PART I

Chapter 1

Positive-Displacement Compressors

Although a fair number of types of large compressors can be found in modern process plants, they can, certainly for our purposes, be separated into positive-displacement and dynamic machines. In the positive-displacement category, reciprocating compressors and twin-helical screw machines are of primary interest. Reciprocating compressors are certainly the "older" of the two. It's on that basis that we will provide an overview of reciprocating compressors before moving on to other process compressor types of interest.

1.1 RECIPROCATING COMPRESSORS*

Some refining and gas-processing facilities (e.g., hydrocracking) are best served by piston-type reciprocating compressors. Accordingly, an overview of these machines is an important complement to this text.

Reciprocating compressors (see Figs. 1-1, 1-2, and 1-5), are the oldest and most widely used process compressor type. They are manufactured in a variety of different configurations, including vertically oriented and skid-mounted ("packaged") (see Fig.1-2). Their application ranges are quite wide and include both lubricated and nonlubricated cylinder models, as shown in the coverage chart of Fig. 1-3. So-called secondary compressors often use plain plungers instead of the more typical pistons shown in Fig. 1-4.

An outstanding reference document, API Standard 618, describes many recommended features of reciprocating-process gas compressors. Fig. 1-5 shows a typical compressor cross section and the customary nomenclature. Other important components are also depicted on this cross-sectional view.

*This section was originally contributed by Ralph James, Jr., and is gratefully acknowledged.

Compression is produced by the forced reduction of gas volume by the movement of a piston or plunger in a cylinder. Suction and discharge valves (see Figs. 1-6 and 1-7), are spring-loaded and work automatically from pressure differentials generated between the compressor cylinder and piping by the moving piston [1].

Reciprocating compressors are manufactured as both air-cooled and water-cooled models. Since we are discussing the compressor as related to petrochemical and oil refining processes, we shall concern ourselves only with the cooled, liquid-jacketed type. This type compresses gas on the forward stroke and also on the reverse stroke of the piston.

Reciprocating compressors have a wide range of applications. Crankshaft speeds may range from 125 to 1,000 revolutions per minute. Piston speeds range from 500 to 950 feet per minute, the majority being 700 to 850 feet per minute. The nominal gas velocity is usually in the range of 4,500 to 8,000 feet per minute, and operational discharge pressures may vary from vacuum to 60,000 pounds per square inch (psi).

1.2 MAJOR COMPONENTS DESCRIBED

1.2.1 Crankcase

The crankcase (Fig. 1-8) is a U-shaped cast iron or fabricated steel frame. The top is left open for installation of the crankshaft. To prevent the top from opening and closing as a result of the forces of the throws, it is held together with torqued bolts and spacers or, alternatively, keyed spacers. These are placed directly above the main bearings. The main bearings, spaced between each throw, have removable top covers for ease of assembly and ready removal of the babbitted bearing-liner shells. The keyed spacers are, therefore, preferred since their removal is easiest for access to bearing covers. Main

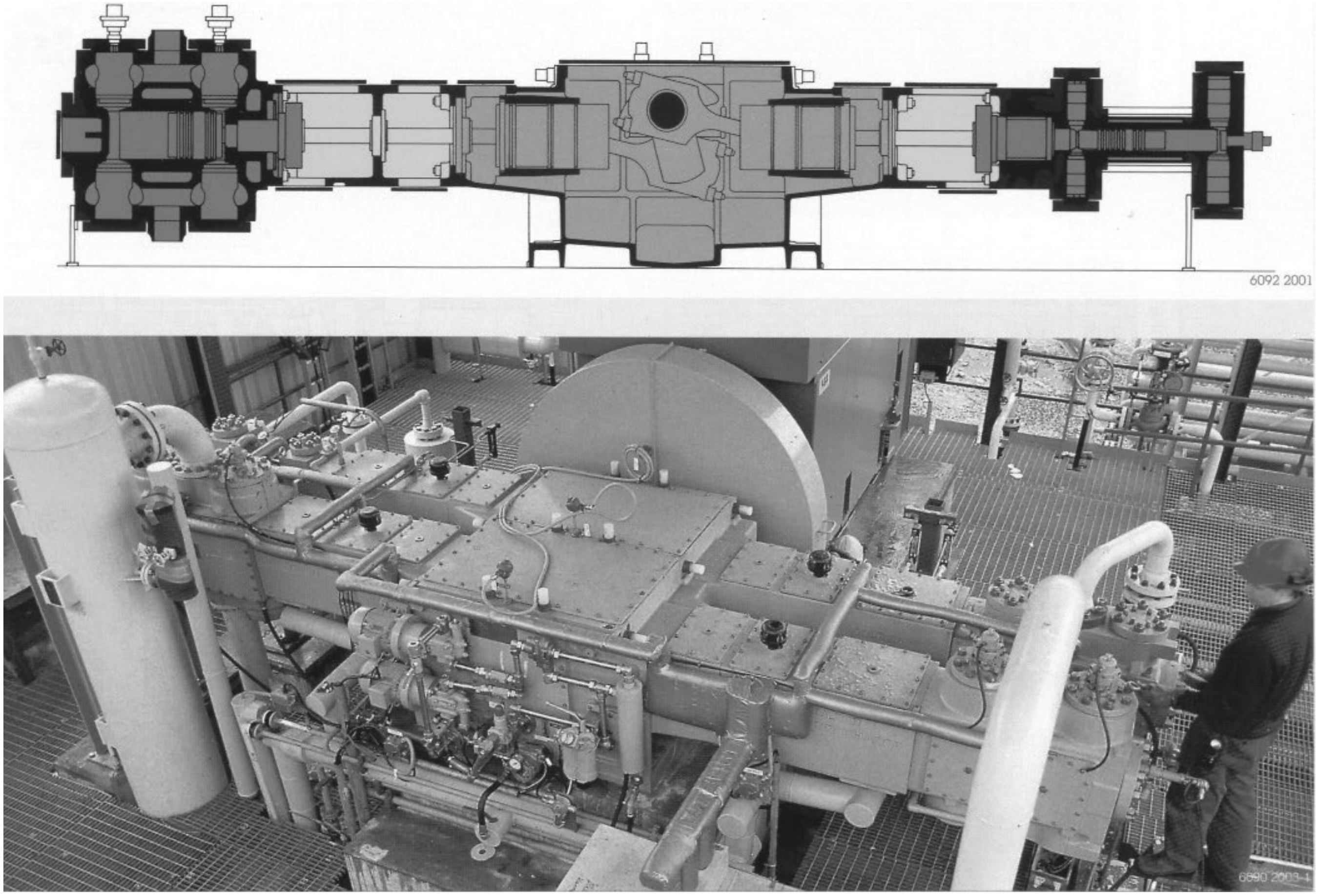


Figure 1-1. Typical horizontal balanced, opposed reciprocating compressor. (Source: Sulzer-Burckhardt, Winterthur, Switzerland.)

bearings, however, in the types of compressors used in petrochemical plants, are so overdesigned for stress that they seldom require removal for rebabbitting.

1.2.2 Crankshaft

The crankshaft (see Fig. 1-8) is the heart of the machine, and usually the most expensive component. Each throw is forged and counterweights are bolted on to balance the reciprocating mass of the crosshead and piston. If the crankcase moves on the foundation, it will cause the throw to open and close through each revolution, resulting in fatigue and breakage. For this reason, the dimension at the open end of the throw must be taken periodically while barring the crankshaft through 360 degrees. This procedure is called "taking crankshaft deflections" and is recommended as an annual check.

1.2.3 Connecting Rod

The connecting rod (Fig. 1-9) is provided with pretorqued bolts fastening cap to body at the crank end. The split is shimmed and shims can be removed as wear progresses. The wrist pin is free-floating and held in place with caps in the crosshead, which allows the connecting rod to find its own center.

1.2.4 Crosshead

The crosshead (Figs. 1-9 and 1-10) runs between two guides with about one mil per inch (0.001″ per inch) of diameter clearance. It is often weighted so that the mass inertia of all reciprocating parts is sufficient to reverse the stress on the wrist pin, even when one end of the piston is under pressure. If this is not done, the wrist pin will wipe all the oil from the side under stress and will bind or "get stuck."

