FLUID MOVERS
Pumps, Compressors, Fans and Blowers

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FOREWORD

This book contains a wealth of practical information on the most common fluid movers—pumps, compressors, fans and blowers—for engineers who work in process operating plants.

For the project engineer—whose combination of managerial and technical skills ensures the viability of "grass roots" plants and the effective major modification of existing ones—there is comprehensive guidance on the specification and selection of fluid-moving equipment, as well as coverage of its installation. Not only is this guidance anchored in thermodynamic and hydraulic fundamentals, it is also channeled by step-by-step procedures that—together with information on the capabilities and limitations of particular fluid movers—will direct the project engineer to the most efficient and economical fluid-moving equipment and circulation system.

For the plant process engineer—the invaluable caretaker whose skill and ingenuity bring about ever-increasing efficiency in plant operation—there are practical insights into how pumps and compressors can be made to function more effectively, and ideas on how this equipment can be modified for changed process demands, such as (to cite only one example) how a compressor can be operated or altered so as to achieve higher, or lower, capacity than that for which it was originally designed. With the understanding that can be gleaned from this book of the intricacies of the wide range of fluid-moving equipment and of the systems in which it operates, the process engineer can more ably solve plant problems that involve the movement of gases, liquids and slurries.

For the operations engineer or supervisor—upon whom depends the continuous, profitable operation of the equipment at his disposal—there are detailed discussions and descriptions of fluid-moving equipment that will enable him to fully comprehend its functioning. There are also insights into operating characteristics, limitations and most-suitable control schemes—all of which will help him achieve the troublefree operation that is the measure of his success. Still other information will enable the operations engineer to quickly pinpoint problems involving fluid movers, to determine the causes of malfunctionings and to arrive at the most practical solutions. For example, a checklist guide to centrifugal pump problems includes eighty-nine probable causes, and is accompanied by discussions on how the malfunctionings might be remedied. Presented also is information on how to cope with troublesome fluids, such as those that are highly viscous, corrosive or abrasive, and how to achieve the closely controlled flows vital to high-yielding operations.

For the maintenance engineer—among whose expertise and efficiency depends the online availability of fluid-moving equipment—there is a vast amount of diagrams and discussion that will provide the familiarity with the external and internal construction of the large variety of such equipment that will promote the expeditious repair of it. There is also practical guidance on how to align shafts and care for bearings, seals, couplings, speed gears, and lubrication and control systems. Such knowledge can be critical. For example, in the modern high-speed compressor, good lubrication is vital to reducing friction, transferring heat, sealing against air leakage and flushing away contaminants. Among the many trouble spots addressed in the book are those that arise from mechanical seals and packing, which particularly plague the maintenance engineer, and a variety of means are presented by which he can avoid or cure such problems, and so contribute immeasurably to the important goal of holding down maintenance costs.

For the plant design engineer—from whose technical expertise must come long-range solutions to persistent process problems—there is detailed information on the systems in which fluid movers perform that will both widen and deepen his understanding of them and so improve his approach to redesigning them. And there is information that is directly applicable, such as how to achieve better equipment layouts and piping arrangements, as well as how to alter the fluid mover itself so as to suit changed process conditions. On this last point, for example, ideas are given on the best mechanical means of revising the output from a pump or a compressor, considering the restrictions imposed by practical reality. Also, because the saving of energy is of paramount importance, particularly valuable to the plant design engineer is an examination of what an inefficient pumping system can cost in terms of energy, and the actions that can be taken to reduce these costs. For many such problems, the book provides guidance to the optimum solutions.

For the safety engineer—upon whom rests the responsibility for guiding equipment design, process operations and maintenance procedures so as to ensure the security of life and property—information vital to the safe operation and maintenance of fluid movers is interspersed through the book. For example, the matter of access for safety is dealt with in the discussion of the optimum layout of pumps and compressors. When, however, the operation and maintenance of a fluid mover bears critically and immediately upon safety, as in the case of oxygen compressors, then safety is focused upon directly.

All the foregoing, of course, only provides the briefest summary of the information on fluid movers contained in the book that can be of value to the process-plant engineer. Understandably, the fluid movers most emphasized are centrifugal, because this type prevails in industry. In the chemical process industries, it is clearly the mainstay of the fluid movers. Its popularity largely stems from its relatively simple design and construction, which causes it to be less demanding in terms of maintenance, and enables it to operate continuously for long periods.

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Section I
Introduction

Compressors and Pumps: the Principal Fluid Movers

Compressors and Pumps: the principal fluid movers

Here's a guide to the structure and operating characteristics of centrifugal and positive-displacement compressors and pumps. It provides information necessary to ensure proper selection and long, trouble-free operation.

ROBERT W. ABRAHAM, The Badger Co.

