9

ROTARY SCREW COMPRESSORS AND FILTER SEPARATORS

9.1 TWIN-SCREW MACHINES

Rotary screw compressors are typically configured as shown in Figs. 9.1 through 9.4. Two counterrotating helical screws are arranged in a compressor casing; gas inlet and discharge nozzles are at opposite ends. Three-, four-, and five-lobe rotors are produced (Fig. 9.5).

9.1.1 Working Phases

The screw compressor is a positive displacement machine and as such has distinct working phases: suction, compression, and discharge. We will limit our description of the working phases to just one lobe of the male rotor and one interlobe space in the female rotor. Once the operation is understood, it is not particularly difficult to envision the relative interaction of all of the lobes and interlobe spaces with resulting uniform, basically nonpulsating, continuous gas flow through the compressor.

The suction phase is depicted in Fig. 9.6*a*. As the lobe of the male rotor begins to unmesh from an interlobe space in the female rotor, a void is created, and gas is drawn in through the inlet port. As the rotors continue to turn, the interlobe space increases in size, and gas flows continuously into the interlobe space. The inlet port is large, and the filling takes place over a large portion of each rotation. Just prior to the point at which the interlobe space leaves the inlet port of the suction end, the entire length of the interlobe space is open from end to end—the lobes and interlobe space being completely unmeshed. The interlobe space is thus completely filled with drawn-in gas.

The transfer phase is a transitional phase between suction and compression where the trapped pocket of gas within the interlobe space is isolated from inlet and outlet ports and is merely transported radially through a fixed number of degrees of angular rotation at constant suction pressure.

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

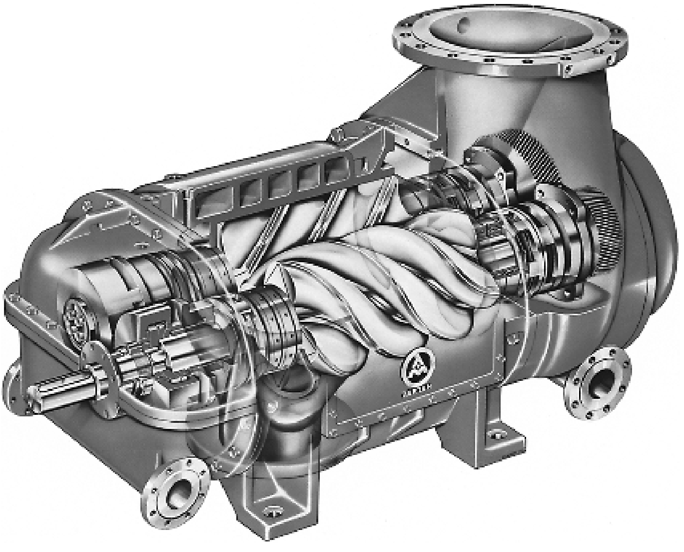


FIGURE 9.1 Rotary screw compressor (double-helical-screw machine). *(Aerzen USA Company, Coatesville, Pa.)*

FIGURE 9.2 Small packaged rotary screw compressor. *(Aerzen USA Company, Coatesville, Pa.)*

Figure 9.6*b* shows the compression phase. As can be seen, further rotation meshes a male lobe (not the same lobe as disengaged previously because of the 4 : 6 relationship) with the gas-filled interlobe space on the suction end and compresses the gas in the direction of the discharge port. The meshing point moves axially from the inlet to the discharge end; thus, the occupied volume of the trapped gas within the interlobe space is decreased and the gas pressure consequently increased.

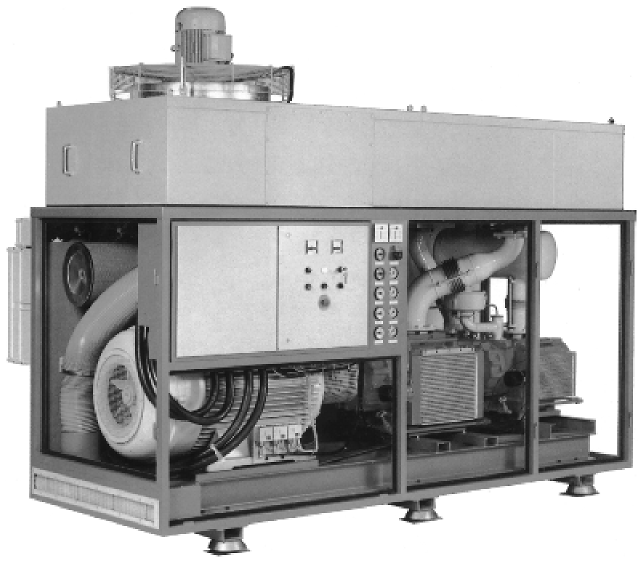


FIGURE 9.3 Medium-sized rotary screw compressor package. *(Aerzen USA Company, Coatesville, Pa.)*

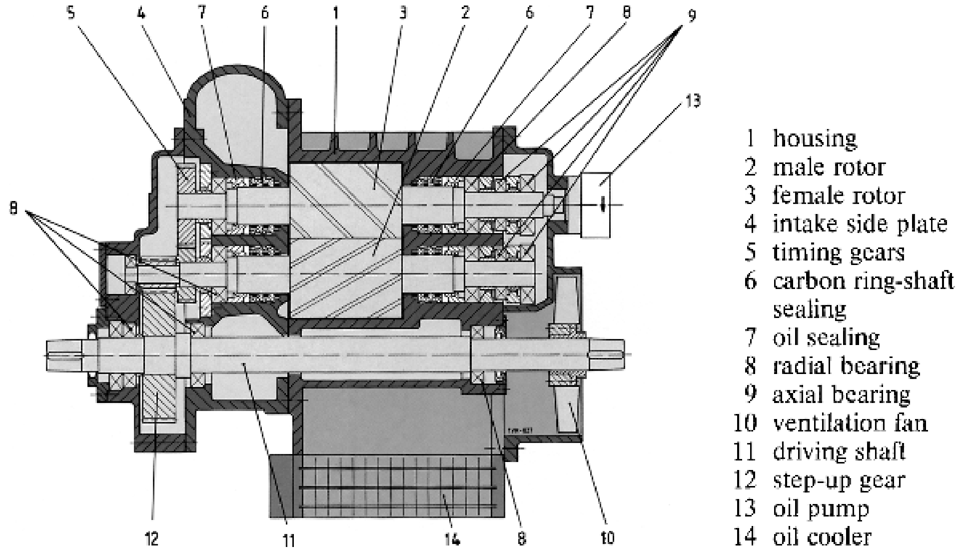


FIGURE 9.4 Oil-free rotary screw compressor with integral step-up gears. *(Aerzen USA Company, Coatesville, Pa.)*

The discharge phase is illustrated in Fig. 9.6*c*. At a point determined by the designed built-in compression ratio, the outlet port is uncovered, and the compressed gas is discharged by further meshing of the lobe and interlobe space. While the meshing point of a pair of lobes is moving axially, the next charge is being drawn into the unmeshed portion, and thus the working phases of the compressor cycle are repeated.

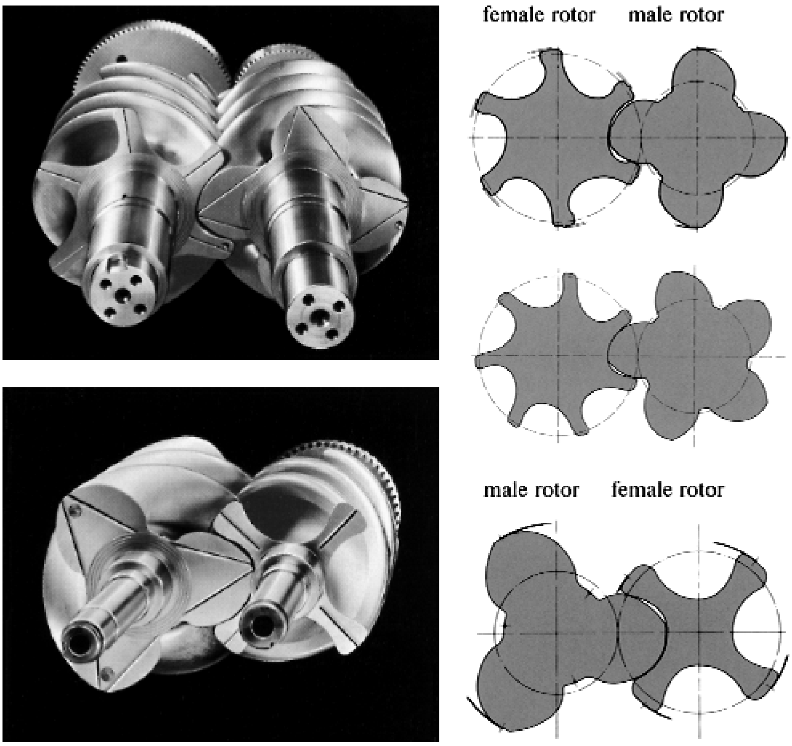
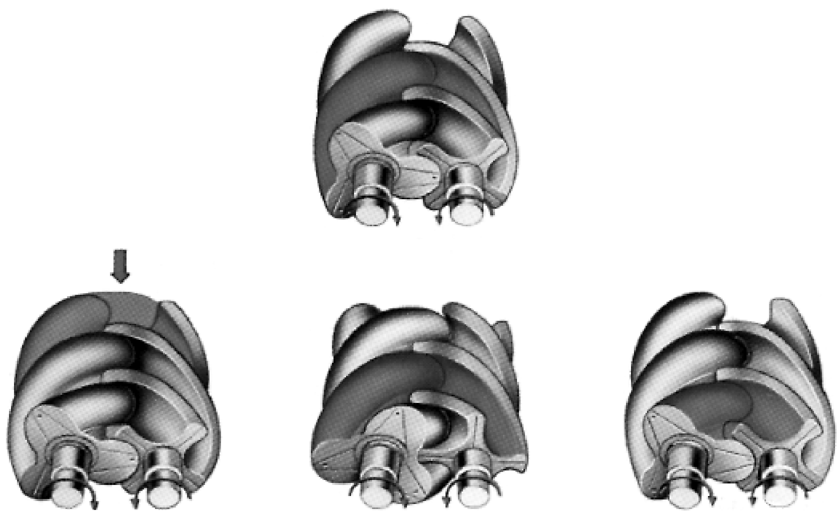


FIGURE 9.5 Typical screw compressor rotor combinations. *(Aerzen USA Company, Coatesville, Pa.)*



(*a*) (*b*) (*c*)

Suction intake Gas enters through the intake aperture and flows into the helical grooves of the rotors which are open.

