the reservoir to the upper portion of the bearing. Here the ring and shaft interface causing some of the oil to be removed. The oil enters through grooves cut into the surface of the bearing, where it is carried to the minimum clearance area by the journal pumping action. The next level of sophistication is to add circulation and cooling to the reservoir. An alternative to the separate

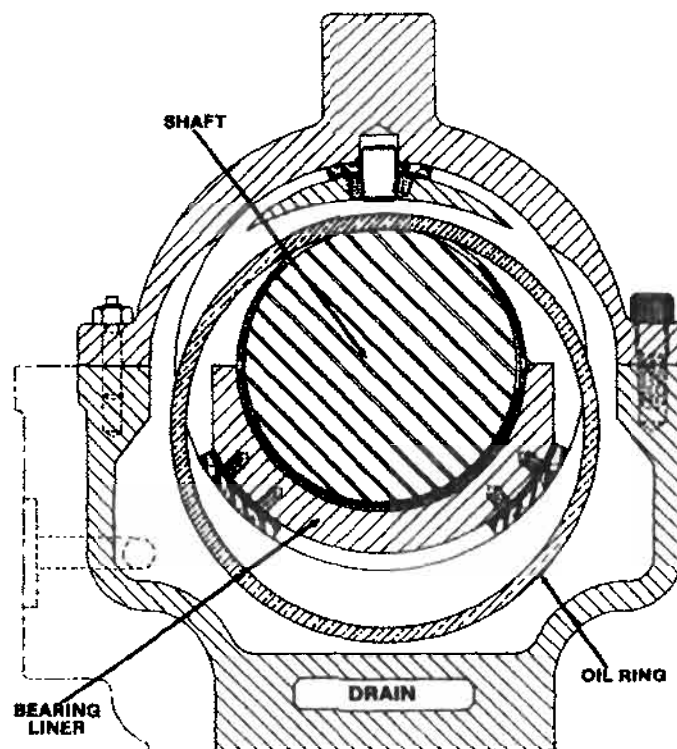


Figure 8-1. Ring-oiled sleeve bearing. (Courtesy of Elliott Company)

circulation system is to connect the reservoir to a pressure-fed external lube system in use with the balance of the compressor train bearings.

Chapter 3, which discussed various reciprocating compressors, stated that many reciprocators use a pressure-fed lubrication system for the frame bearings. This system is built into the crankcase in many applications. The basics of these systems follow the fundamental criteria which will be discussed with the fully separate system. The larger reciprocating compressors may use a separate frame lubrication system.

For the rotary, centrifugal, and axial compressors, a separate lubrication system is used and in some cases, seal oil and control oil are also supplied from this system. Chapter 4 mentioned that the oil used for flooding is taken from the lubrication system. Because the secondary duties of cooling the process stream and timing the rotors overshadow the primary job of lubing the bearings, the lube oil system may be misnamed for this compressor. As a model to guide the discussion, the basic system as covered by API Standard 614, "Lubrication, Shaft-Sealing, and Control Oil Systems for Special-Purpose Applications," [1] will be used. In the opinion of some vendors, this system is an overkill, but it can easily be tailored to fit any system by scaling down as required. The standard then can be fully or partially invoked or in some smaller system, the standard can be used as an outline and guide.

A basic pressurized lube system consists of a reservoir, pump, cooler, filter, control valves, relief valves, pressure and temperature switches, gauges, and piping. The oil is pumped from the reservoir, cooled and filtered, pressure controlled, and directed to the bearings by way of a supply header. A drain header collects the oil exiting from the bearings, and gravity flows it back into the reservoir (see Figure 8-2). If control oil is required for a power positioner on a steam turbine governor valve, additional control valves are used to establish the two levels of pressure needed because the control oil is normally at a significantly higher pressure than that needed by the bearings (see Figure 8-3). Note that an accumulator has been added to improve transient response for the turbine governor. Bearings normally operate in a 15 to 18 psig range with some variation from vendor to vendor. Control oil is generally in the 100 to 150 psig range.

If oil film or mechanical contact seals are used, another pressure level must be established. This pressure level is difficult to generalize as the seal pressure is a differential above the process gas pressure. For a mechanical contact seal, it is in the range of 35 to 50 psid to the gas and must follow the gas pressure from startup to shutdown. This generates an

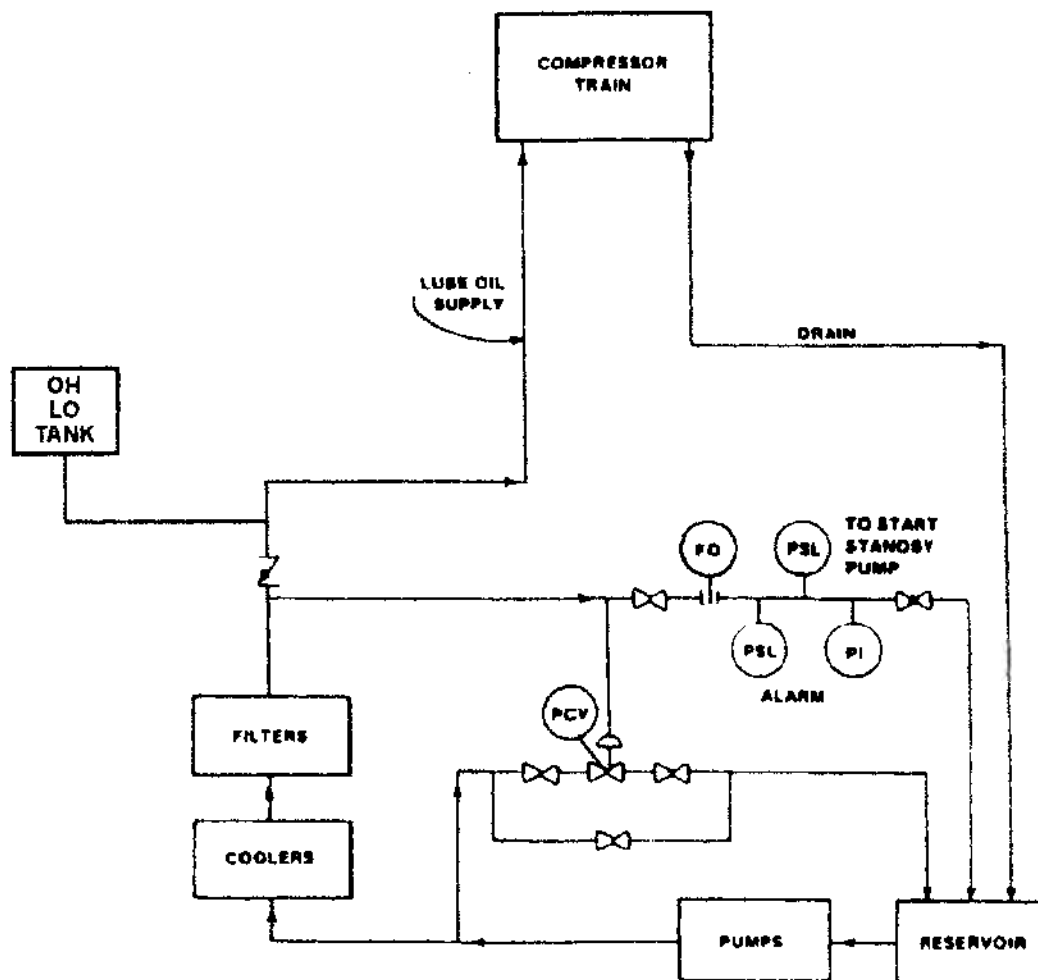


Figure 8-2. Block diagram of a pressurized lube oil system.

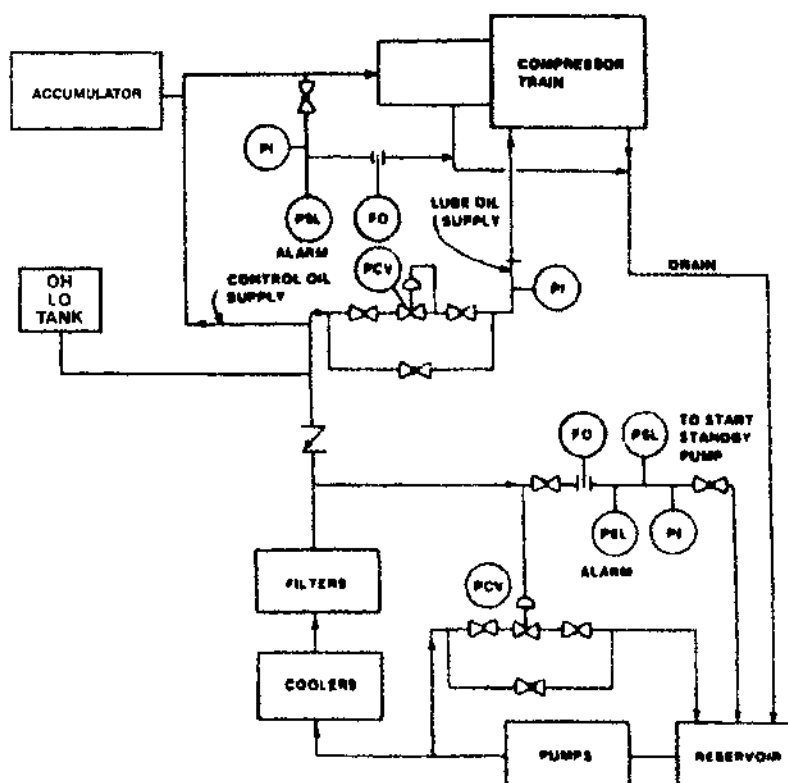


Figure 8-3. Lubrication system for a compressor that requires two levels of pressure.

additional design consideration, which will be discussed in more detail later. For the oil film and pumping bushing seals, the pressure is only a few psi above the gas pressure; however, an elevated tank is required. This tank forms the basis for the manometric differential pressure control for the seals as well as a backup supply reservoir to the seals in the case of seal oil supply failure. Figure 8-4 is a block diagram of a lube oil sys-

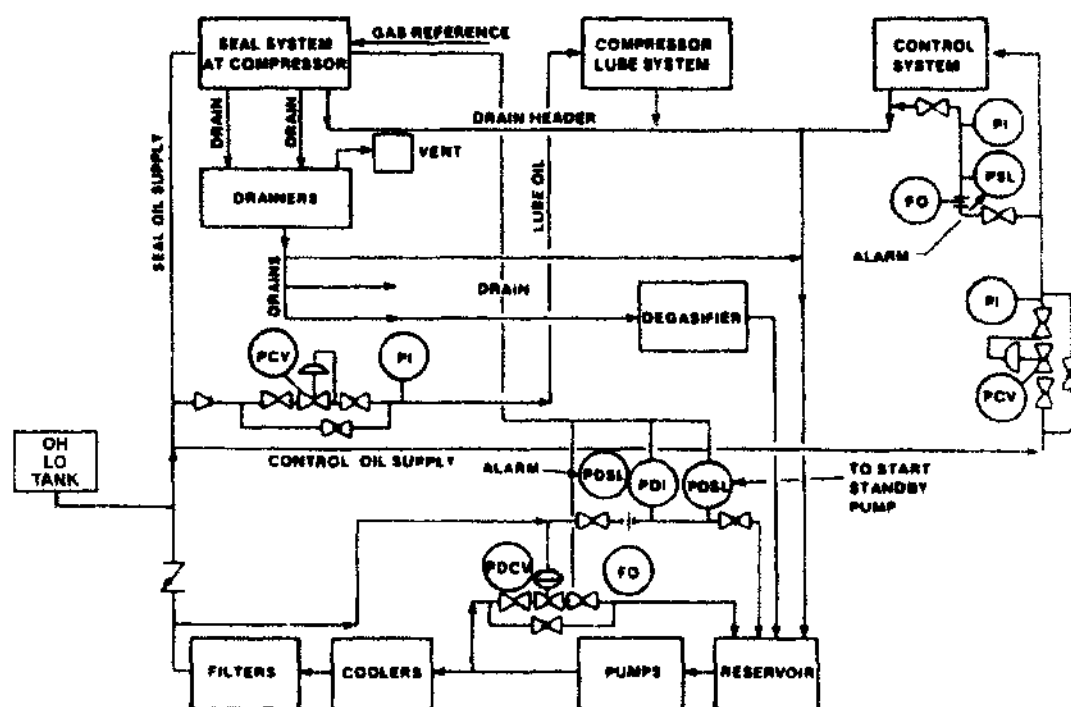


Figure 8-4. Lube oil block diagram used for a compressor lube system with a control system and a seal oil system.

tem similar to the one shown in Figure 8-3 with the seal system added. Figure 8-5 is a schematic drawing of a combined lube and seal system as would be furnished for a compressor with mechanical contact seals.

In the following paragraphs, various available options will be discussed. It is hoped that by using the options best suited for a given application even an inexperienced user might be able to specify a lube system for a compressor.

Reservoir

The reservoir is the lube oil storage tank. In some of the packaged compressors, it is built into the package base, and, in some standardized compressors, it is built into the compressor frame. In a reciprocating

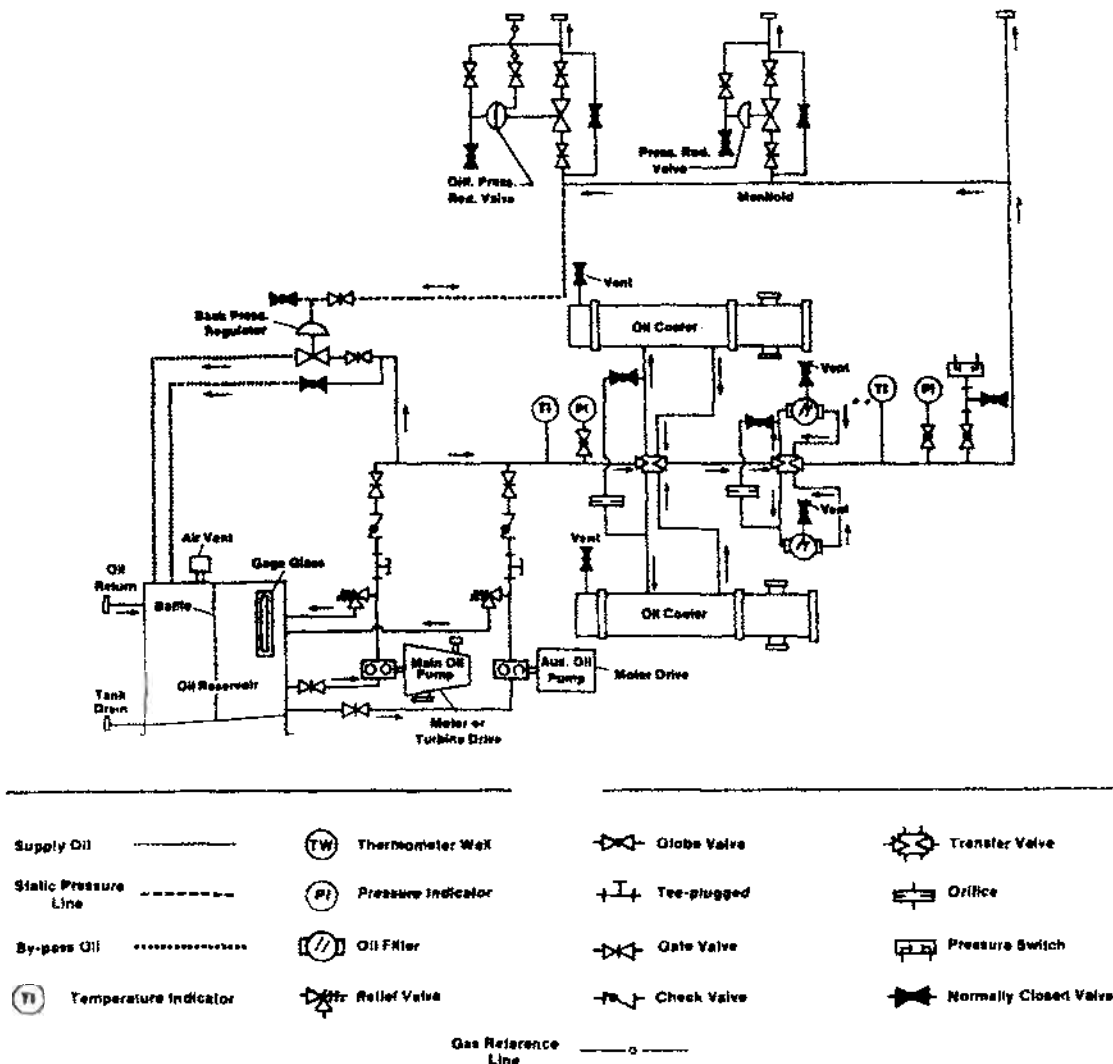


Figure 8-5. Schematic of a combined lube and seal system for a compressor with mechanical contact seals. (Modified courtesy of Elliott Company)

compressor, it is in the crankcase. When there is a choice, which there is for the larger compressors, it is recommended the reservoir be separate from the base (see Figure 8-6). As various reservoir requirements are covered, the reasoning for this will become clear.

The reservoir should be designed to prevent the entrance of dirt and water. This means sealing the top and raising top side openings above the surface. API 614 uses a one-inch dimension. The bottom should slope so the reservoir may be completely drained. The pump suctions should be located at the high side of the sloped bottom. Some form of openings should be provided to permit internal inspection and facilitate cleaning. The gravity return lines should enter the reservoir above the maximum oil level on the side away from the pump suction. The pressurized lines

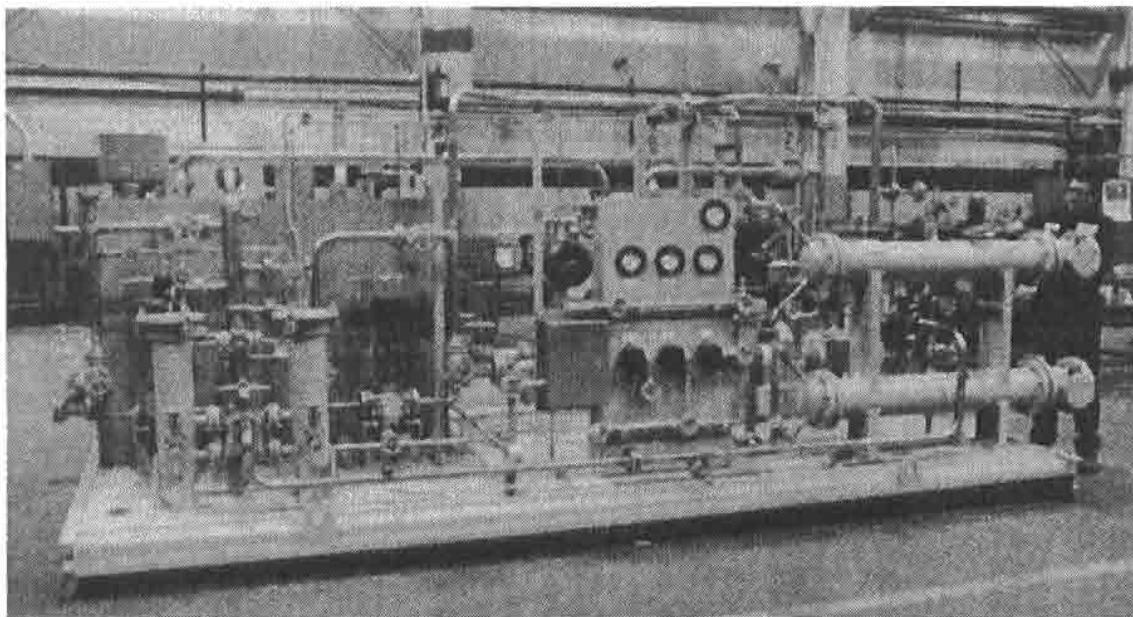


Figure 8-6. Lube oil console. (*Courtesy of Elliott Company*)

should be individually returned, including the relief valves, and piped into stilling tubes that discharge below the suction loss oil level. An automatically closing fill opening should be installed in the top of the reservoir and should include a strainer to prevent entrance of foreign material with the oil during fill operations. A breather-filter cap should be used to cover the fill opening. A flanged opening should be placed on the top and blind flanged for an optional, user-furnished vent stack. Some form of level-indicating device should be provided, mounted on the side of the reservoir. A top-mounted dipstick is also required, with the dipstick marked in liquid units of the type in use at the plant location.

The reservoir should be sized for five minutes of normal flow, with a retention time of eight minutes. The retention time should be calculated using normal flow and total volume below the minimum operating level. Provision must be made for the oil rundown from the field located piping. It should be checked on all systems, but particularly on the larger sizes. It is quite embarrassing to take a new compressor through commissioning, have a shutdown and overflow the reservoir on rundown, especially if all the company executives are there to witness the event. Additional features for the reservoirs and the defined operating levels are shown in Figure 8-7.

