5

HYPERCOMPRESSORS*

Intimately bound to the chemical industry and based on a long evolution, the main stages of which were the liquefaction of air and the synthesis of ammonia, the technique of using very high pressures was eventually perfected from developments in the manufacture of low-density polyethylene. This is now the only industry requiring large reciprocating compressors for very high pressures, because the pressures necessary for other main chemical processes have been reduced steadily since 1945. For this reason, the present segment will be restricted to the ethylene compressors. Considering that the classical designs of reciprocating high-pressure compressors cover an uninterrupted range up to about 1000 atm, very high pressures will imply those greater than 1000 atm.

5.1 INTRODUCTION

A characteristic feature of the high-pressure ethylene polymerization process is that a very large difference in pressure is necessary between the inlet gas entering the reactor and the outlet of the recycle gas. The recirculators, generally called *secondary compressors* (Fig. 5.1), work between two limits: 100 to 300 atm on the suction side and 1500 to 3500 atm on the delivery side, for most of the existing processes. Because the coefficient of reaction lies between 16 and 30%, the secondary compressors have to handle three to six times the fresh gas quantity, thus being by far the most powerful machines in the production stream. When the industrial expansion began, their unit capacities were of 4 to 5 tons/h; now they exceed 50 tons/h, and their power requirement per unit has been increased from 600 hp up to some 20,000 hp.

* Contributed by Burckhardt Compression AG, Winterthur, Switzerland. Based on C. Matile's, Industrial reciprocating compressors for very high pressures, *Sulzer Technical Review*, Vol. 2, 1971.

A Practical Guide to Compressor Technology, *Second Edition*, By Heinz P. Bloch

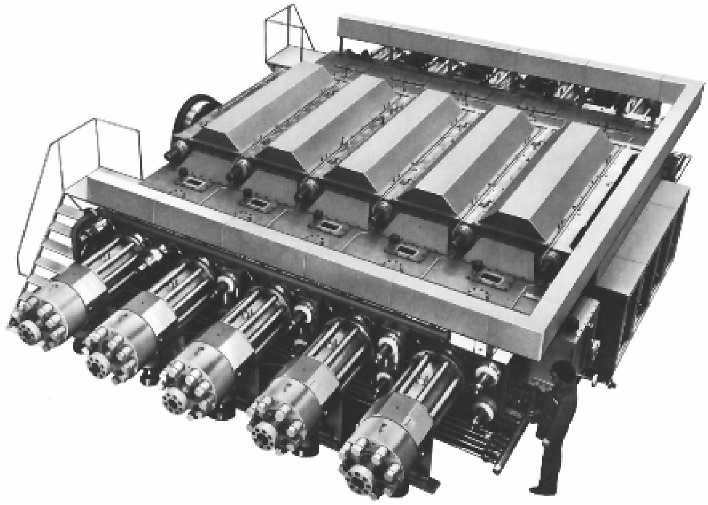

FIGURE 5.1 Large hypercompressor used in ethylene service. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

Because the entire operating range of these secondary compressors takes place well above the critical point of ethylene, the thermodynamic behavior of the fluid lies somewhere between that of a gas and that of a liquid. This peculiar condition has two main effects. The first is a very small reduction of the specific volume with increasing pressure; for instance, at a temperature of 25°C, the specific volume is 3 dm^3/kg at 100 atm, 2 dm^3/kg at 700 atm, and 1.5 dm^3/kg at 4500 atm. The second effect is a very moderate rise in the adiabatic temperature with increasing pressure; for instance, with suction conditions of 200 atm and 20°C, the delivery temperature will reach only 100°C at 2000 atm.

These particular thermodynamic conditions greatly influence the design of high-pressure ethylene compressors. Compared with the conventional reciprocating compressor, the compression ratio is of little practical significance; the important factor is the final compression temperature, which should not exceed 80 to 120°C, depending on process, gas purity, catalyst, and so on, to avoid premature polymerization. The influence of the cylinder clearance on the volumetric efficiency is slight because of the small reduction in specific volume, and very high compression ratios are therefore possible with quite admissible efficiency. In addition, the stability of intermediate pressures depends chiefly on the accuracy of temperatures. For instance, in the case of two-stage compression from a suction pressure of 200 atm to a delivery pressure of 2500 atm, a drop in the first-stage suction temperature from 40 to 20°C will cause the intermediate pressure to rise from 1000 to almost 1600 atm.

