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LABYRINTH PISTON COMPRESSORS

Labyrinth piston compressors represent a very important subset of nonlubricated reciprocating machines. Generally vertically oriented, they are typically configured as shown in Figs. 4.1 through 4.3. Virtually every one of the thousands of machines in service worldwide since 1935 has been manufactured by Sulzer-Burckhardt, now Burckhardt Compression AG, of Winterthur, Switzerland.

4.1 MAIN DESIGN FEATURES

The main design features are highlighted in Figs. 4.1 and 4.2. Labyrinth piston compressors do not use piston rings or rider bands. Unlike the oil-free reciprocating compressors of traditional design with dry-running piston rings, no friction occurs in the cylinder (1). The same applies, as a rule, for the piston rod stuffing box (4). Instead of piston rings, the labyrinth piston (2) is provided with a large number of grooves that generate a labyrinth-sealing action with regard to the cylinder wall. Although the cylinder wall is provided with grooves, they are finer than those of the piston.

The piston moves within the cylinder with a clearance so that even in the warm state, contact-free running is assured. Thermostats at the gas outlet detect any overheating that could lead to the piston scraping the cylinder wall. Thanks to this construction, lubrication of the cylinder and of the piston rod stuffing box is not necessary. Moreover, the suction and discharge valves are designed so that no lubrication is required.

The driving mechanisms of the compressors are usually lubricated by a gear pump (8) driven by the crankshaft. Depending on the requirement, compressors can be equipped additionally with an oil pump for preliminary lubrication (driven by an electric motor), an oil cooler, and a high-efficiency oil filter. The lubricating oil system supplies oil under pressure

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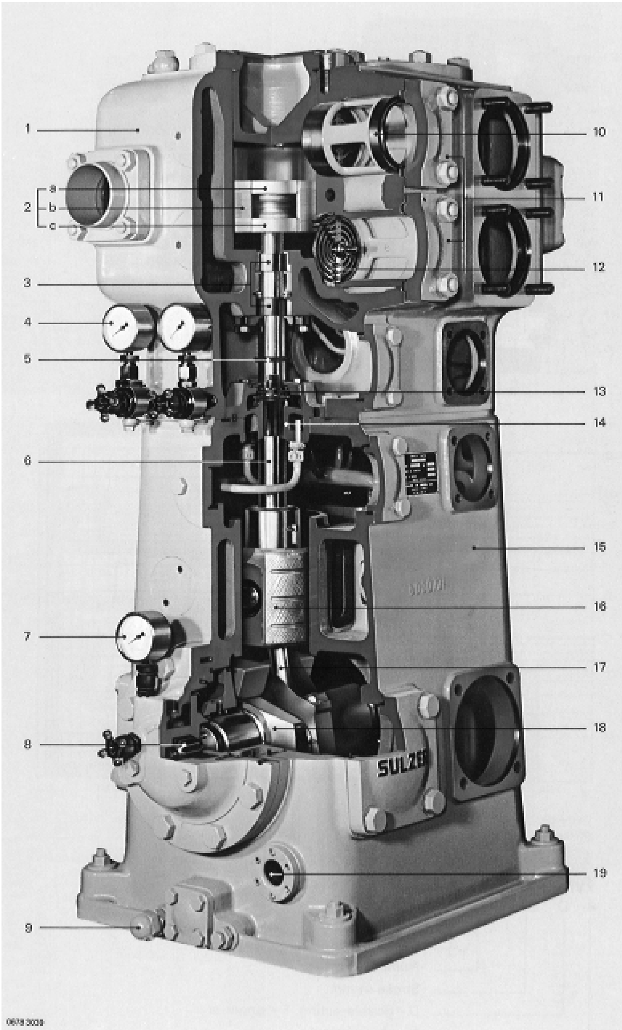

FIGURE 4.1 Labyrinth piston compressor. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

to the main bearings, the lower and upper connecting rod bearings, and the crosshead guide mechanisms. Splash oil lubrication is provided for the piston rod guide bearings (6).