In the chemical process industries, the trend is to build larger and larger plants with larger, more reliable, single-component equipment.

Reliability of rotating equipment for such plants must always be defined in terms of the expected life of a plant and the associated payout time required to earn profits for the company. Many chemical plants may have a life expectancy of five years or less, with the process being outdated at that time, whereas refinery or petrochemical plants may have a payout time of 10 to 15 years or more.

Several, seemingly unrelated, questions are of prime importance when evaluating, selecting and installing rotating equipment. Is the plant to have a continuous or batch process? What premium is placed on operating cost versus capital cost? Are adequate maintenance personnel available or is it the intent to minimize labor and provide more-automatic control of the process?

With such matters in mind, we can now try to evaluate and make use of equipment existing in the field.

The heart of most processes, and potentially the most troublesome, is the compressor. When selecting a type of compressor, it is most important to have all the process conditions available for review. If there are specialists in the company, they should be informed of all these conditions. Failure to do so has been a source of many problems in the past.

Fig. 1 shows the operating range of the most widely used types of compressors in the CPI. Care should be taken when using Fig. 1 because there are no definite boundaries for considering operating conditions where two or more types of compressors can be used. In this case, alternatives must be looked at. The first step is to define the types, and principles of operation, of compressors.

Centrifugal Compressors

A centrifugal compressor develops pressure by increasing the velocity of gas going through the impeller, and then recovering the velocity in a controlled manner to achieve the desired flow and pressure. Fig. 2 shows a typical impeller and diffuser. The shape of the characteristic curve depends on the angle of the impeller vanes at the outer diameter of the impeller and also on the type of diffuser. For technical details on the theory of operation of different impeller types, see Ref. 1. These compressors are normally installed as a single unit, unless flow is too high or process requirements dictate otherwise.

Most impellers furnished in the CPI are the backward-leaning type. This makes the compressor more suitable for control because of a steeper performance curve. The tip speed of a conventional impeller is usually 800 to 900 ft./sec. This means that an impeller will be able to develop approximately 9,500 ft. of head (the head depends on the gas being compressed). Multistage compressors are needed if duties exceed this value. Heavy gases such as propane, propylene or Freon require a reduction in tip speeds due to lower sonic velocities of these gases when compared with air. For these gases generally, the relative Mach number at the side of the impeller is limited to 0.8.

An excellent summary is given in Ref. 2 that discusses in everyday language the reason for the shape of characteristic curves. When evaluating a centrifugal compressor, close attention should be paid to the percent increase in pressure from the normal operating point to the surge point. The surge point is defined as the location where a further decrease in flow will cause instability in the form of pulsating flow, with possible damage resulting from overheating, failure of bearings due to thrust reversals, or excessive vibration.

Because of the high speeds used in centrifugal compressors, greater care must be taken with rotor balance. The following formula is now accepted by the industry generally for allowable vibration limits on compressor shafts:

$$Z = \sqrt{12{,}000/n}$$

where Z is the maximum allowable vibration limit, peak to peak, mil (1 mil = 0.001 in.) and n is speed, rpm. Z has a maximum limit of 2.0 mil at any speed. Because of high speeds, most users specify that vibration monitors of the noncontacting type be provided to sense excessive shaft vibration.

Depending on the type of process system, various anti-surge controls are required to prevent the compressor from reaching the surge limit. Usually, a safety margin of

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from 5 to 10% should be allowed for automatic controls. Simple resistance circuits may not need any antisurge controls because the surge line could never be reached (see Fig. 3).

When a fixed backpressure is imposed on the compressor, special care must be given to selecting a steep performance curve (i.e., a rise in head of approximately 10 to 15% from rated point to surge point) so that a small variation in pressure will not cause a large change in flow and possible surging. Fig. 4 illustrates this process. When recycling gas in the antisurge loop, the gas must be cooled before it is returned to the inlet of the compressor. Furthermore, if variable speed is desired, a pressure control will regulate the speed of the driver.

When both a fixed backpressure and friction drop are required, an antisurge system will be needed—especially if wide variations in flows and pressures can exist (see Fig. 5). Head rise from rated point to surge should be at least 10% for good stability. The control scheme is the same as that shown in Fig. 4, and will usually be based on measurement of flow through the compressor. Again, the bypass flow must be cooled before it is returned to the compressor.