For these reasons, a secondary compressor that has only one or two stages is required, despite the very large pressure differences involved. However, this again compels the designer to face extremely high mechanical strains, due to the high amplitude of pressure fluctuation in the cylinders. Finally, an additional and sometimes disturbing feature of ethylene must be mentioned. If the gas reaches very high pressure and high temperature simultaneously (which can easily occur in a blocked delivery port because of very low compressibility), it will decompose into carbon black and hydrogen in an exothermic reaction of explosive character.

5.2 CYLINDERS AND PISTON SEALS

Sealing of the high-pressure compression chamber is a major problem that could be solved by avoiding friction between moving and stationary parts. This has been realized for laboratory equipment and small-scale pilot plants by the use of either metallic diaphragms or mercury sealants in U-tubes, and such arrangements are still in use for research purposes. In addition, they have the advantage of avoiding any contamination of the compressed gas by any lubricant. Unfortunately, chiefly for economic reasons, they proved to be impractical for industrial compressors, at least for the present state of techniques. Thus, because labyrinth seals are out of the question for very high pressures, friction seals have to be accepted; in fact, two solutions are currently used—moving and stationary seals.

Metallic piston rings are the only sort of moving seals used in the large high-pressure reciprocating type of compressor. They are generally made in three pieces: two sealing rings, each covering the slots of the other, and an expander ring behind both of them, which also seals the gaps in the radial direction. The materials used are special-grade cast iron, bronze, or a combination of both, with cast iron or steel for the expander. The piston, of built-up design, comprises a series of supporting and intermediate rings with a guide ring on top and a throughbolt (two different designs are shown in Fig. 5.2). All parts of the piston are made of high-tensile steel, and particular care must be given to the design and to the stress calculation of the central bolt, which is subjected to severe strain fluctuations.

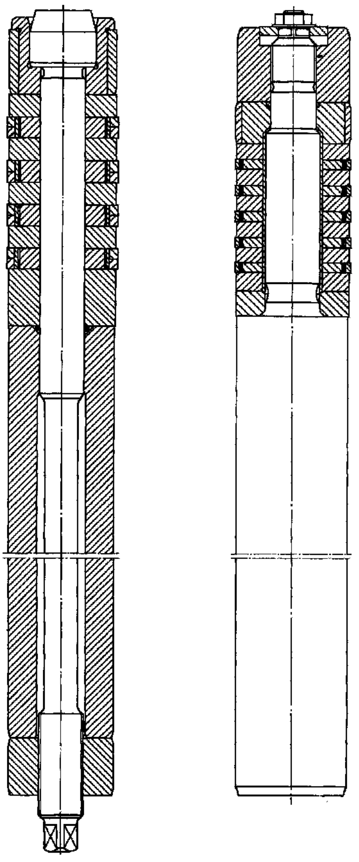

FIGURE 5.2 High-pressure pistons with piston rings. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

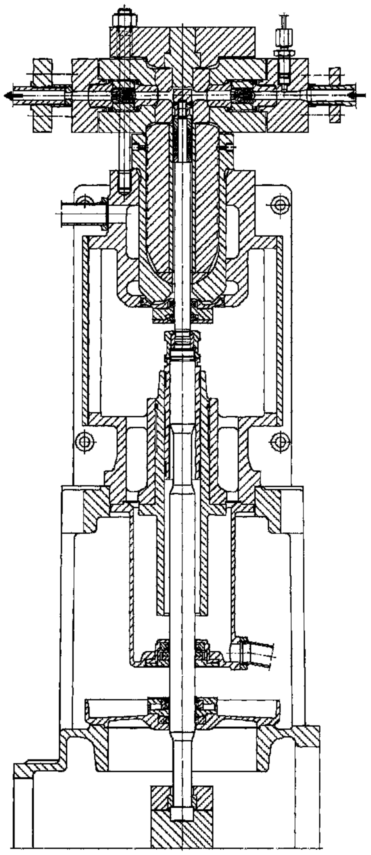

FIGURE 5.3 High-pressure cylinder for moderate end pressures. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