When operating in the normal temperature range, both cylinder and crosshead guide mechanisms are water cooled. This applies also for the piston rod guide bearings. Where low suction temperatures apply, such as in refrigeration applications, cooling of these bearings can be dispensed with.

The piston is guided from outside the compression space by the piston rod, which is located with a relatively small clearance in the guide bearing (6) and by a precise guidance of the crosshead (7). The separation between the oil-free portion and the oil-lubricated crank drive is ensured by oil scrapers (6). To prevent the thin film of oil remaining on the piston rod from creeping upward along it, the rod is provided with an oil slinger. The distance between the drive mechanism and the stuffing box is greater than the length of the piston stroke, so that the oil-wetted portion of the piston rod cannot penetrate the oil-free stuffing box.

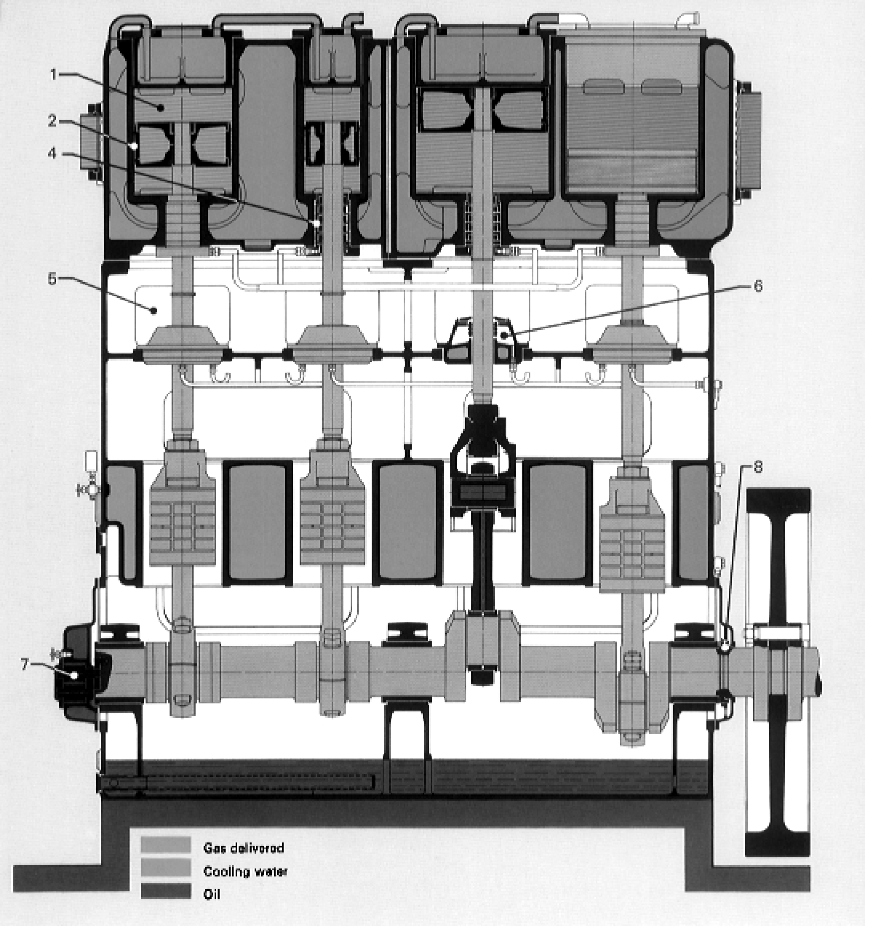

FIGURE 4.2 Longitudinal view of a three-stage labyrinth piston compressor. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

4.2 ENERGY CONSUMPTION

From time to time, the opinion is voiced that labyrinth piston compressors require more energy than dry runners with piston rings. Hence, a question is raised as to whether the energy losses occurring in the labyrinths between piston and cylinder and between piston rod and stuffing box are, in fact, greater than those that take place at these points in dry runners because of friction and (as a matter of fact, smaller) gas losses. Comprehensive tests have shown that an immediate loss of drive power occurs in the case of the ring-sealed piston because of the unavoidable mechanical friction of the piston rings. Moreover, this results in unfavorable wall temperature influences on energy requirement and suction capacity. In the case of labyrinth pistons, where no mechanical friction occurs within the cylinder, any power losses at the piston are to be explained principally by leakage losses.