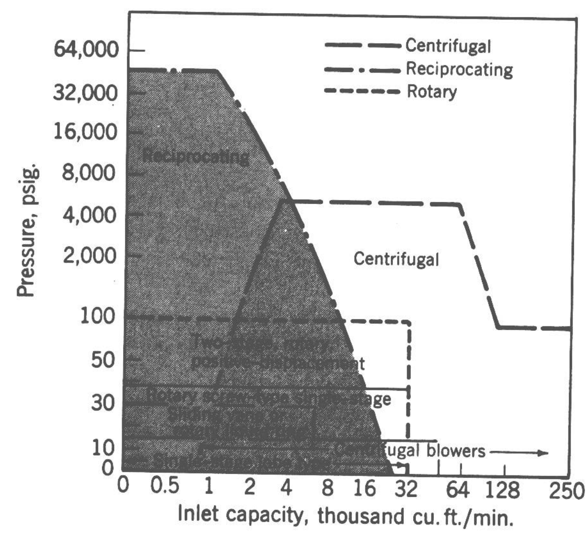
From a process viewpoint, the centrifugal compressor offers the advantage of oilfree gas and no wearing parts in the compression stream. There are several choices of end-seals available. The selection of seals depends on the suction pressure of the compressors since most compressors have the discharge end balanced back to the suction pressure (i.e., both the inlet and discharge ends of the compressor are under suction pressure). The following table gives types of seals and the common range of pressures. For configuration, see Fig. 6.

Type of Seal	Approximate Pressure Psig.
Labyrinth	15
Carbon ring	100
Mechanical contact	500
Oil film	3,000 and higher

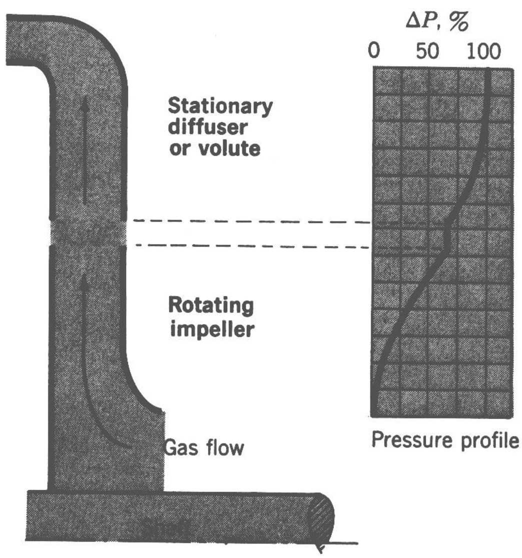
There are several variations of these seals. For example, if the process gas contains a "sour" component such as H_2S, a sweet gas such as nitrogen could be used to buffer the area between the mechanical-contact or oil-film seal and the process gas. (See Fig. 6.) An eductor could be used in conjunction with injection of a sweet gas in order to ensure that the outward leakage is in the direction of eduction.

The advantage of the labyrinth seal is that it is a clearance-type seal with no rubbing parts and is by far the simplest of all seals. The type is also used between stages for multistage compressors. Its disadvantage is that high leakage losses may be encountered which, for valuable gases such as nitrogen or oxygen, cannot be tolerated.

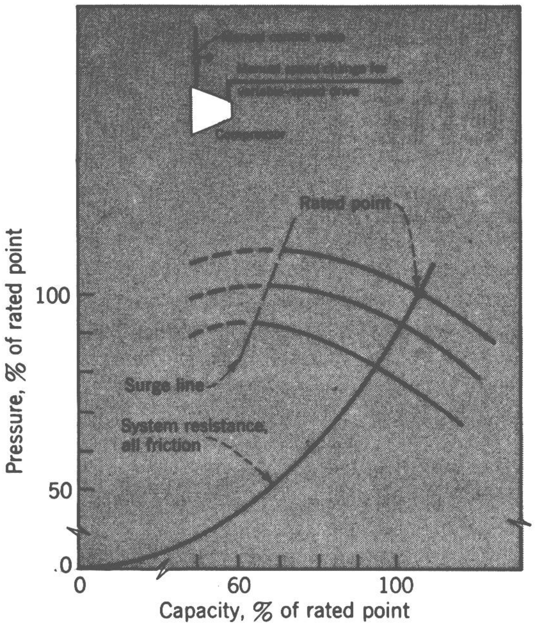
Carbon-ring seals are not commonly used except where the compressed gas is clean or there is a clean buffering medium containing a lubricant. Since these are close-clearance seals, they are subject to wear. They are