The use of piston rings allows for a simple cylinder design, the main part of which is a liner that has been thermally shrunk to withstand the high variations of the internal pressure (see Figs. 5.3 and 5.4). The inner sleeve, which was previously made of nitrided steel, is now generally of massive sintered material such as tungsten carbide. The use of this expensive material is justified by two beneficial qualities: it possesses an extremely hard surface and has a high modulus of elasticity. The first improves the conditions of friction considerably and greatly reduces the danger of seizure. The high modulus of elasticity of sintered tungsten carbide allows the amplitude of the breathing movement under the internal pressure fluctuation to be much smaller than with steel. The stress variations in the expanded outer sleeves are therefore reduced appreciably. However, because these sintered materials have a very poor tensile strength, care must be taken to ensure that the inner sleeve is always under compression, even if the temperature increases. This is the main purpose of external cooling of the liner and not, as is usual, to dissipate the heat of compression.

Packed plungers are the other answer to piston sealing. Although some manufacturers still use packings of hard plastic materials (nylon or similar), the most widely used packings are the metallic self-adjusting type. They are usually assembled in pairs, the actual sealing ring tangentially split into three or six pieces being covered by a three-piece radially cut section. Both are usually made of bronze, kept closed by surrounding garter springs, and held in place by locating and supporting steel plates. These plates must also be thermally shrunk to resist the high variations in internal pressure. Unfortunately, the use of

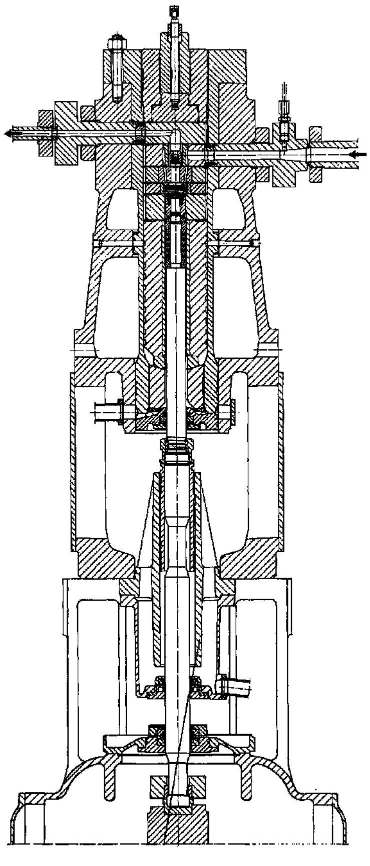

FIGURE 5.4 High-pressure cylinder for medium end pressures. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

sintered hard materials is restricted by the fact that the supporting plates are subjected, in the axial direction, to heavy bending and shearing forces that these materials generally cannot stand. To improve the friction conditions of the packing rings, the high-tensile steel supporting disks are frequently surface hardened or plated with carbide. The plungers are made of nitrided steel for use in moderate pressures, and for higher pressures are of steel, plated with hard materials. For very high pressures the use of solid bars of hard metal is the best wear-resistant solution for both plungers and packings. The disadvantage of the packed plunger design lies in the much larger joint diameters of the static cylinder parts, which require two to three times higher closing forces than the piston ring design. Large cylinders, such as the one shown in Fig. 5.5, require pretensioning of the cylinder bolts to about 10 times the maximum plunger load. This ratio is higher for smaller cylinders.

For piston rings and packed plungers, the optimum number of sealing elements appears to be four or five. In both solutions it is essential that the piston be centered accurately if the seals are to be effective; this is the reason for the guiding ring within the cylinder and for the additional guide at the connection between the piston and driving rod. At the base of the cylinders an additional low-pressure gland allows gas leaks to be collected and the plunger to be flushed and cooled. Other separate glands positioned on the rod connecting the piston to the drive (see Figs. 5.3 and 5.4) prevent the cylinder lubricant from mixing with the crankcase oil, and because the intermediate space is open to the atmosphere, it is impossible for gas to enter the working parts.