The experience gained from labyrinth piston machines indicates that for average to large volumes swept by the piston and for gases that are not extremely light, the energy losses due to leakage along the labyrinth are approximately equal, in some cases even smaller than

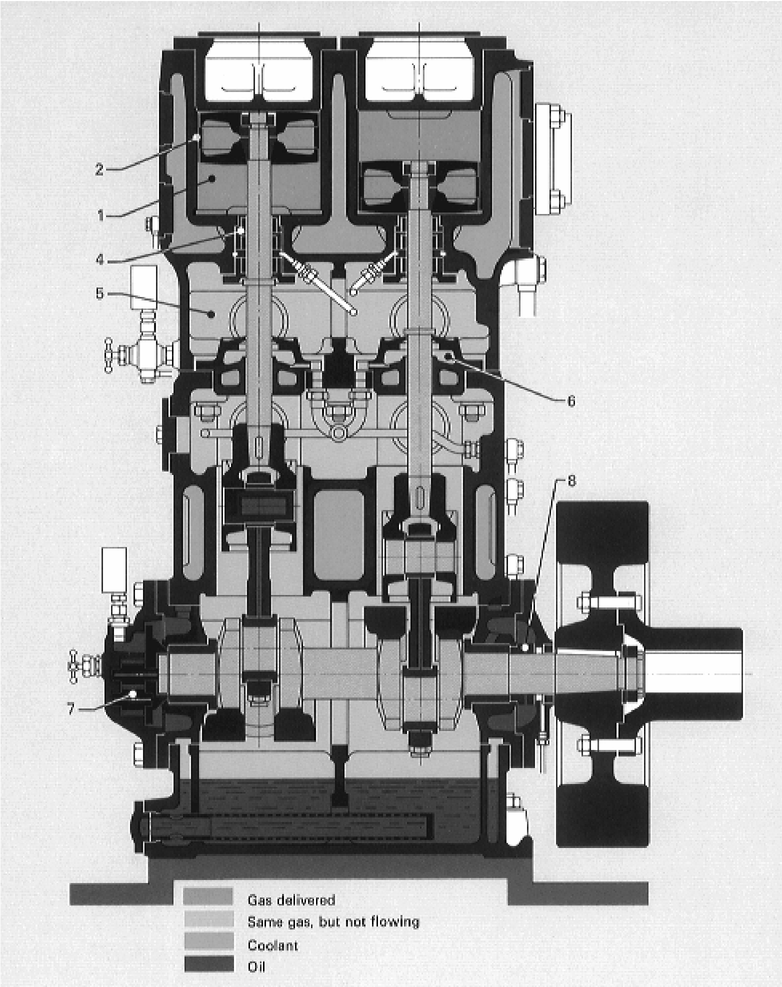

FIGURE 4.3 Longitudinal view of a single-stage labyrinth piston compressor with a completely encapsulated and pressure-tight crankcase. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

those that occur because of friction, sealing leaks, and wall temperature influences in piston ring machines. However, the situation of the labyrinth piston, when light gases and small volumes swept by piston are concerned, is less favorable. As a consequence of the slight quantitative difference for air and similar-density gases, the comparison measurements have to be carried out very carefully and while accurately maintaining the same external conditions.

Two cases are described later where such comparisons were established by precise experiment. Labyrinth and plastic piston ring structures were made for two single-stage double-acting cylinder and piston sets. These were incorporated in a vertical single-throw standard crankshaft motion gear with a nominal speed of 750 rpm. Both cylinders had precisely the same dimensions. The only difference was in the surface configuration of the bores. Whereas one piston was of the standard labyrinth type, the other carried three plastic rings. In both cases the same compressor valves, pipelines, measuring instruments, and drive elements were used for the measurements. Differences in operating characteristics could thus be reliably established without disturbing side effects.