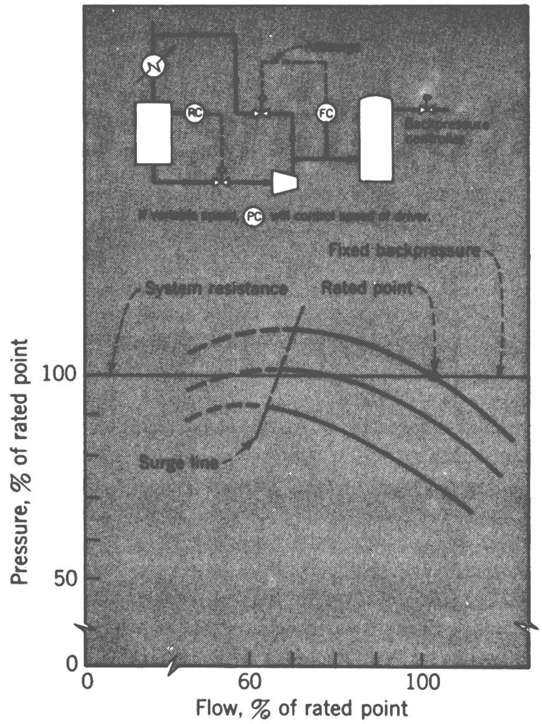
COMPRESSORS have wide ranges for process use—Fig. 1



GAS FLOW through a centrifugal compressor—Fig. 2



RESISTANCE to flow is due only to friction—Fig. 3



FIXED backpressure imposes careful control—Fig. 4

relatively inexpensive compared with oil-film or mechanical-contact seals but still have the advantage of limiting the outward leakage of the compressed gas.

The mechanical-contact seal has an oil film that is maintained between the running and stationary faces. This seal has the advantage of minimizing oil leakage toward the gas side. It is also relatively insensitive to the differential pressure between the compressed-gas suction pressure and the sealing-oil pressure. The disadvantage of the seal is the possible loss of the oil film, which may cause serious damage to the mating faces.

The oil-film seal, like the mechanical-contact seal, relies on an oil film to seal the compressed gas from the atmosphere. Unlike a mechanical-contact seal, however, it is a clearance type and requires a very close differential between suction pressure and sealing pressure in order to minimize inward leakage of oil. When the seal oil is common to the lube system, this could amount to excessive oil losses and become a maintenance problem both in disposing of the contaminated oil and in replenishing the oil for the lubricating system. This type of seal is used for the highest suction pressures commonly found in the CPI.

The disadvantage of both the oil-film and mechanical-contact oil-seal systems is the requirement for sophisticated controls, plus extra pumps and a seal-oil cooler and filter if a separate seal-oil system is required. For further details on seal-oil systems and lube-oil systems, see Ref. 2 and 3.

Casings for centrifugal compressors can be either horizontally or vertically split with relation to the shaft.

From a maintenance viewpoint, it is easier to get at the rotor assembly for a horizontally split casing than for a vertically split one. However, the horizontally split casing is limited in pressure because of the large sealing area of the joint. The American Petroleum Institute's Subcommittee on Mechanical Equipment has recently adopted a guideline that calls for a vertical sealing joint. The proposed guideline for changing to a vertically-split or barrel casing is:

Mol Fraction H_2 %	Maximum Case Working-Pressure, Psig.
100	200
90	222
80	250
70	295

Where vertical-type split casings are used, space must be allowed to withdraw the inner casing and rotor.

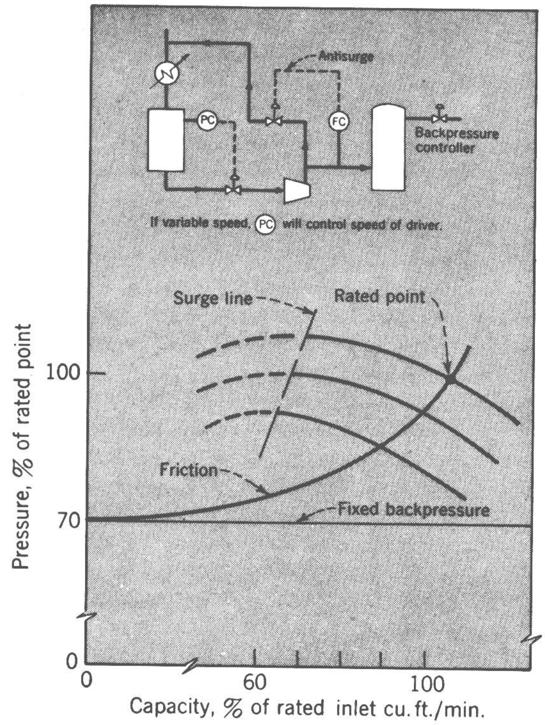
Material selection for casings and rotors depends on the gas being compressed. Recent studies indicate that gases containing H_2S cause stress corrosion in highly stressed areas. In order to overcome this, softer impeller materials are required. This necessitates reduced impeller tip speeds. In some cases, because of this reduction in speed, the next-larger compressor may have to be selected. What this means is that the compressor manufacturer should be made aware of all gas components as well as all operating conditions.

Advantages of using a centrifugal compressor:

1. In the range of 2,000 to 200,000 cu.ft./min. (and depending on pressure ratio), the centrifugal compressor is economical since a single unit can be installed.