Heaters should be considered for the reservoir. While they are normally thought of as cold-weather features, they aid in keeping the oil dry if the compressor is shutdown long enough for the oil to cool. The heater

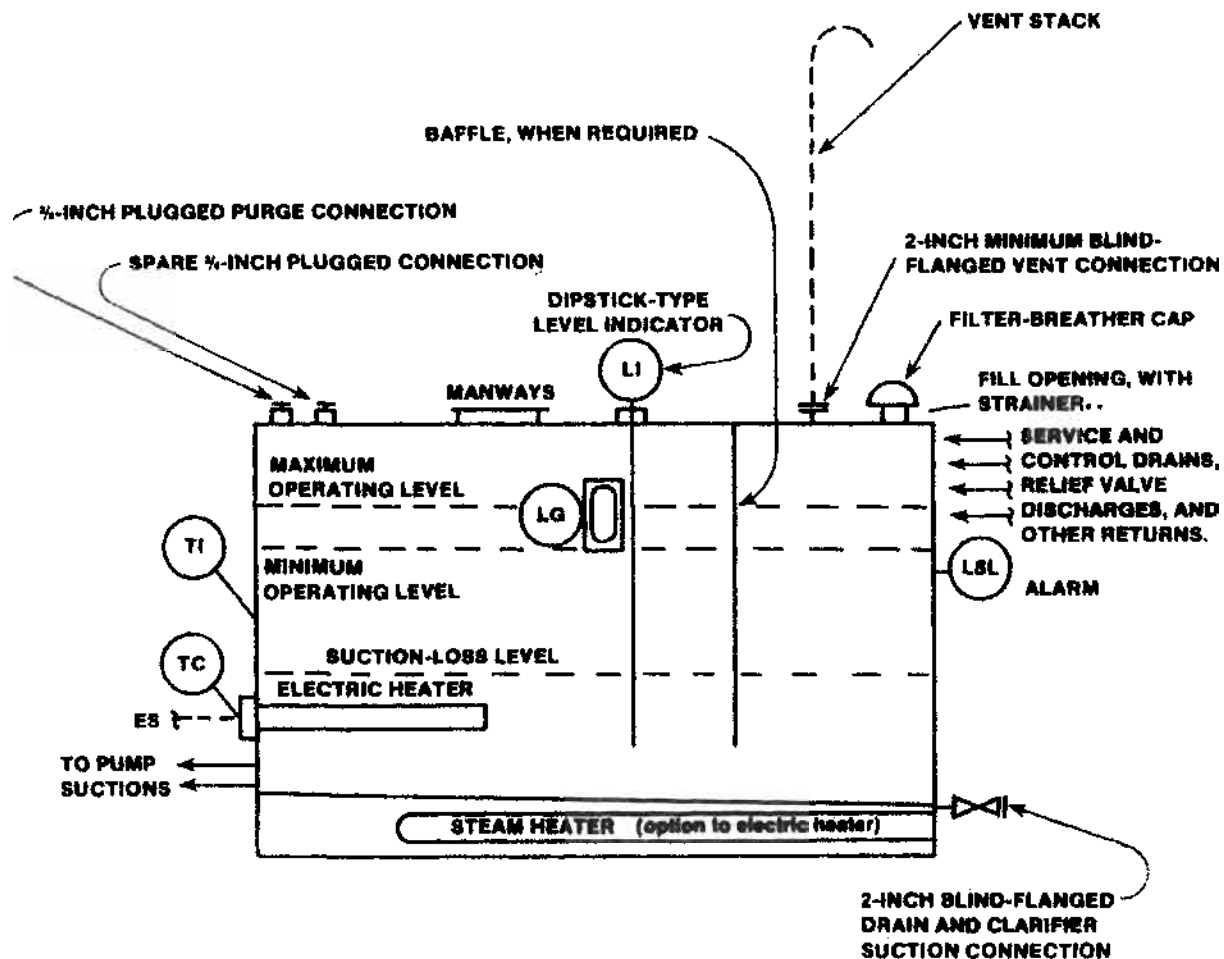


Figure 8-7. Reservoir features. (Modified from [1])

can be either steam or electric. A 15-watts-per-square-inch maximum limit on the watt density should be used in sizing the electric heater.

The internals of the reservoir should be coated unless the reservoir is constructed of 300 series stainless steel. API 614 mandates the use of 304L, 321, or 347 stainless steel processed to ASTM A 240. For the critical equipment units, the stainless steel reservoir is a good idea. A decision must be made by the user relative to the general purpose units. A good coating can keep the internals clean and free of corrosion if applied properly. When the idea of stainless steel reservoirs was introduced, it met with immediate resistance, but as the alternatives were considered, it began to gain acceptance.

Pumps and Drivers

In many ways the pump is the heart of the lubrication system as it is the only active element. It must furnish sufficient capacity at a high

enough pressure to satisfy the entire train connected to its lube system. The pump driver must be sized to be able to start with cold oil, generally 50°F or colder. It must have enough power to operate under all conditions, including the highest seal pressures expected, which, on a refrigeration compressor, is the shutdown stagnation condition. This pressure is equal to the saturation pressure of the refrigerant at ambient temperatures and can be much higher than normal operating pressure. Also the pump should be checked for minimum viscosity operation, particularly at the higher pressures. Pump failures have been traced to violation of the minimum allowable viscosity limit on higher pressure operation resulting in rotor contact.

The pumps can either be rotary positive displacement or centrifugal. The rotary positive displacement pump of the helical-lobe type is recommended. Centrifugal pumps should only be used for large systems, over 500 gallons per minute, if a suitable rotary screw pump is not available. The centrifugal should have as steep a pressure flow characteristic curve as can be found. The normal low head rise centrifugal pump will lose capacity as the filters get dirty and cause the system to starve for oil and go into an unscheduled shutdown. Oversizing the centrifugal and running it normally in the overload region is not an adequate solution to the problem.

On some of the smaller, standardized compressors, the main lube oil pump may be shaft driven. Others use rotary gear pumps. These have been used for quite some time, and the desire for change or modification usually falls on deaf ears. A good quality standby pump should be connected in parallel to the rotary gear pump in case the first one fails. As a general rule, on even the smaller systems, there should be a main and a full-sized standby pump. On the larger systems, as the equipment is put into unsparred service, the two pumps are a must. In some plants a third, smaller pump is used, which is referred to as a coastdown or emergency pump. This pump is used as a contingency in the event of a full power failure when both the main and standby pumps fail due to loss of power or for any other reason. Of course, the main compressor should go into the shutdown mode using the emergency pump to supply oil to the bearings during the coastdown cycle. The whole premise for this course of action is that a source of energy, not affected by a power outage, is available. A better alternative using an overhead tank for compressor coastdown will be discussed later.

The pumps may be driven by any of several drivers. For many years the favorite arrangement has been a steam turbine for the main pump and

an electric motor for the standby. It is a nice combination in that if the steam supply is disrupted by power failure, there is some residual pressure as the system decays, in many cases, allowing the pump to deliver some capacity. This may be the only argument for the shaft-driven pump which delivers oil while the compressor shaft is turning. Unfortunately, even though the pump will turn, it doesn't necessarily deliver oil because of long suction lines, air leaks, or the capacity is just not available at the lower speeds. In some cases there is no steam available and some smaller compressors do not have the steam turbine option, primarily because of cost. In these cases, the two pumps should be motor driven. If possible, the motors should receive power feed from two independent power sources. If an emergency or coastdown pump option is selected, it should be run from a totally independent power system. Some large plants use an uninterruptible power supply (UPS) to operate devices during a power outage. Sufficient additional capacity might be added to the batteries to run a small motor. The motor can be direct current and will not tax the inverter on the UPS. If a separate battery system is installed only for the coastdown pump, the odds are it will not be operational after a period of time due to lack of battery maintenance. That is why it should be combined with other plant functions that require the backup system to be operational. Other power sources, such as air motors, have been used but without great success.

Pump casings should be made of steel if possible, which, on the smaller compressor systems, will not be practical. Most petrochemical plants live in the fear of fire, and the use of cast iron casings in a hydrocarbon plant is not a good idea.

The pumps should be piped with a flooded suction to avoid having priming problems. This is difficult to do on a shaft-driven pump. This procedure precludes mounting the pumping equipment on the top of the reservoir. If top-mounted equipment is desirable to keep the system compact, then vertical pumps, operating below the oil level in the reservoir, should be considered. When this arrangement is used, the need for steel in the pump casing is eliminated. The oil reservoir must be strengthened to carry the extra weight without excessive deflection. While this is one way to maintain a compact system, it is only practical on the smaller systems where component maintenance is not as difficult.

A strainer should be used in the pump suction line temporarily for a centrifugal pump and permanently for the rotary positive displacement pump. For the permanent installations, a Y-type strainer with an austenitic stainless strainer basket should be used. The cross-sectional

area of any strainer should be at least 150% of the normal flow area. In all strainer installations, a compound pressure gauge should be installed between the strainer and pump suction.

Booster Pumps

In some applications (usually high pressure compressors using oil film seals) alternative pump schemes should be considered. It may be that the desired seal pressure is not achievable by one set of pumps or the quantity required by the seal is small relative to the main pump capacity. There are times when booster pumps are needed; however, if the reason is energy, it would be worth reviewing the economics very carefully, because reliability tends to suffer with the booster. The booster pumps are paired into a main and standby and are configured to take suction from the lower pressure system. Sufficient interlocks have to be supplied to the drivers so that if the main pumps shutdown, the boosters come down. Other problems may arise when the controls are set or trimmed because the system is usually quite sensitive to the high system gain caused by the high pressure at the valves.

Pump Sizing

Pumps should be sized for 1.2 times the system's normal flow requirement, with a minimum of 10 gpm above the normal flow. If the booster arrangement is used, an additional capacity is needed equal to the flow of both booster pumps running simultaneously. A centrifugal pump should be within the range of 50 to 110% of the best efficiency point when running at normal capacity. The pump, as was previously mentioned, should have a steep curve. API 614 mandates a minimum of 5% pressure rise to shutoff. This minimum seems to be on the low side. A 15% rise to shutoff would come closer to maintaining minimum oil flow with dirty filters. See Chapter 7 for various driver's sizing guidelines.

Pump Couplings

For pumps above 25 hp, flexible disk, spacer-type couplings should be used. The flexible elements must be selected for compatibility with the plant atmosphere. For the smaller systems, a non-spacer coupling may be adequate, but the coupling should be of good quality. This is not the place to save money. Coupling guards should be furnished as a part of the lube system.

Relief Valves

Whenever possible, external relief valves should be furnished. On the larger systems, this is mandatory. Internal relief valves will lead to premature pump failure if allowed to operate open for more than a few minutes, since the hot oil is returned directly to the suction. An upset period, the time when a pump may open the relief, is probably not the time when an operator would detect the open valve and shut the pump down to reset the valve.

External relief valves should be of the full-flow, non-chattering (modulating) style. In oil systems with relatively low pressure levels, the hydraulic-type relief valve becomes attractive because the plug lift is smooth, and instability during lifting is eliminated. The valves should be located as close to the pump as possible to provide fast reaction time.

Pressure Control Valves

For positive displacement pumps, a bypass-type control valve should be furnished to set the primary lube system pressure. The valve should be able to maintain system pressure during pump startup and pump transfers, which includes relieving the capacity of one pump, while both are running. The valve should provide stable, constant pressure during these transients. Flow turndown of 8 to 1 is not unusual. Multiple valves in parallel should be used if a single valve is not suitable. The valve should be sized to operate between 10 and 90% of the flow coefficient (C_v). Additional pressure control valves should be furnished as required to provide any of the intermediate pressure levels.

On the smaller, non-API type systems, the relief valve is also the pressure control valve. This definitely must be an external valve. While a compromise for the smaller system, the requirements are also not so severe.

Startup Control

A pressure switch located so as to sense falling pressure at the earliest moment should be used to activate the standby pump. The switch should be so connected to the system as to permit testing the startup circuit without shutting down the compressor. Figure 8-8 shows four piping arrangements. The figure at b would be the recommended method.

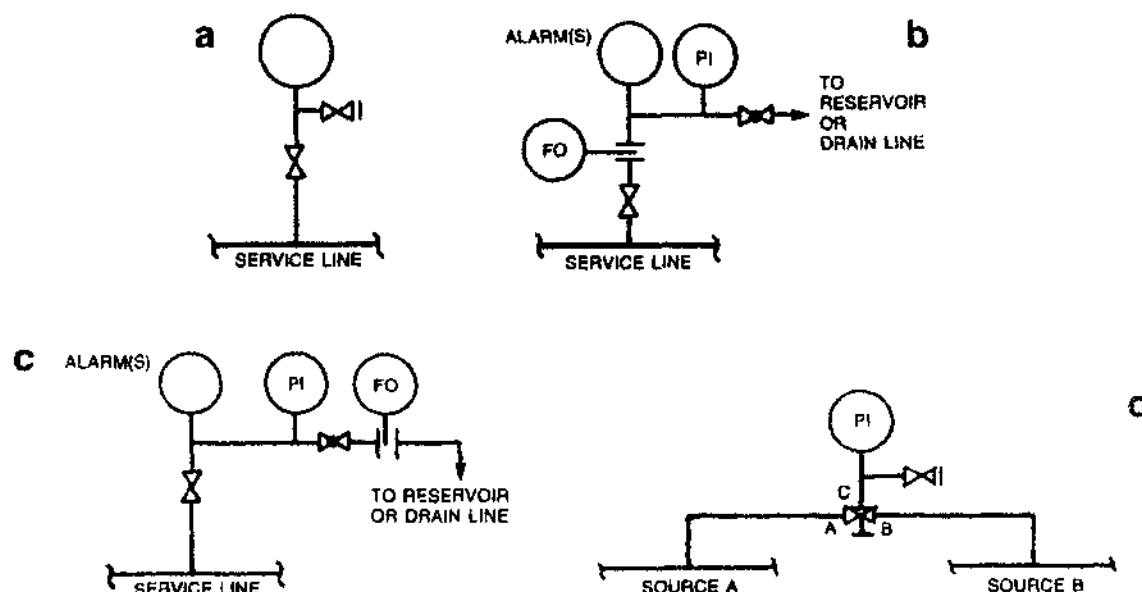


Figure 8-8. Pressure gauge and pressure switch piping arrangements [1].

It should be emphasized at this point that the speed of response is critical. The pressure transient pressure should not fall to less than 50% of the difference in pressure between the standby pump start pressure and the low oil pressure trip pressure. This is normally achievable with good design practice and the use of a switch and direct wiring. There is some tendency to use a transmitter and control through a remote computer. The latter arrangement is difficult to check on a shop test and normally is too slow to meet the requirement. An accumulator can be added and must be used if the requirement cannot be met. This additional hardware contributes to higher initial cost and possible reliability problems in the future. The direct switch method is therefore highly recommended.

Provision must be made in the wiring to maintain power to the standby pump driver. This will prevent power interruption due to the pressure switch contacts opening when lubricant pressure is restored. Cycling of the standby pump will occur if the circuit is not maintained. A reset will have to be furnished to permit shutting down the standby pump when the main pump is back on stream.

Check Valves

Check valves must be used on each pump discharge to prevent back-flow through the idle pump. On those systems where a flooded suction is not furnished, an orificed bypass around the appropriate check valve should be furnished to keep the suction line of the non-flooded suction

pump charged. Sometimes the check valve is drilled to provide this feature at a minimum cost, but this is not the best practice.

Coolers

API 614 mandates twin coolers. For a compressor in critical service or, even in some cases, an unspared compressor, it is important to be able to switch coolers without requiring a main unit shutdown in the case of a problem. For shell and tube coolers using water, a good design outlet temperature is 120°F. The normal expected input from the compressor is approximately 140°F. A removable bundle design is generally recommended above 5 square feet, though this may vary from plant to plant. The classic API tube size has been $\frac{3}{8}$ inch outside diameter with a minimum of 18 BWG tube wall thickness. Again, plant standards may require a larger and heavier tube, which is no problem as the cooler size increases, but causes problems with the smaller sizes. In fact, the API minimum requirement may cause procurement problems and is not generally used on the smaller standard compressor lube systems. Most coolers are of a size which brings them under the ASME pressure vessel code rules. If so, the cooler should bear the ASME pressure vessel code stamp. For all the noise made about it, the extra cost to the vendor is not that severe. Cooler shells, channels, and covers should be steel. Tubes should be inhibited admiralty and the tube sheets should be naval brass, unless the plant standards call for different material, in which case, the vendor must be informed by way of the specification for the equipment. On some packaged standard compressors, the materials of construction are not an option. The materials used should be reviewed to ensure the compatibility with plant operations. If an incompatibility is noted and cannot be changed, the cooler may require changing once the unit is received. The vendor should be fully aware of the plan so as not to void his warranty on the balance of the equipment. If any of this becomes a problem, another vendor should be considered or, if the balance of the equipment is so attractive, it should be owner warranted.

If cooling water is not available, air-cooled exchangers can be furnished, even in relatively small sizes. The cost is higher than a shell and tube-water-cooled exchanger. Also the outlet oil temperature will be higher than that from the water-cooled exchanger. This is no particular problem if the compressor designer is aware of the higher temperature. More oil will have to be circulated to make up for the loss of the temperature differential.

Twin coolers should include a pre-piped vent and orificed fill line to permit filling the idle cooler prior to being put into service. A drain

valve, located at the low point of the cooler, should be furnished for both the oil and the water side (if water is used).

Filters

While there may be reasons to only use one cooler, filters should be dual on all but the smallest standard compressors. The filter should remove particles to a nominal size of 10 microns. A filtration to a smaller particle size should be considered; however, getting the system clean to that level will take more patience. The transfer between filters will occur frequently during the initial operation period and the dual filters will be found quite handy.

The filter elements should be replaceable and should be corrosion resistant. The filter should not contain any type of internal relief valve that would permit the bypassing of the dirty oil.

The filters should be located downstream of the coolers and should be equipped with a vent and orificed fill line to permit air removal prior to being put into service to prevent shocking the system. External lifts for the filter covers should be furnished if the covers are too heavy for an operator to safely handle. A suggested weight is 35 pounds, deferring however to operator safety. An adequate valved drain should be furnished for each filter body to permit easy removal of dirty oil and sludge.

API mandates that the pressure drop across a clean filter element be no more than 15% of the allowable pressure drop when dirty. An upper limit of 5 psi drop is set for clean filters. This is a reasonable criterion. If a little arithmetic is performed, the head rise for a centrifugal pump may be calculated. The specifications on both of these items must be coordinated and made compatible.

As with the coolers, the larger filter bodies come under the jurisdiction of the ASME unfired pressure vessel code. If they do, the filter should receive the code stamp. The filter bodies and the heads should be constructed of steel.

Transfer Valves

For the twin arrangements, a two-way, six-ported, continuous-flow valve will be required. If it is a tapered plug type, a means of lifting the plug will be required. The purpose of the tapered plug valve is to provide

a tight metal-to-metal seat to prevent or at least minimize leakage on reseating after a transfer. Cylindrical bore valves of the two-way, six-port type valves are available. These valves use O-rings to act as seals to prevent leakage. Another arrangement, particularly well suited for use with coolers, is the dual three-way valve arrangement. The valves are oriented so that the stems face each other. The stems are connected by an extension shaft. For smaller sizes, a simple handle will serve as an operator. On the larger sizes, a geared operator may be required. Regardless of construction, under no circumstances should valve failure block oil flow. Valve body materials should be steel with stainless steel trim.

The most versatile arrangement, from an operations point of view, is individual transfer valves, one for the twin coolers and one for the twin filters. The use of one valve for both the filter and cooler results in a loss in flexibility, as the cooler maintenance interval is usually somewhat longer term than a filter changeout. Should one cooler be out of service, and the filter that is paired with the operating cooler be fouled, the compressor will have to shutdown. Each user will have to review the extra cost of a second valve against the operation limit.

Some transfer valves block flow better than others, making it necessary to evaluate the design chosen. Valve design and manufacturing technology has improved to the extent that leakage should not be the problem it once was.

Accumulators

Accumulators can be used to help stabilize the lube system against pressure transients such as that from the turbine power operator during a large correction. For a sizing rule of thumb, the system pressure should not vary by more than 10%, while the turbine servo travels full stroke in a one second interval. The role of accumulators for pump switching was covered earlier in the section on Startup Control.

The preferred accumulator is the bladder type as shown in Figure 8-9. Body material should be 300 series stainless steel, in accordance with ASTM A 240. Either a manual precharge valve or automatic charging system can be used, based primarily on the user's preference. The alternative choice is a *direct contact* accumulator as seen in Figure 8-10. This accumulator has several problems. The gas used to pressure the top of the oil will eventually be absorbed by the oil, which can lead to drain line foam-

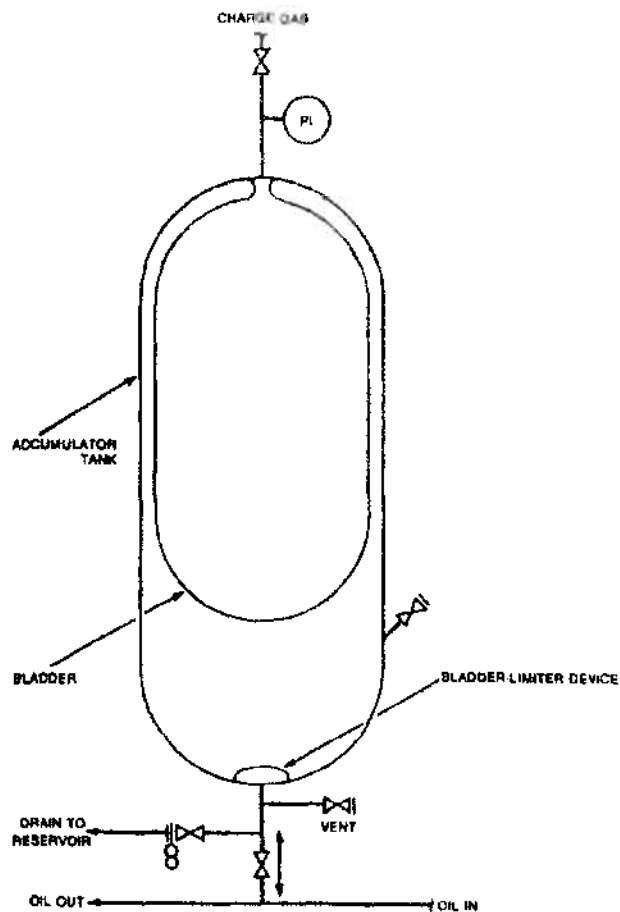


Figure 8-9. Manual precharge bladder type accumulator [1].

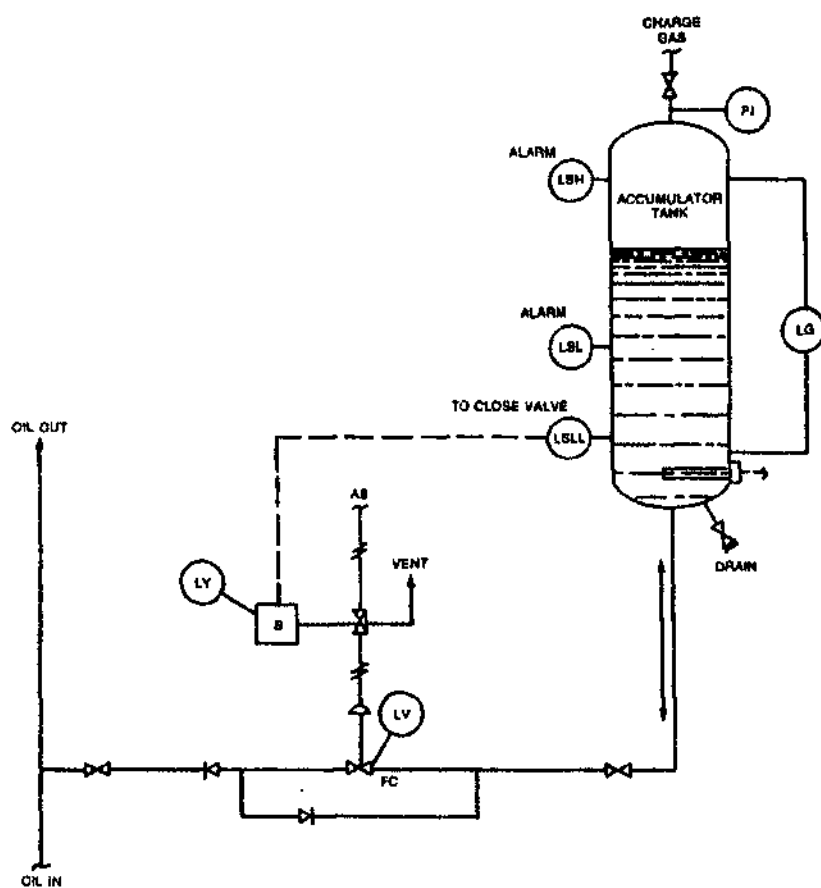


Figure 8-10. Direct contact manual precharge type accumulator [1].

ing. The other problem is that if the oil level isn't carefully controlled, it may dump gas into the oil system. The direct contact accumulator is used primarily when the requirement exceeds the available capacity of the commercial bladder type. With either type of accumulator, additional check valves may be required in the oil system to keep the accumulator from delaying the standby pump start or other similar functions.