1.2.5 Lubrication

Lubrication of the frame is accomplished either by a pump driven from the crank end or by a separately mounted pump. The pump takes oil from the crankcase sump and pumps it through a cooler and filter, usually 25 micron, then through

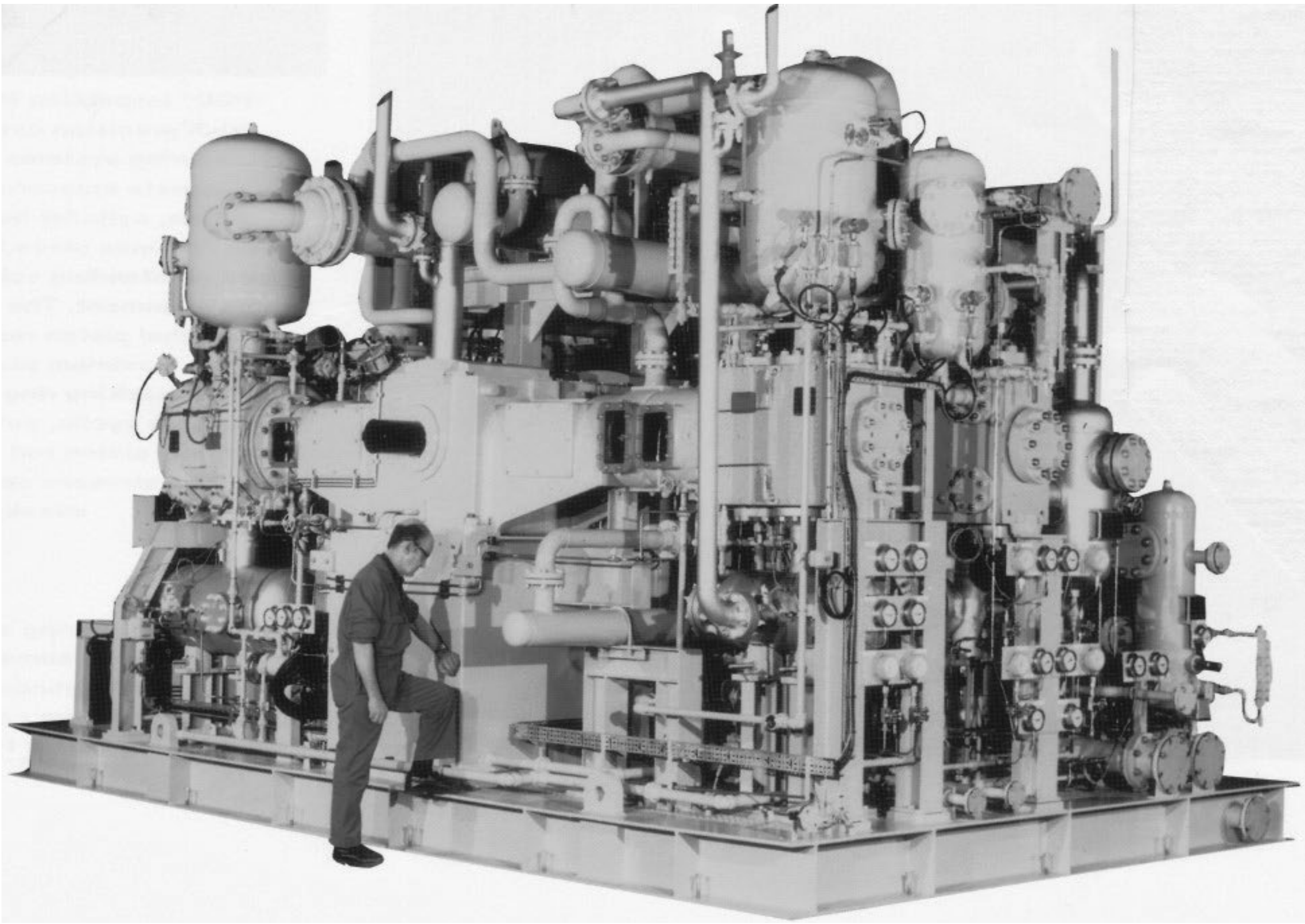


Figure 1-2. Completely packaged five-stage, high-pressure air compressor for a U.K. chemical plant. (Source: Dresser-Rand Company, Painted Post, New York.)

piping to the main bearings. The crankshaft has holes drilled from the main bearing surface through to the connecting-rod bearing face. From here, the oil passes up through a hole in the connecting rod to the wrist pin and from there, through holes to the crosshead sliding faces. Oil scraper rings in the frame end prevent oil leakage out along the piston rod. Because of this tortuous passage of oil, prelubrication is required before start-up. This is accomplished with an auxiliary lube pump.

Crankcase oil heaters are specified for outdoor compressors to keep the oil at required viscosity and to prevent condensation with resultant corrosion. Oil, however, is a poor conductor and local overheating and carbonization have occurred with these heaters. Therefore, when using crankcase heaters while a compressor is not in operation, the auxiliary lube pump should be continuously run.

1.2.6 Cylinder Materials

Up to 1,000 psi, cylinders are normally made of cast iron. Above this working pressure, materials are cast steel or forged steel, at the manufacturer's discretion. Nodular iron castings are sometimes specified in preference to cast iron.

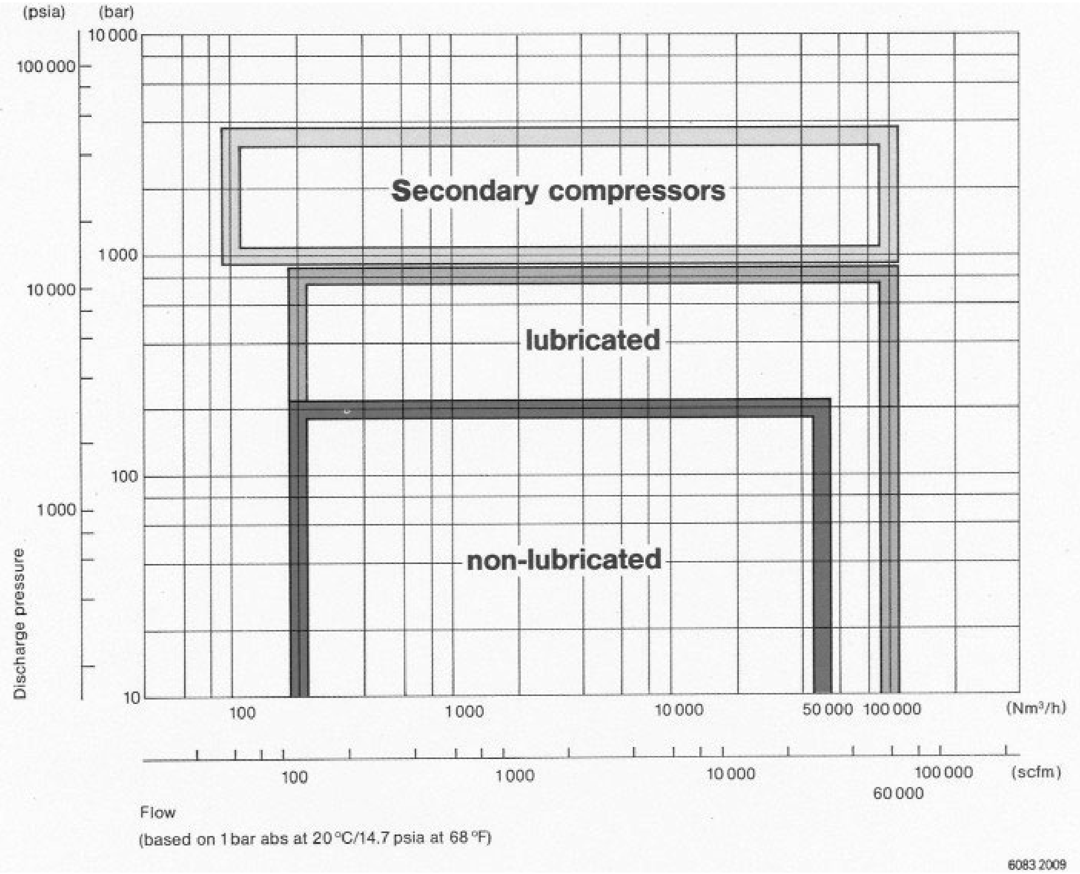
API-618 specifies that all cylinders must have replaceable liners. These are usually of cast iron because of its lubricating and bearing qualities. Liners should be honed to a finish of 10 to 20 microinches.

1.2.7 Cylinder Sizing

Cooling jackets, lubricant, and packing and ring material limit cylinder temperatures. A compressed gas discharge temperature of 275°F is set as ideal maximum, with 375°F as the absolute limit of operating temperatures. Using a common thermodynamic equation with the above limits, the design ratio per cylinder can be set. The design compression and tension load on the piston rod must not be exceeded and, therefore, rod load must be checked on each cylinder application.

Rod load is defined as follows:

$$\text{R.L.} = P_2 \times A_{HE} - P_1 \times A_{CE}$$



Medium

N_2, Ar, He, H_2, CO, CO_2, NH_3, CH_4, C_2H_4, C_3H_6, C_nH_m, LPG, LNG, CNG, Cl_2, HCl, H_2S, SO_2, Synthesis and other mixed gases

Applications

Chemical
Hydrocarbon processing industries
Polyethylene (ldPE, lldPE, hdPE)
Oil and gas production
Refinery
Enhanced oil recovery
Synfuels
Corrosive and toxic gases
Low-temperature and cryogenic applications
LPG on-shore installations

Figure 1-3. Typical application ranges for reciprocating compressors. (Source: Sulzer-Burckhardt, Winterthur, Switzerland.)

where

R.L. = rod load in compression in pounds

A_{HE} = cylinder area at head end in any one cylinder

A_{CE} = cylinder area at crank end, usually $A_{CE} = A_{HE}$ – area of rod

P_2 = discharge pressure, psia

P_1 = suction pressure, psia

The limiting rod loads are set by the manufacturer [2]. Some manufacturers require lower values on the rod in tension than compression. In this case, the above limits are checked by reversing A_{HE} and A_{CE}. Note, however, that the supplemental specifications of reliability-focused user companies often impose a limit on the maximum allowable stress acting on the net thread root area of piston rods. This limit is largely based on field experience and relates to rod attachment limitations rather than stress failures of the rod itself.

Compressors below 500 psig, such as air compressors, are usually sized by temperature considerations. Rod load usually becomes the limiting factor in applications above these pressures.