2. The centrifugal offers a relatively wide variation in flow with relatively small change in head.

3. Lack of rubbing parts in the compression stream



ANTISURGE control handles fixed backpressure—Fig. 5

enables long runs between maintenance intervals, provided that auxiliary systems such as lube-oil and seal-oil are designed properly.

4. Large throughputs can be obtained with relatively small plot size. This can be an advantage where land is valuable.

5. When enough steam is generated in the process, a centrifugal compressor will be well matched with a direct-connected steam-turbine driver.

6. Smooth, pulsation-free flow is characteristic.

Disadvantages:

1. Centrifugals are sensitive to the molecular weight of the gas being compressed. Unforeseen changes in molecular weight can cause discharge pressures to be very low or very high.

2. Very high tip speeds are required to develop the pressures. With the trend to reduce size and increase flow, much greater care must be taken in the balancing of rotors and materials used for highly stressed components.

3. Relatively small increases in process-system pressure drops can cause very large reductions in compressor throughput.

4. A complicated lube-oil system and sealing system is required.

Positive-Displacement Compressors

Positive-displacement compressors can be categorized as rotary and reciprocating for the more common process applications. Unlike the centrifugal, these are essentially constant-capacity machines having varying discharge pressures. Fig. 7 shows a typical performance curve. For this diagram, suction pressure and temperature, and discharge pressure, are assumed constant. Capacity is changed by speed or suction-valve unloading. Furthermore, there is only a slight variation in flow over a wide pressure change.

Reciprocating compressors operate on an adiabatic principle whereby the gas is drawn into the cylinder through inlet valves, is trapped in the cylinder and compressed, and is then passed through the discharge valves against discharge pressure. These compressors are seldom used as single units unless the process is such that intermittent operation is required. For instance, if catalyst must be regenerated every two or three months, or a backup supply is available from another source, this would allow time to make valve or piston-ring repairs or replacement if required. Reciprocating compressors have contacting parts such as piston wear-rings mating to cylinders, valve springs and plates or disks mating to valve seats, and piston-rod packing mating to piston rods. These parts are all subject to wear due to friction.

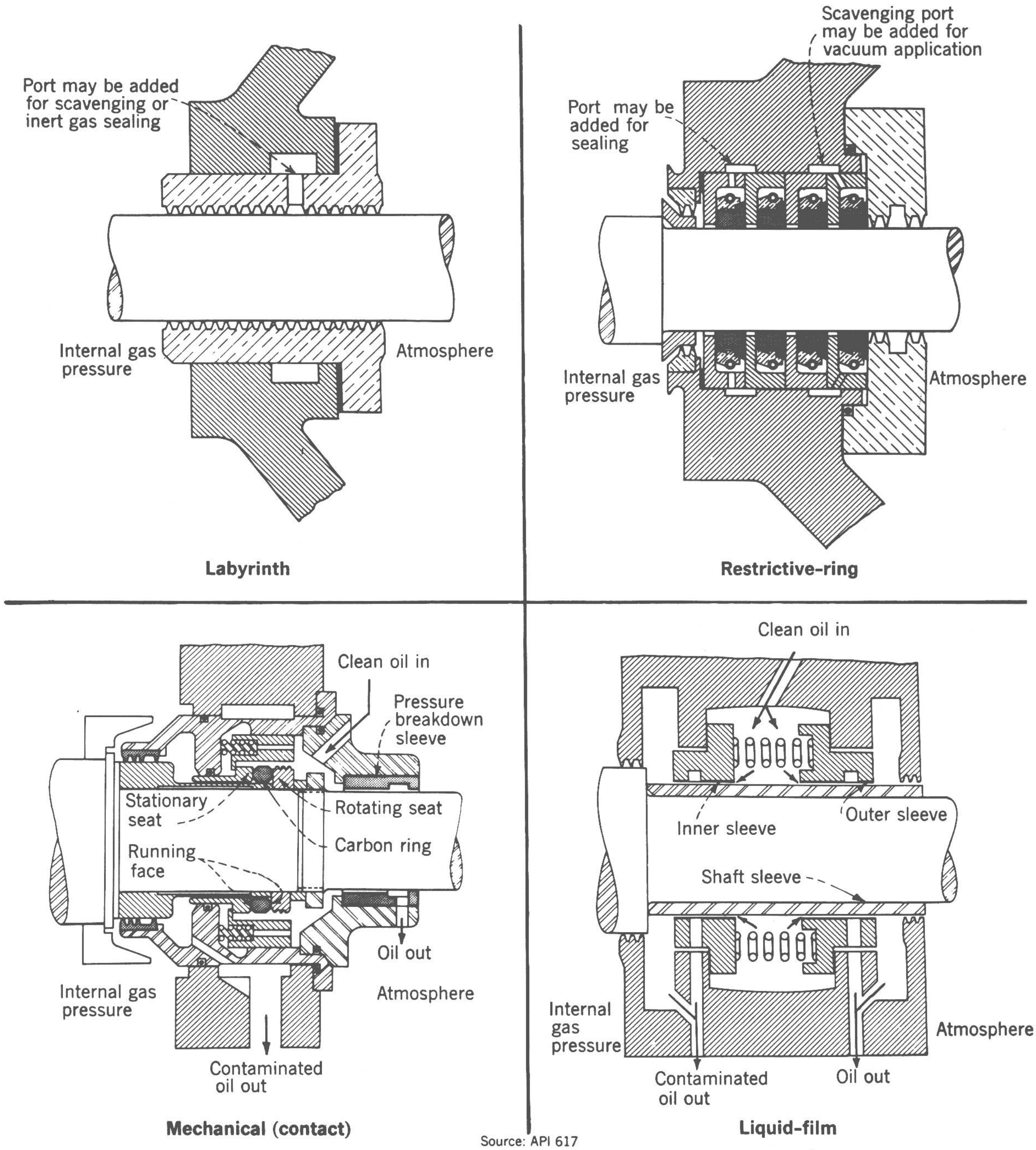
Reciprocating compressors can be furnished in a lubricated or nonlubricated design. If the process permits, it is usually better because of increased life of wearing parts to have a lubricated compressor. However, care must be taken to avoid overlubricating since carbonizing of oil on valves can cause sticking and overheating. Also, oil-saturated discharge lines are potentially a fire hazard, so an adequate separator should be placed downstream to remove the oil. The biggest problems for lubricated-cylinder compressors are dirt and moisture, both serving to destroy the oil film created in the cylinder.