As is usual in the compressor sector, the comparison between the two test series was made using the efficiencies:

$$\eta_{\text{adiabatic}} = \frac{P_{\text{adiabatic}}}{P_{\text{effective}}} \tag{4.1}$$

$$\eta_{\text{isothermic}} = \frac{P_{\text{isothermic}}}{P_{\text{effective}}} \tag{4.2}$$

These values are shown in Fig. 4.4 as a function of the pressure ratio, determined for three different speeds of rotation.

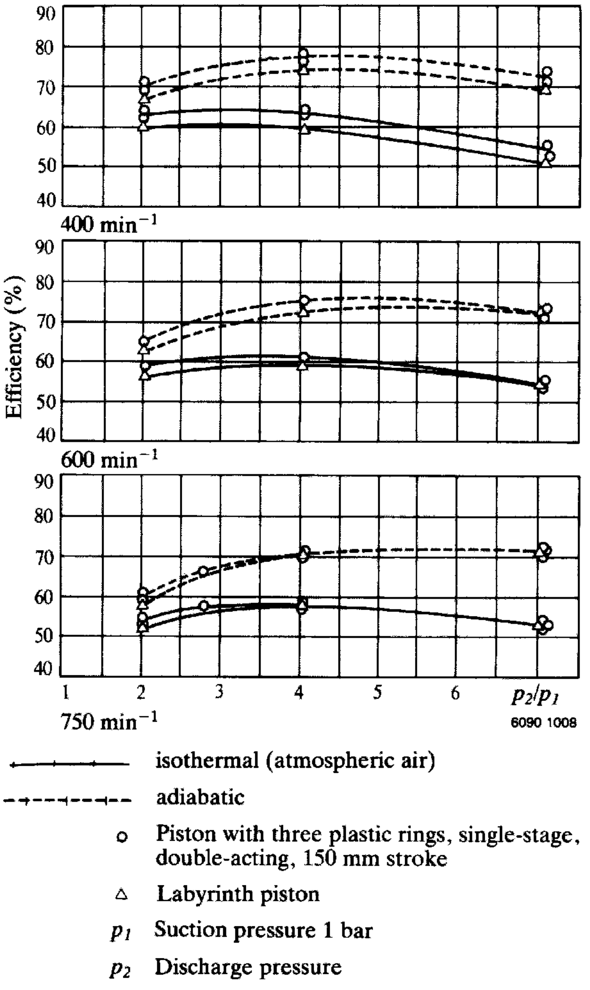

FIGURE 4.4 Efficiency comparisons of a conventional nonlubricated compressor and a labyrinth piston machine. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

4.3 SEALING PROBLEMS

Sealing questions are of interest as well. Logically, where a compressor is not fully encapsulated, the piston rod stuffing boxes have to seal the cylinder to the outside. Where these are oil lubricated, the lubricating oil acts as a sealing agent. Dry-running packing rings that rub on the piston rod are not as effective as seals, and friction-free labyrinth packings are even less so.

As far as most gases are concerned, sealing problems play a big role in the choice of the compressor design. The labyrinth stuffing box of the labyrinth piston compressor consists principally of a number of graphite rings; these are fitted to the piston rod with longitudinal and transverse clearance in the annular chamber. The rings are self-centering with regard to the piston rod. Labyrinth grooves on the inner surface provide the required sealing effect. Graphite is an ideal material for packing rings, since it has good dry-running characteristics, high chemical stability, low thermal expansion, and is not hygroscopic. Moreover, packing rings made from graphite cannot run hot.

One- or three-part designs of graphite rings are used. Three-part rings, whose constituent parts are held together radially by two garter springs, have the advantage that they can be replaced without the piston and the piston rod having to be pulled. In addition, they can be reworked if they become somewhat worn on the inner surface. The purpose of these springs is not, however, to press the three parts of the ring onto the piston rod, but to facilitate fitting and dismantling. These rings also have clearance with respect to the piston rod.