Seal Oil Overhead Tank

For liquid film seals, an overhead tank is normally used. The tank functions as part of the differential pressure control. The process gas is referenced to the top of the tank, and the tank's physical height becomes a manometric leg. The oil level is controlled in the tank. Figure 8-11 shows an alternate arrangement for the overhead seal tank, including the recommended operating levels and volumes. The material of the seal oil tank should be stainless, the same as that recommended for the accumulator.

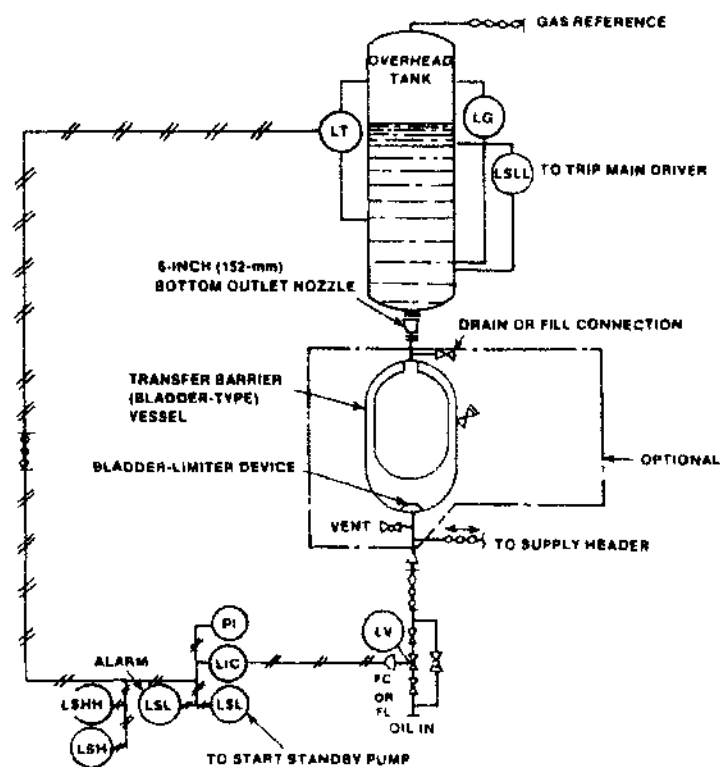


Figure 8-11. Direct contact type overhead tank module for a seal oil system, including an optional transfer barrier (bladder type) vessel [1].

Lube Oil Overhead Tank

An alternative to the coastdown pump approach is the overhead lube oil tank. It has the valuable feature of being motivated by gravity. If reasonably maintained, it should be one of the more reliable methods available. It will safely permit the coastdown of a compressor without losing the bearings due to lack of lubrication during power failure. Figure 8-12 shows two schemes for the tank arrangement. While both systems are in successful operation, the method at (b) requires less instrumentation and would be considered somewhat less prone to problems.

The tank should be placed at an elevation that provides a static head less than the low oil pressure trip switch setting so as to not interfere with the normal trip function. Normally this means that the static head should not be more than the equivalent of 5 psi. The minimum sizing should be based on a three-minute rundown cycle. Adding a bit more to the time is probably not all that costly.

For this tank, stainless steel should be specified considering its importance, and because it is elevated, it will not receive the inspection for main-

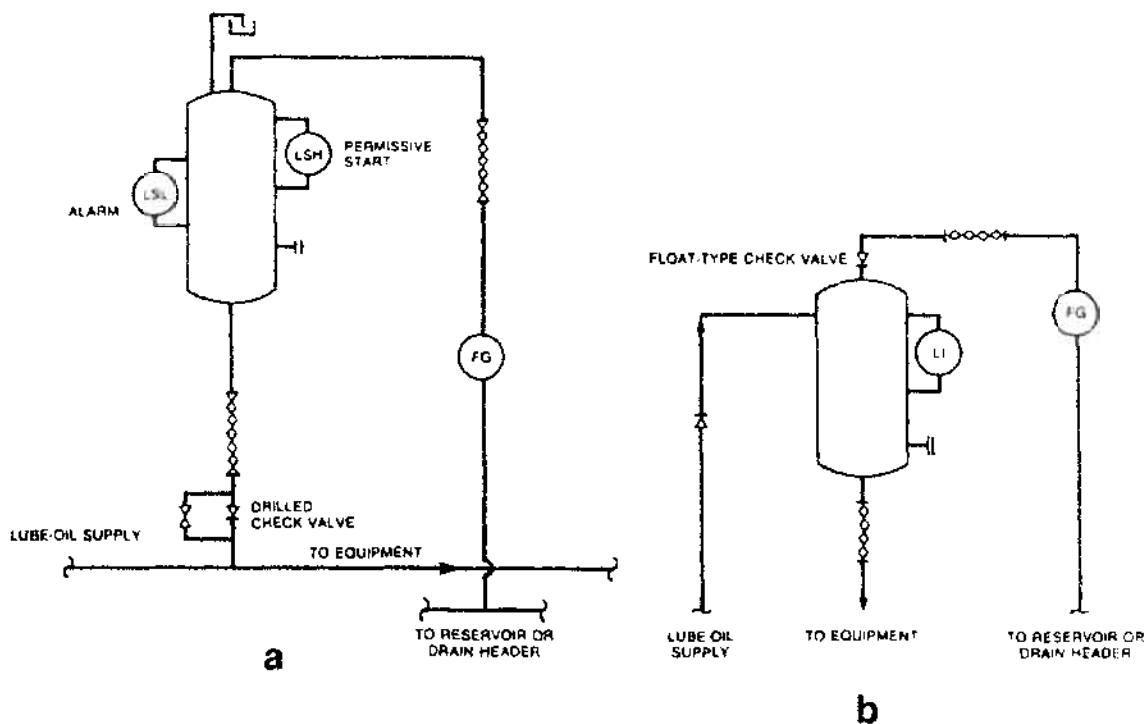


Figure 8-12. Two arrangements for a lube oil coastdown tank. Arrangement "b" is recommended [1].

tenance as the grade elevation equipment. Provision should be made for the cleaning of the tank. If the alternate arrangement in the figure at (b) is selected, the tank will be a code vessel and will require an ASME stamp.

Seal Oil Drainers

When oil buffered seals are used, oil will move past an inner seal toward the process side of the compressor. The oil is prevented from moving into the compressor by a set of labyrinths and is captured in an inner drain cavity. From the cavity, it is piped to the outside where it is collected in either a pot or trap. Figure 8-13 shows several alternative arrangements and equipment. The user must choose between automatic or manual drainers. If the gas from the top of the drainers is to be directly returned to the compressor, it is important that mist eliminators be used. The oil collected in the drainers is reclaimed or disposed of, based on the level of contamination and the user's disposal practices.

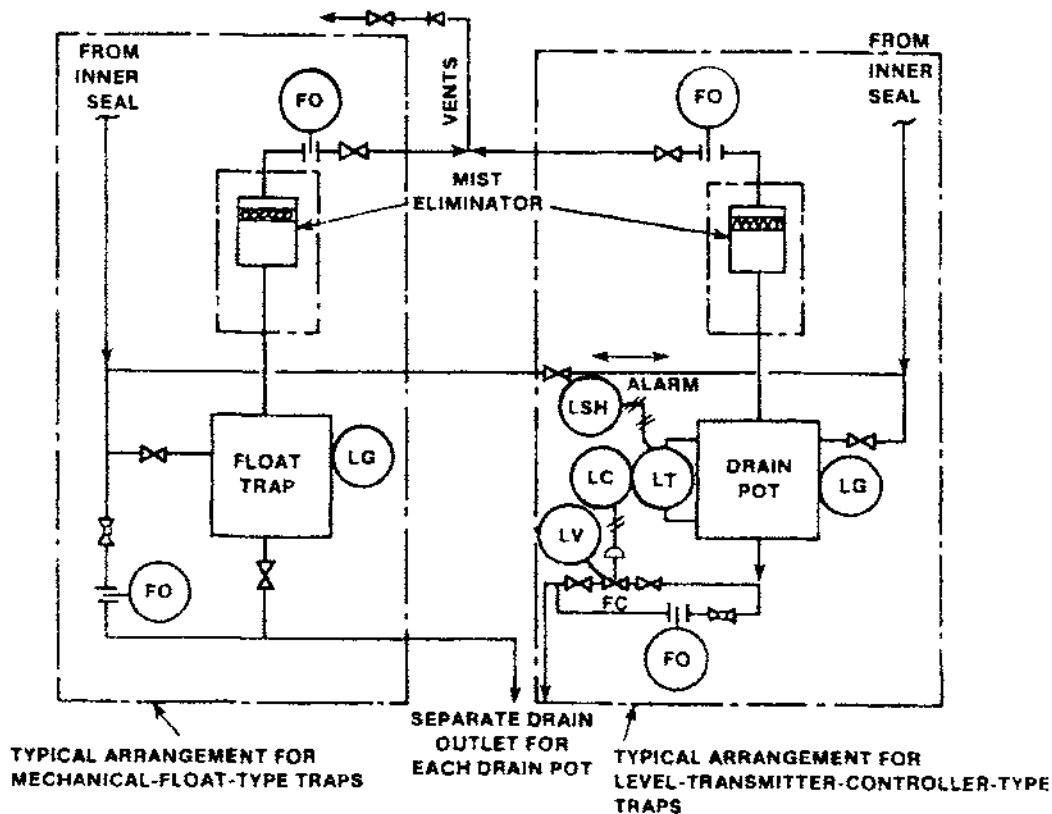


Figure 8-13. Arrangements for inner seal drain traps [1].

Degassing Drum

If the oil from the drainers is not contaminated, it can be returned to the reservoir for immediate reuse. If this be the case, a *degassing drum* is recommended to remove the entrained gases prior to returning to the reservoir. For flammable gases, this practice helps eliminate the explosion hazard in the reservoir. If a suitable low pressure recovery system is available, the gases removed can also be recovered. The alternative is to go to the flare with hydrocarbons. Figure 8-14 is a diagram of a typical

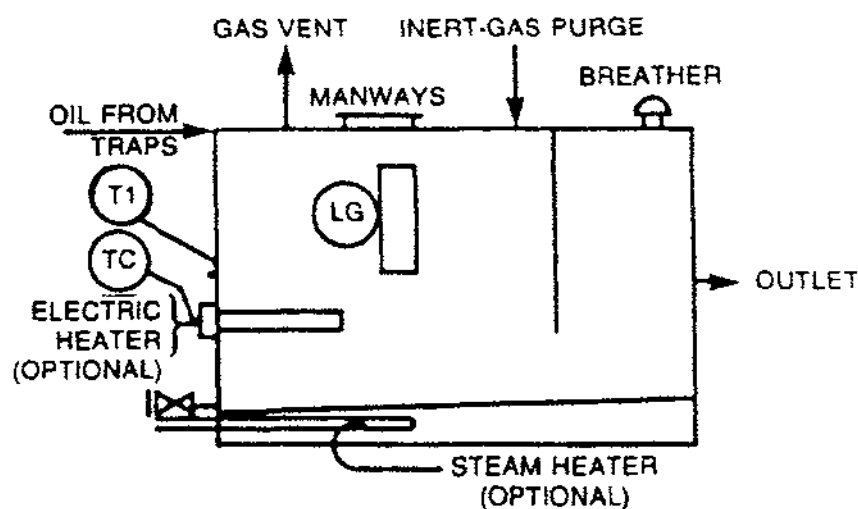


Figure 8-14. Typical degassing drum arrangement [1].

degassing drum. Materials of construction should be consistent with the balance of the lube system. Sizing of the drum should be three times the guaranteed total inner seal leakage from all seals over a 24-hour period. If required to operate the drum at a positive pressure to match a gas recovery system, the suggested design may eliminate the breather and operate as a pressure vessel. The vessel will, of course, need to follow the appropriate pressure vessel design practices.

Piping

Interconnecting piping should be stainless, consistent with the balance of the system. In critical equipment applications, this is definitely recommended. The user must decide for the smaller system to what extent stainless is feasible. For all systems, the piping should follow the recommendations and mandates in API 614. Fundamentally, this means mini-

mize fittings, bend when possible, and fabricate by welding. It really makes no difference how small a lube system is; screwed pipe lube systems leak. On the small systems where welding or seal welding of the screwed fittings is not feasible, fabrication using the thickest walled tubing that the vendor will furnish will make a better system. With a little persuasion, the vendor may consider stainless tubing.

System Review

In the course of obtaining a new lube system, or when revising an existing one, there is a point in the design when the overall system should be reviewed. With a lube system, in particular, it is very easy to get totally engrossed in the individual parts and forget the functioning of the parts together. The possibility of an accumulator overriding the standby pump start function was previously discussed as an example of this problem. The system should be reviewed keeping the basic compressor operating parameters in mind at all times. The startup function should be reviewed in a systematic manner, from the time the pump is initially started to the time the overhead tanks automatically fill. From startup, the review should look at the running operations and the steady state conditions of each item. Some questions that should be answered include: Is the accumulator charged with oil? Did it charge properly to the desired gas pressure? Is the standby pump interlocked to prevent it from cycling if the main pump drops off the line? After looking at the possible upsets that could occur during normal operation, shutdown should be reviewed. If there are seals, do the pressures appear to track the process pressure, particularly if this pressure rises on shutdown? Is there enough power in the driver to take the additional head? Do the valves sequence properly? Are the valves properly sized for each step? This is a sample of the questions that might be reviewed, and not intended to be a comprehensive list. The review should be tailored to the system being considered and the length of the list should reflect the complexity. The review of existing systems for operational improvement can also be considered.

Dry Gas Seal Systems

System Design Considerations

One of the main functions of the seal gas system is to supply clean (filtered) dry gas to the primary seal. Figure 8-15 shows a typical seal gas

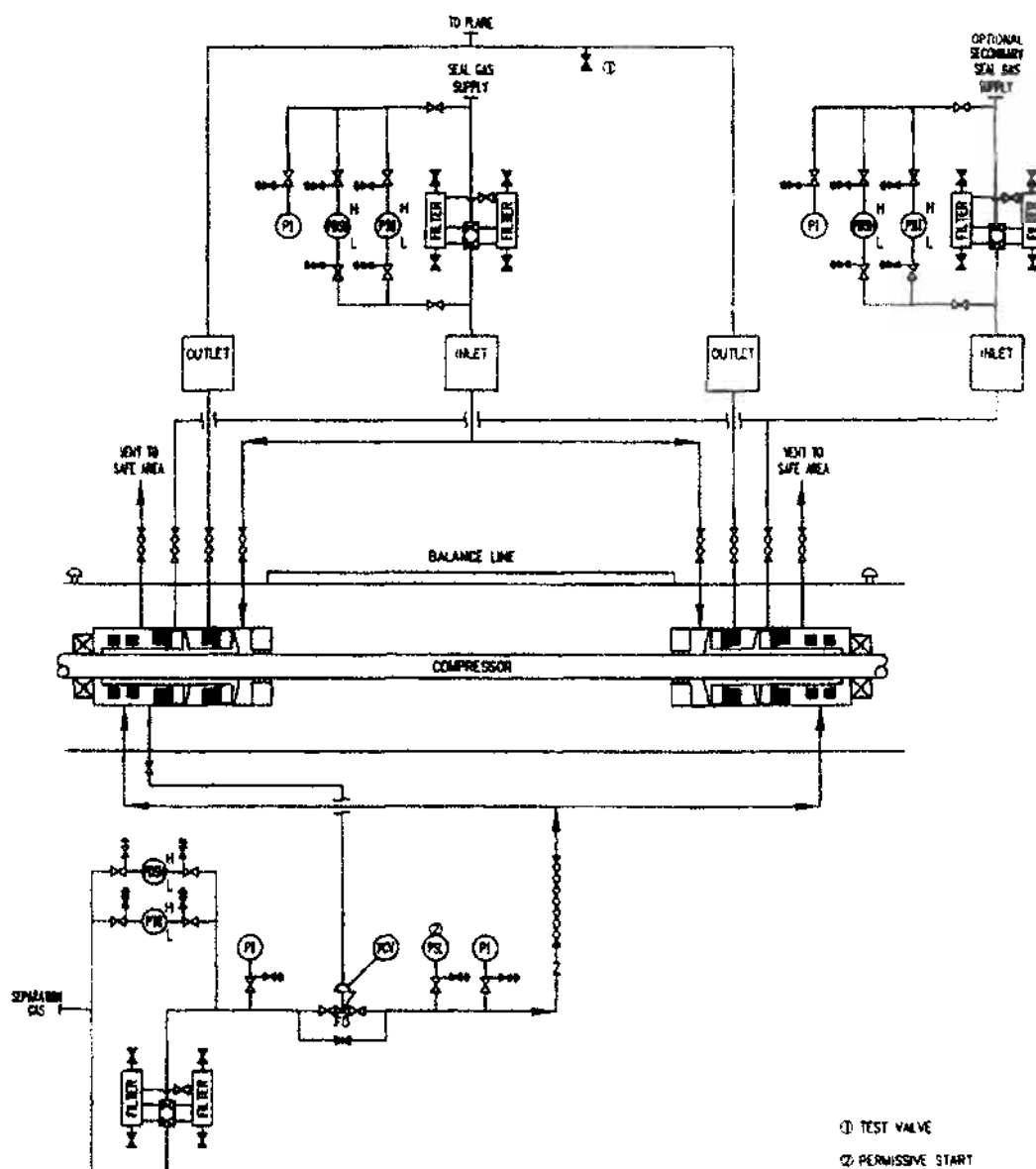


Figure 8-15. Typical dry gas seal system schematic for a tandem arrangement dry gas seal.

system for a tandem seal arrangement. Note the optional secondary seal gas supply to be used with tandem seal having an internal baffle. Figure 8-16 shows a typical seal gas system for a double-opposed seal arrangement. The source of the seal supply (tandem) and buffer gas (double) is normally compressor discharge gas, but is not limited to that source. An alternate supply of gas may be required for startup, if the suction side of the compressor can go sub atmospheric. The system design pressure must be able to accommodate the higher of either settle out pressure or system relief pressure. Care must be taken to exclude all liquids from any of the

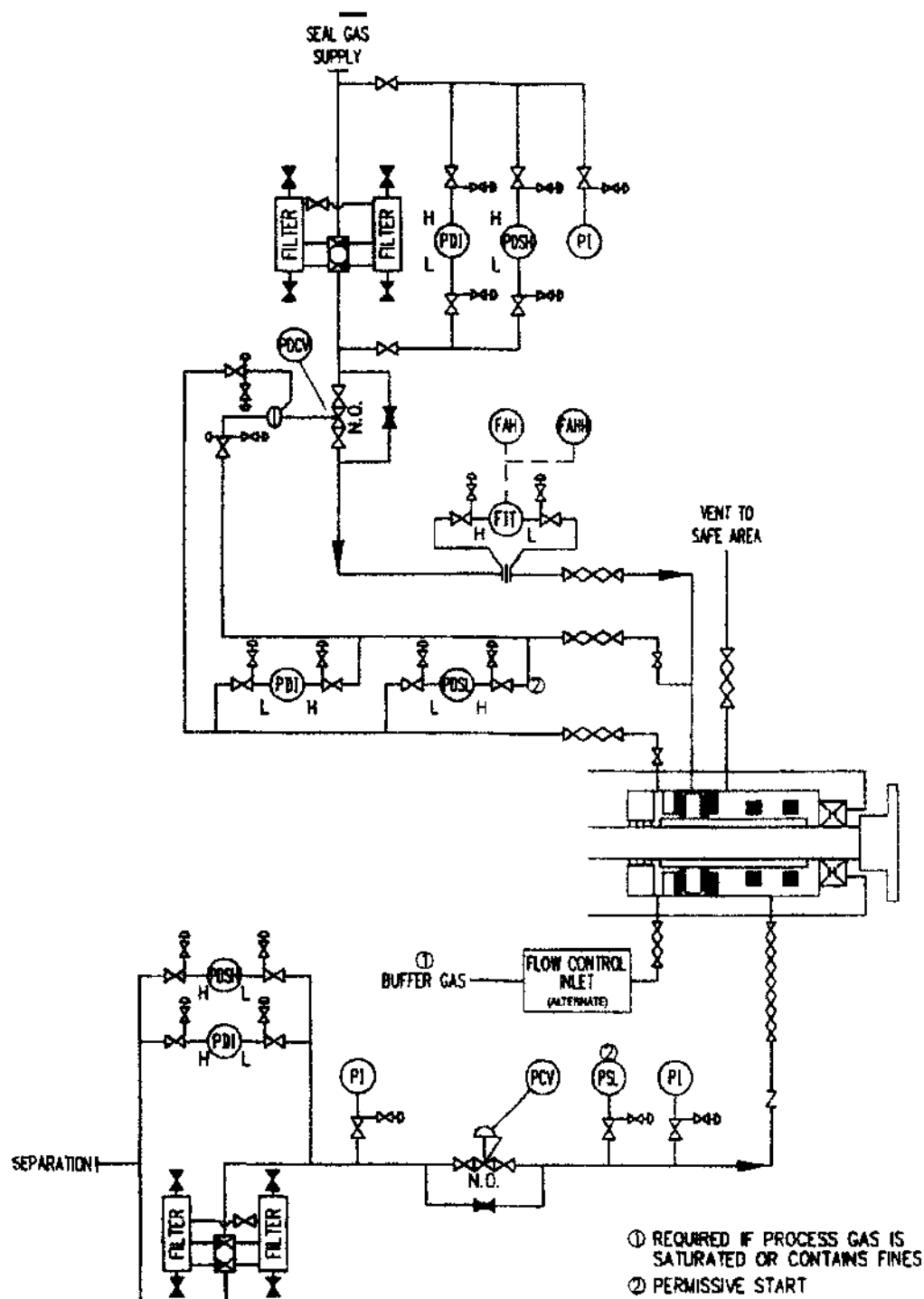


Figure 8-16. Typical dry gas seal system schematic for a double-opposed arrangement dry gas seal.

buffer streams. This may require heat tracing or liquid knock-out provisions and sometimes both. The material of construction is normally austenitic stainless steel. For many of the systems, stainless tubing will be adequate (see Figure 8-17). When pressures are high or the system is very large, rigid stainless pipe is required.