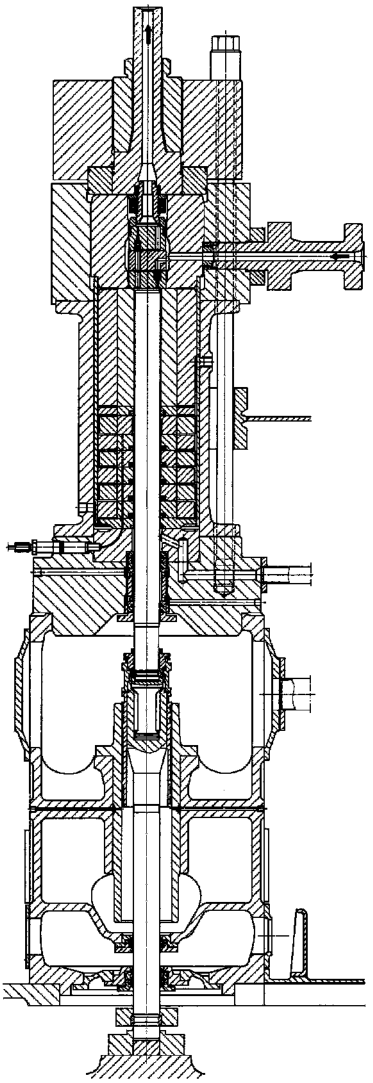

FIGURE 5.5 Gas cylinder for very high end pressures. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

From the point of view of design and maintenance, piston rings would appear to be the most adequate solution, and they are currently used for pressures up to 2000 atm, or in some circumstances up to 3000 atm. The choice between them and the packed plungers depends largely on the process and type of lubricant used. One difficulty is that normal mineral oils are dissolved by ethylene under high pressure to such an extent that they no longer have any lubricating effect. The glycerine used in earlier machines has been widely replaced by paraffin oil, either pure or with wax additives, which is much less diluted by the gas than other mineral oils. However, it is a rather poor lubricant and is inferior to the various types of new synthetic lubricants, which are generally based on hydrocarbons. The basic difference between piston rings and plunger packings is that the latter may be lubricated by direct injection, while piston rings are lubricated indirectly. This may be an advantage since the low polymers carried by the return gas back from the reactor are reasonably good lubricants. However, too large an amount of low polymers causes the rings to stick in their grooves, and some types of catalyst carriers also brought back by the gas are excellent solvents for lubricants. Thus, the most convenient solution has to be selected for each specific case. In general, for higher delivery pressures (greater than 2000 to 2500 atm), better results are obtained with the use of packed plungers.

5.3 CYLINDER HEADS AND VALVES

It is relatively simple to construct a vessel that will resist 2000 atm, but the problem becomes more intricate when the vessel must withstand, for years, a pressure that fluctuates between 300 and 2000 atm at a frequency of 3 to 4 Hz. The leading idea of the designer must be to divide a complicated problem into a series of simpler problems, each of which is then accessible to accurate methods of investigation. If this is done properly, it is possible to divide a large piece at the very places where inadmissible changes of stresses would occur and to keep the combined strains in each item within tolerable limits. The examples of cylinders shown in Figs. 5.3 through 5.5 illustrate the result of this method. The striking feature is the very simple shape of all pieces subjected to high pressure.

A first obvious result is that suction and delivery valves have to be located in a separate cylinder head. Figure 5.3 shows one type of cylinder head that can be used for moderate pressure fluctuations (up to amplitudes of about 1200 atm) and moderate cylinder dimensions. The intersection of the gas passages with the main bore is located in a small forged core, shrunk in a heavy outer flange, and pressed by the upper cover in the axial direction. This piece has a symmetrical shape with carefully rounded internal edges. By dismantling only the upper cover, it is possible with this design to pull out the complete piston with its rings through the central hole without disconnecting the gas pipes and without removing the valves. A typical valve for this type of cylinder head is shown in Fig. 5.6*a*. The same valve is used on the suction and delivery side, the two end pieces being shaped differently to avoid incorrect assembly. The valve is held against the central head piece by the connection flange of the gas piping as a type of composite lens.

For higher amplitudes of pressure and larger cylinders, cross bores and duct derivations must be taken away from the area of large pressure fluctuations. This is effected by the use of central valves, combining suction and delivery valves into one concentric set. For cylinders of moderate size, this can be done as shown in Figs. 5.4 and 5.6*b*; the different valve elements are located in a succession of simply shaped disks, with the same diameter as the cylinder liner and piled up on top of it. The lower two disks, which have been produced by the shrinking technique, receive the pressure fluctuation in their central hole, while the upper two, which contain the radial bores for the gas connections, are subjected only to static pressure.