The advantages of the pure labyrinth stuffing boxes during operation are so convincing that the designer accepts the unavoidable losses via the labyrinth and deviates from the friction-free principle only in special cases. These labyrinth losses are taken up in the lower part of the stuffing box and returned to the suction side of the compressor, so that as far as the environment is concerned, no or only negligible gas losses occur. However, the higher the value of suction pressure above atmospheric, the greater the gas losses through the lowest sealing elements. To keep these losses as low as possible, the lowest ring can be designed as a sliding contact-sealing element. Such a sliding ring is in three parts, smooth on the inner surface, and pressed lightly against the piston rod by garter springs. Special arrangements have been developed for higher suction pressures.

A machine design as illustrated in Fig. 4.5 can be used for the compression of gases that are neither poisonous nor flammable, such as air, carbon dioxide, oxygen, and nitrogen. Slight leakage of such gases to the outside can be tolerated. However, this does not apply for helium and argon. Although inert and nonpoisonous, their loss is not acceptable because of the high price.

Machines as illustrated in Fig. 4.6 are used for gases that are poisonous, flammable, and incompatible with lubricating oil. These have special stuffing boxes with gas sealing (mainly nitrogen), since strict separation between crankcase and distance piece (4) is absolutely necessary. The adaptor is flushed out with scavenging gas, and the crank mechanism is filled either with air or with scavenging gas.

Completely encapsulated compressors (Figs. 4.7 and 4.8) are used for gases that are compatible with lubricating oil but have characteristics that do not allow even the smallest amount of loss to the outside (e.g., all hydrocarbons, carbon monoxide, hydrogen, helium, and argon). The machine shown in Fig. 4.8 is of special pressure-resisting design—the K series. Originally, these machines were developed as refrigeration compressors (K is the abbreviation in German for *refrigeration*), so that they are also suitable for the compression of all refrigerants.

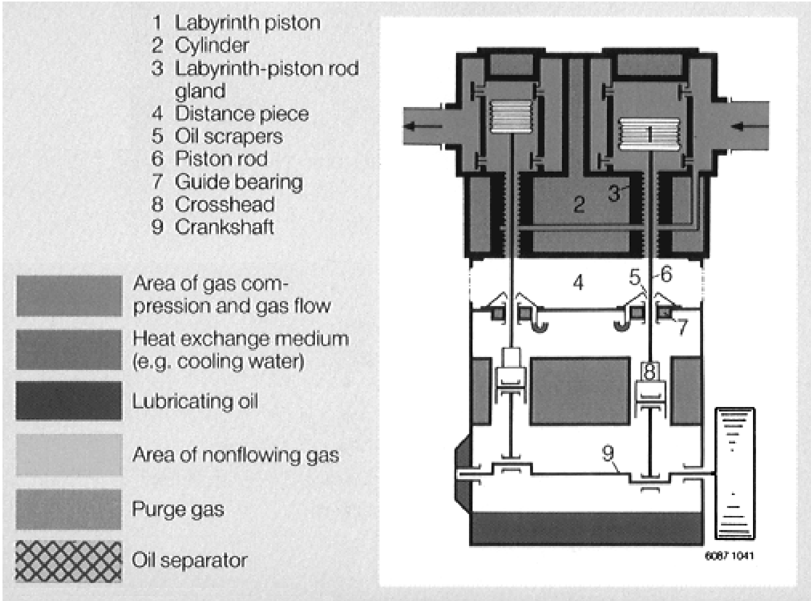

FIGURE 4.5 Labyrinth piston compressor with an open distance piece and a nonpressurized crankcase. Typically used for compression of gases, where strict separation between cylinder and crankcase is essential and where process gas is permitted in the open distance piece (e.g., for O_2, N_2, CO_2, process air; generally in the industrial gas industry). *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