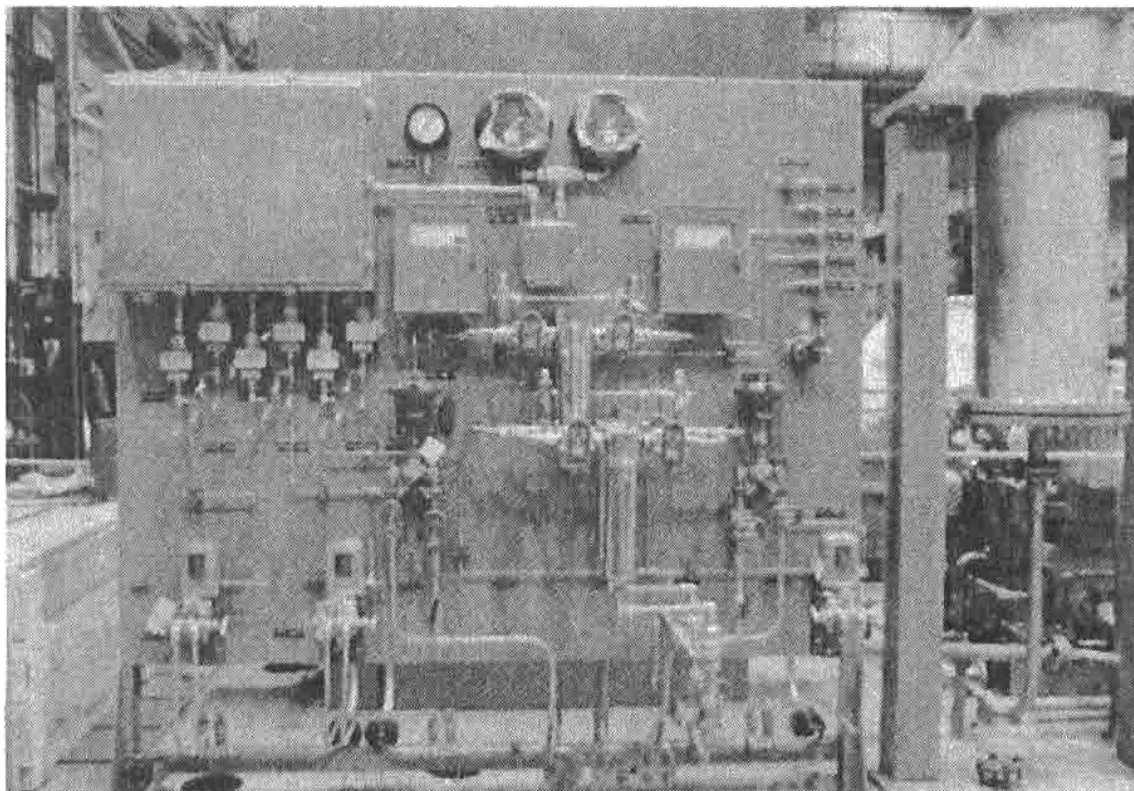


Figure 8-17. Dry gas seal console. (*Courtesy of A-C Compressor Corporation*)

Dry Gas Seal System Control

The control system should be kept as simple as possible to maintain reliability. For tandem seals, the system control for the seal gas supply has traditionally been a differential pressure control, using a direct-operated control. The differential pressure usually was set at 10 to 25 psid. The flow using this control is generally much higher than needed. The differential pressure control can be modified to measure the pressure drop across an orifice and the control converted to a volume control. A velocity across the inner labyrinth of 5 fps would be considered sufficient to prevent back diffusion. This arrangement does have the advantage of reducing the seal gas requirement. For a practical design, the velocity should be increased to at least 10 to 15 fps to allow for wear. Many of the compressor manufacturers will recommend a higher value for more margin.

For the double-opposed seal, normally an inert gas is injected between the two opposed seals. This gas is pressure controlled to maintain a differential pressure higher than the process side pressure. The supply to this seal is critical because a failure will permit the differential pressure across the outer seal to reverse, which will result in a seal failure. This seal arrangement usually incorporates a buffer to the process side of the

seal to keep any dirty process gas away from the dry gas seal. The leakage to process from the gas seal is too low to keep any dirt from reaching the seal. However, there is somewhat of a problem. If the buffer is differential pressure controlled, this differential pressure must be added to the process pressure which in turn sets the injection pressure. For this reason a volume control is a better choice because the differential pressure involved is not as great.

The dry gas seal has one of two types of barrier seal (seal between the bearing and dry gas seal) either a labyrinth or single or double carbon rings. Normally the seal system includes provision to supply buffer gas to the barrier seal, also known as a separation seal. The gas to this seal is referred to as separation gas. One reason for choosing the carbon ring style barrier seal is to keep the separation gas usage to a minimum. The gas is normally nitrogen. The basic control is by a direct-operated pressure control valve.

For the tandem arrangement gas seal, a primary seal vent must be provided to vent the leakage across the process side seal. This vent may be to flare or other suitable gas disposal point. The back pressure under normal conditions should be kept to a low value. A small amount of back pressure is recommended to keep a positive differential across the secondary seal. Leakage measurement may be provided in the vent line to provide health monitoring of the primary seal. Unfortunately, the rotameter, which would be the obvious choice, should not be used because of its lack of reliability. If an orifice or needle valve is used to set the back pressure to the seal vent, pressure upstream of the restriction can be measured for a relative flow measurement. This type of reading does provide trend data that may be used to judge the seal's performance.

The balance of the controls consists of the required pressure switches and/or transmitters to provide monitoring of the system and alarm and shutdown functions for the critical buffers. Filter differential normally is also monitored and alarmed on high filter differential pressure.

Dry Gas Seal System Filters

Filters must be provided for all gas buffers in places where the gas will pass through the gas seal. The gas should be filtered to a nominal value of 2 microns. On most critical systems, dual filters will be required to permit servicing the filters without having to shut down the compressor. This requires the use of a transfer valve that can be switched without causing a flow interruption. If liquids are anticipated, heat tracing or an alternate gas source should be the first consideration. Should this not prove to be practi-

cal, a liquid separation system must be included with the filters. On systems with heat tracing, liquid separation may be desirable as a backup.

Gears

Whenever there is an inherent speed mismatch between a compressor and driver, several solutions are available. For small sizes, V-belts offer advantages in flexibility. For compressors where the power levels approach 100 hp, a more positive drive should be considered. Depending on the application, this value may move up or down by 25 hp.

Most of the discussion has been about compressors over 75 hp and has leaned toward critical or semi-critical equipment. This type equipment requires a gear unit external to the compressor with the gears arranged for increasing or reducing speed as dictated by the application (see Figure 8-18). Alternately, the gear may be integral to the compressor as shown in Figures 8-19 and 8-20. API has two gear standards, API 613 [9] for special purpose gears and API 677 for general purpose gears.

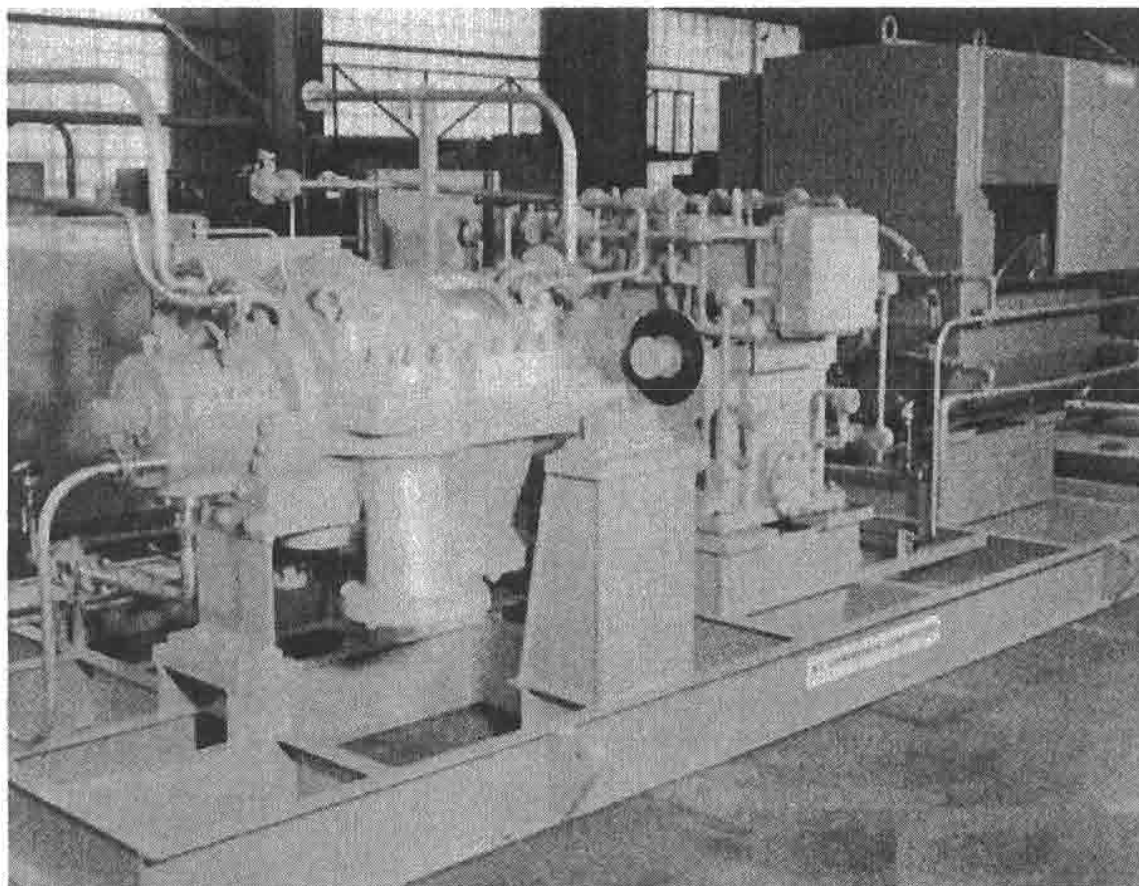


Figure 8-18. Compressor driven by an electric motor through a speed increasing gear. The motor enclosure is a WP2. (Courtesy of A-C Compressor Corporation)

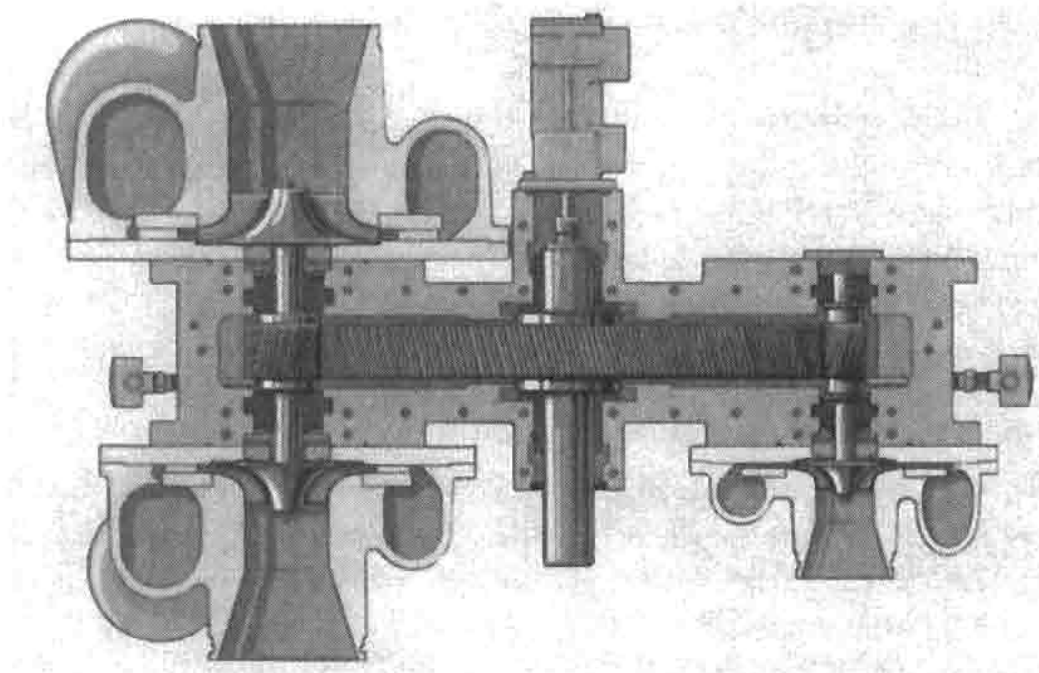


Figure 8-19. Section illustration of the gearing of an integral geared compressor.
(Courtesy of Elliott Company)

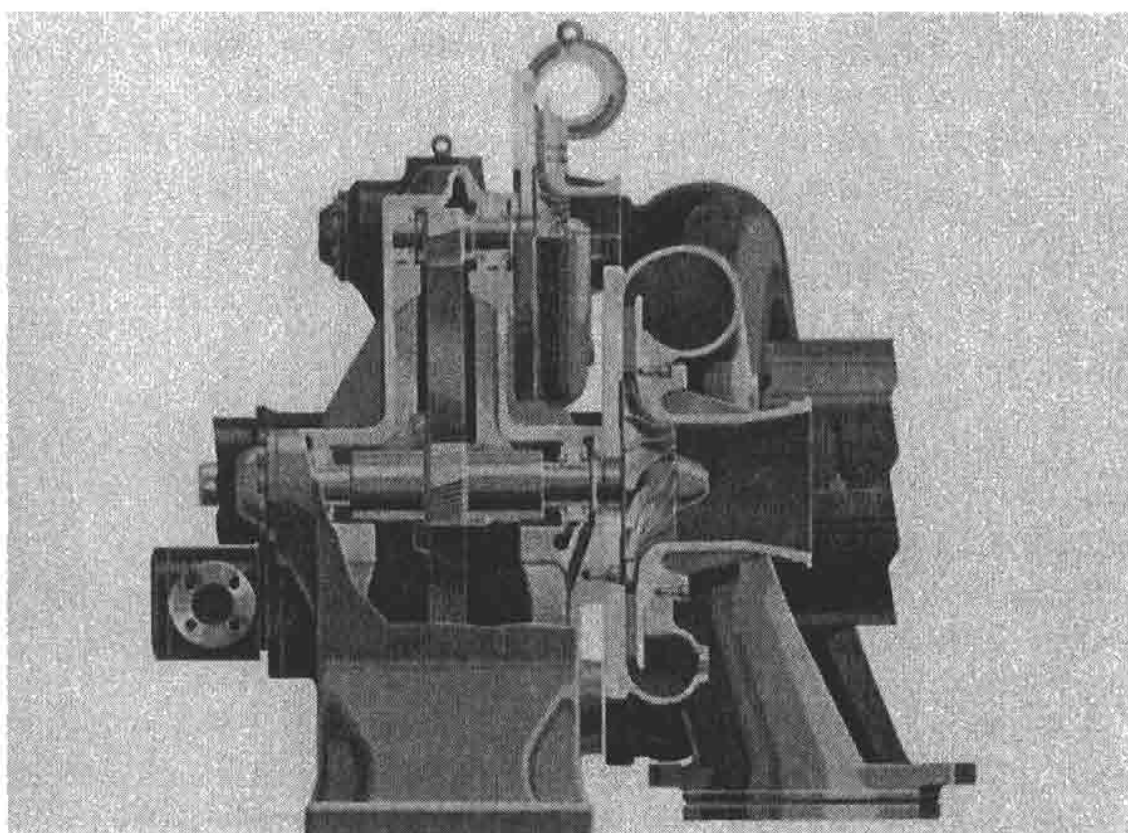


Figure 8-20. Cutaway of a gear-mounted multistage centrifugal compressor.
(Courtesy of Nuovo Pignone)

Gear Design and Application

The special purpose gear unit should have a specified API 613 service factor of 1.4 as a minimum. The service factor used in the gear design should be selected to meet the requirements of API for the application.

The gear unit must be tuned to the system in which it operates. The vendor with the train responsibility must analyze the system to be certain the gear unit will not transmit torsional or lateral vibrations to the rest of the system or be damaged by system excitations.

If the gear unit is retrofit, for example when replacing a turbine with a motor and gear unit, the system, in like manner, must be thoroughly analyzed. In either case, the gear supplier must furnish the user with adequate data on the mass elastics of the unit, inertias, bearing characteristics, and the like, to permit the user's own or a consultant's analysis.

Resonant responses must not coincide with excitation frequencies of rotational shaft speed, especially gear meshing frequency (the speed of a shaft times the number of teeth of the gear on that shaft), or other identified system frequencies; otherwise, a self-excited system will exist. Lateral response criteria should conform to API 613.

An important consideration of a gear unit is the *pitch line velocity* (PLV), which is the product of the gear or pinion pitch diameter and the shaft speed of the gear or pinion.

A general classification of pitch line velocity and corresponding American Gear Manufacturers Association (AGMA) quality values are:

- 20,000 fpm or less—moderate service—AGMA 12
- 20,000 to 30,000 fpm—medium service—AGMA 13
- 30,000 fpm or higher—severe service—AGMA 14

To promote even wear, a hunting tooth design is desirable. In this design, a pinion tooth does not contact a given gear tooth more than once until it has meshed with all the other gear teeth.

For maximum reliability, a gear-unit ratio must be within the capability of a single-step application. Practically, ratios may approach 6:1, but the preferred ratios are in the 3:1 to 5:1 range.

A double-helical design should be considered, particularly for critical service. Rotation of the helices should be such that they will be apex leading, that is, teeth engage at the centers with mesh progressing to the tooth outer ends. Additionally, rotation should permit the force from the gear mesh to cause the gear to load the lower half of its bearings. The

double-helical gear generates no thrust load of its own and can be operated with no thrust bearing if flexible disc or limited end-float type couplings are used on each shaft. The advantage of this approach is the elimination of the horsepower loss associated with a thrust bearing.

The low speed gear shaft and the housing must be designed to permit installation of a stub shaft for a torsigraph unit if an operational problem occurs. API 613 gives the details of the shaft end requirements for attaching a torsigraph. This should be done on all synchronous motor compressors and on multiple driver or multiple compressor case trains.

Rotors and Shafts

The shaft and gear teeth of a pinion should be of an integral design. The gear should be a forging with the gear shrunk on a keyless shaft for critical service with a pitch line velocity over 20,000 feet per minute. Under 20,000 feet per minute, the gear may be "rimmed," that is, design-welded in place.

The gear teeth should be heat-treated for proper strength and through hardened. Alternatively, surface hardening by carburizing or nitriding is used. Flame and induction surface hardening are also alternative hardening methods but normally are used less than carburizing or nitriding. For new gears, through hardening is preferred, using the surface hardening for later upgrades. Requirements for strength and hardness must result in an adequate durability and tooth stress as determined by API 613. The surface finish of the teeth must be 20 microinches Arithmetic Average Roughness Height (Ra) or better.

Gear elements with adequate hardness and surface finish have a greater resistance to initial scoring, a destructive phenomena found in meshing teeth.

For adequate operation, there are many features that should be incorporated in the rotors. They include: (1) good surface finish, (2) modified tooth forms, (3) high AGMA quality number, (4) hunting tooth design, (5) no resonant responses, (6) high damping journal bearings, (7) good contact pattern between teeth, and (8) good rotor balance. Other factors to be considered for quiet gear operation are proper diametral pitch, pressure angle, helix angle, overlap ratio, backlash, and minimum apex runout. (For the meaning of these terms, refer to API and AGMA standards.) API 613 has a requirement for maximum apex run-out under subject of "Axial stability of the meshing pair."

Shafts must be stiff enough to prevent deflection that would adversely affect the tooth contact in operation. Internal alignment should be carefully performed. This is another area that can directly affect tooth contact.

Each shaft should have an integral coupling flange rather than a removable coupling hub. If removable hubs must be used for some reason, they should be of the keyless, hydraulically dilated design.

Bearings and Seals

The journal bearings must have stiffness and damping properties sufficient to prevent bearing contributed vibrations and to result in proper gear contact. The stiffness and damping properties of the journal bearings affect the rotor system dynamics. Normally, stabilizing bearings, such as tilting pad and three-lobe, are needed to prevent shaft oil whirl. The tilting pad bearing can be seen in Figure 5-38. Figure 8-21 shows a typical three-lobe journal bearing.

If a thrust bearing is used, it should be of a tilting pad, self-leveling design. With a double-helical gear unit, thrust bearings, if used at all, should be on the low speed shaft only to accommodate loads in both axial directions. Journal and thrust bearings must be split for removal and installation without having to remove the coupling hubs.

Seals should be non-contacting, multiple point labyrinths. The housing and seals should be drilled and the housing tapped to accommodate a dry

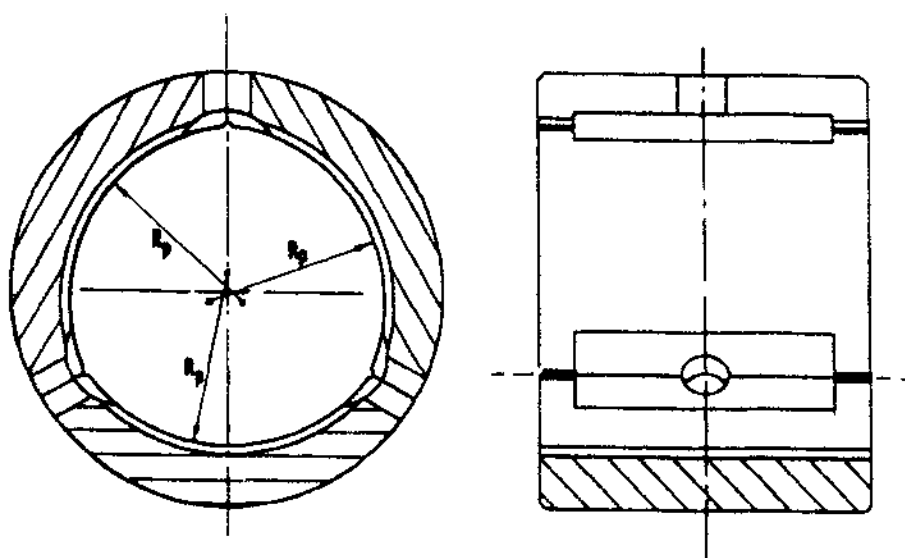


Figure 8-21. Three-lobe journal bearing. (Courtesy of Turbocare, A Division of Demag Delaval Turbomachinery Corp., Houston facility)

gas purge connection. Appropriate shoulders or shaft fingers should be immediately inside the seal to aid in the prevention of oil leaks.