1.2.8 Cylinder Cooling

In process compression, higher pressures are normal. Therefore, compression ratios are low. Low compression ratios

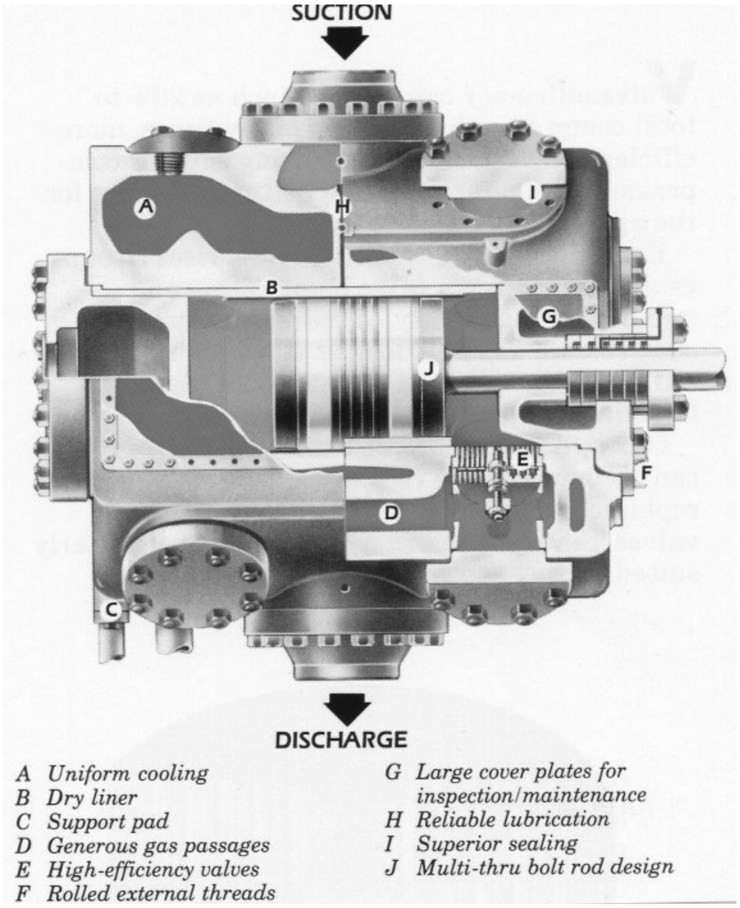


Figure 1-4. Typical reciprocating compressor cylinder for long-term reliable operation. (Source: Dresser-Rand Company, Painted Post, New York.)

give low temperature rises. Thermosiphon cylinder cooling is, therefore, specified, with a discharge temperature limit of about 200°F. Thermosiphon cooling consists of filling the jackets with an appropriate liquid such as water or a suitable radiator fluid and letting the heat radiate from the outer cylinder walls. The purpose of filling the jackets is to obtain an even heat distribution throughout the cylinder.

Above 200°F, coolant circulation is applied. Raw water is to be avoided because it leaves deposits in the cooling jackets that are extremely difficult to clean out. A closed system is specified, therefore, which consists of a reservoir, circulating pumps, and heat exchanger.

The old saying "cooler is better" is really incorrect here. Overcooling of the cylinder to a temperature below the dew point of the compressed gas must be avoided to prevent cylinder corrosion, or liquid build-up that results in a slug of liquid. Therefore, the coolant must be bypassed around the exchanger under temperature control. It is also recommended that a thermostat and heater be mounted on the reservoir and the coolant circulated when the compressor is stopped on standby, to maintain the cylinders at a temperature above the gas dew point.

The coolant is usually 50% ethylene glycol and water, to prevent freezing. This has a lower specific heat than water. Therefore, the manufacturer must always be asked to size heat exchangers on this basis, even if the initial coolant is intended to be only water.

Compressor valves are the most critical part of a compressor; they generally require the most maintenance of any part. They are sensitive both to liquids and solids in the gas stream, causing plate and spring breakages. When the valve lifts, it can strike the guard and rebound to the seat several times in one stroke. This is called valve flutter and leads to breakage of valve plates. Light-molecular-weight gases such as hydrogen cause this problem mostly, and it is controlled in part by restricting the lift of the valve plate, thus controlling valve velocity.

API valve velocity is specified as:

$$V = D \times 144/A$$

where

V = average velocity in feet/minute

D = cylinder displacement in cubic feet/minute

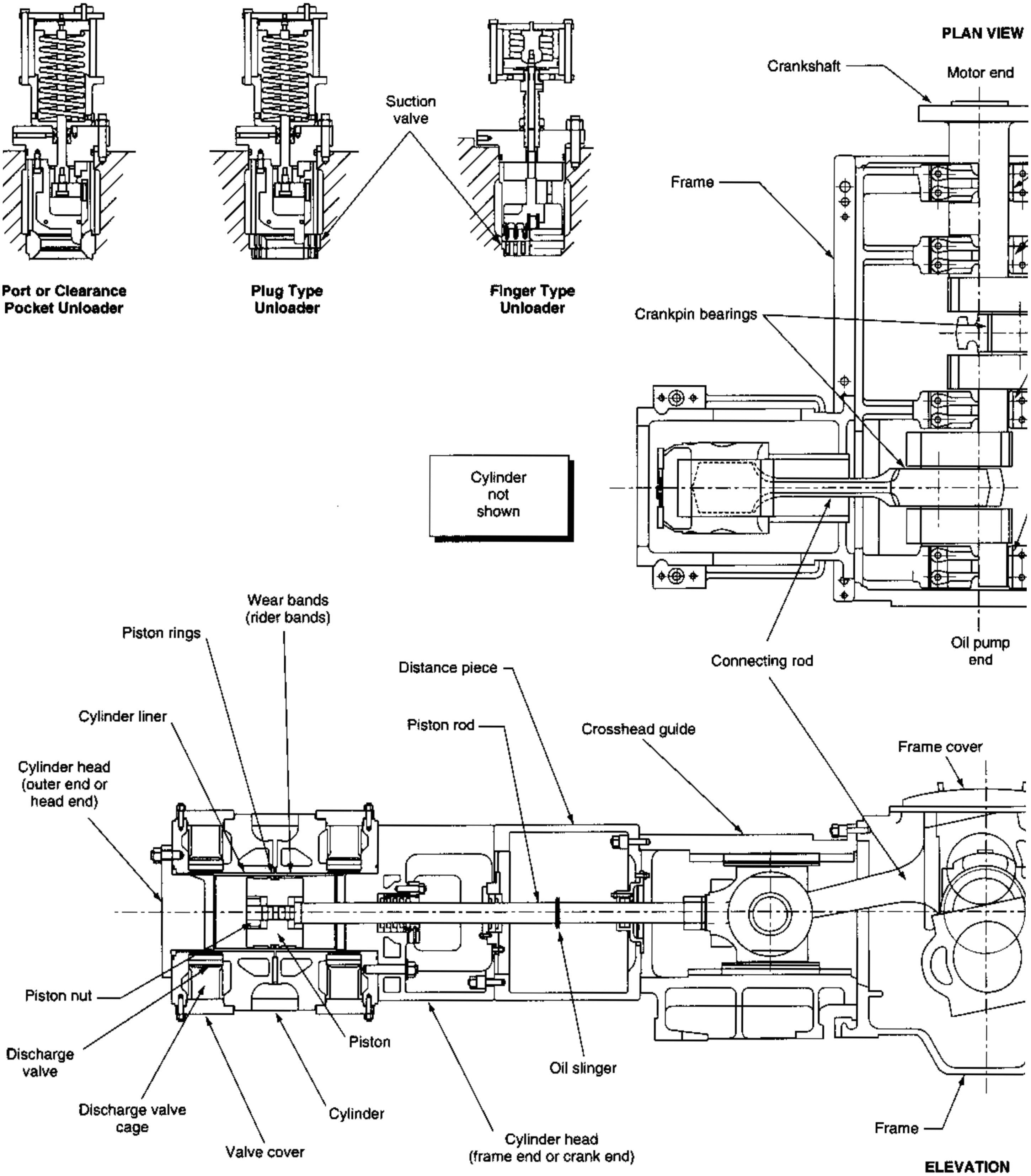
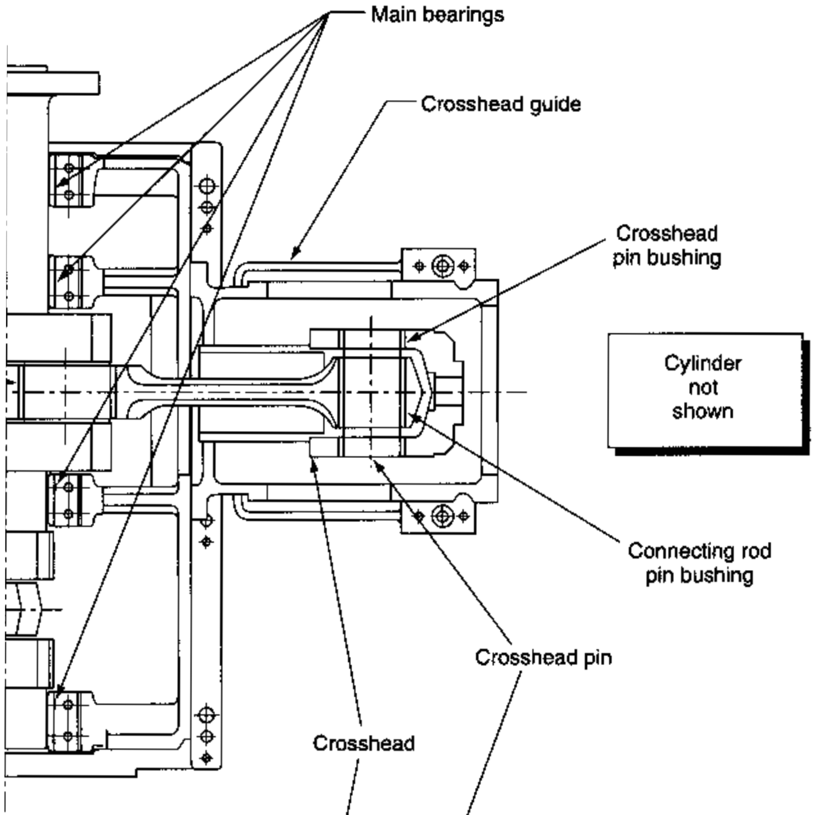
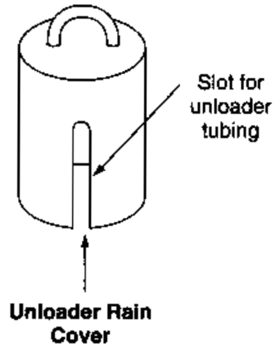
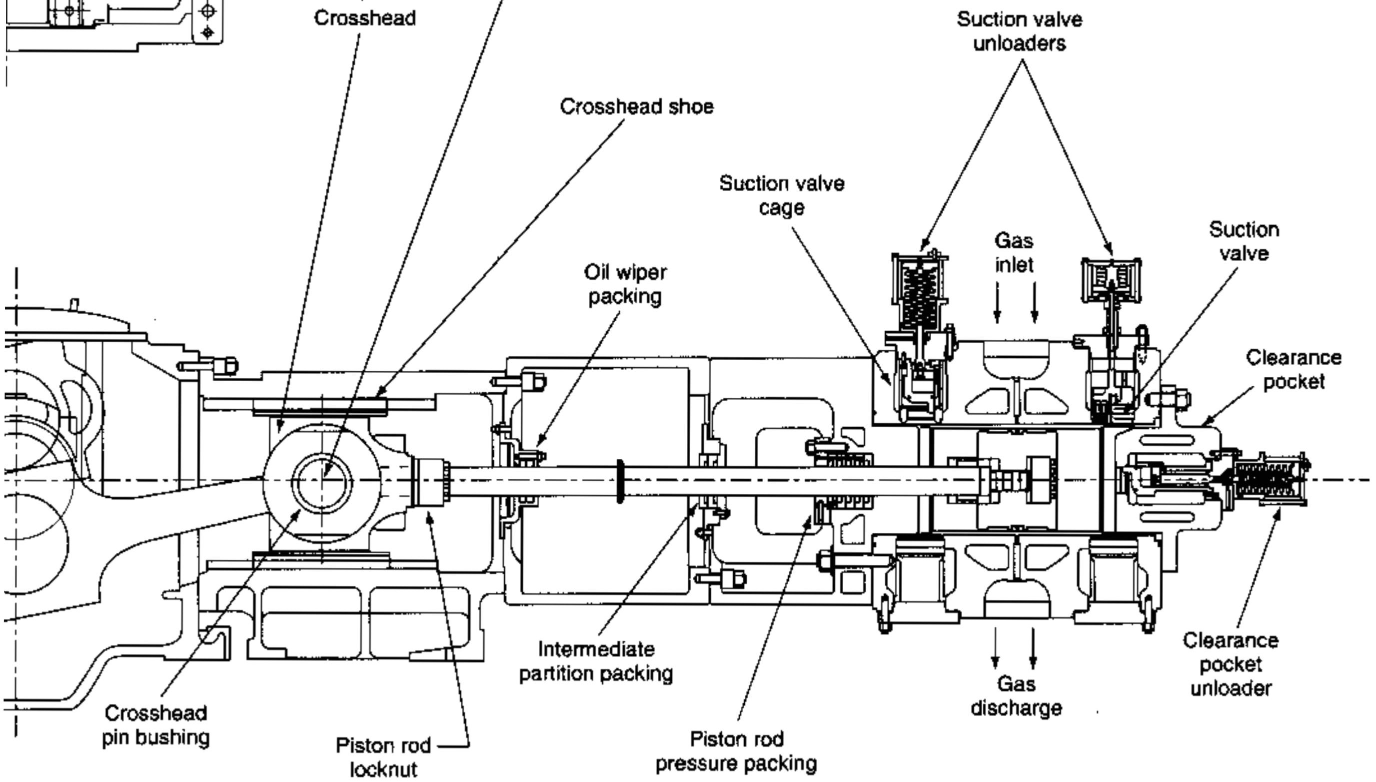