Compression process As rotation of the rotors proceeds, the air intake aperture closes, the volume diminishes and pressure rises.

Discharge The compression process is completed, the final pressure attained, the discharge commences.

FIGURE 9.6 Working phases of rotary screw compressors. *(Aerzen USA Company, Coatesville, Pa.)*

9.1.2 Areas of Application

Rotary screw compressors have been around for many decades and are very likely the equipment of choice for either oil-free or oil-wetted compression of air in mining, construction, industrial refrigeration, and a host of other applications where their relative simplicity, general reliability, and high availability are appreciated.

What is less well known is that rotary screw machines are equally suited to compress such process gases as ammonia, argon, ethylene, acetylene, butadiene, chlorine, hydrochloric gas, natural and synthetic pipeline gases, flare gas mixtures, blast furnace gas, swamp and biomass gases, coke oven or coal gas, carbon monoxide, town gas, methane, propane, propylene, flue gas, crude or raw gas, sulfur dioxide, nitrous oxide, vinyl chloride, styrene, and hydrogen.

Modern sealing and liquid injection technology has been partly responsible for making rotary screw units capable of competing in applications previously reserved for other compressor types. As a result of sophisticated contour machining and enhanced metallurgy, single- or multistage rotary screw compressors today cover a range of suction volumes from 300 to 60,000 std m^3/h (176 to 35,310 scfm), with discharge pressures up to 40 bar (580 psi). For vacuum applications, an absolute pressure of 0.09 bar (1.3 psia) is achievable.

9.1.3 Dry vs. Liquid-Injected Machines

Two slightly different types of rotary screw compressors can be employed in process plants: dry machines and wet liquid-injected units. Liquid-injected rotary screw compressors are further divided into oil-injected machines and machines using other liquids.

Dry compressors typically use shaft-mounted gears to keep the two rotors in proper mesh. Prevalent in the pharmaceutical and high-purity chemical industries, these machines may also be used in aeration services in the brewing industry and other applications where complete absence of entrained air and other contaminants is mandatory.

Oil-injected rotary screw compressors are generally supplied without timing gears. Other liquid-injected compressors usually require gearing to keep the two counter-rotating screws in the proper mesh. The injected liquid could be water, a heat-removing fluid, or some other liquid. In oil-injected machines, the lubricant provides a layer separating the two screw profiles even as one screw drives the other. All liquid-injected machines offer the following advantages:

- The liquid injected provides internal cooling. Certain gases are thus kept from polymerizing or from operating in an explosion-prone temperature range
- Compared to their dry counterparts, these units achieve considerably higher compression ratios. This capability is, in part, attributable to the fact that in many services, liquid-injected machines do not require seals between the rotor chamber and the bearings. This reduces the bearing span, and therefore the rotor deflection. For example, a single liquid-injected compressor stage can do the job of two or more stages of dry compression.

9.1.4 Operating Principles

High-performance screw compressors use a twin-shaft rotary piston to combine positive displacement with internal compression (Figs. 9.5 and 9.6). Gas entering at the suction flange is conveyed to the discharge port and entrapped in continuously diminishing spaces between the convolutions of the two helical rotors. The result is compression of the gas to the final pressure before it is expelled via the discharge nozzle.

The position of the edge of the outlet port determines the inherent or built-in volume ratio, v_i, which is the ratio of the volume of a given mass of gas at the discharge and suction ports. The corresponding built-in compression ratio, π_i (i.e., gas pressure at discharge over the pressure at the suction port) is calculated using the following equation:

$$\pi_i = v_i^k \tag{9.1}$$

where k is the ratio of specific heats of the gas at constant pressure and volume, respectively.

The compression process is shown in the theoretical pressure–volume diagram (Fig. 9.7). A rotary screw compressor is designed for an anticipated compression ratio, π_i. If the machine discharges into a receiver with a compression ratio in excess of π_i, the compressor end wall will be exposed to that pressure.

When operated at compression ratios higher than the designed value, a centrifugal compressor is likely to undergo *surging* or periodic reverse flow, causing significant decline in machine performance. The screw compressor, on the other hand, is subject only to the constraints of machine component strength and input power. Thus, it can easily produce the increased compression ratio or discharge pressure. Rotary screw compressors can also accommodate less than built-in compression ratios. In this case, however, some efficiency will be sacrificed. These efficiency losses are identified as shaded areas in Fig. 9.7.

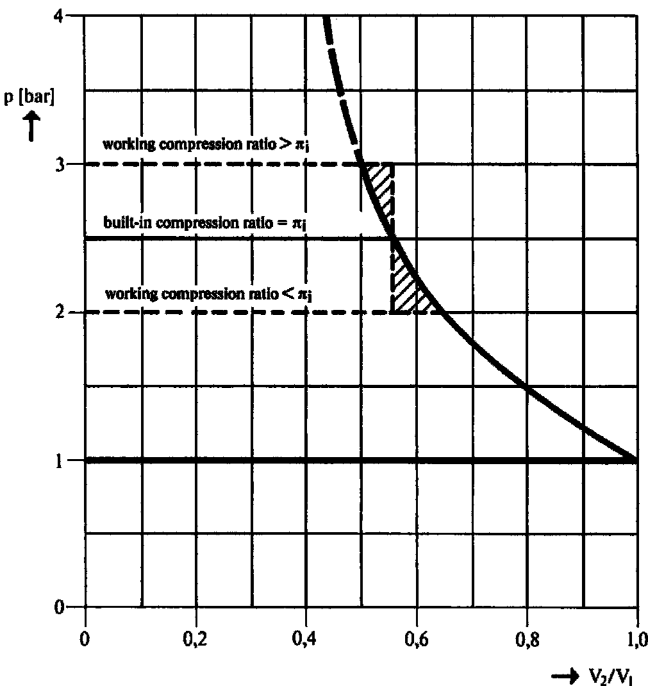


FIGURE 9.7 The *p–V* diagram of a modern helical screw (rotary screw) compressor. *(Aerzen USA Company, Coatesville, Pa.)*

9.1.5 Flow Calculation

The induced volume flow of the gas may be calculated from any compression ratio if the data applicable to the particular compressor being considered are known. One revolution of the main helical rotor conveys the unit volume q_0, L/rev. From this, one calculates the theoretical volume flow, Q_0, in std m^3/min for the compressor running at n rpm as follows:

$$Q_0 = \frac{nq_0}{1000} \tag{9.2}$$

The actual volume flow Q_a is lowered by the amount of gas Q_v flowing back through the very small clearances between machine components. Thus,

$$Q_a = Q_0 - Q_v \tag{9.3}$$

Q_v (also known as the volume flow lost via component slippages) is mainly dependent on the following factors:

- Total cross section of clearances
- Density of the medium handled
- Compression ratio
- Peripheral speed of rotor
- Built-in volume ratio

9.1.6 Power Calculation

The volumetric efficiency, η_v, is expressed as

$$\eta_v = \frac{Q_a}{Q_0} = 1 - \frac{Q_v}{Q_0} \tag{9.4}$$

The theoretical power input, W_0 (in kilowatts) required to compress the induced flow volume Q_a is given by

$$W_0 = \frac{10^{-3}}{60} \rho_a Q_0 H_a \tag{9.5}$$

where ρ_a, expressed in kg/std m^3, is the gas density at inlet conditions; and H_a represents the amount of energy required for the adiabatic compression of 1 kg of gas from pressure P_1 to P_2.

Alternatively, the theoretical power input could be obtained from

$$W_0 = \left(\frac{10^4 Q_a P_1}{6000}\right)\left(\frac{k}{k} - 1\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \tag{9.6}$$

where Q_a is expressed in std m^3/min and P_1 is in bar.

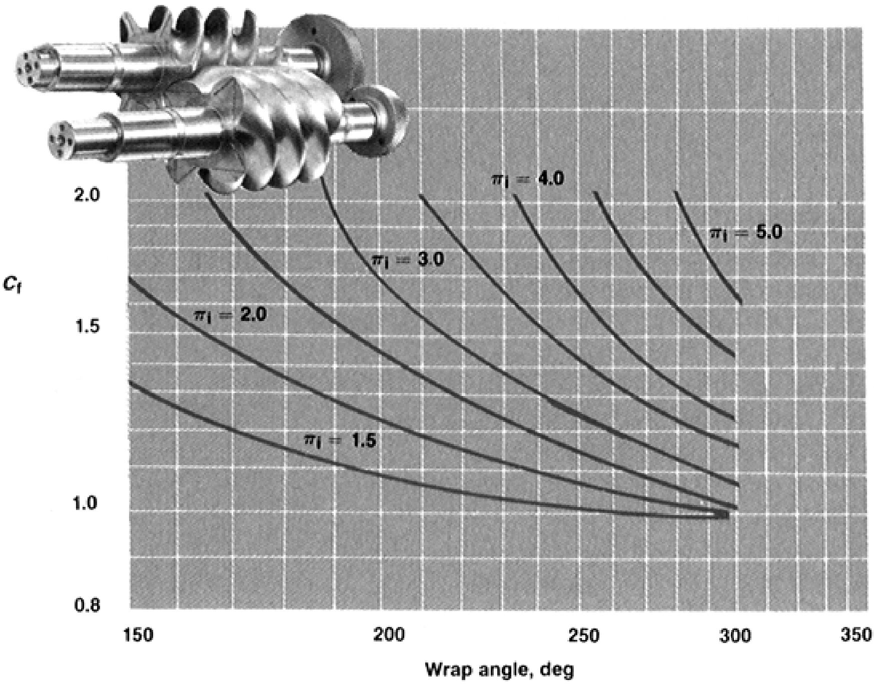


FIGURE 9.8 Empirical loss factor C_f as a function of the compression ratio π_i and the wrap angle of screw compressor rotors. *(Aerzen USA Company, Coatesville, Pa.)*

In practice, the theoretical power input is just a part of the actual power, W_a, transmitted through the compressor coupling. W_a should include the dynamic flow loss, W_d, and the mechanical losses, W_v. The mechanical losses—typically amounting to 8 to 12% of the actual power—refer to viscous or frictional losses due to the bearings, the timing, and step-up gears.