For still larger cylinders, the combined valve is assembled as a separate unit to keep compact dimensions and weights and is inserted into the central hole of the cylinder head core, as shown in Fig. 5.5. The two valves illustrated in Fig. 5.6*c* and *d* are designed on the same basic principle; the last one, used in very large cylinders, is fitted with multiple suction and delivery poppets to reduce the moving masses. The entire valve body is subjected to suction pressure on the outside and only to the pressure fluctuations in the longitudinal hole. The suction pipe is connected to the radial bore of the cylinder head core as shown in Fig. 5.5. Separation of suction and delivery pressures is ensured by the circumferential self-sealing ring of hard plastic material, as shown in Fig. 5.6*c* and *d*. The entire valve is pressed on the end of the cylinder liner by the difference of pressures; the plate springs visible in

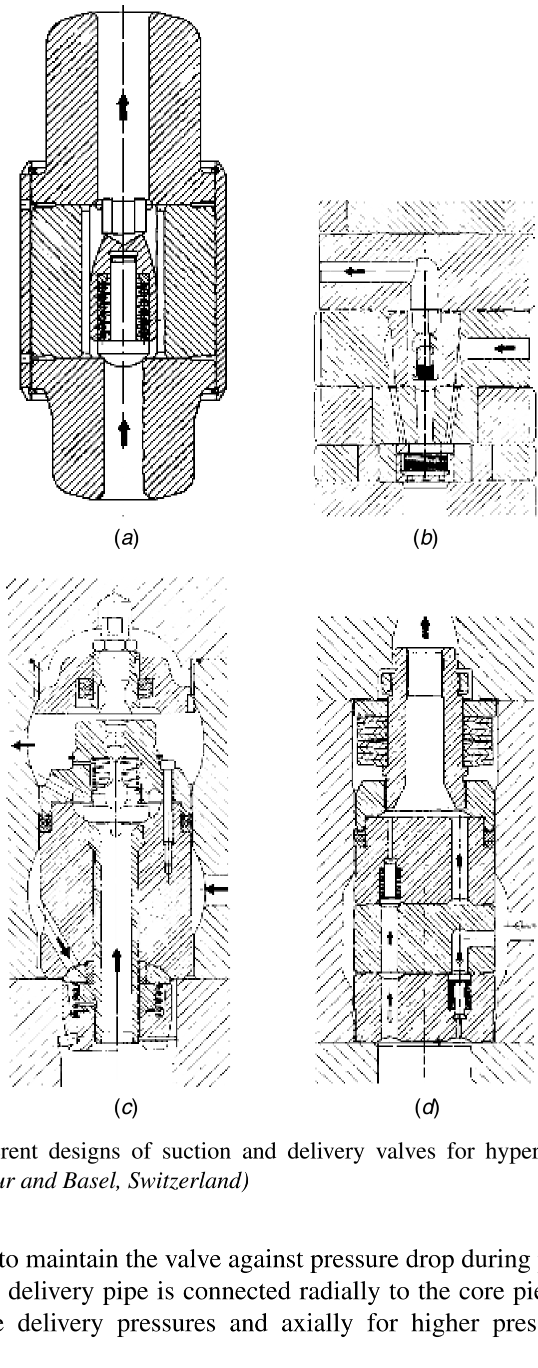

FIGURE 5.6 Different designs of suction and delivery valves for hypercompressors. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

the figure has only to maintain the valve against pressure drop during periods of operation on bypass. The gas delivery pipe is connected radially to the core piece (like the suction pipe) for moderate delivery pressures and axially for higher pressures (as shown in Fig. 5.5).