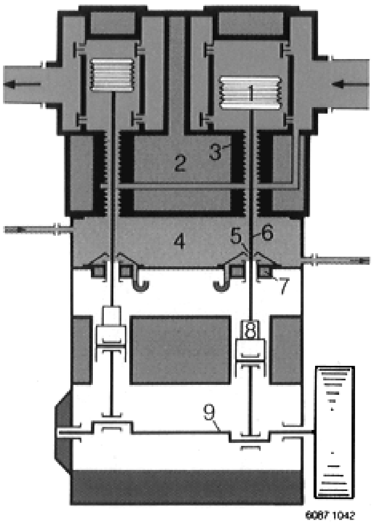

FIGURE 4.6 Labyrinth piston compressor with a closed and purged distance piece. Used for compression of gases, where a strict separation between cylinder and crankcase is essential and where no process gas may leak to the surroundings or no ambient air may enter the distance piece (e.g., for weather protection). *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

Attention has to be paid to the standstill pressure where closed refrigeration circuits are concerned. The crank mechanism of these compressors is thus designed for at least 15 bar internal pressure. For certain gases, however, the solubility in lubricating oil imposes a lower limit on the permissible internal pressure. The machines in Figs. 4.7 and 4.8 each feature mechanical seals. Unlike the pressure-resisting crankcase shown in Fig. 4.8, the

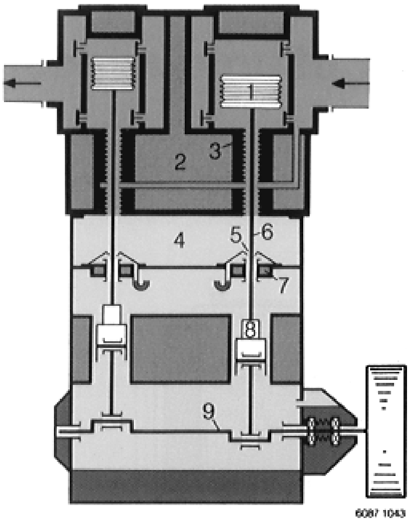

FIGURE 4.7 Labyrinth piston compressor with a gastight crankcase and a mechanical crankshaft seal. This design is used for compression of gases that are compatible with the lubricating oil (e.g., for hydrocarbon gases, CO, He, H_2, Ar) and where no process gas may leak to the surroundings. The suction pressure is limited by the design pressure of the crankcase. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

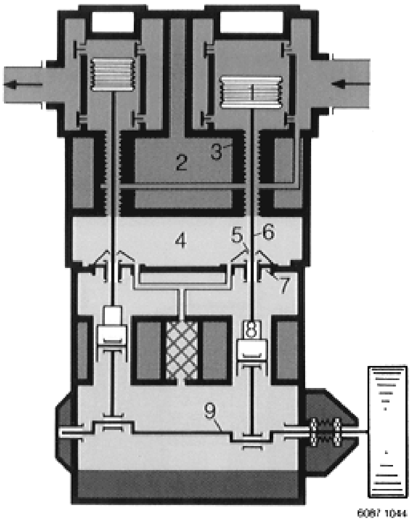

FIGURE 4.8 Labyrinth piston compressor with a gastight and pressure-tight crankcase and a mechanical crankshaft seal. Used to compress gases that are compatible with the lubricating oil and where no process gas may leak to the surroundings. Suction pressure may range between subatmospheric and crankcase design pressure. This machine finds its applications in closed cycles, for hydrocarbon gases, refrigerants, VCM, CO, N_2, CO_2, He, H_2, Ar, etc. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

crankcase in Fig. 4.7 can only accept a low internal pressure. In closed machines, the gaseous medium usually fills the crankcase, where it can mix with the oil mist.