The dynamic flow losses typically amount to 10 to 15% of the actual power. A crucial factor in determining these losses is designated N_{id}, which is a function of the built-in compression ratio and the Mach number (ratio of gas velocity over the velocity of sound) at compressor inlet conditions. One can use the following formula to estimate dynamic flow power loss:

$$W_d = C_f \frac{L}{D} \frac{k}{1.4} \frac{P_1}{1.013} \left(\frac{Q_0}{60} N_{id} \right) \tag{9.7}$$

where C_f is an empirical factor (obtained from Fig. 9.8), L is the rotor length, D is the rotor diameter, and N_{id} is another empirical factor (obtained from Fig. 9.9). The reference conditions assumed in Eq. (9.7) and the associated charts are $Q_0 = 60\,\text{std m}^3/\text{min}$, $L/D = 1.0$, wrap angle described by a point on the thread of a screw as the point travels from the bottom to the top of the rotor (inset in Fig. 9.9) = 300°, $P_1 = 1.013$ bar, and $k = 1.4$ (corresponding to air).

Thus, the actual power requirement for the compressor is given by

$$W_a = W_0 + W_d + W_v \tag{9.8}$$

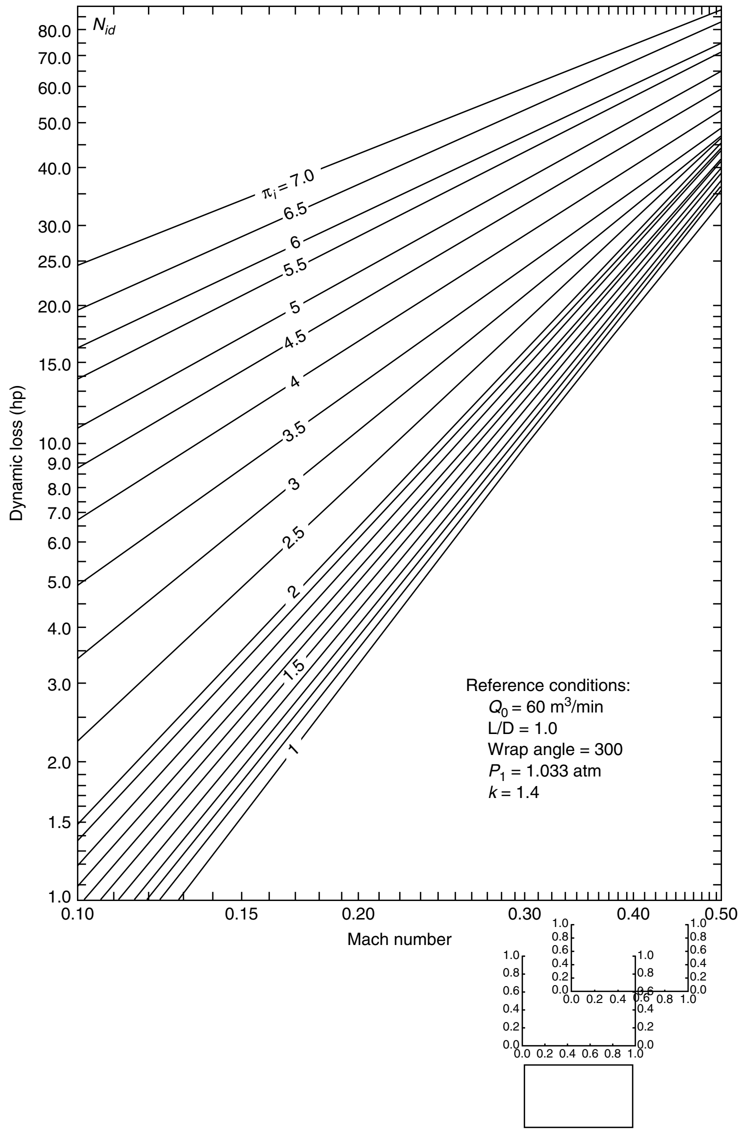


FIGURE 9.9 Empirical loss factor N_{id} vs. Mach number at different compression ratios π_i. The effect of compressor inlet conditions on the dynamic-flow power losses is shown as a function of the Mach number of the gas. *(Aerzen USA Company, Coatesville, Pa.)*

In North America, screw compressors for important process applications are typically built in compliance with the American Petroleum Institute (API) Standard 619. No negative tolerance is permitted on capacity, and the power requirement may not exceed the stated horsepower by more than 4%.

Screw compressors made in Europe can easily comply with this requirement. However, their customary Verein Deutscher Ingenieure (VDI; Society of German Engineers) Specification 2045 would allow a different margin of deviation to accommodate tolerances resulting from the usual operational limits of the manufacturing process.

9.1.7 Temperature Rise

For a dry compressor, the temperature (in °C) of the compressed gas at final compression is calculated as follows:

$$\Delta T_0 = \frac{T_1}{\eta_v}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \tag{9.9}$$

$$T_2 = T_1 + \Delta T_0 \tag{9.10}$$

When operating under oil-free, dry-running conditions, a screw compressor may come up to a maximum final compression temperature of 250°C. When air is the compressed medium, this temperature (with adiabatic exponent $k = 1.4$) corresponds to a compression ratio $P_2/P_1 \approx 4.5$. On the other hand, within the same temperature limits, gases with $k = 1.2$ will permit a compression ratio as high as 7.0.

In an oil-injected screw compressor (Figs. 9.10 and 9.11), most of the heat of compression is carried away by the oil. The amount of oil injected is adjusted to ensure that final discharge temperatures do not exceed 90°C (194°F). If air is taken in under atmospheric pressure, compression ratios as high as 21 are obtainable.

9.1.8 Capacity Control

Because screw compressors are positive displacement machines, the most advantageous method of achieving capacity or volume flow control is that obtained by variable speed. This may be done by variable-speed electric motors, a torque converter, or a steam turbine

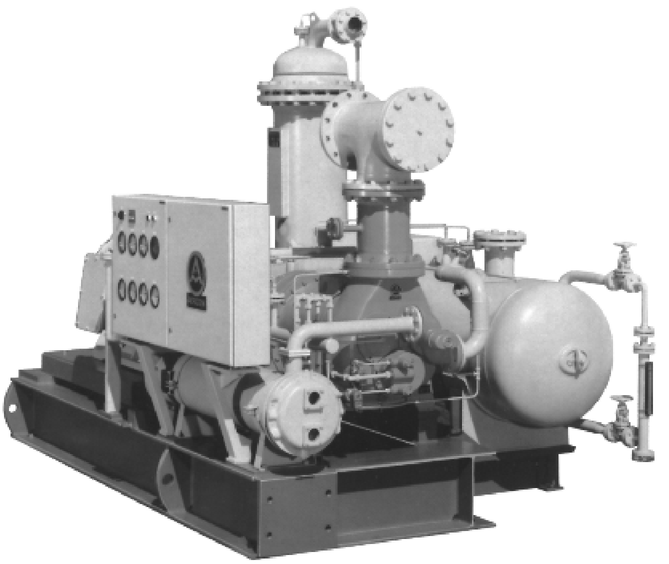


FIGURE 9.10 Oil-injected rotary screw compressor. *(Aerzen USA Company, Coatesville, Pa.)*

drive. Speed may be reduced to about 50% of the maximum permissible value. Induced volume flow and power transmitted through the coupling are thus reduced in about the same proportion.

Another method of capacity control is by using a bypass, in which the surplus gas is allowed to flow back to the intake side by way of a device that is controlled by the allowable final pressure. An intermediate cooler reduces the temperature of the surplus gas down to the level of the inlet temperature.

Use of a full-load and idling-speed governor is yet another means of capacity control. In this mode, as soon as a predetermined final pressure is attained, a suitable transducer operates a diaphragm valve (Fig. 9.12) that opens up a bypass between the discharge and suction sides of the compressor. When this occurs, the compressor idles until pressure in the system drops to a predetermined minimum value. This will cause the transducer to initiate closure of the diaphragm valve, and the compressor will again be fully loaded.

Control by suction throttle and discharge unloading is particularly suitable for air compression in the manufacture of industrial gases. As in the case of the full-load and idling-speed control method, a predetermined maximum pressure in the system (e.g., in a compressed-air receiver) causes pressure on the discharge side to be relieved down to atmospheric pressure. At the same time, the suction side of the system is throttled down to about 0.15 bar (2.2 psia). When pressure in the entire system drops to a predetermined minimum value, full load is restored.