All components subjected to high stresses, particularly the internal cylinder elements under high tridimensional fatigue strains, are generally investigated at the design stage by three different methods. The first is a conventional calculation of combined stresses, based on the classical hypotheses, using computer programs as far as convenient. The second approach is that of the frozen stress technique of photo-elasticity applied on resin models cast either on full scale or on slightly reduced scale: It supplies accurate information about the course of the two main stresses in every plane section within the material. The third method is a direct measurement of the superficial stresses by means of strain gauges on the actual component subjected to the full prestressing and internal pressure. A variation of this last method consists of stress measuring on an enlarged model made of a low-modulus material such as aluminum: It provides better information through strain gauges on small rounded edges and allows progressive modification of such places in an attempt to reach an optimum. Comparison of results of these different methods gives a very useful reciprocal check on their exactness and accuracy.

5.4 DRIVE MECHANISM

Different types of driving mechanism are illustrated diagrammatically in Fig. 5.7. The first two (Fig. 5.7*a* and *b*) have been used extensively during the initial period of development and are still applied to smaller units. They are characterized by the fact that high-pressure cylinders have been fitted to frames of conventional design, without substantial modification of the existing equipment. Some manufacturers did not take into account the purely unilateral loading of the crosshead pins—and they generally got into trouble. Others tried to balance the forces by getting additional pistons set under constant or variable gas pressure in the reverse direction; this may work but is a rather unsatisfactory solution because it is expensive and introduces supplementary wearing elements. The best means of application is to use special high-pressure lubrication pumps, fastened to the crossheads and driven by the rocking movement of the connecting rods, which inject the oil directly into the crosshead bearings, thus lifting the pins against the load. This arrangement is well known from the design of large diesel engines, but because the requirements called for higher delivery pressures and larger capacities, the solutions shown in Fig. 5.7*a* and *b* appeared to be increasingly unsatisfactory. Since it is impossible to use double-acting pistons on very high pressures, these designs load the driving mechanism with the full gas pressure (instead of the difference between delivery and suction pressures) and are working only on each second stroke. Although this was still admissible for small units, it proved to be uneconomical for larger ones, and there was obviously a need for more specialized constructions.

The widespread design represented in Fig. 5.7*c* is still based on a conventional application of the horizontally opposed reciprocating compressor, but it avoids the foregoing difficulty by having, on each side of the frame, an external yoke that is connected rigidly to the main crosshead by means of solid connecting bars. A pair of opposed pistons (or plungers) is then coupled to each yoke, which is shaped as an outboard crosshead. Because of the long flexible connecting bars, the movement of the yoke is not disturbed by any transverse force, and it allows a full loading of the drive; thus, this is not a bad solution. However, the compressor is becoming extremely wide, and the accessibility of some of these high-pressure cylinders is rather poor.

All the other systems illustrated in the figure are specially designed solutions: Fig. 5.7*d* and *f* use a rocking beam bound to a fulcrum by a lever, which gives a linear translation of the rotary movement. Figure 5.7*e* uses a moving frame surrounding the crankshaft to connect the crosshead to the piston on the opposite side—a solution already applied for more than half a century to high-pressure pumps; and Fig. 5.7*g* is based on the idea of hydraulic transmission of the driving power. It should be noted that Fig. 5.7*d* may modify the stroke of the crankshaft in a fixed predetermined ratio, that Fig. 5.7*f* reduces the stroke in a fixed ratio, and that Fig. 5.7*g* can perform a variable reduction of the stroke. Although Fig. 5.7*e*