In nongastight machines, there is no danger that oil penetrates into the oil-free zone of the compressor since the pressure in the crankcase is very low. However, where pressure-resisting machines are concerned, gas flow from the crankcase into the cylinder part has to be expected if the suction pressure decreases. Such a flow, nevertheless, passes through the oil separator shown in the center of the crankcase (Fig. 4.8), where the oil mist is retained. An external pressure balancing line (with molecular sieve) can be used, if required, to improve the separation effect still further. Hence, this also can be regarded as an oil-free functioning configuration. Very low leakage rates can be achieved with the pressure-resisting machine design by virtue of a purpose-oriented construction (e.g., baseplate and frame as a one-piece casting, round frame openings with O-ring seals) in conjunction with an

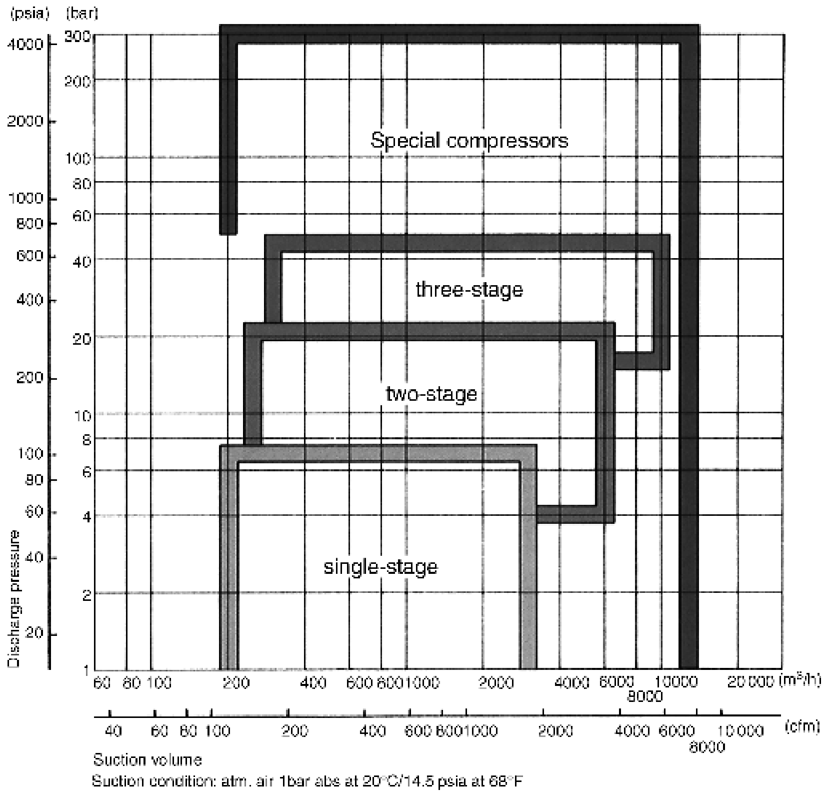

FIGURE 4.9 Typical application limits for labyrinth piston compressors. *(Sulzer-Burckhardt, Winterthur and Basel, Switzerland)*

especially carefully accomplished casting process. At machine standstill, these leakage rates for helium are in the range 10^{-3} to 10^{-4} cm^3/s. In many cases, even a leakage rate of 10^{-1} cm^3/s will meet the requirements. Special procedures have been developed to confirm such low leakage rates.

Labyrinth piston compressors are made in at least 40 frame sizes, with one, two, three, four, and six cranks for piston strokes of 65 to 375 mm and matching cylinders for one-, two-, three-, and four-stage compression. Available suction capacities range from 20 to 11,000 m^3/h and discharge pressures of up to 300 bar (Fig. 4.9). The permissible power input for the driving mechanism ranges from 20 to somewhat more than 2000 kW. The loadability of the crank mechanism is, however, limited by the permissible loading of the piston rod.

Since no oil can penetrate the cylinder and no temperature-sensitive materials are used, comparatively high compression ratios and final temperatures up to more than 200°C are possible. This means that in many cases, one compression stage less is required than would be necessary for compressor designs with plastic piston rings. It should be noted that the stage pressure ratios are often limited because of safety aspects (e.g., when compressing oxygen) and energy consumption considerations.