Figure 1-5. Reciprocating compressor cross section and nomenclature, per API 618.


Main bearings
Crosshead guide
Crosshead
pin bushing
Cylinder
not
shown
Connecting rod
pin bushing
Crosshead pin
Crosshead


Slot for
unloader
tubing
Unloader Rain
Cover


Suction valve
unloaders
Crosshead shoe
Suction valve
cage
Gas
inlet
Suction
valve
Oil wiper
packing
Clearance
pocket
Intermediate
partition packing
Gas
discharge
Clearance
pocket
unloader
Crosshead
pin bushing
Piston rod
locknut
Piston rod
pressure packing

VIEW

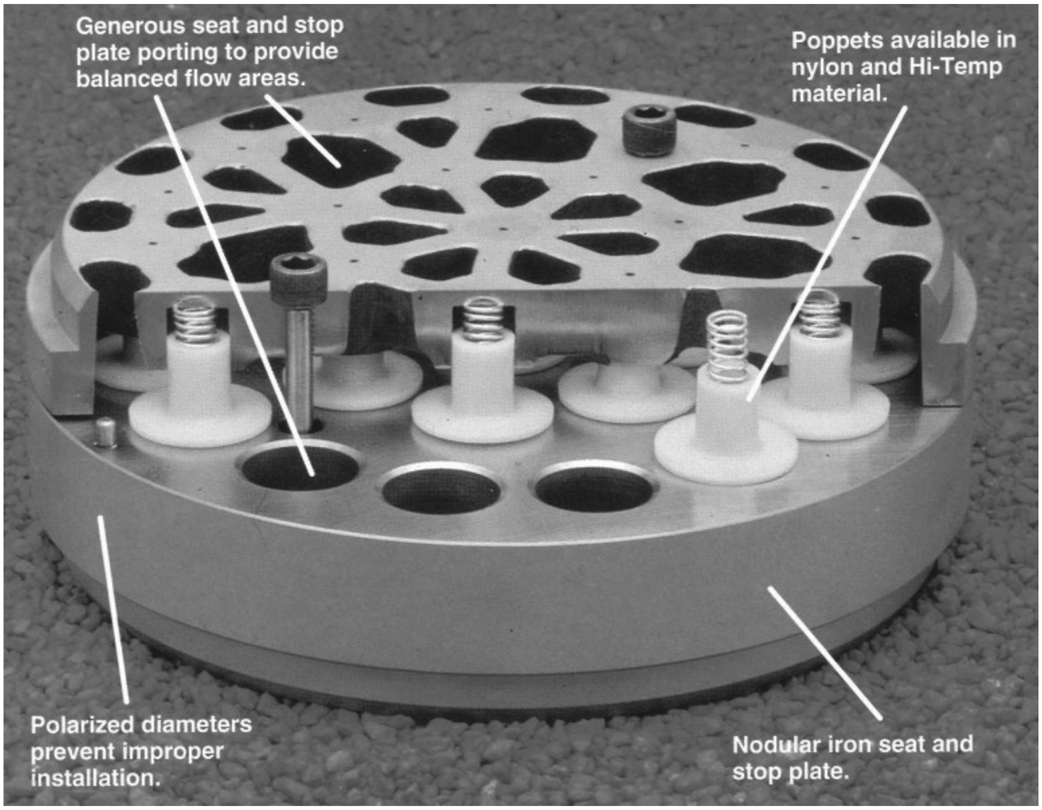


Figure 1-6. Poppet valve. (Source: Dresser-Rand Company, Painted Post, New York.)

A = total inlet valve area per cylinder, calculated by valve lift times valve opening periphery, times the number of suction valves per cylinder in square inches

Compressor manufacturers sometimes object to the above specification, since it gives a valve velocity for double-acting cylinders one half the value compared to equivalent single-acting cylinders. Therefore, manufacturers' data on double-acting cylinders often indicate a valve velocity double the API valve velocity and care must be taken to find out the basis upon which valve velocity is given. For heavier molecular weight gases (~20), API valve velocities of about 3,580 fpm are selected, and for lighter molecular weight gases (MW = 7), 7,000 fpm.

Manufacturers often use interchangeable suction and discharge valves. This can lead to putting valves in the wrong port, which can result in massive valve breakage or broken rods or cylinders. Reliability-focused users specify that valves must not be interchangeable. However, this feature can be lost or broken off, so correct valve placement should always be checked.