Housing

Either cast from cast iron or carbon steel or fabricated from carbon steel, the housing (gear box) must be rigid to prevent deflection under loads. The housing must be thoroughly stress-relieved to maintain accuracy. The housing should be horizontally split at the shaft centerlines. The horizontal joint should be finished to prevent oil leaks with a metal-to-metal fit. Ample side clearance between housing and rotors will prevent oil pumping and will permit free fall of oil to the sump. Sump depth, adequate windage baffles, and generously sized return lines prevent oil level buildup or foaming in the housing. Uneven impingement of hot oil on the housing causes differential thermal growth of the unit and must be avoided.

All piping and instrument connections should be made on the lower half of the housing to aid in removal of the top half. Other connections, if they must be made in the top half, should have easily accessible joints.

Lubrication

A *dry sump* design should be employed. The gear unit for a train with a central lube oil system should be designed for the turbine grade oils of the system. Typically, 150 Saybolt Seconds Universal (SSU) oils at 100°F (ISO 32) with an inlet temperature of 110°F to 120°F are adequate.

If the gear unit application must have its own oil supply, such as with some retrofits, the intent of the API 614 should be employed.

All lube piping should be stainless steel. The stainless steel should be one of the 300 series, preferably "L" or low carbon grade, such as type 304L.

Couplings

Introduction

The purpose of a flexible coupling is to transmit torque from one piece of rotating equipment to another, while accepting at the same time a small amount of misalignment. Flexible coupling misalignment is expressed, as an order of magnitude, in thousands of an inch. Actual misalignment, expressed in coupling terms, is angular in nature and expressed in angular units, that is, degrees. How many is a function of the coupling type and

installation. An installation variable is the equipment movement due to the temperature changes taking place in the machines as they go from the non-operating state to operation. Some angular values will be used in the discussion of the various types, but again, these are for reference only. Each application must be reviewed using the type coupling selected and the specific design proposed by the vendor.

Ratings

It is recommended that couplings and coupling-to-shaft juncture should be specified for a capability of 150% of normal torque for flexible element couplings and 175% for gear couplings. The coupling rating should be based on the maximum continuous misalignment capability of the coupling. Further, the coupling and coupling-to-shaft juncture should be rated for a capability of 115% of any specified maximum transient torque. Special purpose couplings are covered in API Standard 671 [6].

Spacers

There are a number of features that all couplings have in common. One is the need for a spacer. API 671 calls for an 18-inch spacer minimum. This is reasonable for smaller units, say to 5,000 hp; however, as the size of train increases to 15,000 to 20,000 hp, a 24-inch spacer should be considered. Above that size, longer spacers, 30 to 36 inches, are in order. The spacer first of all provides for unit separation and maintenance space. Secondly, the longer the spacer, the less the angular deflection of the coupling at its flexure point for a given offset. This makes absolute equipment alignment less critical.

For the smaller units, under 200 hp, this requirement may be relaxed. Space for service is not as critical, but a minimum of 5 inches should be used.

Hubs

The attachment of the coupling to the shaft has been the source of many problems in the industry. The following practices are recommended for critical equipment.

1. If a coupling hub does not have to be removed for maintenance, seal removal, or impeller removal, it should be integral with the shaft.

2. On slow-speed equipment, where the hub does not have to be removed for maintenance and integral hubs are not available from the supplier, a straight fit can be used with a reasonably heavy shrink—1.25 mil/in. of diameter. Tolerances may allow this value to vary from 1.0 to 1.5 mil/in. of diameter; however, 1.0 mil/in. may be too loose for some applications, while 1.5 mil/in. over-stresses hubs at maximum bore sizes. In light of this, each application should be carefully evaluated if there is no experience at hand to use as a guide. Of course, field removal at the higher shrink fit is more difficult. Keys should not be used.

Where removal is necessary, keyed hubs on straight shafts should use 0.5 mil/in. to 0.75 mil/in. shrink fit. Caution should be exercised in these applications.

3. If the hub removal is necessary, such as required on compressor with non-split seals, a tapered hub fit on the shaft should be used. The removable hubs should have tapped puller holes. The shaft should be keyless with the preferred method of installation and removal by use of hydraulic dilation. Two injection ports 180° apart should be used whether injection is through the shaft or through the hub. Shrink fits should be 2 to 2.5 mil/in. of diameter. API 671 recommends 1.5 mil/in. minimum, but experience indicates the heavier shrink may be required. For the juncture rating calculation, a friction value of .12 is recommended.

In the NEMA standard, $\frac{1}{4}$ in./ft taper has been used for keyed tapered hubs. However, API 671 recommends a $\frac{1}{2}$ in./ft taper for hydraulic release, keyless hubs (see API 671, Appendix E). The $\frac{1}{2}$ in./ft seems a reasonable compromise, but there is still a strong following for the NEMA taper.

Contact area for hydraulically removable, keyless hubs should be a minimum 85% when using bluing and a taper gauge. It is recommended that a set of gauges, ring and plug, be purchased with a new unit for maintenance and replacement purposes. A male and female cast iron lapping block set should also be made at the same time to help achieve the required fits by lapping.

For installation, care should be used to install the hub at the proper location on the shaft taper to provide the interference needed. A jig should be used and allowance made for hub shrinkage on cooling.

4. If keyed tapered shafts are furnished, as may be true with some installations, particularly for machines under 1,000 hp with small shafts, two keys are recommended. A shrink fit of 1 to 1.5 mil/in.

should be used. Unfortunately, heavier shrinks are impractical because of hub stress.

For maintenance and stocking, it is sometimes desirable for plants with a large number of pieces of equipment to minimize the number of coupling sizes. Also, plant-to-plant coupling exchanges may be made if the size permits. However, caution should be exercised because what appears interchangeable may not be. Because of the tuning of torsional criticals, the original equipment vendor may have had a coupling custom-made for a system. In the fine tuning of torsional criticals, couplings are the logical focus as the other components must be established before a system torsional study can even be made. A coupling tuned for one system may not be a logical choice for another. Therefore, before arbitrary trades are made, this aspect should be reviewed. The same comments apply to lateral criticals; however, a new critical speed analysis may not be necessary if the replacement coupling weight is within 10% of the original. If heavier, a new lateral analysis should be performed.

Gear Couplings

Gear couplings are no longer the primary coupling of choice, having been superseded by the flexible element coupling. When used for critical equipment, the gear coupling should be specified as the high speed type regardless of speed for most applications. This class of coupling can be balanced and has the highest grade material. Plain hub style gear couplings are usually called a marine type. When the application permits, this is the preferred type. Since first preference is integral hubs, it follows that the marine type must then be used. Another advantage to this type is that all wearing parts are removable with the center section. The disadvantage of the marine type coupling is that it has a higher overhung moment because the teeth are on the spacer and the weight is further removed from the bearing centerline. This may cause a problem with lateral criticals. When this is true, teeth on the hub must be considered. This requires a removable hub design because the hub itself now becomes a wearing part. Teeth on the hub can be furnished as a standard arrangement with teeth near the outboard end or a low moment arrangement with the teeth on the inboard end of the hub. The latter, which is attractive in reducing overhung weight, tends to have problems with clearance for sleeve withdrawal and, therefore, should only be used if rotor dynamics problems cannot be solved in another manner.

It should be noted that a mix and match has been used with a marine type plain hub on one end of the coupling and a toothed hub on the other. This, however, is most extraordinary and is mentioned only to indicate a possibility for problem solving.

Gear teeth hardness should be a minimum of 45 on the Rockwell C scale. Hardness of the teeth having the greater face width (generally the sleeve teeth) must be equal to, or preferably greater than, the hardness of the mating teeth.

Lubrication may be continuous from the lube oil system or grease packed. For longer continuous operation and where the coupling speeds are high, continuous lubrication is preferred. Where maintenance intervals permit and the separation forces on grease are not too high, grease is a good solution. There are newer coupling greases on the market whose oil and thickener are the same density and, therefore, are not subject to centrifugal separation. Run times on conventional grease lubricated couplings are 8–18 months.

When continuous lubrication is called for, the couplings should be supplied with oil filtered to a minimum of 2 microns. Experience shows that filtering to $\frac{1}{2}$ micron is possible and desirable. Usually this is done with a separate filter; however, the whole system may be filtered to this level. The oil quantity furnished, per gear mesh, should be 3 gpm minimum.

Alignment

The absolute angular misalignment capability of a gear coupling is a function of the tooth form and backlash. The total angular misalignment at each mesh will be the total of the angular component and the angular result of parallel offset, both of which are the vector sums of horizontal and vertical misalignments. Values from $\frac{1}{2}^\circ$ /mesh to 6° /mesh may be found in the catalogs. However, this higher number is the value at which the teeth exceed the clearance and the coupling truly locks up.

Equation 8.1 is based on a maximum sliding velocity during misalignment of 5 ips and Equation 8.2 is based on 8 ips. Research by the Naval Boiler and Turbine Laboratory [7] developed these values. Experience indicates that the more conservative range of 1–3½ ips is most desirable.

$$\alpha = 5,500/dN \quad (8.1)$$

$$\alpha = 8,800/dN \quad (8.2)$$

where

α = angular misalignment per mesh, deg

d = pitch diameter of the teeth, in.

N = shaft speed, rpm

Flexible Element Couplings

Flexible element couplings transmit torque and accommodate misalignment by use of flexible diaphragms or discs. These elements are of metal construction and may be used singularly or in packs. Refer to Figure 8-22 for a multiple element coupling of the diaphragm type. A single element, diaphragm type is shown in Figures 8-23 and 8-24. A flexible disk is illustrated by Figure 8-25.

The flexible element coupling requires no lubrication, which is a distinct advantage. The need for lubrication is always a problem with gear couplings. This advantage is partially offset in the flexible element by the need for corrosion-resistant materials for the normal chemical plant atmosphere. Inconel 718 and 15-5 PH have been used and have been rea-

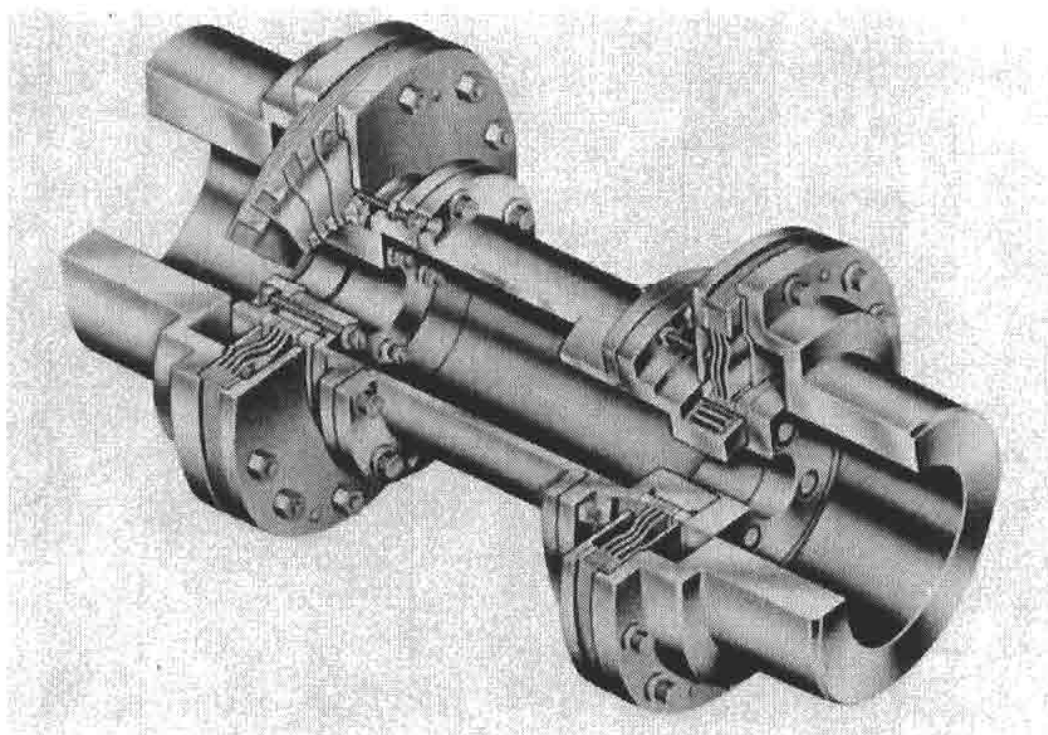


Figure 8-22. Cutaway of a multiple diaphragm flexible element coupling. (Courtesy of Zurn Industries, Inc., Mechanical Drives Division)

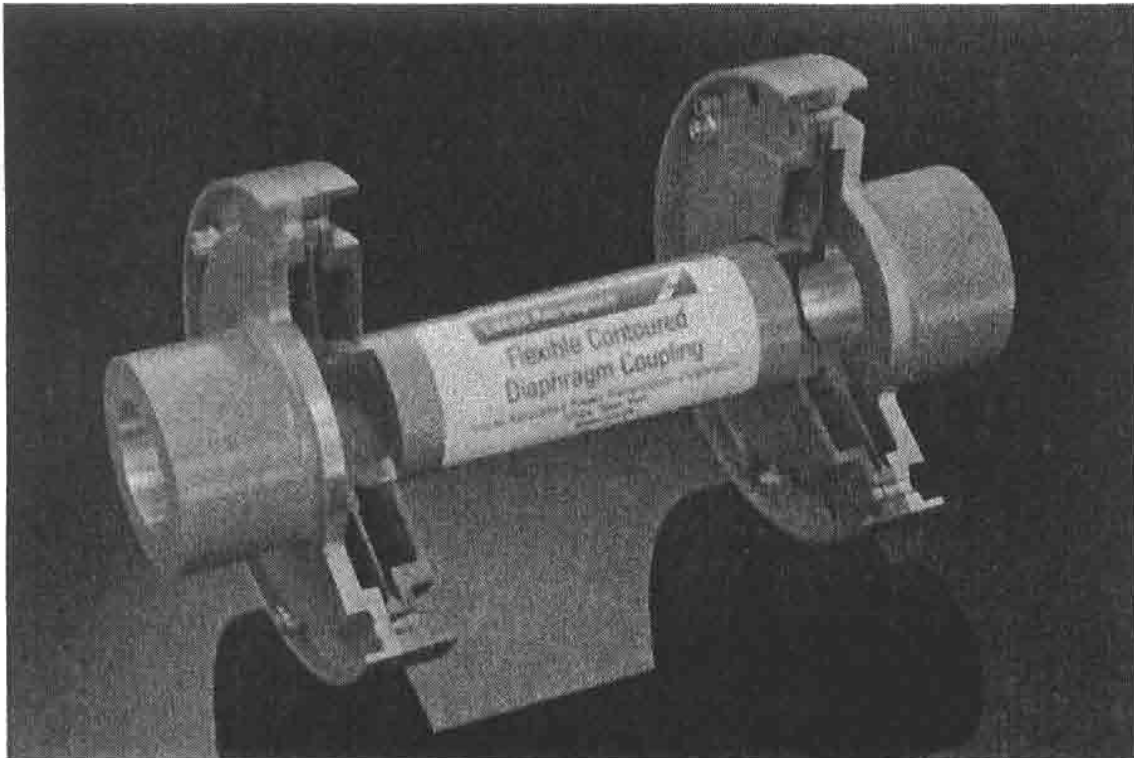


Figure 8-23. Cutaway of a single diaphragm flexible element coupling. (Courtesy of Lucas Aerospace Power Transmission)

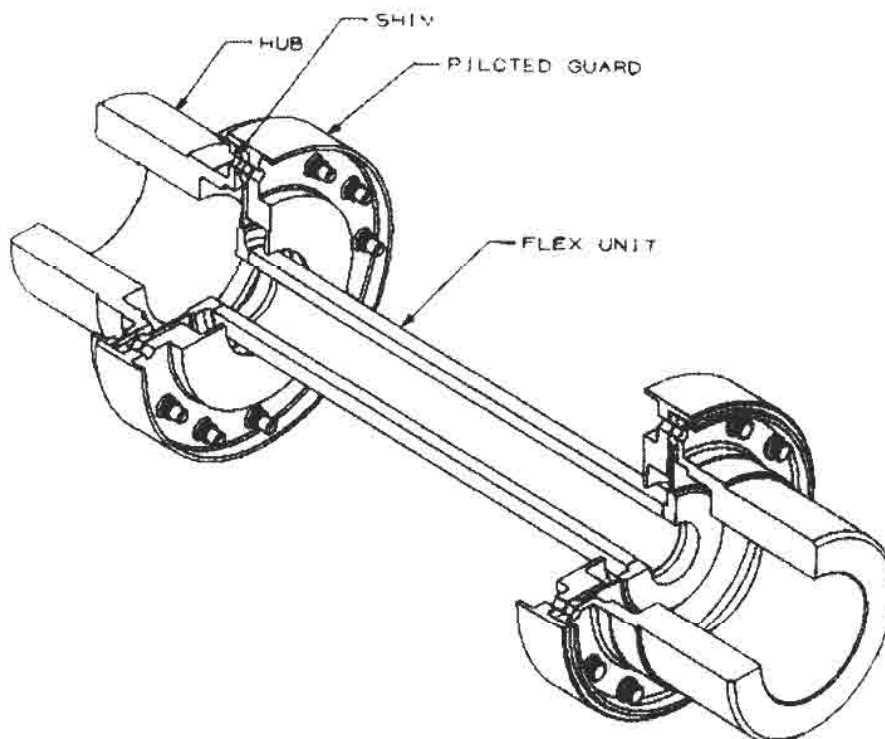


Figure 8-24. Section drawing with parts identification of a single diaphragm flexible element coupling. (Courtesy of Lucas Aerospace Power Transmission)

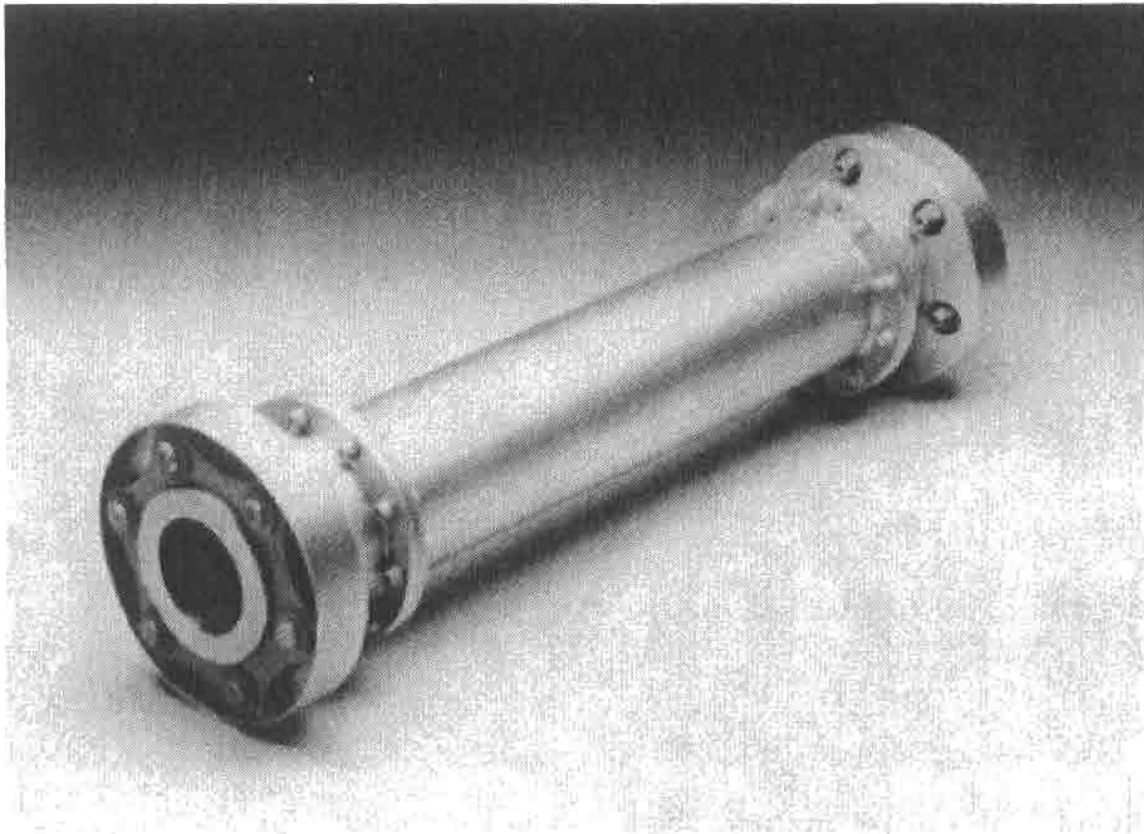


Figure 8-25. Multiple disk flexible element type coupling. (*Courtesy of Kop-Flex Power Transmission Products*)

sonably successful in chloride atmospheres, and 300 series stainless steel has been used successfully in hydrogen sulfide environments.

Another advantage of the flexible element coupling is the lower bending moment imposed on shaft ends. This can be a problem with the gear coupling, particularly if the loading is high or lubrication poor. The flexible element, because of the elastic member, has a predictable bending moment which is normally much lower than the comparable gear coupling [8]. Axial loads transmitted are much less than for a gear type coupling. This greatly reduces thrust bearing loads. Absence of radial clearances through the major components makes it possible to obtain a precision repeatable balance. Flexible element couplings are available in both single and multiple element form.

Flexible element couplings tend to be somewhat heavier than the comparably rated gear coupling. On a retrofit, heed the earlier warning about lateral criticals. The coupling can handle axial misalignment but is more restrictive than the gear type.

Limited End-Float Couplings

The flexible element coupling is by the nature of its design a limited end-float coupling. In gear applications where flexible element couplings are used on both the low and high speed sides, the gear, if double-helical, may be operated without need of a thrust bearing. The gear elements are allowed to float between the compressor thrust bearing and the motor magnetic center.

For gear couplings, to guard against transmitting thrust continuously from the motor to driven equipment bearings, use gear-type flexible couplings with the limited end-float provision. The following requirements are specified for all large motors:

1. The coupling is required to have a no more than $\frac{1}{4}$ -inch end-float (if a gear type).
2. The motor end-float must exceed the coupling end-float by $\frac{1}{2}$ inch, making the minimum permissible motor end-float $\frac{1}{2}$ inch.
3. The motor magnetic center must be within $\frac{1}{2}$ inch of the motor's geometric center.
4. The motor centering force must be sufficient to return the rotor to magnetic center against friction of a new, properly lubricated gear coupling.
5. The momentary end thrust transmitted by the motor to the coupling and thus to the driven equipment must not exceed 100 pounds per thousand horsepower. (This provision covers thrust transmitted when breaking away the engaged coupling gear teeth from their rest position or when the coupling separators touch when starting.)