The best way to prevent dirt is to start up with a clean system by using temporary suction strainers. Moisture or condensables carrying over into the compressor suction can be avoided by using an efficient separator located as close to the compressor as possible. Steam tracing or preheating of the inlet gas should be considered downstream of the separator if a wet gas is being compressed.

For nonlubricated machines, dirt is usually the most severe problem. Other problems can arise from the gas itself. For example, a bone-dry gas can cause severe ring wear. The manufacturer should be consulted in cases such as this, since new test data are constantly being developed. For nonlubricated machines, the piston and wear rings are usually made of Teflon-filled, bronze, glass or carbon materials, depending on the gas being compressed. Honing of the cylinder to 12 μ (rms.) finish usually prolongs ring life. (See Ref. 4.) Packing is subject to the same wear as the piston rings.

If the gas being compressed is a sour one, or the lubrication used for the cylinder is not compatible with the oil used in the crankcase, or vice versa, then an extra-long distance piece should be specified. See Fig. 8 for various distance-piece configurations. In cases where the gas is a safety hazard, a double-distance piece should be specified, and the distance-piece next to the cylinder should be purged with an inert gas.

Packing leakage should be vented to a flare system or returned to suction. Lubricated compressors may require separate feedlines for lubricating packing, though smaller-size cylinder diameters may not require them. Teflon, nonlubricated, packing usually requires water-



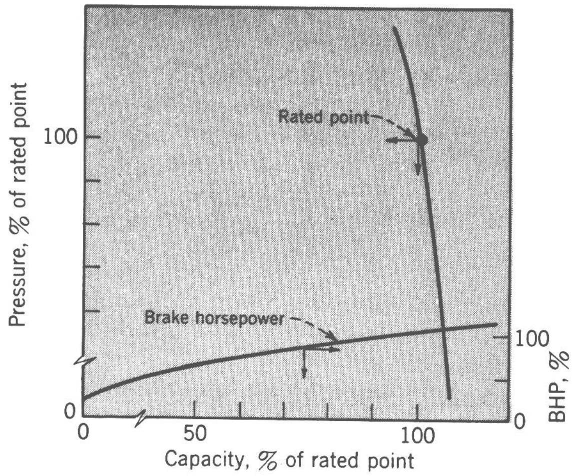
SHAFT end-seals for centrifugal compressors handle a range of pressures for various gases—Fig. 6

cooling, since its thermal conductivity is very low. If low-temperature gases are being handled below 10 F., the manufacturer should calculate the amount of preheat in the gas from internal bypassing. This will mean that a slightly larger cylinder is required to pass the same weight of flow.

Reciprocating compressors are best suited to low-speed, direct-drive motors, especially above 300 hp. These machines are normally constant-speed. Capacity control is accomplished by unloading valves. These valves should be either the valve-plate-depressing type or the plug-unloader type. Unloaders that lift the entire valve off its seat can introduce sealing problems. Unloading can be done automatically or manually. Normal unloading steps are: 0-100%, 0-50-100%, 0-25-50-75-100%. Intermediate steps can be obtained by clearance pockets or bottles. However, these pockets should not be used if polymerization can take place, unless adequate precautions are taken.