Because the temperature at the final compression stage is governed by the injected oil, it is possible to operate an oil-flooded screw compressor over a wider range of compression

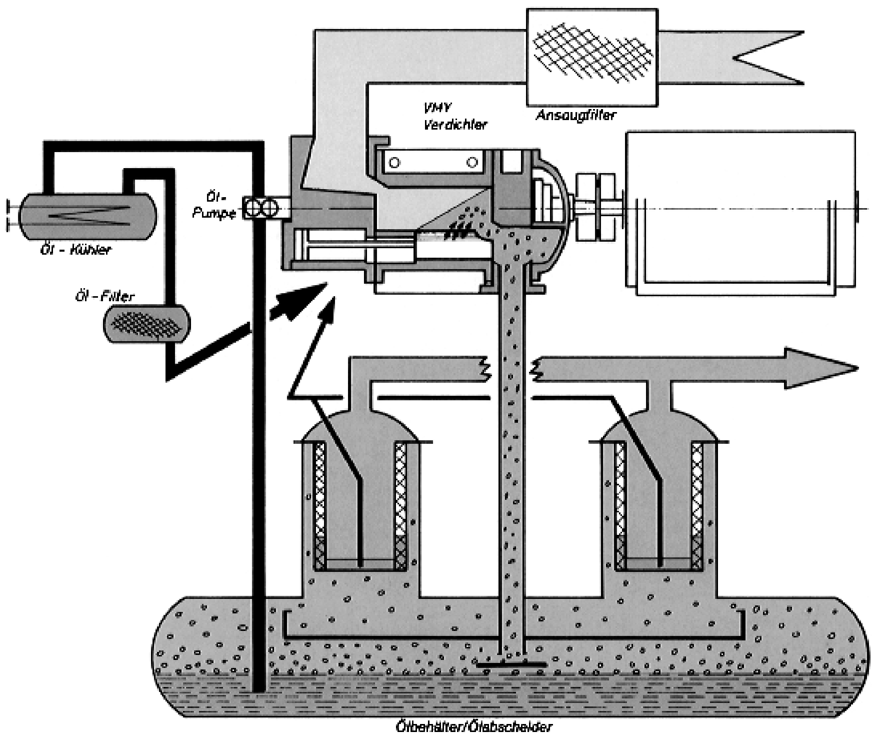


FIGURE 9.11 Operating principle of oil- or liquid-injected rotary screw compressors. *(Aerzen USA Company, Coatesville, Pa.)*

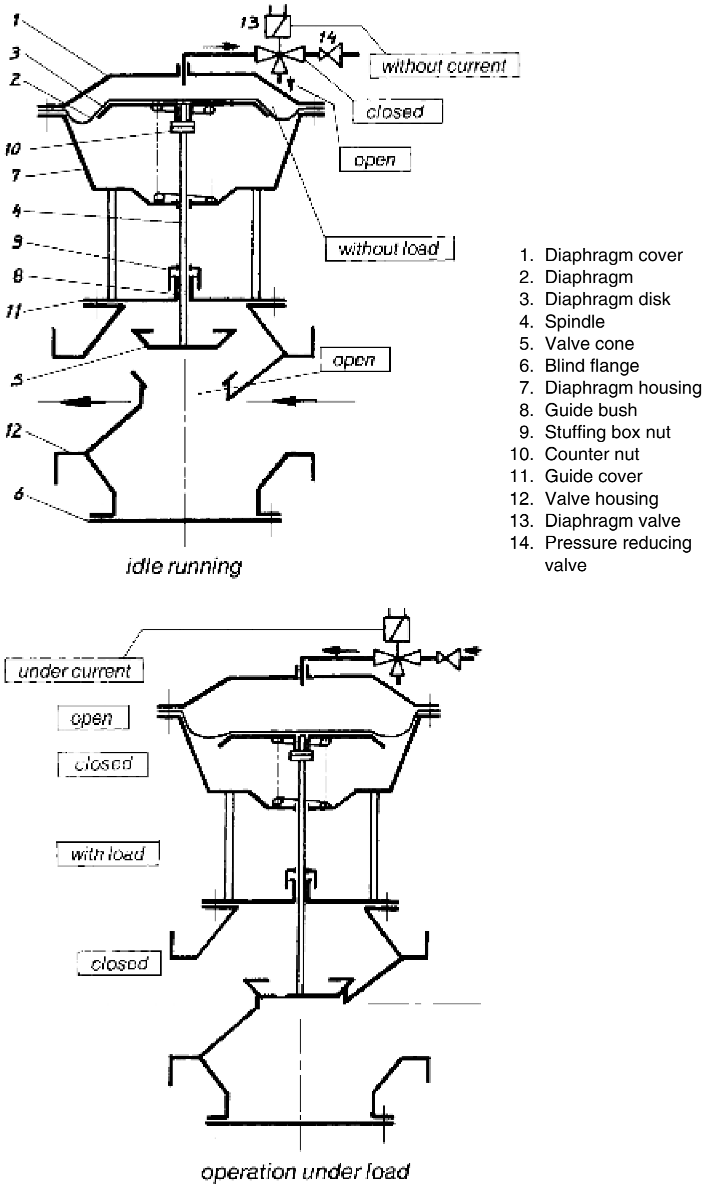


FIGURE 9.12 Diaphragm valve associated with constant-speed unloading devices in rotary screw compressors. *(Aerzen USA Company, Coatesville, Pa.)*

ratios than would be feasible with dry machines. In addition, suction throttling in a screw compressor brings about a drop in the inlet pressure, thereby increasing the compression ratio.

Consequently, oil-injected machines can achieve smooth adjustment of volume flow with relative ease. Some machines, however, may not be designed for this increased compression ratio. It is thus necessary to ensure that achievable pressures do not exceed the mechanical limitations of a given compressor.

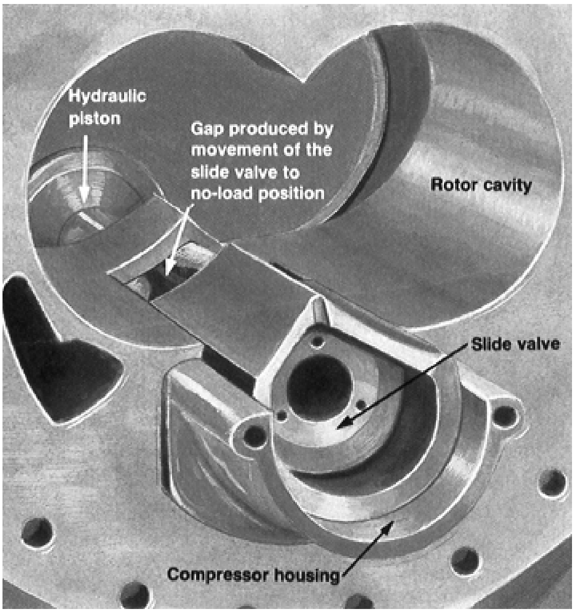


FIGURE 9.13 Internal volume regulating device for oil-injected rotary screw compressors. The position of a slide valve can be shifted in a direction parallel to the axes of the rotors. This provides control of the volume flow of the compressed gas. *(Aerzen USA Company, Coatesville, Pa.)*

Larger compressors can readily be equipped with an internal volume-regulating device (Fig. 9.13). It consists of a slide that is shaped to match the contours of the housing. By moving the slide in a direction parallel to the rotors, the effective length of the rotors can be shortened. The range of this smooth, infinitely variable control extends from 100% down to 10% of full compressor capacity. Also, slide controls offer stepless flow adjustment combined with power savings. These controls are applied primarily on screw compressors where the injected liquid has lubricating properties.

9.1.9 Mechanical Construction

Rotary screw compressors designed for high speeds and pressures incorporate sleeve bearings and self-adjusting multisegment thrust bearings (Fig. 9.14). These machines can also be equipped with the type of sealing system best suited for a particular process gas service. For example, carbon ring seals are used in conjunction with buffer gas injection and leak-off ports that are connected back to compressor suction (Fig. 9.15).

Floating ring seals containing barrier water (Fig. 9.15*b*) allow a certain amount of water to reach the compression space. This water functions as a sealing, cooling, flushing, or gas scrubbing medium. Typically, most of the barrier water is returned to its supply system for reuse.

Stationary double-mechanical seals, lubricated with pressurized water or a suitable oil (Fig. 9.15*c*) are used in many applications where emissions must be minimized. Alternatively, a stationary single seal combined with a floating sleeve element (Fig. 9.15*d*) works very well in machines that feature high-differential pressures. The more traditional

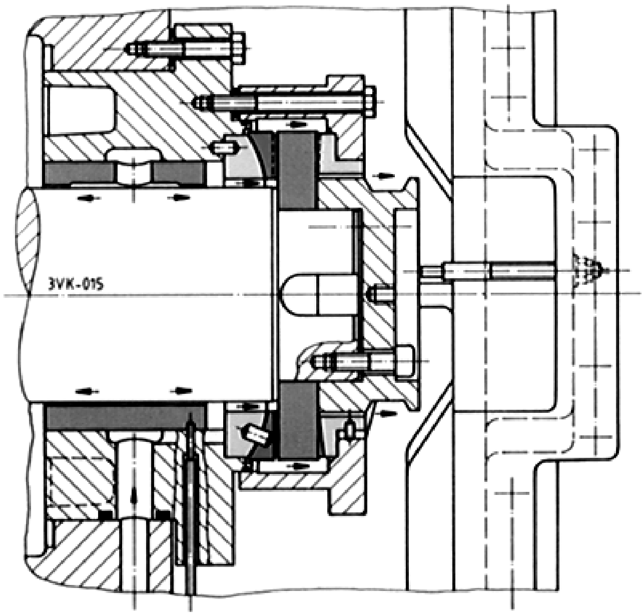


FIGURE 9.14 Radial and thrust bearings furnished with large rotary screw compressors. *(Aerzen USA Company, Coatesville, Pa.)*

labyrinth or mechanical seals can be used on the transmission side of the casings for geared rotary screw units (Fig. 9.15*e* and *f*).

9.1.10 Industry Experience

The capabilities of modern two-stage screw compressors that use water injection are being exploited in several European coke gas–producing plants. Conventional centrifugal compressors are highly vulnerable to performance degradation due to rapid polymerization of this relatively dirty, hydrogen-rich gas. In fact, centrifugal units require frequent cleaning of the internals, causing costly downtime every 6 to 8 weeks.

For example, in one coal gasification plant in Europe centrifugal compressors have been replaced with the three two-stage water-injected screw units shown in Fig. 9.16. The track record (in terms of availability and reliability) of these multistage screw compressors, driven by a 5.5-MW (7400-hp) electric motor, has proven to be remarkably good. Each of the three machines compresses about 33,000 std m^3/h (19,420 scfm) of coke-oven gas varying in pressure from 1 to 12 bar (14.5 to 174 psia). They also operate at considerably lower cost than the centrifugal compressors they replaced. Both power and overall maintenance costs have been reduced.