The gear coupling must be modified to provide a limited end-float feature. This feature is desirable when sleeve bearing machines without thrust carrying capability are used. On shutdown, the units can float to their limit stops, for example, on a motor. On shutdown, the motor will float to a bearing face. On restart, the friction in the teeth keeps the rotor on a thrust face, which is not capable of carrying load and causes a bearing failure. To prevent this, a feature is added to a gear coupling limiting this end-float. Happily, the flexible disc coupling is an inherently limited end-float design. On a motor gear compressor arrangement, the parasitic power used by the gear thrust bearings can be eliminated by omitting the thrust bearing entirely and using both a high-speed and low-speed limited end-float coupling.

Instrumentation

Overview

Compressor instrumentation has moved from some simple pressure and temperature gauges to a field all of its own. Systems now in the marketplace appear to rival the telemetry used by NASA on the space flights. Thirty plus years ago, vibration monitoring was non-existent for all but some hand-held velocity type of pickup equipment. The name Bently was known to only a few friends of Don Bently, who were aware of his experiments with proximity probes. Today the name is associated with a large vibration monitoring equipment company with several competitors. The advent of the vibration probe and monitor opened up a new world to the compressor engineer. It was wonderful and frightening all at the same time. This little probe saw things that had been present in compressor operations for a long time but were unknown to both the design engineer and the operator. The measurement data came in faster than the technologists could analyze the information. An awful lot of time was spent worrying about the insignificant data, only because it was there. With the maturing of the measurement, part of the vibration story can be the rotor dynamics field. The ability to observe the shaft dynamics in real time by way of orbits was truly fascinating.

The subject of vibration instrumentation can fill many pages, but is only one phase of the world of compressor instrumentation. Therefore, the coverage here can only be brief.

Pressure

The most fundamental aspect of basic instrumentation for monitoring compressor performance is pressure. That is what the compressor does, it delivers gas at a pressure. In most cases, the measurement instrument is a common gauge. In covering these items, the basics of how the instrument works or why will be left to the reader. As was mentioned, space permits only a superficial treatment. In the era of computer monitoring, the pressure transducer or transmitter is becoming more prominent. This instrument converts the input signal into another form proportional to the input. For example, an electrical transducer or transmitter would convert a pressure signal to a proportional electrical output signal, a logical form for input to a computer or data logger. Chapter 10 discusses the basic requirements of pressure instrumentation in more detail.

It should be pointed out that it is probably not necessary to make as many redundant taps as called for in a code test. However, as long as taps

are made, they can be made to provide information as accurately as possible. There is no point in putting a calibrated gauge or transducer into a pressure connection that is incapable of delivering a reliable reading.

Differential pressure is included in the pressure instrument class. Good differential readout gauges are still not all that common. Fortunately, in the transducer category, they are more readily available. Figure 8-26 covers some installation details for pressure-oriented instrument piping, supplementing the information presented earlier in Figure 8-8.

Temperature

Verifying temperature is the second most important aspect of any compressor operation. As with pressure, the basic form of measurement is a simple temperature gauge. The construction of the gauges is quite varied, ranging from a bimetallic device to the filled systems. When transmission is involved, the sensor becomes quite simple, taking the form of a thermocouple or a resistance temperature detector (RTD). The monitor does the translation from the native signal to a temperature readout or signal proportional to temperature.

Temperature switches are also part of the temperature instrument family. Although not sophisticated, entire compressors are influenced by the temperature switch. It is now becoming more popular to take a thermocouple signal to the computer and generate a switch closure as part of the computer signal.

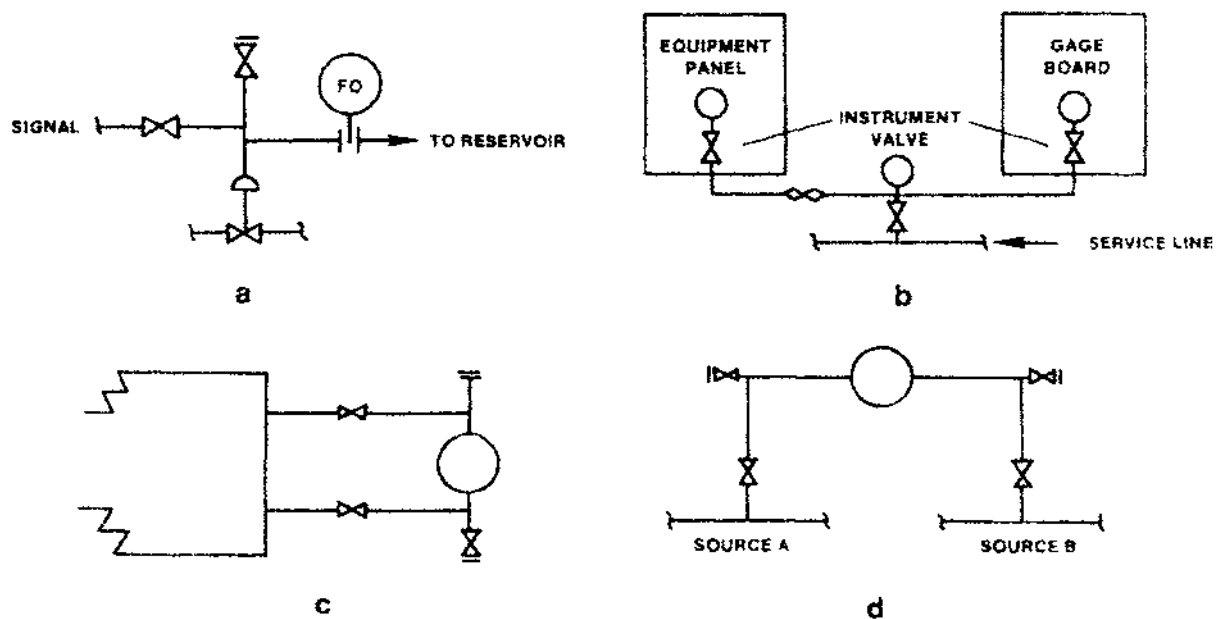


Figure 8-26. Installation details for pressure oriented instrument piping [1].

API 670, "Vibration, Axial Position, and Bearing Temperature Monitoring System" includes temperature in its scope. For several years, radial and thrust bearings have been instrumented using either thermocouples or RTDs. Each user specifying the instrumentation had the bearings fitted in his own way. While this gave good data in some instances, it was not consistent. Furthermore, the data from one user could not be compared with the data from another. In fact, because of different influences by the various compressor vendors, one large user could not correlate his own experience. While the addition of bearing temperature monitoring to the API standard has numerous benefits, the more immediate one is the establishment of a standard method of installation. Figures 8-27 and 8-28 show the recommended installation position on the radial and thrust bearing respectively.

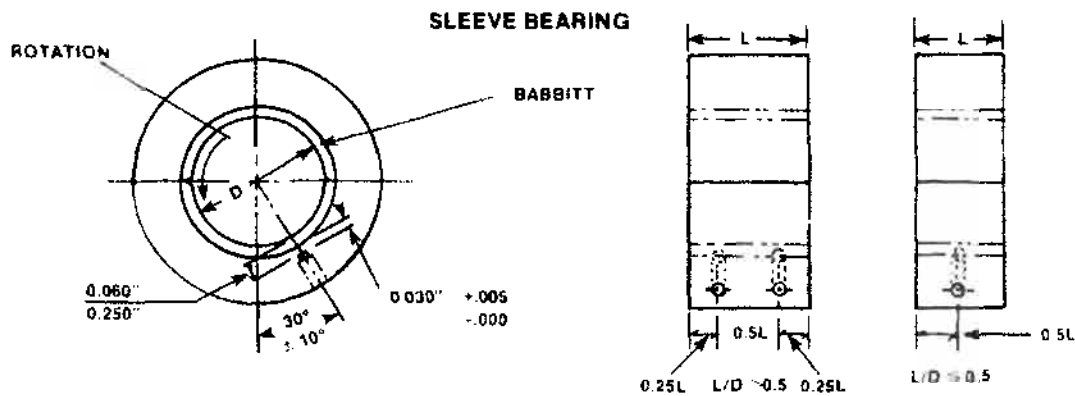


Figure 8-27. Typical radial bearing temperature sensor installation.

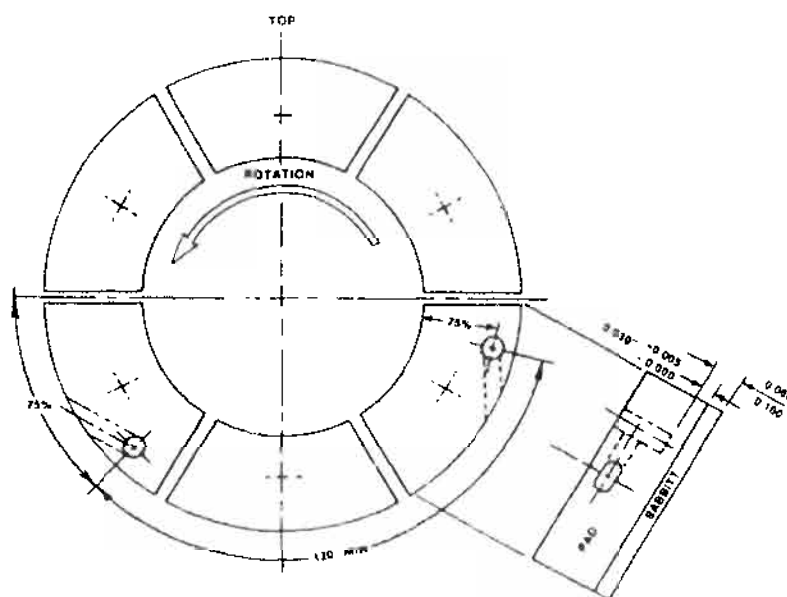


Figure 8-28. Typical thrust bearing temperature sensor installation.

The two recognized standard sensors are the ISA type J thermocouple (iron-constantan) and the 100 ohm at 0°C platinum 3-wire RTD. Additional attributes such as TFE insulation and stainless overbraid are specified. The sensors are installed in a drilled hole at the location shown, with the objective being to place the sensor approximately .030 inches to the rear of the base of the babbitt. Surprisingly, steel conducts at approximately the same coefficient as the babbitt, so there is no significant temperature drop at the metal interface. The sensor is potted in place, with some of the overbraid included, to provide strain relief. An alternate to potting is to use a spring and clip arrangement, which has the advantage of an easy sensor replacement. Figure 8-29 shows a radial bearing with a temperature sensor installed. Figure 8-30 depicts an instrumented thrust bearing.

Flow

Flow is another one of the basic compressor parameters. It can be deferred back to pressure, since most of the flow involving compressors is measured flow by a primary device such as an orifice and a differential pressure sensor as discussed in the section on pressure. For plant use,

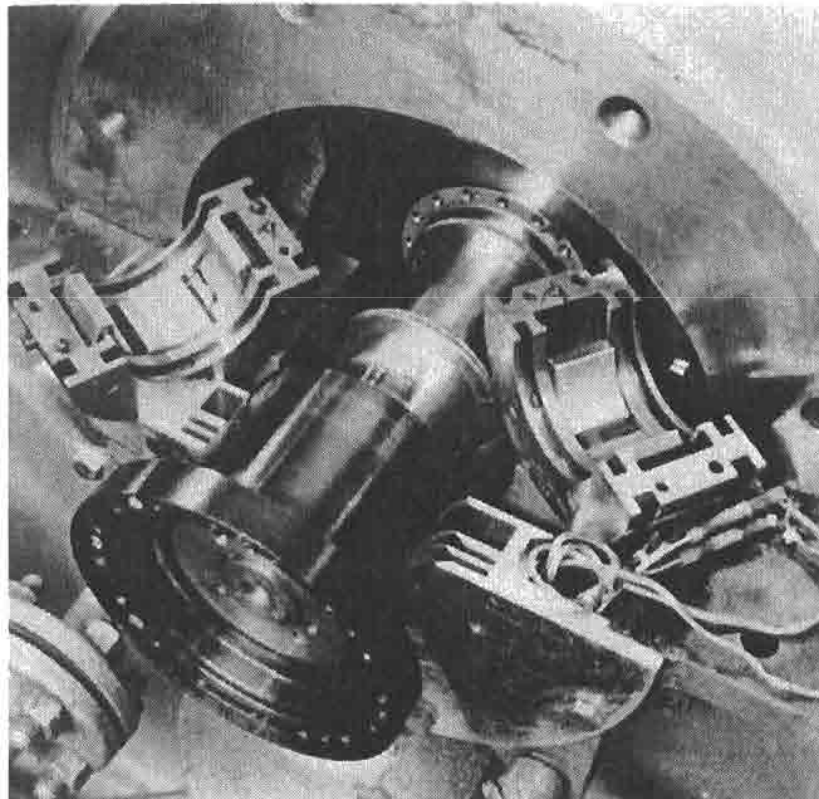


Figure 8-29. Radial bearing with a temperature sensor installed. (*Courtesy of A-C Compressor Corporation*)

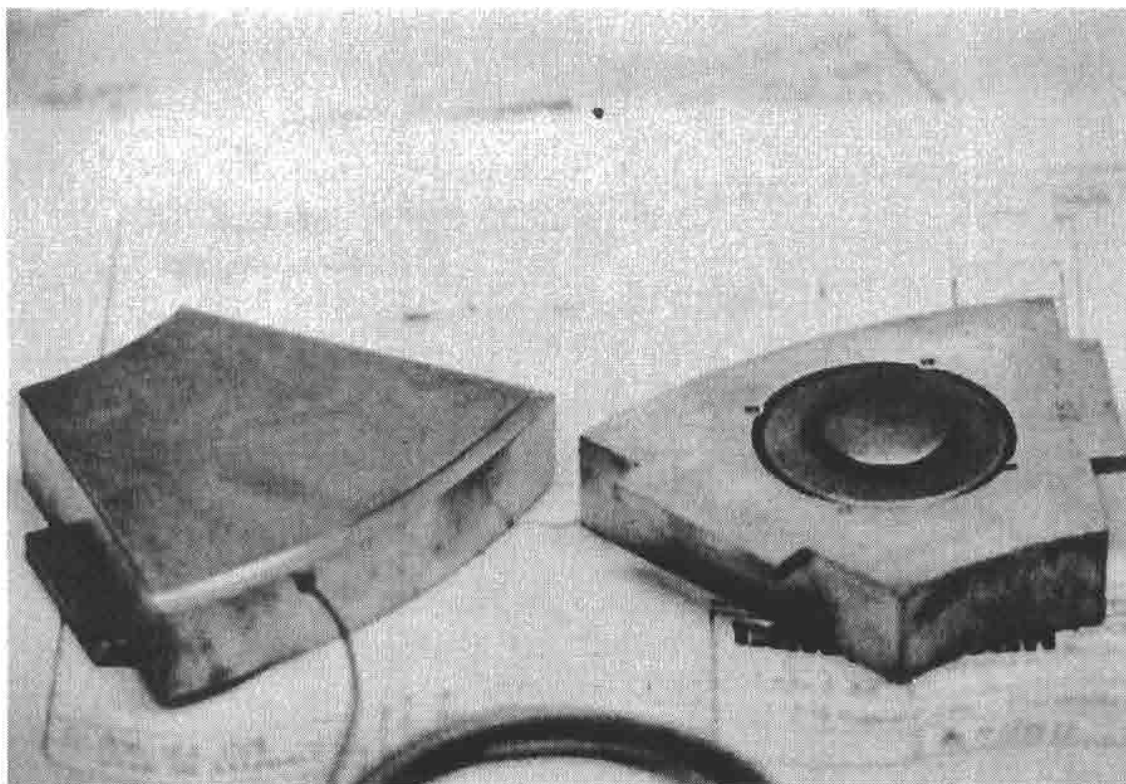


Figure 8-30. Thrust bearing pad with a temperature sensor installed. (*Courtesy of Turbocare, A Division of Demag Delaval Turbomachinery Corp., Houston facility*)

where relative flow is generally of more interest, the flow is usually recorded on a flow chart. The chart is calibrated in square root units. Unfortunately, unless all the constants can be located, the relative flow is not of much value. If a differential transducer is connected across a primary element with a known bore and pipe size, and if the element was calibrated, the calibration chart will provide the raw data that will permit the generation of meaningful flow information.

Torque

Torque meter instrumented couplings are available using the *strain gauge* for the measuring element. A standard coupling is sent to the torque meter supplier for application of the electronics and the strain gauges. The torsional strain, as measured by the gauges, is telemetered to a stationary cylindrical receiver placed concentric to the coupling. The received data are processed and converted to a signal proportional to the transmitted torque. It may be displayed locally and/or transmitted to a

remote location such as a control room. Figure 8-31 shows a coupling instrumented for torque measurement.

Speed

With the introduction of the new instruments, speed is basically taken for granted. It is a very important parameter for reciprocating compressors, however, because speed is one of the factors in generating displaced volume. For the axial and the centrifugal compressor, speed offers a multiple influence. In the fan laws stated in Chapter 5, speed was the common parameter in both capacity and head. In fact, since head is proportional to speed squared, it becomes quite important that the speed be accurate.

The electronic counter circuit contributed to the development of modern tachometers. By using a toothed wheel and a magnetic pickup with the counter, a direct reading digital speed output may be derived. While not too common, the signal can be put through a digital-to-analog converter and an analog meter reading made available. The digital readout is useful for performance testing because it requires no interpolation. The analog meter reading is good for startups or gross adjustments where the rate of response is part of the information.

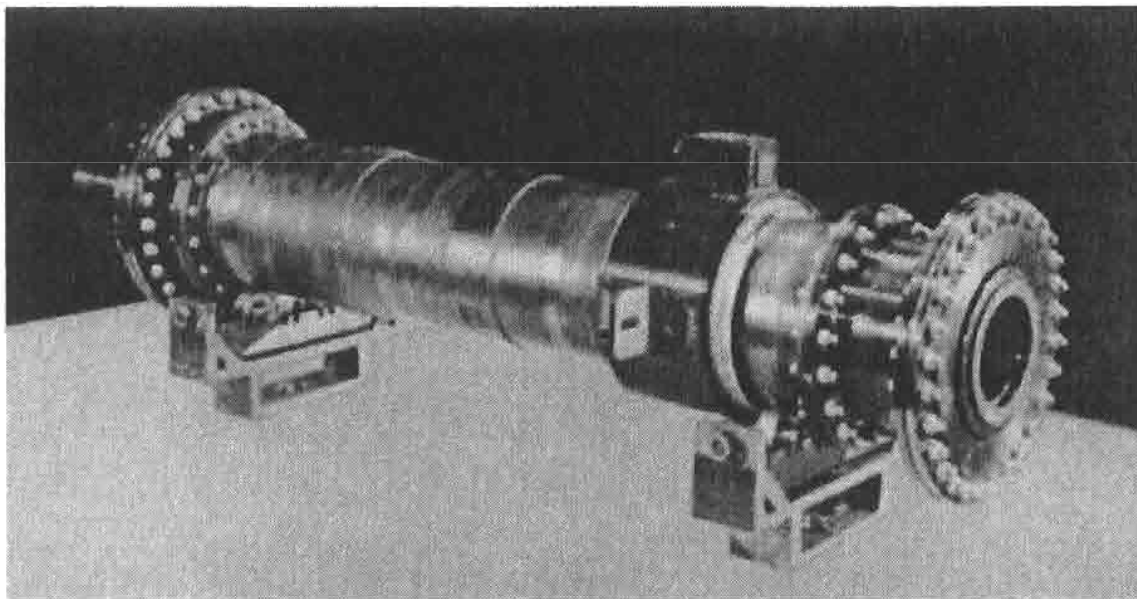


Figure 8-31. Multiple diaphragm flexible element coupling with a strain gauge type torque meter. (Courtesy of Zurn Industries, Inc., Mechanical Drives Division)

Rod Drop Monitor

Rod drop monitoring is a monitoring system used on horizontal reciprocating compressors. Most horizontal reciprocating compressors use wear bands also known as rider rings to support the piston in its cylinder. The wear bands are generally made of a plastic material which in time abrades and requires replacement. Should the wear go unchecked, contact with the cylinder wall by the piston will occur with the potential for serious damage to the cylinder and the piston. Also the packing may be damaged which could lead to a gas release.

Compressors should be stopped periodically to check for wear. The inspection may occur every few months. To accomplish this the compressor must be stopped, blocked in, and purged if the gas is hazardous. A valve is removed from the cylinder and a feeler gauge inserted through the valve opening. A measurement is taken between the lower side of the piston and the cylinder wall.

An alternate method of measurement is the use of a rod drop monitor. The monitor consists of a proximity probe located in a vertical position in the packing case. Another probe is used to develop a once-per-revolution timing pulse. The probe gap voltage is read with a remote monitor. The timing pulse is used to gate the probe reading to allow for an instantaneous rod position reading. The advantage of taking the reading at a discreet point in the stroke is that the effect of scratches or coatings on the rod can be minimized. Also, the reading can be timed to occur at or near bottom dead center (BDC). Here the dynamic forces have a minimal effect. Also the location of the center of the piston is known at that instant. With this information, the monitor can make a geometric correction to rod position reading providing a display that indicates the amount of wear band wear. Figure 8-32 shows a diagram of the rod drop sensors.

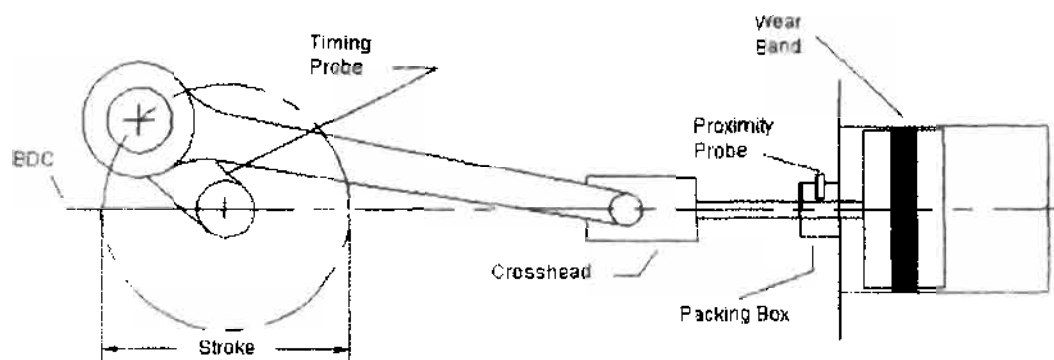


Figure 8-32. Line drawing showing instrumentation locations for a rod drop monitor.

The rod drop monitor is recommended for all non-lubricated compressor applications, because there is no lubricant to act as a buffer to prevent piston-to-cylinder contact on the loss of the wear band. Lubricated compressors handling gases with traces of water or gas components that can degrade the local lubricant are candidates. Hydrogen compressors should be considered for monitors because hydrogen is a difficult gas in itself and may contain trace quantities of water. While sweet gas compressors, as are found in pipeline service, would normally not be considered a problem, the rod drop monitor may be used to signal a loss of lubricant and the compressor can be shutdown before damage can occur.