Cylinder Cooling

If pressure ratios are low, resulting in discharge temperatures of 190 F., or lower, a static closed system or



POSITIVE displacement compressor curve—Fig. 7

thermosyphon cooling system can be used. Here again, care should be exercised not to run at prolonged periods of no load. Otherwise, a forced, closed-loop system should be used. The inlet temperature of the cooling water should always be maintained at least 10°(F.) above the suction temperature of the incoming gas in order to prevent condensation from forming in the compressor cylinder.

Discharge temperature for nonlubricated machines should be held to a maximum of 350 F. for process compressors. Lubricated-compressor discharge temperatures should be held to 300 F. With synthetic lubricants, these temperatures can be raised to 350 F.; however, care must be taken to see that these lubricants do not cause problems such as paint removal.

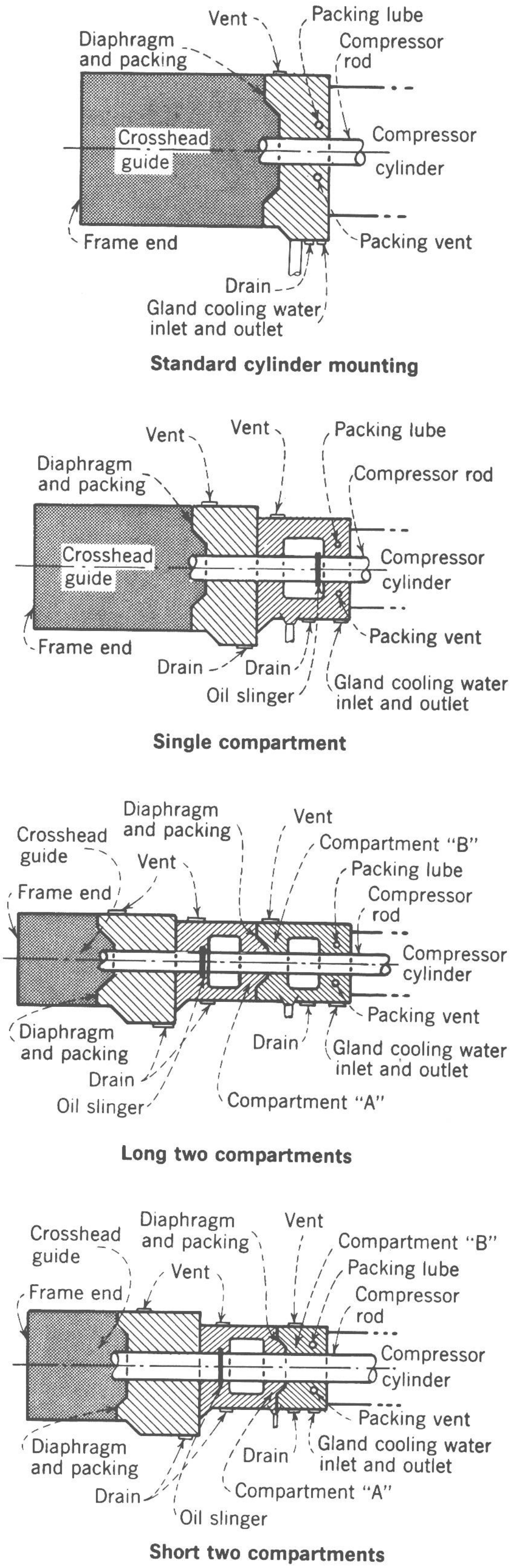
The above limitations may be reduced. For example, oxygen in nonlubricated service should be limited to a discharge temperature of 300 F., while chlorine compressors should be limited to 225 F. to prevent fouling (see Ref. 9).

Compressor Loadings, Speeds and Pulsations

Compressor ratings are based on rod loadings. Longer strokes usually mean higher rod-load ratings and a greater differential-pressure and horsepower capability. Most manufacturers have established proper frame sizes. It is important that the frame and rod loadings are not exceeded even during relief-valve operation.

Average piston speeds for nonlubricated compressors should be approximately 700 ft./min. maximum, while lubricated compressors could run at approximately 850 ft./min. maximum. Rotative speeds for heavy-duty compressors should be held to below 600 rpm., and even lower for high horsepower (over 400 hp.).