The decision to use screw compressors with water seals must be based on sound technical and thermodynamic considerations. In dry compression, for example, the discharge temperature may be well in excess of 100°C (212°F). As a result, the higher hydrocarbons may evaporate, leaving asphaltlike residues in the gas to form a coating on the rotors and the housings. These deposits can adversely affect throughput and efficiency. In particular, the sticky residue may fasten the two rotors together in a dry screw compressor whenever the machine is brought down and cooled for any reason.

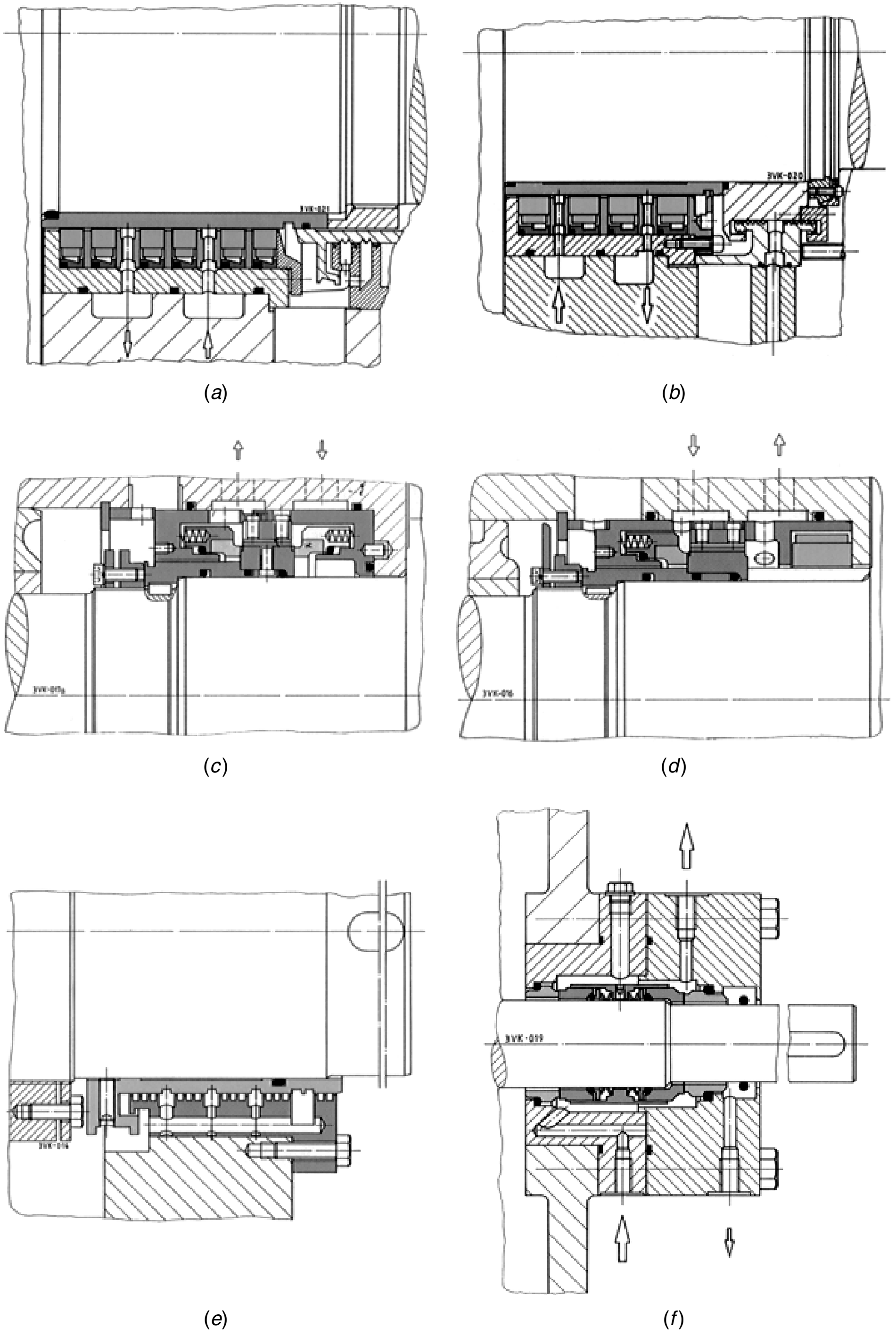


FIGURE 9.15 Sealing arrangements for rotary screw compressors. At the conveying chamber: (*a*) carbon labyrinth seal, (*b*) water-sealed floating rings, (*c*) double-acting slide ring seal, (*d*) combined floating ring and slide ring seal. At the drive shaft: (*e*) labyrinth seal, (*f*) double-acting slide ring seal. *(Aerzen USA Company, Coatesville, Pa.)*

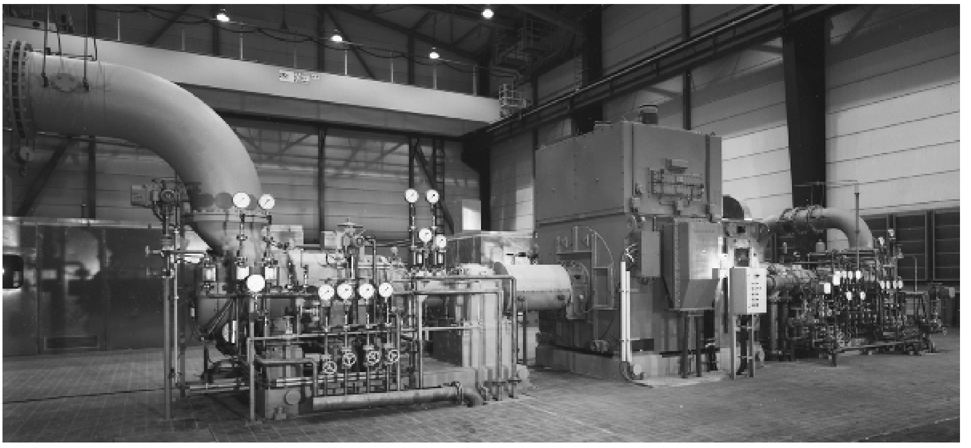


FIGURE 9.16 Large (approximately 7000 hp) rotary screw compressor installation at a German coal gasification plant. *(Aerzener Maschinenfabrik, Aerzen, Germany)*

Water injection limits the final gas temperature to 100°C. In addition, feeding the right amount of water into the compression space of the rotary screw compressor prevents polymerization by removing the heat of compression with the evaporating water.

Much of the water is supplied through the four shaft seals of each stage. The remainder is sprayed into the inlet nozzle of each stage, with the injection rate controlled by a temperature transducer at the compressor discharge. Refer to Fig. 9.17 for a flow schematic of a two-stage unit with water barrier seal.

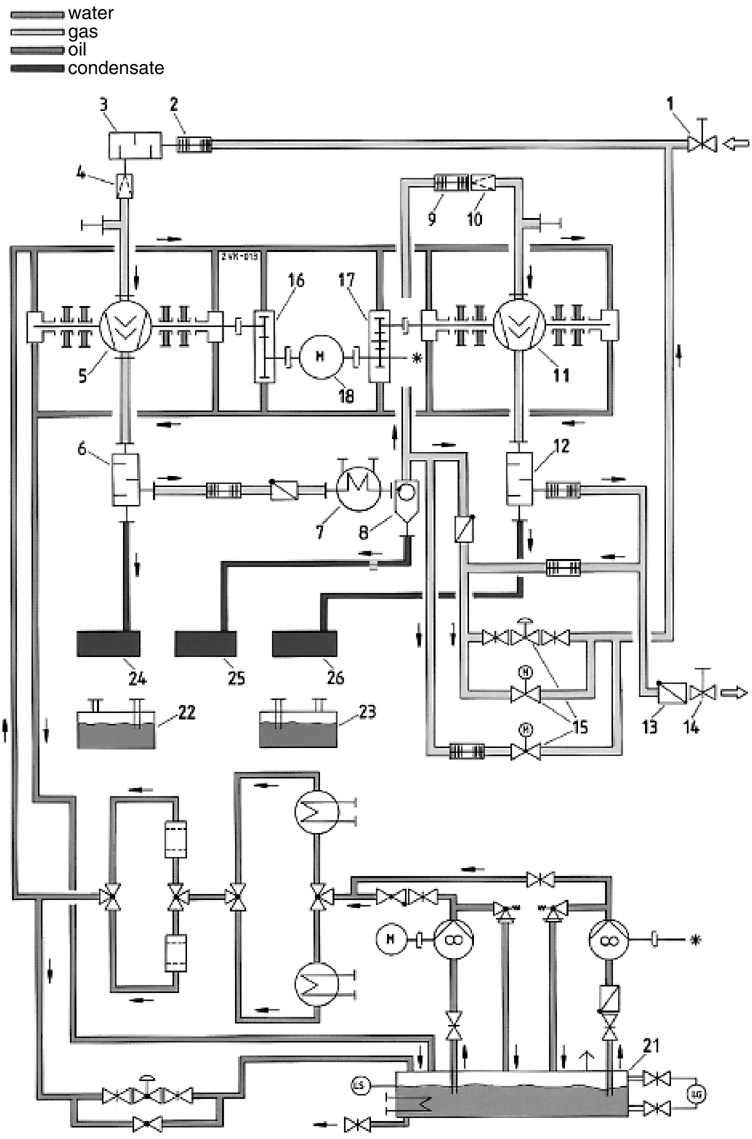
The gas temperature is thus regulated to just below the dew point. This ensures that the deposits are flushed away by the excess water present. The water is finally drained off at the discharge silencer and in the water separators of the intercoolers and aftercoolers of each stage.

Because the gas contains corrosive components, such as hydrogen sulfide, ammonia, hydrogen cyanide, and carbon dioxide, the materials of construction of choice for the compressor have been chromium–nickel alloy steels. Although ensuring erosion resistance, these steels are also resistant to chemical attack by the gas. However, precautions must be taken to avoid corrosion-related wear that can occur over a period of time as a result of the injection of water.