1.2.9 Pistons

Pistons (Fig. 1-9) are usually cast iron and are often hollow, to reduce weight. This space can fill with gas and is an explosive hazard when removing the piston from the rod. One should, therefore, specify that an easily removable plug must be supplied to vent this space before handling. Larger pistons are made of aluminum to reduce weight. These pistons have large clearance in the bore to allow for thermal expansion, on the order of 20 mils per inch of diameter. Rider rings are often supplied on cast iron pistons and are necessary on all aluminum pistons. These, or the whole piston, are rotated 90° about once per year to reduce wear. Bearing loads of pistons and rider rings are based on a unit stress calculated from half the rod-plus-piston weight in pounds, divided by the diameter of the piston or rider ring, times the width in square inches. Normal limits are 5 psi for Teflon®, 12 psi for cast iron, 14 psi for bronze, and 22 psi for Allen metal. Teflon compression rings are specified and are useful up to 500 psi ΔP. Above these loadings, copper-bearing material or Babbitt metal are used for wearing rings and bronze for compression rings. Teflon compression rings are often offered with steel expander rings underneath. These should be avoided because when the compression rings wear, the expander rings can score the cylinder. Designs of pistons and rings are available that will hold the compression ring out against the cylinder without expander rings.

1.2.10 Piston Rod

The piston rod (Fig. 1-10) mounts into the crosshead, and must be locked, either by a locknut or a pin, to prevent back-

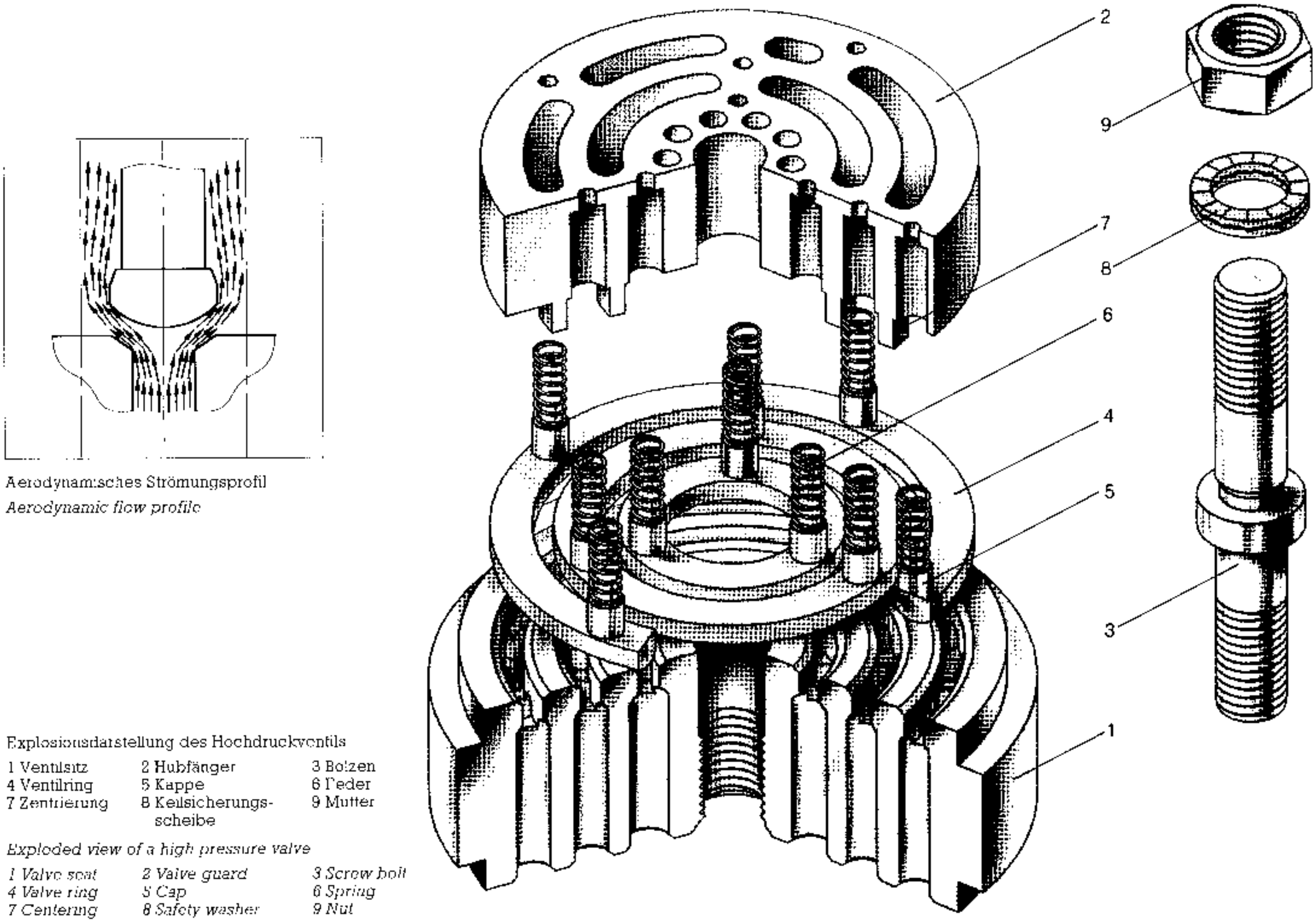


Figure 1-7. Typical high-pressure plate valve. (Source: Babcock-Borsig, Berlin, Germany.)

ing off. The rod is adjusted in the crosshead to equalize the end clearance of the piston in the cylinder. This is checked by barring over the machine, crushing a piece of soft lead, and measuring the remaining thickness. This is called the bump clearance.

The rod must be hardened where it passes through the packing. Some rods are chrome plated, but problems have occurred with them, especially on high-pressure machines with a high heat buildup. This can easily cause spider web cracks in the chrome which, in turn, can flake off and destroy the packing. The best arrangement is to purchase a flame-hardened rod. As wear occurs, the rod could be plated with tungsten carbide, which should last the life of the machine. High-pressure machines often have the rod extended through the piston and out the cylinder head to balance the pressure load (rod load) on the piston. Such an extended-through rod is called a tail rod. Tail rods [1] have been known to break off and be ejected from the cylinder like a missile. Reliability-focused users specify that all tail rods must be housed in a capture box strong enough to contain the tail rod should breakage occur.

1.2.11 Packing

Compressor packing (Fig.1-11) is made up of two rings in pairs, mounted in steel or cast iron cups with the open end of the cups facing away from the pressure. The cups are bolted together and have vent, oil, and drain holes drilled in them where required. These must be correctly aligned each time the packing is opened. If Teflon is specified for packing for pressures above 500 psi, generally an additional metallic backup ring is used to prevent the Teflon from extruding out of the cups.

Compressor manufacturers supply a distance piece (Fig. 1-5) between the cylinder and the crankcase for access to the packing. This is usually good for a three- or four-cup set. In process applications, however, especially above 2,000 psi, packing sets run to 8 or 18 cups or more. Extra long distance pieces must be specified in order that sufficient space is available to dismantle the cups and change the packing rings. If any doubt exists, the extra long distance piece should be specified, as the extra cost is minor. It is generally not possible to change to a longer distance piece after the machine is built.

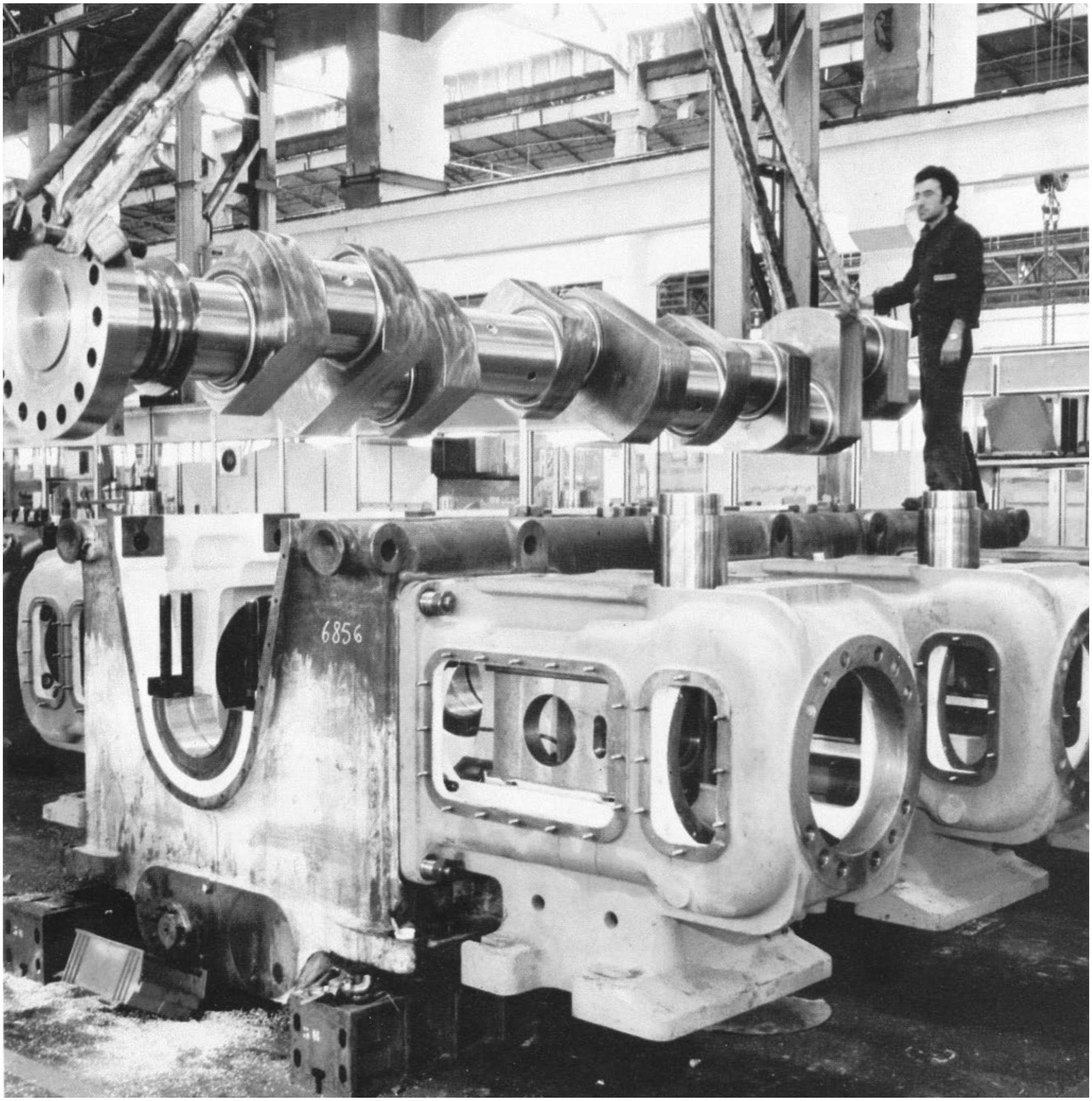


Figure 1-8. Crankshaft being lowered into compressor crank case. (Source: Nuovo Pignone, Florence Italy.)