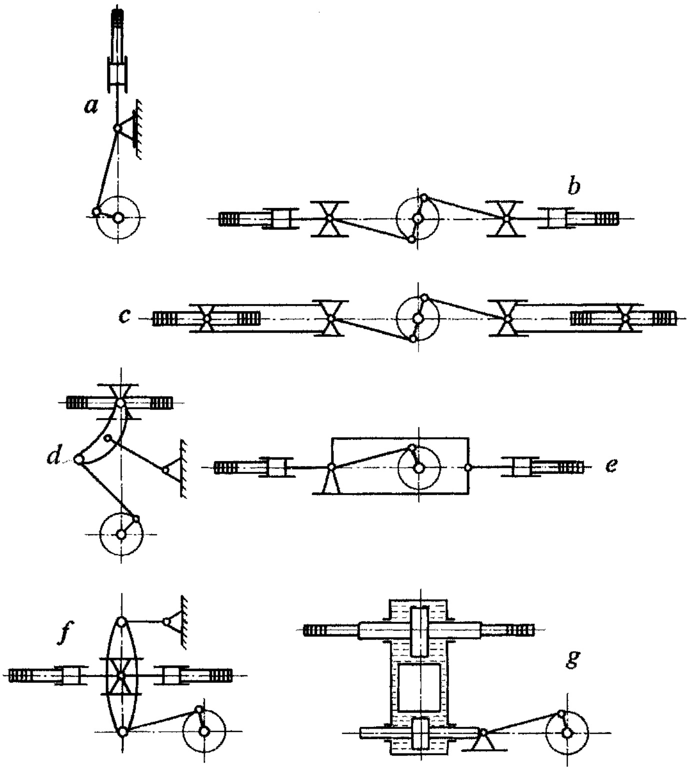

FIGURE 5.7 Various types of driving mechanisms. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

appears to be the best specific design for a large production compressor, the very special solution of Fig. 5.7*g* is worth further explanation.

Figure 5.8 shows greatly simplified diagram of a basic operation. By means of two reciprocating columns of fluid, a double-acting primary piston operates a secondary piston located above it. A pair of opposed high-pressure gas pistons are coupled to the latter. Although the hydraulic transmission of power could theoretically work as a closed system, it is actually necessary to renew the fluid continuously through forced-feed recirculation, both for the purpose of cooling and to compensate for possible seal leaks. Figure 5.8 shows a low-pressure feeding system; it has also been made as a high-pressure feed. Since this transmission may be built as a hydraulic intensifier, it is possible to use a comparatively light primary mechanism at rather high speed and to reduce the linear speed and increase the forces on the secondary part. Furthermore, by opening a bypass valve between the two fluid columns, the secondary stroke may be reduced. In this manner, stepless output control can be achieved down to zero. Because the fluid pressures on both sides of the pistons vary according to two opposed indicator diagrams, there are two points on each stroke where they will balance. If such a bypass is opened wide on the first of these points, the fluid will transfer theoretically without losses, and the secondary piston will stand still until the valve is closed. In fact, this is one of the very few ways of realizing power-saving capacity control of reciprocating compressors for very high pressures. This output control can be governed automatically, and applying it separately to each compression stage makes it possible to control the intermediate pressure exactly.

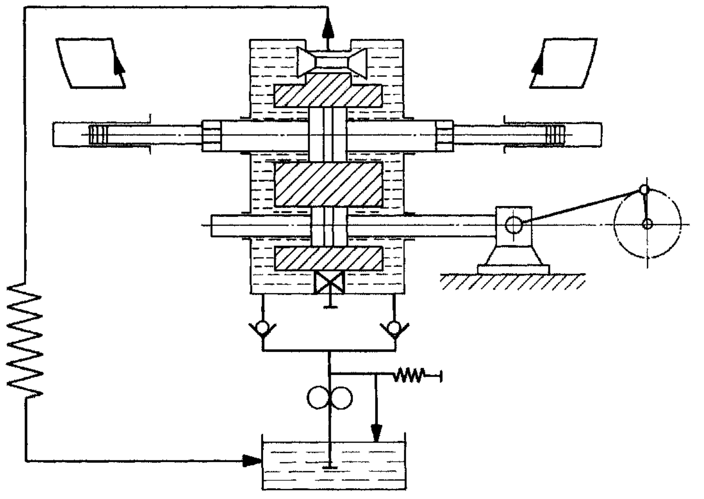

FIGURE 5.8 Hydraulic transmission of power. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

5.5 MISCELLANEOUS PROBLEMS

The number of problems posed by industrial reciprocating compressors for very high pressures is almost unlimited. Nearly every question of installation or maintenance needs a special study and an original answer, and all elements and accessories require special design and calculation. Only a few of them are mentioned here.

When designing large reciprocating compressors, it is common practice, to take into account the three different types of strains for selecting the most favorable crank-angle arrangement: (1) the resulting forces and moments of inertia acting on the foundations, (2) the resulting torque diagrams under different conditions of operation (important for the cyclic variations of current consumption of the driving motor), and (3) the forces due to pressure pulsations in the gas piping. For most compressors working at lower pressures, this last consideration may be deleted or answered in a summary way at the initial stage of design, because it may be solved by the use of surge drums. In the case of very high pressures, the gas pulsations, which are capable of destroying the piping system, have to be given first priority in the basic investigations, even if it sometimes leads to acceptance of higher-inertia forces.