An inspection carried out after 1 year of continuous service in the coking plant has revealed that the rotors and housings are free of dirt deposits. In addition, they have shown no signs of physical or chemical damage due to erosion or corrosion.

Employing what is known as *intermediate pressure regulation,* the screw compressor matches the volume flow to the variable, momentary requirement of the process gas. Gas not required by the final receiver or downstream process is returned after the first stage via a bypass to the intake side. This results in significant power savings while providing a continuous regulation of volume flow ranging from about 20,000 to 33,000 std m^3/h (11,770 to 19,420 scfm). For example, at 20,000 std m^3/h, the power consumption is about 3200 kW (4290 hp), compared with 3950 kW (5295 hp) at 30,000 std m^3/h.

The compressor inlet system has been fitted with separate superchargers that offer the additional capability of boosting the inlet pressure to about 1.6 bar (23 psia). This results in



1. Gate valve
2. Lateral compensator
3. Intake silencer 1st stage
4. Starting strainer 1st stage
5. Screw compressor 1st stage
6. Discharge silencer 1st stage
7. Intercooler
8. Separator
9. Safely relief valve 1st stage
10. Starting strainer 2nd stage
11. Screw compressor 2nd stage
12. Discharge silencer 2nd stage
13. Non-return valve
14. Gate valve
15. Control devices
16. Gear box 1st stage
17. Gear box 2nd stage
18. Safety relief valve 2nd stage
21. Oil system
22. Barrier water system
23. Water injection system
24. Condensate tank 1
25. Condensate tank 2
26. Condensate tank 3
27. Drive motor

FIGURE 9.17 Flow diagram of a two-stage rotary screw compressor unit with a barrier water seal. *(Aerzener Maschinenfabrik, Aerzen, Germany)* an intake volume flow of about 46,000 std m^3/h (27,070 scfm). Thus, combining intermediate pressure regulation with supercharging provides a continuous range of regulation from 20,000 to 46,000 std m^3/h. With this wide range of operability, the coking plant can adapt at any time to changing gas requirements.

9.1.11 Maintenance History

The maintenance history of the three two-stage 5.5-MW rotary screw compressors indicates that they went through partial dismantling every five years. At that time, only the carbon seal rings needed replacement because they suffered some physical damage. All other parts, including bearings, have been reinstalled in the compressors without modification or repair.

In 2006, maintenance costs were estimated at $140,000 for a typical five-year period. This estimate, which includes labor and materials, is orders of magnitude below the expenditures incurred with centrifugal compressors in this type of service.

A second installation that operates two three-stage rotary screw compressors has had its first turnaround inspection after 35,000 hours of uninterrupted service. Several carbon seal rings show traces of wear, and all other parts are in excellent condition. However, because the carbon rings were still quite serviceable, this installation has increased its typical turnaround intervals to 45,000 to 50,000 operating hours.

A third installation (also in Europe) has been less than satisfactory, however. The screw compressors in this plant have experienced accelerated erosive wear of carbon seal rings. This is attributed to unacceptable water quality. The lesson to be learned is that water-injected rotary screw compressors require demineralized water. In this respect, they resemble the common steam turbine.

Of course, applications of screw compressors are in no way limited to coking plants. They can be used for delivery and compression of contaminated gases as well. Moreover, these units are ideally suited for compression of gases that tend to polymerize at relatively low temperatures.

9.1.12 Performance Summary

The performance of a screw compressor is influenced by factors such as gas properties, internal clearances, length/diameter ratio of the rotors, built-in compression ratio, and operating speed. Although compressor manufacturers have not yet found practical ways to present compressor performance for various machine sizes or inlet and outlet conditions in a single graphical layout or diagram, one can still make rule-of-thumb estimates using the graphs and charts shown later for conventional centrifugal compressors.

Typical adiabatic efficiency is almost always between 70 and 80%. The maximum allowable compression ratio for one stage of a screw compressor corresponds to the value that will not cause the final compression temperature to rise above the permitted value of 250°C (482°F). To a large extent, this will depend on the k value (the ratio of specific heats) of the gas to be compressed.

Compressor speeds can vary from about 2000 to 20,000 rpm, depending on unit size. The peripheral speed of the rotor determines the magnitude of the rotational speed. This peripheral speed ranges from 40 to 120 m/s (131 to 394 ft/s) up to a maximum of 150 m/s (492 ft/s) for gases of low molecular weight.

As would be the case with virtually any other specific type or category of fluid machinery, rotary screw compressors embody both advantages and disadvantages in relation to other equipment competing for market share. Listing the advantages first, the application engineer would wish to consider the following:

- Available as "wet screw" compressors with an oil loop serving (a) both bearing lubrication and compression space, or (b) bearings and compression space separately (dual circuit).
- Considerably reduced sensitivity to molecular-weight changes compared to centrifugal machines.
- Much greater tolerance for polymerizing service than other compressors, except perhaps liquid ring machines.
- Capability of accepting more liquid and fine solids entrainment than other compressors, except liquid ring compressors.
- Higher efficiency and less maintenance than liquid ring machines.
- Estimated availability in excess of 99.5%. This may approach or, in certain services, exceed that of centrifugal and axial compressors.
- Smaller size and lower cost than reciprocating compressors in the same capacity range.
- Lower cost than centrifugal compressors in the small and moderate-sized ranges (below approximately 3000 kW, or about 4000 hp).
- Higher pressure capability than other types of rotary positive displacement machines.

Among the disadvantages found are some that are perceived and others that are real. They thus merit more detailed examination.

- Sensitivity to discharge temperature that could affect close clearances and hence operability and availability: Proper temperature control instrumentation and generous sizing of cooling water or liquid injection facilities make this a "non-issue" for modern liquid-injected screw compressors.
- Performance affected by rotor and casing corrosion or erosion. Increased clearances promote internal recycle or gas slip effects—not a serious concern with water- and oil-injected rotary screw compressors.
- Noise level is high enough to require silencing—a factor that must be taken into account. Capable rotary screw compressor manufacturers are fully equipped to provide well-engineered means of reducing environmental noise to meet even the most stringent requirements (see Fig. 9.18).
- Rotary screw compressor systems require pulsation suppression. Although not as severe as piping pulsations encountered with equivalent reciprocating compressors, a properly engineered screw compressor system would incorporate appropriate pulsation bottles and, for high discharge temperatures, pipe expansion loops.
- Choice of rotor and casing materials more limited than for centrifugal compressors. This observation is related to the intricacies and close tolerance requirements of the machining process. Also, a knowledgeable manufacturer is cognizant of certain nonlinearities in the coefficients of expansion of different stainless steels. This might impose experience-based temperature limitations on certain metallurgies and service conditions.
- Maintenance cost and duration of downtime higher than for centrifugals—highly service dependent and not always so; merits closer investigation.

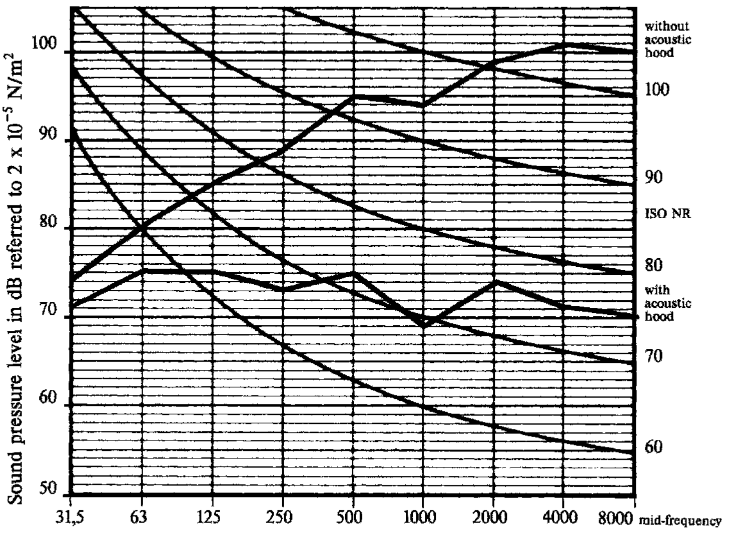


FIGURE 9.18 Typical sound levels obtained from rotary screw compressors with and without acoustic enclosures. *(Aerzen USA Company, Coatesville, Pa.)*

- Flow control flexibility inferior to that of centrifugals and reciprocating compressors—a serious misconception that neglects to take into account the full spectrum of available options given earlier.

9.2 OIL-FLOODED SINGLE-SCREW COMPRESSORS

Oil-flooded single-screw compressors are available for gas flows in the vicinity of 1000 cfm (1700 m^3/h) and discharge pressures approaching 800 psi (55 bar). As can be seen from Fig. 9.19, these machines incorporate features that make them a hybrid between other, sometimes competing compressor types. Cooling oil, which circulates through the compressor to absorb the heat of compression also provides sealing of the gas in the compression spaces and lubrication of the rolling element bearings. Synthetic or special lubricants are used for applications with corrosive gases or when high condensation rates are encountered.

The intermeshing of three principal rotating parts accomplishes the continuous compression process in oil-flooded single-screw machines. The gas flow can be visualized from Fig. 9.20. Suction gas flows into the inlet passage and fills a screw groove. The inlet gas is trapped in the groove when the gate rotor tooth meshes with the screw groove and seals the groove. As the screw continues to rotate, the trapped gas is compressed as the length and volume of the groove is reduced. Injected oil seals the running clearances to prevent leakage of gas.

When the screw rotates far enough, the groove passes the discharge port, delivering gas to the discharge manifold. The location of this port determines the internal volume reduction and thus the internal compression ratio.

Since there are two gate rotors, compression occurs simultaneously on both sides of the screw rotor. Thus, compressive forces are radially balanced. Thrust forces are minimal since suction pressure is ported to both ends of the screw. Because the gate rotor axes are perpendicular to the screw axis, virtually no torque is transmitted to the gate rotors, so wearing forces are kept low.