1.2.12 Gaskets

Cylinder end covers, valve covers, and valves are gasketed to the cylinder. Metallic gaskets are specified. Manufacturers often offer nonmetallic gaskets, which have led to considerable trouble with leaks, especially on low-molecular-weight gases.

Often, the gasket seats must be lapped to obtain a good seal. Soft iron or metallic V-ring sets have given the best service. O-rings confined four ways on valve cover plugs have also given good service, but should never be used where they can be crushed when pulling down the cover. Plug holes must be chamfered to prevent cutting of O-rings upon entry.

1.3 COMPARISON BETWEEN RECIPROCATING AND CENTRIFUGAL COMPRESSORS

In the final analysis, economic factors, including repair and failure projections, will govern the selection of a compressor [2, 3]. Both the user and compressor manufacturer should

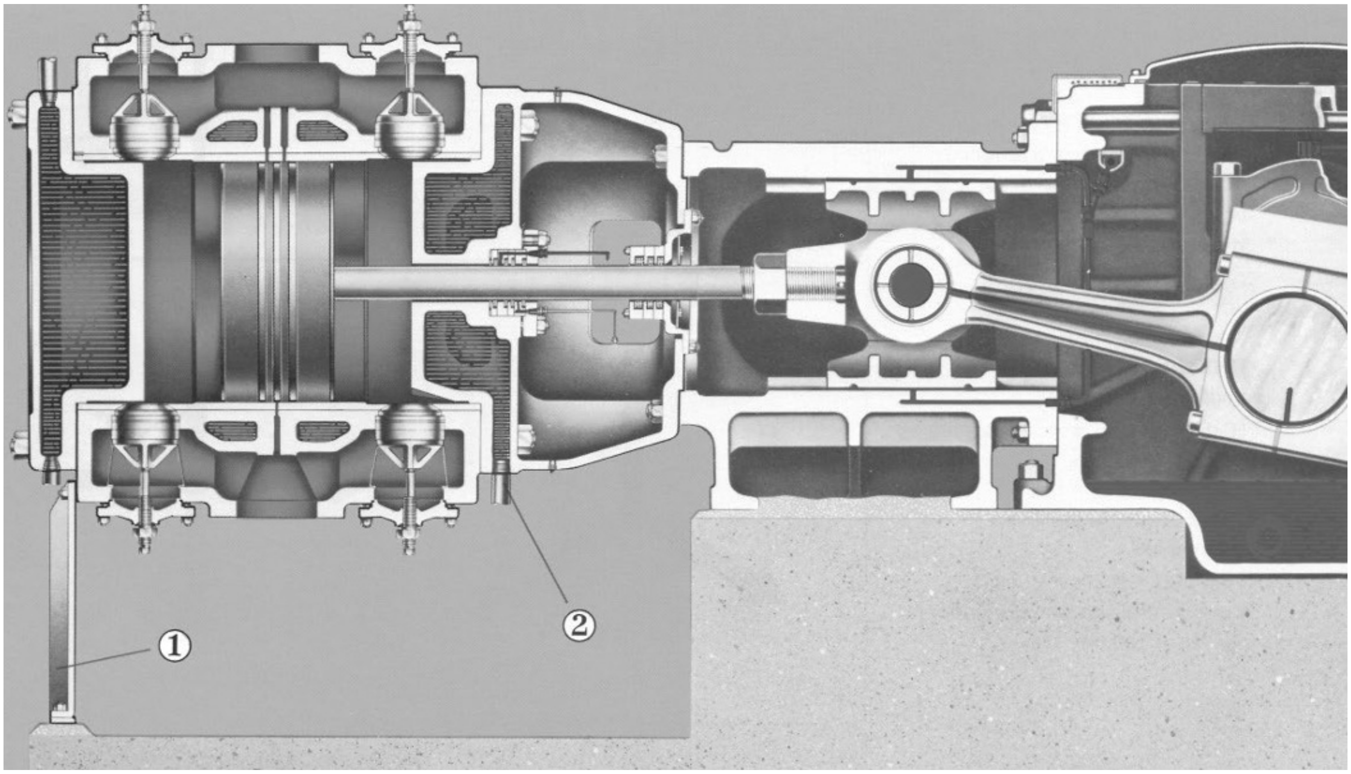


Figure 1-9. Partial cross section showing "cold" cylinder support (1), crank-end cylinder head and drain (2), piston and piston rod, sliding crosshead, and connecting rod. (Source: Dresser-Rand Company, Painted Post, New York.)

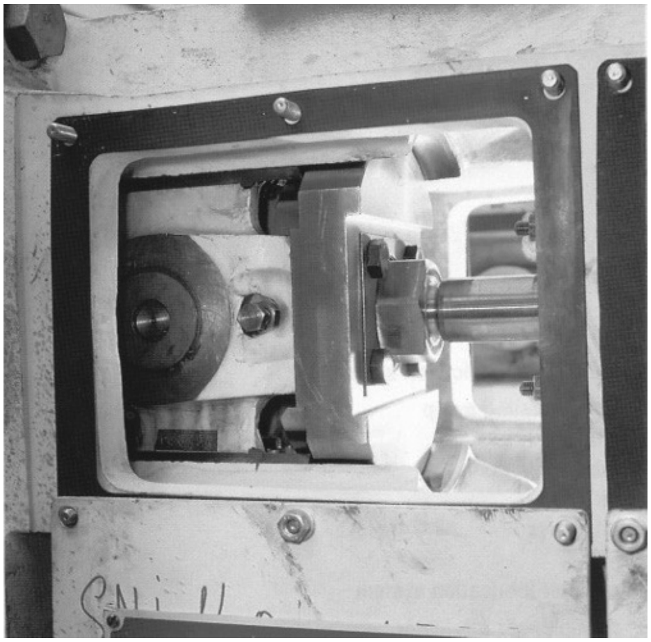


Figure 1-10. Compressor distance piece with sliding crosshead (left) and piston rod (right). (Source: Nuovo Pignone, Florence Italy.)

have full knowledge of operating conditions and factors governing selection. As energy costs increase, the higher efficiency of the reciprocating and rotary screw (double helical screw) compressor becomes increasingly important. To what extent energy considerations influence equipment selection is, interestingly enough, also a cultural issue. In a house in which lights are habitually left on in every room, or even the unoccupied rooms are cooled in the Summer and heated in the Winter, the introduction of energy conservation is neither a popular nor high-priority subject.

Obviously and in some instances, certain tangible and even intangible factors dictate the use of one or the other machine, without question. In other cases all factors must be analyzed carefully in order to make a choice between the two types. Occasionally, a reciprocating and centrifugal compressor operating in series is indicated.

1.3.1 Gas Properties and Process Conditions

Gas Analysis

A complete and accurate gas analysis is the starting point in compressor selection. Percentage by volume of component gases, entrainment of liquids and solids, and percent water vapor should be recorded. Even minute quantities of contaminants should be reported in the analysis. Trace amounts of