Molecular Weight

This is another tough one, as there are no direct molecular weight meters. In many plants molecular weight can be obtained indirectly and in most instances does not change all that rapidly. Actually, gas analysis is a more fundamental piece of data. There are "on stream" gas chromatographs but they are quite expensive. With an equation of state, which is geared to handle a broad line of gases by having a large stored base of gas constants, the gas analysis is the input of choice. Getting all this together into a compressor analysis program is somewhat of a chore, but is in the realm of reason.

A control room readout of molecular weight or, more correctly, a control room computer readout of molecular weight coming in at real time is currently available for the newer computer controlled plants.

Vibration

The vibration equipment available is probably best approached by discussing the basic sensors and what they do or don't do. Because monitors now run from A to Z, the reader can obtain vendor literature or dig into the multitude of papers and articles on the monitoring, hardware, software, and, in some cases, philosophy of instrumentation.

Vibration Sensors

Sensors are divided into two general classes:

1. Seismic transducers
 - (a) Accelerometers
 - (b) Velocity Transducers
2. Proximity Transducers

Sensors respond either to amplitude or to displacement of the vibration. Seismic sensors are also frequency sensitive. Figure 8-33 shows a comparison of various methods of sensing vibration. Velocity sensors have a sensitivity directly proportional to frequency and amplitude:

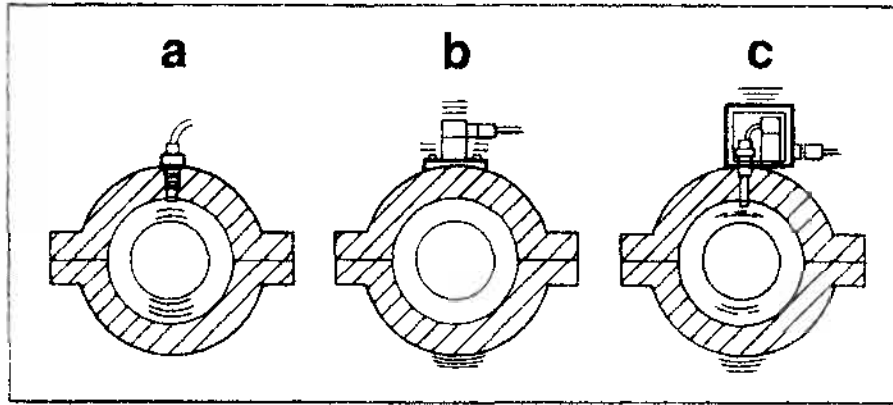


Figure 8-33. Three methods for measuring shaft vibration. (*Courtesy of Bently Nevada*)

whereas, accelerometers show a sensitivity directly proportional to the amplitude as the square of the frequency. It is appropriate to use seismic sensors when the frequency of vibration is high, since high acceleration forces are involved at high frequencies and the sensitivity of seismic sensors increases with the frequency. Conversely, if the frequency is low, the proximity sensor would be favored (see Figure 8-34).

Often, two or more types of sensors may be used in conjunction with each other to give more “visibility” to what is going on. Monitoring of a turbine generator is a good example. Information may be collected about shaft thrust, eccentricity, rpm, bearing wear, gearbox wear, and others for maintenance and protection analysis.

Sensors may be used with a number of monitoring devices. Most sensors give a voltage output signal proportional to the vibration level. The output signal is interpreted as either peak-to-peak voltage, peak voltage, or rms voltage signals. Figure 8-35 illustrates these values on a sine wave.

Seismic Sensors

Accelerometers

Characterized by high frequency response, accelerometers are compact and rugged, ideal for mounting on machinery cases, foundations, piping, etc. Applications to gear trains and rolling element bearings are typical.

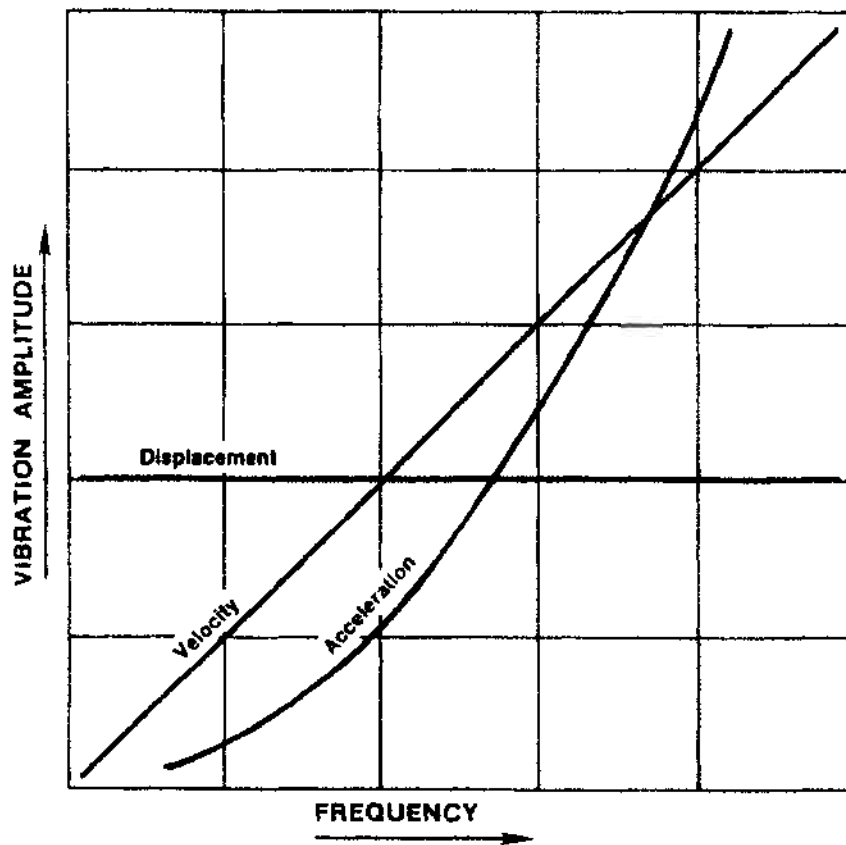


Figure 8-34. The relationship of displacement, velocity, and acceleration to vibration amplitude and frequency.

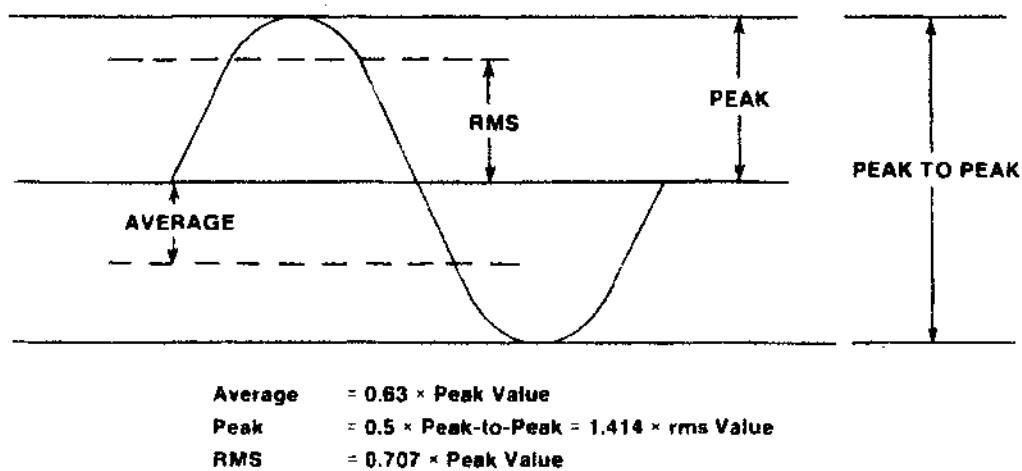


Figure 8-35. The relationship between peak, peak-to-peak, and average amplitudes for a sine wave.

A crystal material is excited by the force imposed on it by an internally mounted mass. A voltage is produced by the crystal proportional to acceleration. This voltage is then amplified by a charge amplifier type signal conditioner from whence the signal can be transmitted long distances (1,000 feet is not uncommon) to the monitor/readout unit. It is calibrated in terms of gravitational units (g), which are proportional to force. Force is one of the most reliable indicators of equipment distress.

The accelerometer output is measured in terms of pico coulombs per gravitational unit. The nominal signal level output of the charge amplifier/signal conditioner is 100 mv/g.

The size of mass within the accelerometer determines the self-resonant frequency of the sensors. The smaller the mass, the higher the frequency. Accelerometers are usually operated in a range below this self-resonant frequency.

Velocity Sensors

Since acceleration is the second derivative of displacement, a piezoelectric accelerometer sensor with an integrator becomes a velocity transducer. This arrangement is gradually superseding the self-generating moving-coil velocity sensor (where a coil of wire moves relative to a magnetic field).

The signal levels from these various velocity sensors are comparable. An advantage of the self-generating moving coil type is that no excitation voltage is required to drive it. However, it could be affected by high magnetic fields generated around heavy electrical equipment, a problem that the accelerometer type is immune to.

Proximity Sensors

Proximity sensors are non-contacting devices that measure the relative distance between the probe tip and the conductive surface that it is observing. In operation, a modulator/demodulator generates a high frequency RF signal that is then applied to a coil on the tip of the probe. The signal is radiated into the area surrounding the probe tip. If there is no conductive material within the range of the RF field, the entire signal is returned to the modulator/demodulator. As a conductive material begins to intersect the RF field, eddy currents are set up in the conductive material, resulting in an energy loss and, consequently, a decreasing signal return to the modulator/demodulator. At some point approaching zero gap, the modulator/demodulator drops to zero.

For most applications, the modulator/demodulator is calibrated to a scale of 200 millivolts per mil. This means that for each one mil of gap change, there will be a corresponding 200 millivolt change. Five mils of gap change will produce a one volt change in output of the modulator/demodulator.

If the observed surface is moving, the modulator/demodulator output varies in direct proportion to the peak-to-peak movement of the observed surface. Having a flat frequency response from DC to 10,000 Hz, the transducer is able to accurately follow motion at frequencies in excess of those typically encountered.

There are several types of information available from each probe and modulator/demodulator. Average gap, or position, data are available from the DC output. Thus, the system can be used for measurement of average position, eccentricity, concentricity, thickness, etc.

The AC component of the modulator/demodulator output is an indication of dynamic motion, or vibration. This signal provides data relating to the peak-to-peak amplitude, frequency, and form of the dynamic action of the observed surface.

Radial Shaft Vibration

Two vibration sensors, mounted at 90° to the axis of a metallic shaft and 90° apart around the shaft, can provide orbital information and display vibration eccentricity (see Figure 8-36).

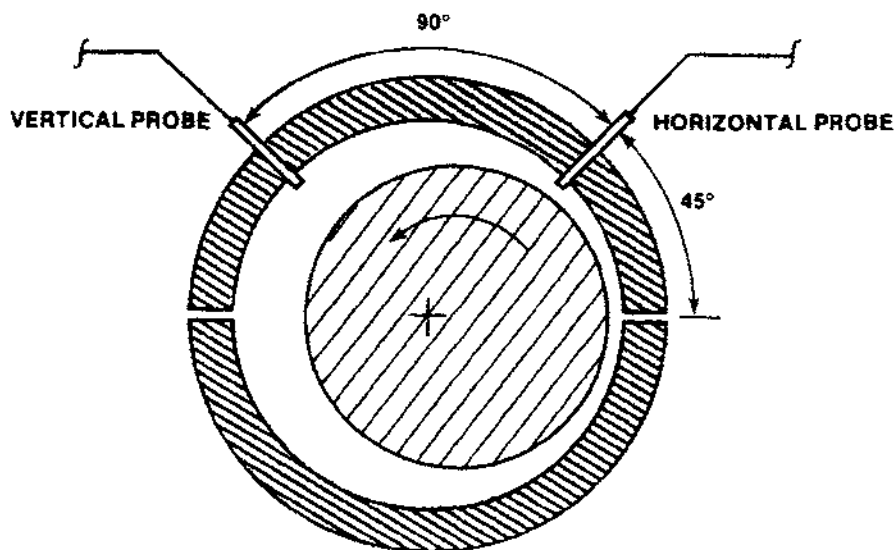


Figure 8-36. Radial shaft vibration probes.

Axial Shaft Motion

By using the positional information available from a proximity probe, another valuable parameter can be measured. The axial position or axial shift detection can be added to the monitored information. Probes are generally installed at the shaft end in pairs to provide redundancy. The probes, preferably, should be sensing the axial position from two different surfaces. A typical arrangement can be seen in Figure 8-37.

Continuous vs. Periodic Monitoring

There are two types of condition monitoring: continuous and periodic. Continuous monitoring, as its name implies, examines measurements taken on a continuous basis. Periodic monitoring is based on measurements taken at regular time intervals.

With the advent of computerized real-time systems, the distinction between continuous and periodic condition monitoring must be modified. Though technically periodic, a scanning system operating fast enough to protect against catastrophic failure is considered continuous. Most people consider one second or faster scan rates as continuous. A scan rate of one second is defined as monitoring each point once each second.

The argument that an analog system dedicated to each measurement parameter provides better protection than a scanning system simply is not true. An analog system has inherent time delays that result in finite

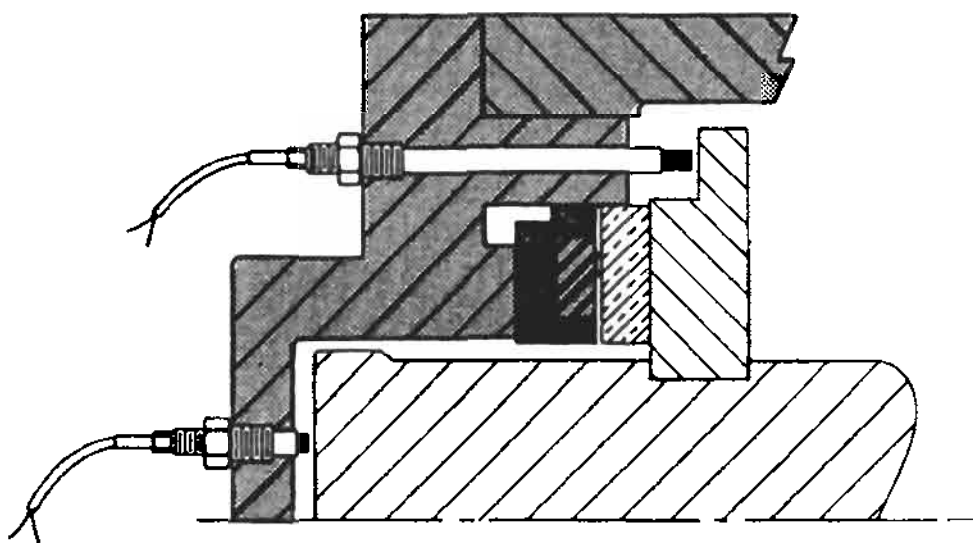


Figure 8-37. Typical axial proximity probe installation. (Courtesy of Turbocare, A Division of Demag Delaval Turbomachinery Corp. Houston facility)

response time to a step increase. Some users have demanded computer systems with exceptionally high scan rates to duplicate the response they thought they had with analog systems. In actuality, a relatively low scan rate of 1–2 seconds is normally sufficient to duplicate the performance of analog condition monitoring systems. Periodic monitoring once meant taking measurements manually at intervals that varied from one week to three or six months. Periodic monitoring can now be redefined as measurements taken at intervals that are too long to provide protection against a sudden failure. In general, measurements taken at intervals longer than 5 seconds can be considered periodic.

What are the advantages and limitations of continuous and periodic monitoring? Continuous monitoring requires a relatively large initial expenditure. But once installed, cost of operation is quite low. Periodic monitoring has a low initial cost, but is manpower intensive and therefore has a relatively high continuing cost.

Continuous Monitoring

Continuous monitoring is necessary on critical machines where problems can develop rapidly and have severe financial consequences. Typical machines in this category are unspared process compressors. Remotely located machinery such as pipeline gas compressors also require continuous monitoring. Also, continuous monitoring may be dictated by safety considerations. Even though the cost of a failure is small, machines should be continuously monitored if a failure will result in hazards to personnel. Figure 8-38 depicts a typical continuous monitoring system.

Periodic Monitoring

Periodic monitoring is typically applied to less critical machinery where advance warning of deteriorating conditions will show a positive return on investment. Another form of periodic monitoring is the detailed analysis of dynamic data from critical machines. Signals from sensors installed for continuous monitoring of overall vibration level, together with additional temporary sensors, are spectrum analyzed and compared against previously accumulated data. This is sometimes referred to as signature analysis. In some cases, changes in vibration signatures will provide earlier warning of deterioration than changes in overall vibration level.

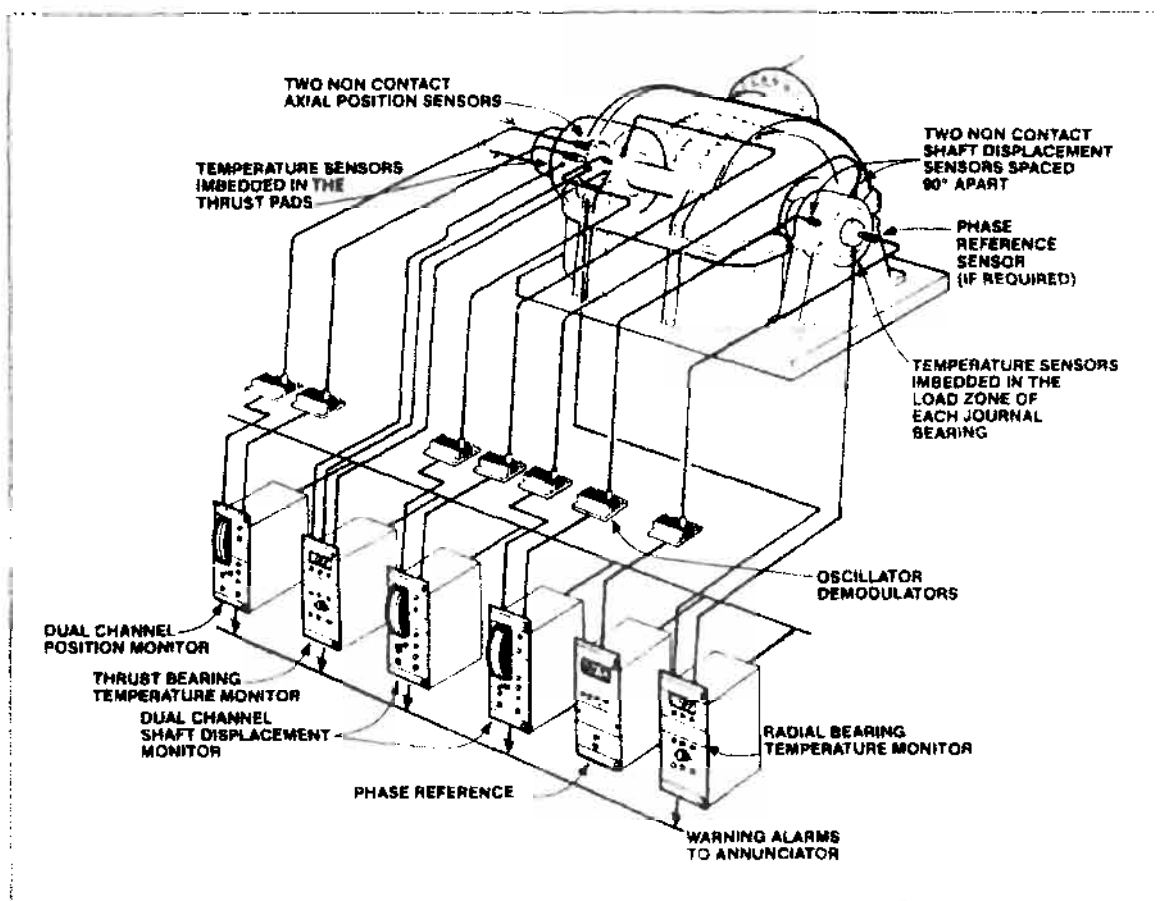


Figure 8-38. Typical continuous monitoring system for a centrifugal compressor.
(Reprinted by permission of Sound and Vibration)

Control

Compressors are normally one part of an overall process train. When a compressor is selected for use in a process, consideration must be given to the required operating range. It will become quickly apparent when a compressor with a fixed capacity is installed in a process with a variable capacity requirement. Modifying capacity of the individual compressors was discussed in the earlier chapters.

A few basics on manual and automatic control will be covered to familiarize the user with some of the considerations necessary to acclimate a compressor to the process environment.

Control systems for process compressors become an absolute necessity whenever deviations from a single operating point occur in the process or system. These systems consist of a mechanism that can sense performance and, by making a calculation, adjust the process controls with satisfactory speed, accuracy, and stability.

Almost all compressor operations will be more satisfactory if a control system is included. Hence, it should always be considered during the original planning phase. Manual operation will result in an operating pattern that differs considerably from a controlled one. Proper evaluation of patterns is required before the need for a control system can be established.

Figure 8-39 illustrates the operating characteristics for a simulated manual and automatic control. In any specific case, the process engineer and the compressor design engineer must cooperate in establishing some of the needs for and capabilities of the compressor.

The design of a control system is a job of considerable magnitude and a general understanding of the engineering approach is necessary to prevent underestimation of the work to be done.

Analysis of the Controlled System

An automatic control system is made up of four blocks: (1) process, (2) transducer, (3) controller, and (4) final element.

The process block produces the required control parameter. The transducer block converts this signal and transmits it to the controller block, which sends the signal to the final element where it is translated into a usable parameter of the proper type to modify the process. Hence, the process block can be described as being derived from the complete physical system.

In the same manner, the transducer block can be defined as a signal selector-converter. The main function is that of interpreting a signal to the controller. The controller block compares this signal to a reference signal called the set point, and issues a corrected signal to the final element. The final element physically changes the process as directed by the controller block. The determination of the final element depends upon the control parameter selected. The elements may be suction throttle valves, discharge bleed valves, governor speed changers, inlet guide vanes, slide valves, variable volume pockets, movable stator vanes, or other compressor elements.

These basic blocks may be arranged to control various parameters as shown in Figure 8-40. Actual applications of the various block arrangements should help in understanding them.

Pressure Control at Variable Speed

In a specific application, a large, mixed hydrocarbon gas system may be set up with a turbine-driven compressor as shown in Figure 8-40(a).