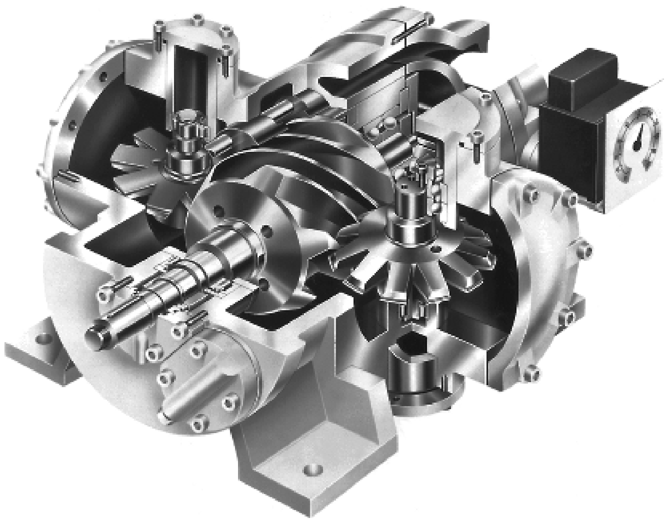


FIGURE 9.19 Oil-injected single-screw compressor. *(Dresser-Rand Company, Broken Arrow, Okla.)*

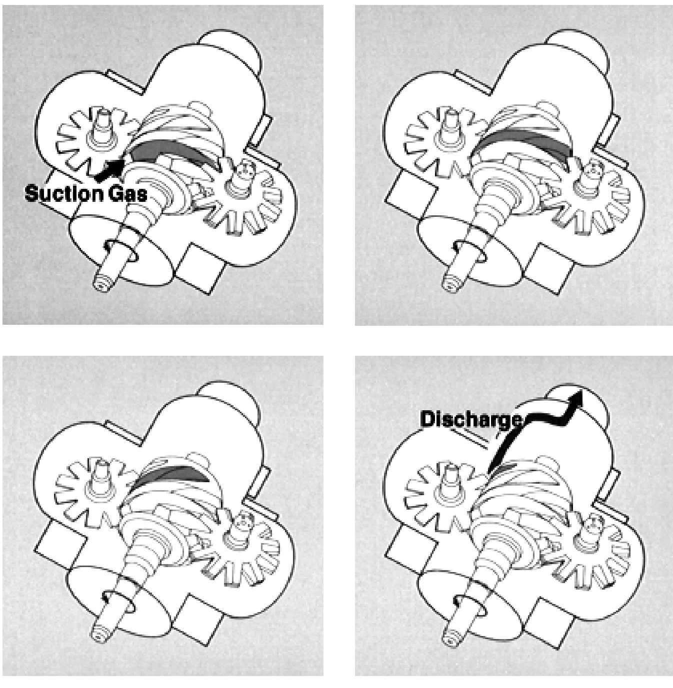


FIGURE 9.20 Gas flow in an oil-injected single-screw compressor. *(Dresser-Rand Company, Broken Arrow, Okla.)*

The schematic of Fig. 9.21 shows the basic relationship between the gas to be compressed and the cooling-lubricating oil. Inside the compressor, oil and gas are mixed and then delivered to a high-efficiency gas–oil separator. Clean gas is then delivered to the skid edge, usually aftercooled. The oil collected in the separator is cooled, filtered, and reinjected

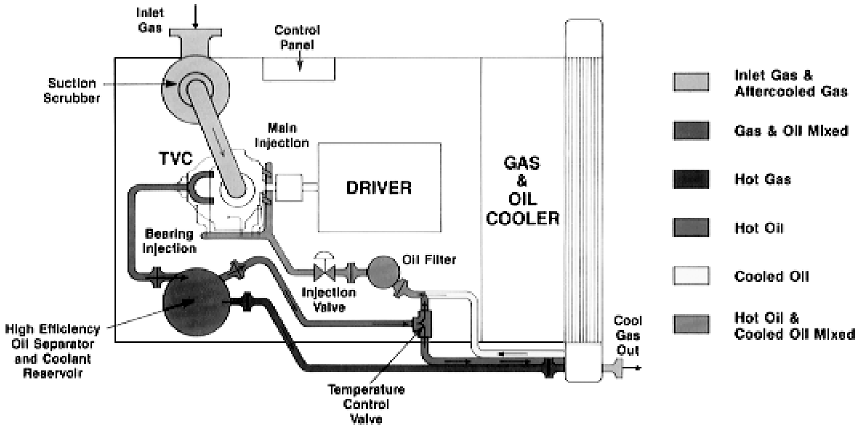


FIGURE 9.21 Process flow schematic showing an oil-injected single-screw compressor. *(Dresser-Rand Company, Broken Arrow, Okla.)*

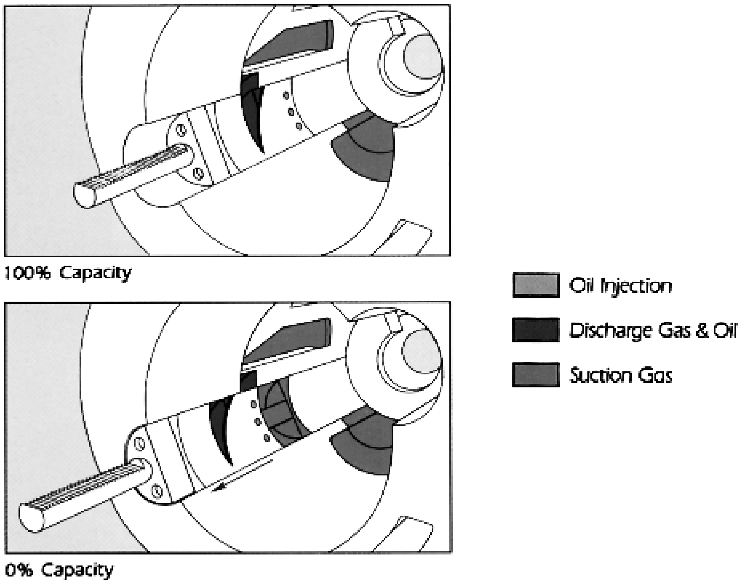


FIGURE 9.22 Capacity control slides in an oil-injected single-screw compressor. *(Dresser-Rand Company, Broken Arrow, Okla.)*

into the compressor. The compressor discharge temperature is held constant during the compression cycle, but the operating discharge temperature selected will vary according to the application.

Capacity control is accomplished through slide pistons contained in the compressor casing of Fig. 9.19. By means of rack-and-pinion gears, the two slides are moved axially and controlled via a stub shaft which protrudes from the side of the casing near the discharge flange.

Figure 9.22 shows one capacity slide at 100% capacity (top) and 0% capacity (bottom). The screw rotor has been removed for clarity. In the 100% position, the slide is positioned so that no leakage of gas can occur during compression; thus, all the gas that enters the

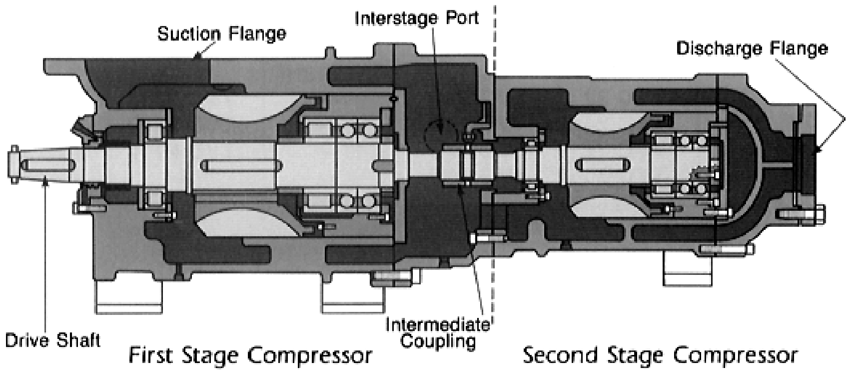


FIGURE 9.23 Two-stage version of an oil-flooded single-screw compressor. *(Dresser-Rand Company, Broken Arrow, Okla.)*

screw groove is delivered to the triangular discharge port. At less than 100% the slide valve is moved so that some of the gas that had entered the groove returns to suction prior to compression. In the same motion, the discharge port moves away from the return port. This delays the groove from passing the discharge port, preserving the internal volume reduction in the compressor.

Oil-flooded single-screw compressors are also available in single-shaft two-stage designs for high-compression ratio service. The second stage of compression is connected to the first through a gastight transition piece that delivers gas and oil between stages and a through shaft with a splined coupling that transmits torque to the second-stage screw (Fig. 9.23). Two-stage compressors are equipped with sidestream capability (e.g., refrigeration economizing) and 40 to 100% infinitely variable capacity control.

It can be assumed that the cost of process interruptions and frequent maintenance incurred with single oil circuits in dirty gas services is often prohibitive. In view of this, it would be difficult for a reliability-focused purchaser–owner to allow anything other than separate oil circuits. The buyer must be prepared to budget funds for the commensurate higher cost. The term *dirty gas* includes even trace quantities of H_2S. Also, if solids are allowed to enter with the gas, they will somehow have to be removed from the oil or other medium that is used with liquid-filled machines. Clearly, this leads to considerations involving our next topic, filter-separator technology.

9.3 SELECTING MODERN REVERSE-FLOW FILTER-SEPARATOR TECHNOLOGY*

Reverse-flow filter-separator technology is a profit generator for best-of-class refineries and petrochemical plants. First applied in the mid-1970s, these flow-optimized self-cleaning coalescers (SCCs) represent mature low-life-cycle-cost best-technology solutions for reliability-focused users. A reliability-focused user is far more interested in low life-cycle costs than in lowest-possible purchase price. However, since aggressive marketers are known to have clouded the issue with advertising claims, a thorough examination and explanation of facts and underlying principles are in order.

* Contributed by King Tool Company, Longview, Tex.