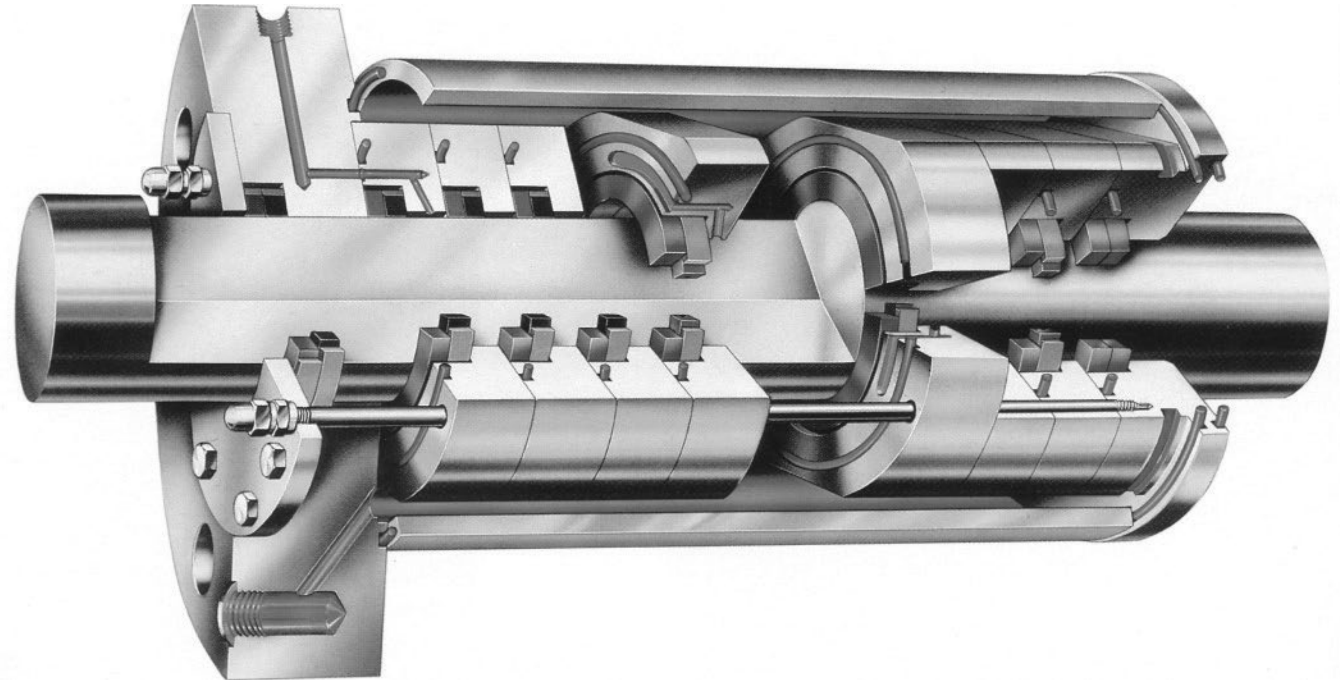


Figure 1-11. Cutaway view of packing area. (Source: Babcock-Borsig, Berlin, Germany.)

sulfur compounds and chlorides can cause corrosion or other mechanical difficulties. Even slight corrosion can produce failure of cyclically stressed parts of either type of compressor.

In a reciprocating compressor, solid particles will cause high maintenance by accelerating wear of valves, pistons, cylinders, piston rods, and packing. Solids passing through a centrifugal compressor may erode impellers and casings severely. If at all possible, any solid particles should be removed from the gas stream before the gas reaches the compressor. In some cases, a wash fluid may be used to carry fine solids through a compressor.

Water vapor or other vapors entrained in a gas use up compressor capacity. This volume must be allowed for when sizing the compressor.

Molecular Weight

Except for a change in efficiency due to valve-related losses, the reciprocating compressor is not affected by the molecular weight of the gas. Periodic changes in gas composition will have little effect on compression horsepower and pressure. On the other hand, the pressure developed by a centrifugal compressor at a particular speed is directly proportional to the gas density or molecular weight.* Also the internal flow passages of a centrifugal compressor are designed to best handle the change in density of a particular gas as it passes through the compressor.

For equal impeller peripheral speeds it will take many stages compressing hydrogen to equal the pressure created by a single stage compressing a dense gas such as butadiene or methyl chloride.

If the percentage of constituents in a composite gas varies from time to time, so will the molecular weight. A centrifugal compressor, to generate the desired pressure when handling the full flow of such a gas, would have to be designed for the lowest expected molecular weight. Excess pressure resulting from increased molecular weight would have to be reduced by throttling, change in speed, or by some other means. In other words, it is difficult to design a centrifugal unit that will economically and practically handle changing gas densities.

Polytropic Exponent

As a gas is being compressed, the polytropic exponent determines the pressure–volume relationship and temperature change. If the polytropic exponent is not known, the average ratio of specific heats may be used for calculating the theoretical adiabatic compression temperatures, volumes, and horsepower.

Temperatures are important in a reciprocating compressor. Temperatures within a conventionally lubricated cylinder should not exceed 350°F. Theoretically, a gas with a low exponent can be compressed with much higher ratios of compression per stage, yet hold temperatures within desired lim-

*Centrifugal force = (mass)(tangential velocity)/(radius). Note also that the impeller of a centrifugal compressor will develop the same head for various gases but the pressure is a function of gas properties its. One drawback to this, however, is that volumetric efficiency decreases with high ratios, and a low exponent may make it uneconomical to compress to high ratios.

The compression exponent also influences the design of centrifugal compressors. Pressure developed by an impeller will be less for a high-exponent gas than for a low-exponent gas of equal density at inlet conditions.

Flow Rate

The centrifugal compressor is essentially a large-capacity machine. If, say, 1000,000 cfm of gas were to be compressed from near-atmospheric suction conditions, a centrifugal compressor would probably be used for the lower stages of compression. If the final discharge pressure is high, the last stages of compression could be handled by reciprocating compressors.

In general, reciprocating compressors have a cost advantage up to perhaps 10,000 cfm capacity. Another rule of thumb for process applications is that comparison of centrifugal compressors with reciprocating and rotary screw machines is recommended if the volume of a gas stream at discharge pressure is over 600 cfm. Also, custom-built centrifugal compressors are generally a parity investment compared with reciprocating and/or helical screw compressors if the absorbed power is in the vicinity of 1,500 kW.

If the flow rate varies widely, reciprocating compressor and certain screw compressors can operate with a reasonable sustained efficiency. Flow variation may be handled by suction valve unloaders and/or clearance pockets (Fig. 1-12) or, in the case of certain screw compressors, speed changes.

A suction valve unloader operates by depressing the valve feathers or plates to hold the valve open during both suction and compression strokes. So, on the compression stroke, the gas is not compressed, but flows back into the suction manifold. The unloader can be operated either manually or by remote automatic means.

Clearance pockets are volume chambers built into the cylinders or heads, or may be attached to the cylinder by a piping connection. Ordinarily, a built-in pocket is fitted with a plug valve so that the additional volume may be added or removed at will, either manually or automatically.

Generally, clearance pockets are more flexible in application than are valve unloaders. They can be sized to accomplish any degree of per-stage unloading. Two or more smaller pockets commonly are placed in one cylinder end. Some

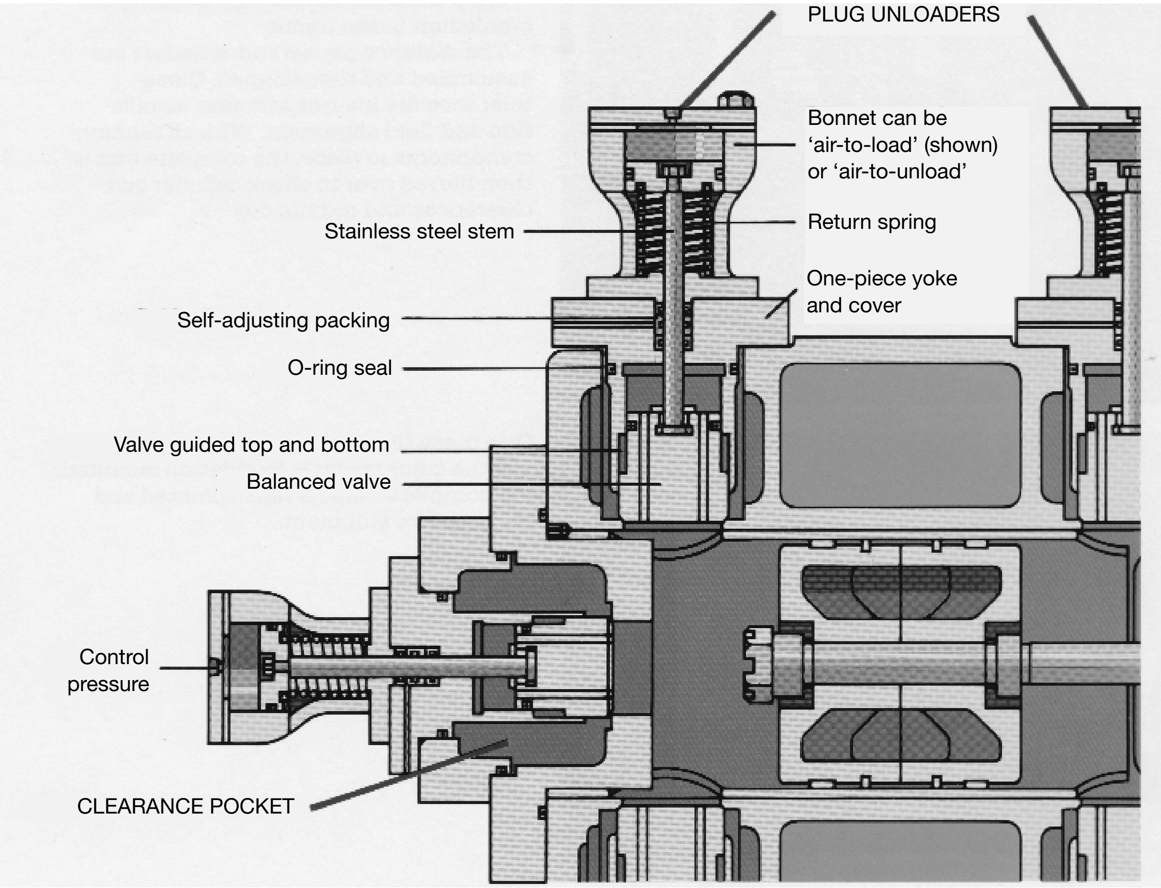


Figure 1-12. Plug unloaders (top) and clearance pocket (left). (Source: Dresser-Rand Company, Painted Post, New York.)

compressor manufacturers are able to offer variable-volume clearance pockets. Variable-volume clearance pockets plus suction valve unloaders permit continuous (infinitely variable) capacity control with minimum power consumption. (As will be seen later, with centrifugal compressors operation is not feasible below the surge point. This flow limit ordinarily is between 50% and 75% of rated capacity.)