The most practical ways of studying gas pulsations are to use either an analog computer, which is, in fact, an electroacoustic analogical system where every part is individually adjustable or replaceable, or alternatively, a digital computer study. The first purpose of the analysis is to avoid any resonance between the active systems (the compression cylinders) and the passive systems (the entire piping network); the second purpose is to reduce the amplitudes of the remaining pressure pulsations as far as possible. Theoretically, the means available are (1) change of diameter and of length of the gas piping, (2) removal of pipe connections or adjunction of additional piping, and (3) use of pulsation snubbers and of orifices at well-selected places. In reality, the possibilities are restricted because of the high speed of sound in the gas (1000 to 2000 m/s), because of the very low compressibility of the gas and the very high price of vessels and piping. However, in many cases it has proved easily possible to reduce a dangerous pulsation (e.g., from 25% down to 5%) by inexpensive and simple means.

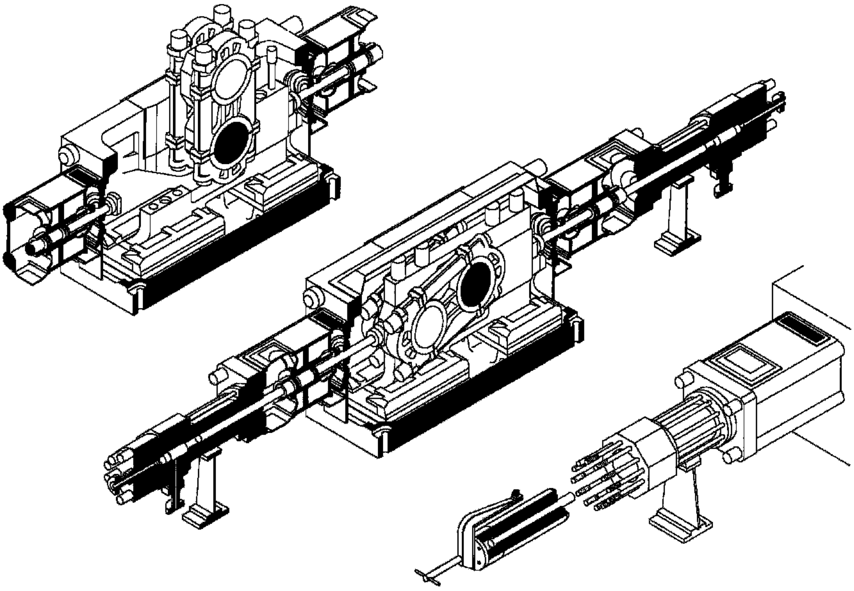

FIGURE 5.9 Large mechanically driven compressor for very high pressures (sectional views). *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

Designers dealing with compressors for very high pressures need to keep in mind at least three basic ideas: (1) safety, (2) large forces (how to apply them), and (3) accessibility. The last two are, of course, chiefly economic, but they are often combined with the aim of safety. For instance, in the design of large compression cylinders, as shown in Fig. 5.5, the long throughbolts connecting the base with the cylinder head are an important safety factor. If, by chance, the gas decomposed in the cylinder, these long bolts, acting as springs, would be elastically lengthened by an appreciable amount without increasing the stresses significantly and would allow the gas to escape between the liner and the head. They must all be equally pretensioned with a very high force. If done by hand, this would be an extremely tiring and time-consuming exercise, and for this reason a hydraulic piston has been incorporated within the cylinder head, which allows, when set under oil pressure, a very quick, easy, and regular tightening and loosening of the bolts. After removal of the outer flange, the entire inside of the cylinder can be removed with the help of a lifting device. It resembles a closed cartridge, as shown in Fig. 5.9. The same figure also shows how major parts of the driving mechanism may be dismantled without removal of the cylinders.

5.6 CONCLUSIONS

Although closely related to other reciprocating compressors, industrial compressors for very high pressures require the construction of a separate group of machines, different in many ways, and call for much greater research, development, and calculation than the others. Being compelled to employ all materials very near their limits of resistance, designers are bound to keep in close contact with the latest developments in material science and many related technologies.