Inherent with the reciprocating compressor is pressure-pulsation. This is caused by the reciprocating motion of the piston. In order to avoid it, dampers are fitted to the compressors as closely as possible. The following formula is presented as a guide for maximum limitation of



DISTANCE-PIECES protect area from leakage—Fig. 8

peak-to-peak pulsations in compressor suction and discharge lines:

$$P_1 = \frac{1.5}{\sqrt[3]{0.001p}}$$

where P_1 is maximum allowable pulsation, %; and p is mean effective line pressure, psia. The value for P_1 is that obtained from the formula, or 1%, whichever is greater. Compliance with these limits not only ensures reduced pulsations but also better valve performance from the compressor.

In order to ensure that the whole compressor system is adequate, including piping and associated vessels, an analog study should be done by the manufacturer. In case of complicated systems, Southwest Research Institute[5] has the capability of doing both a mechanical and acoustical vibration check.

An excellent source of information on reciprocal compressors is API 618.[6] Following the API recommendations does bring the cost of the equipment up; however, these specifications represent many years of experience and can mean reduction of costly startup or onstream repairs.

Rotary Positive-Displacement Compressors

Various types of rotary positive-displacement compressors are available. Among these: the lobe-type blower (such as the Rootes design), SRM rotary-screw type, water-ring design, and sliding vane. All of these have the same type of performance curve as a reciprocating compressor; i.e., they are essentially fixed-capacity machines with varying backpressure. Rotary compressors are most readily adaptable to variable-speed drives, such as steam turbines, than are reciprocating compressors. Normally, these compressors are limited to a maximum of about 25,000 cu. ft./min. with present technology. This applies to the rotary-screw and lobe types. The water-ring design offers the advantage of not having the compressed gas in contact with the rotating metal parts. The critical areas are the vapor pressure of the incoming gas vs. the vapor pressure of the liquid forming the water ring, and the temperature rise of the liquid ring. The vapor pressure of the fluid used for sealing should be well below the boiling point, otherwise the sealing ring will evaporate and cause loss of capacity and possibly serious damage from overheating.

Since sliding-vane compressors require lubrication, they are limited to applications where the process can tolerate the lubricant. The oil in the compression chamber does tend to lower discharge temperatures. Oil consumption is fairly high compared to a reciprocating compressor. The sliding-vane compressor is very compact—although it has the same disadvantage as a reciprocating compressor in that rubbing parts are required in the gas stream. Loss of lubricant could cause overheating of the cylinder. High-water-temperature and high-air-temperature switches are necessary for these compressors. Speed reduction is limited to about 60% of normal, since decrease in centrifugal force results in loss of sealing efficiency.

The more common types of rotary, positive-displacement compressors in the CPI are the rotary screw and rotary lobe. These units offer the advantage of oilfree gas, since contact is not made with any parts in the compression area. Their rotary design gives them a throughput capability much greater than the reciprocating compressor, and without pulsation problems.

Timing gears are used to maintain rotor separation and to prevent contact with rotors. For the Rootes-type blower, these gears transmit about 30% of the torque, whereas the rotary-screw type transmits about 10% of the torque. Because these compressors are positive-displacement types, a relief valve should be placed between the compressor and the block valve.

The Rootes lobe-type has a low differential-pressure capability, usually limited to about 15 psig. The rotary-screw compressor can take differentials of much higher values. Both have a fixed slippage rate, causing internal bypassing and a preheat of suction gas. The slower the speed for a given compressor size, the greater the internal bypass that will occur. If the speed is too low, overheating will occur, with possible damage to the rotors. The vendor should stipulate a minimum speed of operation in anticipation of this.

If the discharge temperature of rotary-screw compressors exceeds 350 F., oil-cooled rotors should be specified. It is also well to determine whether the vendor has a minimum backpressure requirement, in order to prevent backlash of timing gears. Another wise precaution is to require the vendor to make a torsional analysis of the compressor and driver.

The first lateral critical speed for these types of machines is usually above the operating speed. This critical speed must be established for both compressor and driver, and should be at least 20% above the highest operating speed and always above the trip speed if a turbine drive is used.

Since noise levels are of increasing interest, it should be noted that these compressors are quite noisy, and are without acoustical provisions such as inlet and discharge silencers, or complete acoustical enclosures. More details can be found in Ref. 8.

Pumps for Processes

In most processes, liquid is transferred from one vessel to another by means of a pump. The type most common to the CPI is the centrifugal pump that operates on the same principle as centrifugal compressors except that the fluid handled is essentially incompressible. The large clearance and lack of rubbing parts, except for bearings and seals, have been responsible for the wide use of the centrifugal pump.

Some guidelines for specifying and evaluating centrifugal pumps are:

1. In the range of 20 to 500 gpm., standard AVS horizontal pumps[10] are used. Vendors making this type of pump have extended the range to about 1,500 gpm., though no official standards have been approved as of this writing. These pumps are normally limited to about 500 F. maximum. Pumps below 20 gpm. will probably require a maximum-flow bypass. Vertical, inline pumps