Inlet and Discharge Pressure

In a multistage reciprocating compressor, a variable suction pressure while maintaining a constant discharge pressure may necessitate pockets and/or valve lifters in order to maintain satisfactory operation. Lowering the suction pressure will lower the overall horsepower, lower the differential pressure on all stages except the last stage, increase the differential on the last stage, and, very often, the horsepower on the last stage. Raising the suction pressure will raise the horsepower of the complete machine, raise the differential pressure on all stages up to the last stage, and probably lower its horsepower.

In centrifugal compressors, an increased suction pressure will raise the discharge pressure and increase the horsepower. If the suction pressure is lowered, the centrifugal machine will not compress to the desired discharge pressure.

Temperature

In both reciprocating and centrifugal machines, the compressor recognizes only capacity at the actual inlet conditions. Therefore, the inlet temperature must be specified and compressors are usually rated at suction conditions. Conversion to standard conditions are made for reference and comparison only.

As regards mechanical operation of equipment, centrifugal compressors are less affected by high or low temperature extremes than reciprocating compressors. Centrifugal compressors have been used to circulate gas at 800°F. With conventional lubricants, such temperatures are impractical in a reciprocating compressor. Very low temperatures also cause lubrication problems. However, reciprocating compressors have been operated at suction temperatures below minus 100°F.

A nonlubricated reciprocating compressor developed by a Swiss company in the 1940s—the Burckhardt Labyrinth Piston Compressor, Figs. 1-13 and 1-14—will often solve both of these lubrication problems. A grooved piston that does not contact the cylinder walls is used. Close clearances are employed to control leakage between the cylinder and piston. Outstanding results have been obtained with this design.

A centrifugal compressor designed for a given capacity at a specified inlet temperature will fail to deliver the required discharge pressure if the suction temperature is increased significantly. Or, if the suction temperature is lowered, the discharge pressure will be increased. Consequently, the centrifugal compressor must be designed to deliver the desired capacity and pressure under the maximum inlet temperature conditions.

Heat Balance

Process heat balance sometimes has a bearing on the selection of a driver. In turn, the driver selection may determine the compressor type. For example, if a backpressure steam turbine driver is selected because of low-pressure steam requirements, a centrifugal compressor would be a logical choice. Steam turbines are ideally suited for centrifugal compressors for several reasons. First, the rpm match between centrifugal compressors and turbines permits a direct drive. In the size range of centrifugal compressors, turbines are one of the least expensive drivers. They are mechanically efficient and lend themselves well to speed control by which compressor pressure and gas flow may be indirectly controlled. Moreover and if properly selected, they rate favorably on the list for high reliability and low maintenance cost. They may be used in continuous service of perhaps three or more years. In clean services and at best-practices user companies, uninterrupted run lengths of eight years have sometimes been achieved. Steam turbine drivers are limited, of course, to use in plants where steam is available at reasonable cost.

If economics or other reasons dictate the use of electric-motor or gas-engine drivers, reciprocating compressors are sometimes the most logical choice. One exception is that gas engines drive large single-stage centrifugals through speed-increasing gears in some gas pipeline stations. This particular combination is less expensive than gas-engine reciprocators of the same size. A second exception is in applications that favor a step-up gear between the electric motor driver and centrifugal compressor.

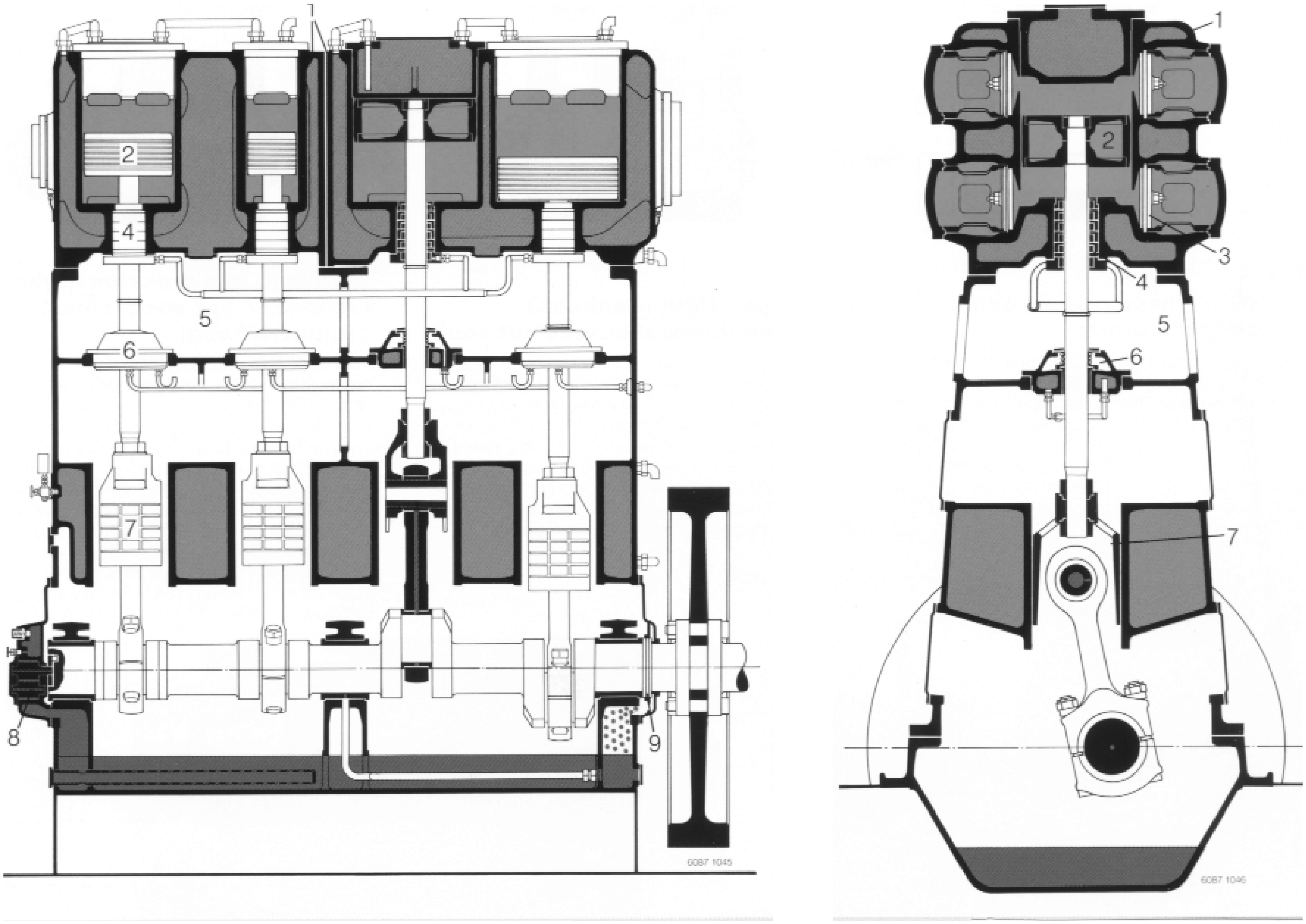
Under certain conditions, such as low fuel cost plus unavailability of moderate-pressure steam, a combustion gas turbine driver may be selected. Credit for steam produced by a waste-heat boiler will generally offset the higher first cost of the gas turbine. In any event, driver selection is governed by economic factors that may vary greatly from site to site.

1.4 SERIES AND PARALLEL OPERATION

Within certain limits, compressors can be operated in series or parallel. In positive-displacement machines operated in series, the discharge flow rate of the first must equal the inlet flow rate of the second machine. If not carefully matched, a series system may produce an appreciable vacuum or excessive pressure between the two compressors. With turbocompressors in series, a flow mismatch will cause surging of one or the other, and flow instability.

Positive-displacement compressors can be operated in parallel if the discharge pressures of the two machines are about the same; flow through one compressor has little or no

Figure 1-13. Principle of labyrinth piston compressor. (Source: Sulzer-Burckhardt, Winterthur, Switzerland.)



Design features

1. Cylinder with jacket for heat exchange medium such as cooling water
2. Labyrinth piston
3. Compressor valve
4. Labyrinth-piston rod gland
5. Extra long distance piece separating the oil-lubricated crankcase from the oil-free cylinder
6. Oil-lubricated guide bearing with oil scrapers
7. Crosshead guide surrounded by jacket for heat exchange medium such as cooling water (supersedes the oil cooler on small compressors)
8. Crankshaft-driven lubricating-oil pump for forced-feed lubrication of bearings and crossheads
9. Crankshaft seal

Area of gas compression and gas flow

Area of nonflowing gas

Heat exchange medium (e.g. cooling water)

Lubricating oil

Figure 1-14. Four-throw labyrinth piston compressor. (Source: Sulzer-Burckhardt, Winterthur, Switzerland.)

effect upon the other. However, turbocompressors operated in parallel do present a matching problem; one compressor will frequently pick up the entire flow while the second idles or becomes unstable. Consequently, an automatic flow-control system is used by reliability-focused users.

Turbo and positive-displacement compressors can be combined; a turbocompressor can be used in series with a positive-displacement machine, or vice versa. A turbocompressor also could be used in parallel with a positive-displacement compressor as a flow booster.