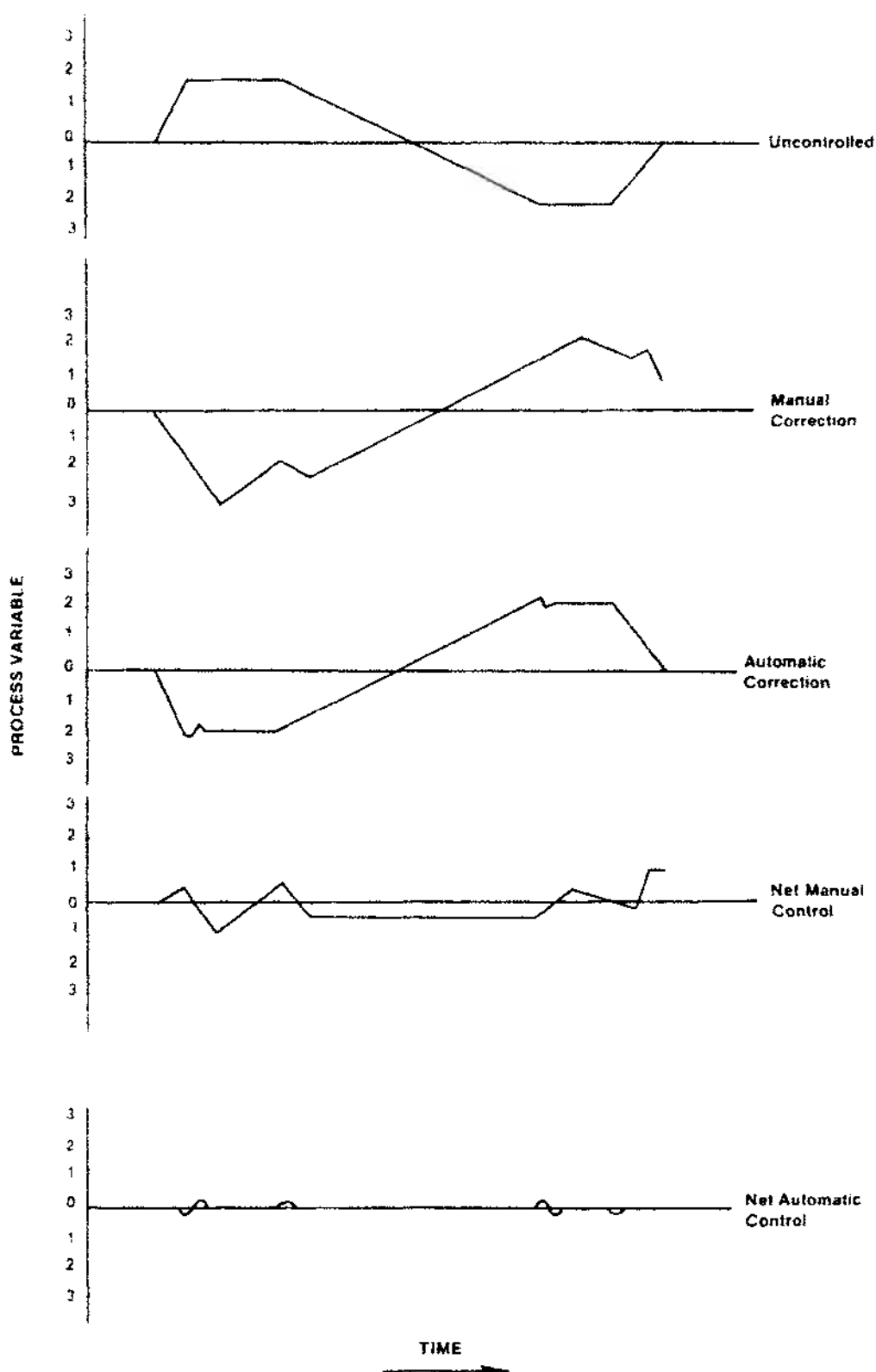


Figure 8-39. Operating characteristics for a simulated manual and automatic control.

The process demands that a constant pressure be held on the header at various loads (flow). Specifically, the control system operates as follows:

1. The pressure transmitter (transducer block) senses the process pressure. It converts this signal to a signal proportional to the process pressure and sends it to the pressure controller.
2. The pressure controller (controller block) amplifies the transmitter signal and sends a modified signal to the final element. Depending on the system requirements, the controller block may include additional correction factors, integral and derivative (reset and rate). This is called a three-mode controller.
3. The final element in this case is speed control. This mechanism varies the turbine-governor speed setting over a predetermined range.

A pressure rise in the reactor would cause an increase in process pressure above the set value, send a signal to the governor, and thus, reduce the speed, lower the flow, and thereby, maintain the desired system pressure.

Volume Control at Variable Speed

A turbine-driven screw compressor might be applied to a catalyst regeneration process. The nature of the process then will require constant volume control to maintain a required output temperature in the regenerator. The arrangement is shown in Figure 8-40(b) and would occur as follows:

1. The flow transmitter (transducer block) senses the flow element differential pressure, converts this signal to a signal proportional to the process flow, and sends it to the flow controller.
2. The flow controller (controller block) amplifies the transmitter signal and sends a modified signal to the final element. The three-mode controller will probably be used.
3. The final element is speed control, which is accomplished by a mechanism that varies the turbine-governor speed setting.

An increase in flow over set point would cause a signal to reach the governor and reduce the speed to maintain the desired system flow.

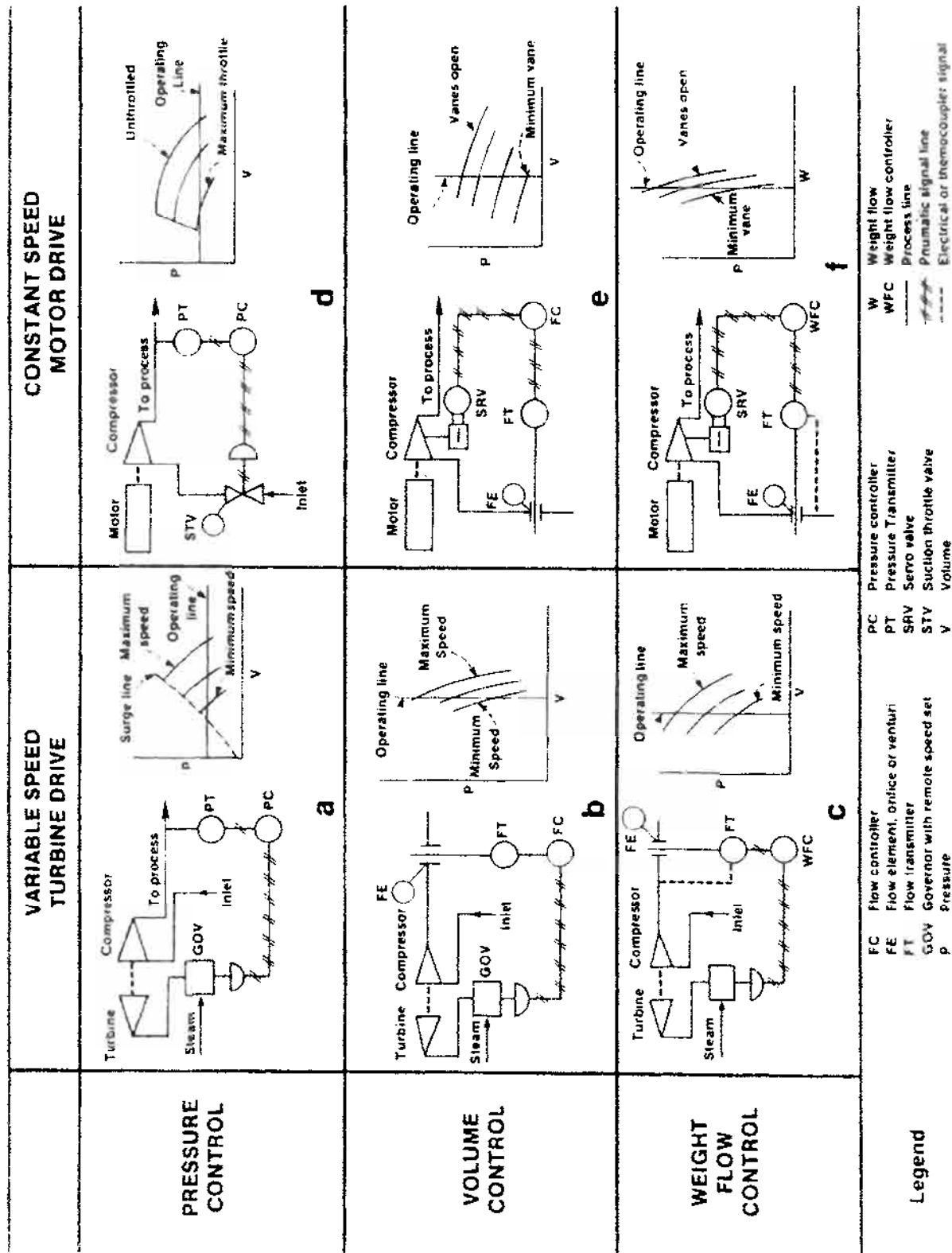


Figure 8-40. Control arrangements for compressors. (Modified from [5])

Weight Flow Control with Variable Stator Vanes

A turbine-driven axial compressor is used to supply air to a blast furnace. The furnace requires a certain constant weight flow, depending upon the size of furnace and the size of the charge. An example of a possible arrangement is shown in Figure 8-40(c) and can be described as follows:

1. The compensated flow transmitter determines the process flow. It converts this quantity to a signal that is proportional to the process flow and sends it to the flow controller. The transmitter could be a pneumatic device using a venturi primary element, with compensation for pressure by a pressure element and compensation for temperature by a thermocouple. The output would be a pneumatic signal that is proportional to weight flow.
2. The flow controller amplifies the transmitter signal and sends a modified signal to the final element. Rest and rate correction factors may also be required.
3. The final element is speed control. This is accomplished with a mechanism that varies the turbine-governor speed setting.

An increase in weight flow over the set point would cause a signal to reach the governor, which would reduce the turbine speed to maintain the desired flow.

Pressure Control at Constant Speed

A plant air system could resemble Figure 8-40(d) where a motor-driven centrifugal compressor is used.

1. The pressure transmitter senses the process pressure and converts it to a signal that is proportional to it.
2. The three-mode pressure controller amplifies the transmitter signal and sends a modified signal to the final element.
3. The final element is a suction throttle valve that reduces the flow of air into the compressor.

A process pressure increase over a set value would cause a signal to reach the suction throttle valve and would close the valve in order to reduce the inlet pressure.

Volume Control at Constant Speed

A motor-driven compressor is used as an oxidation system for a chemical unit. The control scheme for this arrangement is shown in Figure 8-40(e). Description of the control blocks follows.

1. The flow transmitter senses the process flow, converts this to a signal that is proportional, and sends this signal to the flow controller.
2. The flow controller amplifies the transmitter signal and sends a modified signal to the final element.
3. The final element is the compressor guide-vane mechanism. The guide vanes are adjusted by means of a positioning cylinder. This cylinder is operated by a servo valve that receives a signal from the flow controller.

Here, an increase in flow above the set point causes a signal to reach the final element, which will result in the closing of the guide vanes to decrease flow.

Weight Flow Control at Constant Speed

Motor-driven reciprocating compressors are sometimes used in tonnage oxygen plants. To maintain a uniform output, the plant must be supplied with a constant weight flow of air. As ambient conditions change weight flow, a control system as shown in Figure 8-40(f) can be used to keep the plant supplied with the proper quantity of air. The necessary steps in this system are:

1. The compensated flow transmitter senses the process weight flow. It converts this signal to one that is proportional to the process flow and sends it to the weight flow controller.
2. The weight flow controller amplifies the transmitter signal and sends a modified signal to the final element.
3. The final element is the compressor guide-vane mechanism. The variable clearance points are adjusted by means of a positioning cylinder that is operated by a servo valve in response to a signal from the flow controller.

An increase in flow over the set point causes a signal to reach the final element and will result in the closing of the guide vanes to decrease flow.

Anti-Surge Control

Surge is part of the inherent operating characteristics of centrifugal and axial compressors. It is probably one of the most misunderstood of all the characteristics of these compressors. Regard for surge seems to vary from complacency to terror based sometimes on fact and experience and other times on rumor. While surge was described in Chapters 5 and 6 for the centrifugal and axial compressor, a quick review to introduce anti-surge control is included here.

As flow is reduced in an axial or centrifugal compressor, there is a minimum limit when the geometric form of the internal blading can no longer move gas forward through the machine in a stable fashion. Unfortunately, the flow cannot remain static, so if it no longer moves forward, the residual volume in the machine moves in the reverse direction. The audible sound, normally associated with surge, comes from the reverse-flowing gas meeting forward-moving gas in the inlet. The noise is similar to thunder being caused by pressure waves of air colliding due to the local heating from lightning.

Is the surge phenomenon harmful? This can be compared to a wasp or bee sting to a human. If the human is healthy and not allergic, there is only temporary discomfort but no permanent damage. If the human is allergic, has other health problems, or is hit by a whole nest of wasps or bees at one time, we have another story. Similarly, a low-stressed, conservatively designed machine with no large amount of process temperature sensitivity can withstand surge occasionally with the only problem being the disturbance in the process from unsteady flow. If, however, the machine has highly stressed parts or exhibits other marginal design parameters, then we have a problem. Likewise, if the gas is sensitive to large temperature rises, where either process gas decomposition or temperature reactions can occur, we have an allergy. Finally, if any machine is surged long enough (wasp nest), problems are likely to occur. To understand the mechanics, the following occur in surge.

1. Temperature—during the back flow, gas at discharge temperature is introduced to the inlet. If more than one cycle occurs, the same gas is reheated and returned, getting hotter with each cycle.
2. Load—during the back flow the shaft torque is reduced, then restored with feed forward giving a torsional pulse with each cycle.

3. **Component stress**—during the back flow and forward reversals, all the components involved with the gas propulsion are loaded and unloaded, placing blading in a cyclic loading mode.

By evaluating the three physical parameters above as they apply to any compressor, one may anticipate the possibilities for problems.

Whenever normal operating limits or startup require the compressor flow to be less than the minimum flow for surge, an automatic anti-surge control should be considered. Process variables that affect the surge flow limit are flow, differential pressure, inlet temperature, molecular weight, other gas properties, and speed. To evaluate the possibility of surge, the variation of the above limits must be considered together with a compressor performance curve, normally polytropic head plotted against inlet flow in actual operating conditions. This curve can also be plotted in terms of pressure and flow for different inlet temperatures and flows if the variations are not too great. The head capacity curve, while not as straight forward, is somewhat more useful if many variations must be considered. Figure 8-41 is a centrifugal compressor curve. Included is the unstable region normally not shown.

To control or prevent surge, gas must be bypassed around the compressor in order to increase inlet flow. On a gas turbine or in a multistage compressor, the bypass may have to come from more than one point in the compressor. In air service with atmospheric inlets, bypass means dumping back to atmosphere. In gas application or closed systems, bypassing must be done by returning discharge gas to the inlet with provision made to remove the heat of compression either with a shell and tube exchanger or by direct contact such as flashing liquid to vapor.

There seems to be one problem with anti-surge control, which is its name. Both surge control and anti-surge are used. Anti seems to be what the control is supposed to do, so it will be used here.

The best anti-surge control is the simplest and most basic that will do the job. The most obvious parameter is minimum-flow measurement, or if there is a relatively steep pressure-flow characteristic, the differential pressure may be used. The latter parameter allows for a much faster response system, as flow measurement response is generally slow; however, the speed of response need only be fast enough to accept expected transients. One major problem with the conventional methods of measurement and control is the need to move the set point for initiation of the control signal away from the exact surge point to allow some safety factor for control response time and other parameters not directly included

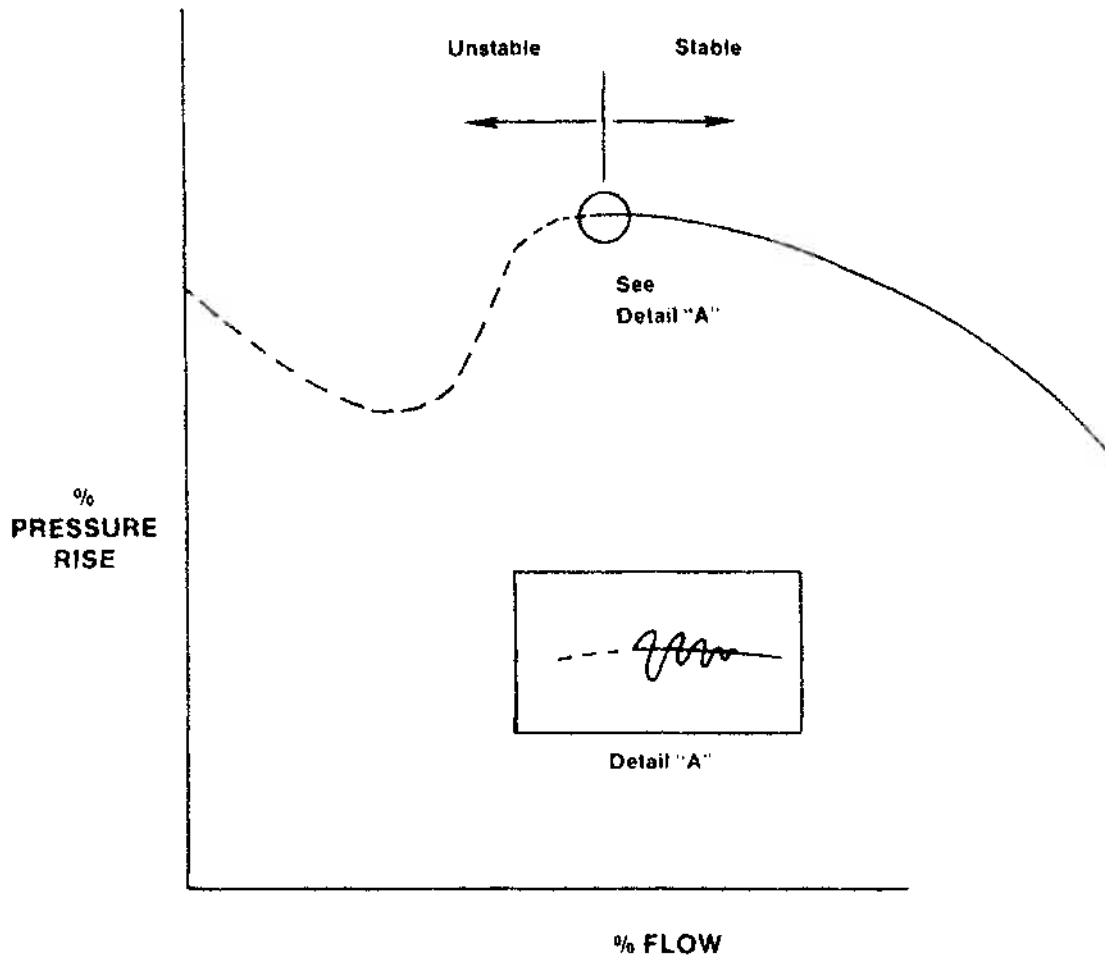


Figure 8-41. Unstable flow region of a centrifugal compressor curve.

in the surge control. A proper speed of response and other more exact variable compensations are ways of narrowing this margin, which represents wasted power and wasted capacity.

Variables such as speed, temperature, and molecular weight can be included as part of the compensation. These may be linearized and put into the system as analog weighting functions or, with computer control, be put into the equation in a more exact fashion. Inlet temperature and speed present little problem as signals proportional to these parameters are commonly measured. A well-designed, properly compensated anti-surge control with a good response can probably function to a 5% flow margin on a conventional centrifugal compressor. On an axial compressor, differential pressure is normally used and a limit can only be defined for a given application. Needless to say, close control can be held. The computer controller also adds intelligence to permit the system to selectively narrow the margin based on plant conditions. With the high cost of power there has been a desire to operate without surge margin or to the

level of incipient surge. Incipient surge is a narrow margin between full stability but before full complete flow reversal. This is represented at detail A of Figure 8-41. Various methods have been devised, such as measuring stage differential pressure and looking at the unsteady component of the signal. The unsteady component does increase with the onset of surge; however, on multi-impeller compressors, either all impellers must be instrumented or a guess must be made, based on probability, as to which impeller is the one most likely to go into surge first. Other methods have been proposed that require internal instrumentation which, however, have the same limits just presented, as well as the problem of simply existing in a hostile environment.

References

1. API Standard 614, *Lubrication, Shaft-Sealing, and Control Oil Systems for Special-Purpose Applications*, Third Edition, Washington, D.C.: American Petroleum Institute, 1992.
2. API Standard 670, *Vibration, Axial Position and Bearing Temperature Monitoring System*, Third Edition, Washington, D.C.: American Petroleum Institute, 1993.
3. Gilstrap, Mark, "Transducer Selection for Vibration Monitoring of Rotating Machinery," *Sound and Vibration*, Vol. 18, No. 2, February 1984, pp. 22–24.
4. Mitchell, John S., "How to Develop a Machinery Monitoring Program," *Sound and Vibration*, Vol. 18, No. 2, February 1984, pp. 14–20.
5. Brown, Royce N., "Control System for Centrifugal Gas Compressors," *Chemical Engineering*, February 17, 1964, pp. 135–138.
6. API Standard 671, *Special-Purpose Couplings for Refinery Services*, Second Edition, Washington, D.C.: American Petroleum Institute, 1990. Reaffirmed 1993.
7. Boylan, William, *Marine Application of Dental Couplings*, Paper 26—1966, Society of Naval Architects & Marine Engineers, May 1966.
8. Bloch, Heinz P., *Less Costly Turboequipment Uprates Through Optimized Coupling Selection*, Proceedings of the 4th Annual Turbomachinery Symposium, Texas A&M University, College Station, TX, 1975, pp. 149–152.
9. API Standard 613, *Special-Purpose Gear Units for Refinery Services*, Fourth Edition, Washington, D.C.: American Petroleum Institute, 1995.

10. Brown, Royce N., *An Experimental Investigation of a Pneumatic Closed Loop Anti-Surge Control for Centrifugal and Axial Flow Compressors*, Master's Thesis, University of Wisconsin, Madison, Wisconsin, 1966.
11. Bloch, Heinz P., "Use Keyless Couplings for Large Compressor Shafts." *Hydrocarbon Processing*, April 1976, pp. 181-186.
12. Feltman, P. L., Southcott, J. F., and Sweeney, J. M., *Dry Gas Seal Retrofit*, Proceedings of the 24th Turbomachinery Symposium, Texas A&M University, College Station, TX, 1995, pp. 221-229.
13. Schultheis, S. M. "Rider Band Wear Measurement in Reciprocating Compressors," *Orbit*, Vol. 16 No. 4, Bently, Nevada, December 1995, pp. 12-14.
14. API Standard 677, *General-Purpose Gear Units for Petroleum, Chemical, and Gas Service Industries*, Second Edition, Washington, D.C.: American Petroleum Institute, 1997.