9.3.1 Conventional Filter-Separators vs. SCCs

To understand how SCCs work, we must first recall how most *conventional* filter-separators (CFSs) function. In a CFS (Fig. 9.24), the gas enters the first-stage filter elements, where its velocity is reduced as it passes through a large filter element area. Initially, the various and sundry contaminants (e.g., iron sulfides) are caught by the filter, but the gas forces gradually sluff it to a particle size that will pass through the filter elements.

The gas, solid particles, and liquids coalesced on the inside of the filter element undergo reacceleration and are being reentrained in the collector tube before being led to the next separator section. With wire mesh or vanes in this section typically allowing passage of fine mist droplets and particles—let's call them *globules* of liquid—in the size range below 3 to 8 μm, a good percentage of liquid and small solids (particulates) remain entrained in the gas stream leaving the CFS.

In contrast, self-cleaning coalescers (SCCs) (Fig. 9.25) vastly reduce this entrainment and send much cleaner gas to the downstream equipment. However, SCCs do not accomplish this task merely by making the inlet into an outlet, changing the outlet to the inlet, and calling the "new" device a reverse-flow unit. Instead, consideration had to be given to internal configuration, flow pattern, and—most important—the characteristics of both the liquids

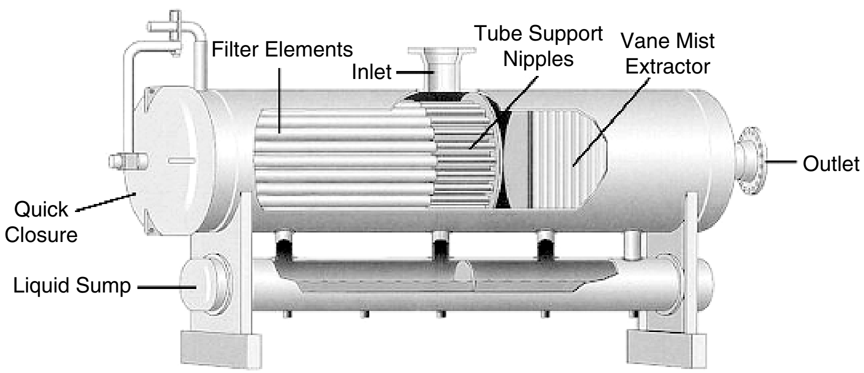


FIGURE 9.24 Conventional filter separator. (*King Tool Company, Longview, Tex.*)

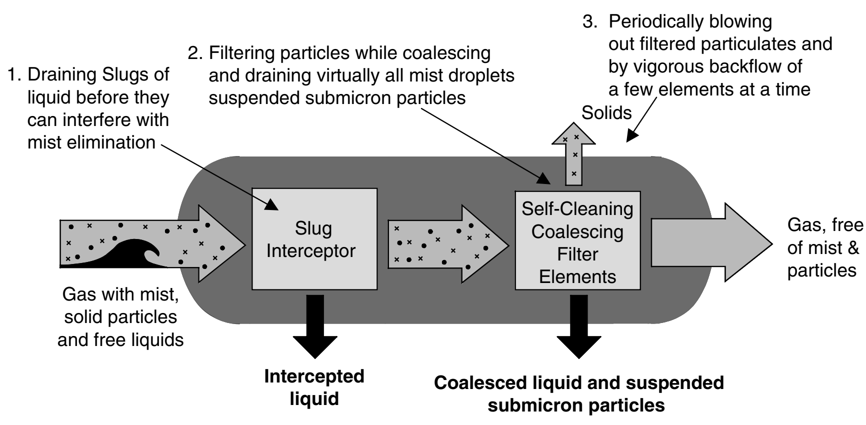


FIGURE 9.25 Self-cleaning coalescer. *(King Tool Company, Longview, Tex.)*

and solids to be removed. The designers had to adjust their thinking from only pressure-drop concerns to considerations dealing with liquid specific gravities, liquid surface tensions, viscosities, and reentrainment velocities.

In properly designed SCCs, gas first passes through the plenum, then through collection tubes to the filter elements. The front end of an SCC represents a slug-free liquid knockout. The deentrainment section is sized to reduce the gas velocity so as to allow any particulates that might have made it through the filter either to drop out or to attach themselves to the coalesced liquid droplets that fall out at this stage. Over three decades of solid experience have proven the effectiveness of this design. Essentially all entrained particulates and mist globules are removed, as are free liquids and large agglomerated materials.

9.3.2 Removal Efficiencies

Some CFS configurations and models claim removal efficiencies with their "coalescers" that are much better than those actually achieved. These claims are often made for vessels that are much smaller than the well-proven SCCs and are virtually impossible to achieve with single-stage CFS models. Also, these CFS designs are vertically oriented and their manufacturers or vendors sometimes state—incorrectly—that effective coalescing cannot be achieved in a horizontal vessel.

Upon closer examination, one may find certain CFS configurations to have high pressure drops with "moist" gases or high velocities, shorter filter elements, and virtually never any slug-handling capacity. Moreover, unless a vendor or manufacturer uses the high-efficiency particular air (HEPA) filters mandated for use in nuclear facilities and hospital operating rooms, filtration effectiveness down to 0.3 μm—considerably less than one hundredth of the width of a human hair—is simply not achievable.

9.3.3 Filter Quality

Keep in mind that a conventional forward-flow filter separator is considered to be a coalescer. It incorporates filter elements that operate on the coalescing principle. The filter elements coalesce liquid droplets into globules of 10 μm and larger to be removed by the downstream impingement vane mist extractor (vanes are guaranteed to remove 8- to 10-μm particles). It is not reasonable to use simple piping insulation as a filter medium and guarantee the removal of droplets in the 0.3-μm size range. Multistage configurations are needed and the ultimate filter has to be "HEPA-like" (i.e., it has to far exceed the quality of piping insulation).

A good design typically embodies long fiberglass filter elements using certain microfiber enhancements that are known to modern textile manufacturers. Low-velocity technology is extremely helpful, and surface area is not as important as the depth of the media through which the gas has to pass. The thicker the filter element, the longer the gas takes to pass through it, resulting in more and better coalescing of the liquids.

Some SCCs are offered with thin high-pressure-drop pleated-paper elements, representing very low contact times and high exit (reentrainment) velocities. As dirt builds up, exit velocities rise even higher, resulting in more and more reentrainment of liquid mists and any associated shearable solids exiting the cartridges. This process goes on as the reentrained particles get smaller and smaller, thus meeting an artificial guarantee as velocities become higher and higher. Others offer fibers of high-density and high-depth media which result in a high pressure drop and high exit velocity, and which reentrain immediately after passing through the cartridges. Both of these approaches, as well as the downsizing of vessels and internals, contribute to marketing strategies geared to high consumption of elements and thus high sales volume and profitability for the vendor.

A competent SCC manufacturer's approach should be just the opposite—to give the user–purchaser maximized reliability, maximized cartridge life, and the lowest possible maintenance expenses. Years ago, the concept of "self-cleaning" vessels was transferred successfully from oil bath separator scrubbers. They are still offered for specific applications and incorporate rotating cleanable bundles. This technology evolved to filter vessels with a rotating cleaning mechanism and to the present state of the art: the back-flushing of individual elements while remaining onstream. Further, competent manufacturers still offer maximized performance even from conventional vessels by utilizing tried-and-true designs with maximized internals. They will not advocate the use of downsized versions that violate certain velocity and pressure-drop criteria, thereby incurring high maintenance and nonsustainable or nonoptimized performance.

This takes us back to HEPA filters. Designed and developed for air filtration, HEPA filters recycle the air many times within a closed system and add fresh makeup air periodically to achieve the desired air quality. In the hydrocarbon processing industry there is usually only a single-pass opportunity to achieve clean gas. It is rarely feasible to recycle process gases several times to obtain the desired gas purity. Since absolute beta-rated filter elements are simply not able to achieve these results, many inferior designs call for one or more "conditioning" filters, or vessels to be placed upstream of their coalescer.

Also, be on the lookout for offers that allude to the advisability, or just the merits, of installing downstream vessels to clean up certain liquid streams to which the gas has been exposed. A relevant question to ask is why the liquid has to be cleaned up if the upstream vessel(s) have done their job of, say, protecting the treating tower. Without fail, the answer will point to liquids, or mists, or corrosion products in the form of small solids particles that were not adequately removed upstream of the tower. Hence, foaming and treating agent contamination were not eliminated. This results in tower upsets, additional filtration for liquids, and even the possible need for carbon beds or filters to remove trace liquid aerosol contaminants. SCCs have been implemented successfully to protect such process streams and to eliminate or prevent contamination-related upsets. Time and again, bottom-line results show that self-cleaning coalescers protect equipment and safeguard reliability.

9.3.4 Selecting the Most Suitable Gas Filtration Equipment

Superior self-cleaning coalescers can remove iron sulfides, viscous fluids, and slugs because of their inherent low pressure drops (4 to 6 in., or 100 to 150 mm H_2O). Moreover, low velocities and other important considerations conducive to good separation and low life-cycle costs must be taken into account here. With input from the user or destination plant, a competent vendor can assist in drawing up a good inquiry specification. Within the specification there are many options to consider. The choice quite obviously depends on process conditions and related parameters, some of which are as follows:

- *Dry filter:* gas with associated solids
- *Dry filter, self-cleaning:* gas associated with solids
- *Line separator:* gas containing entrained liquid mist
- *Vertical or horizontal separator:* gas with entrained liquid globules (mist, aerosol); gas with entrained liquid particles (mist) and free liquid (slug) removal

- *Vertical or horizontal filter-separator:* gas with entrained liquid globules (mist, aerosol) and stable solids
- *Reverse-flow mist coalescer:* gas with entrained liquid globules (mist, aerosol); removal to submicrometer particle size and extremely high efficiency
- *Reverse-flow mist coalescer with slug chamber:* gas with entrained liquid globules (mist, aerosol), slugs and (stable or unstable) solids; removal to submicrometer level or better, at high efficiency (can be furnished in self-cleaning configuration while in full service)
- *Oil bath separator-scrubber:* gas with liquid globules (mist) and solids (stable or unstable); removal to 3 μm at 97% efficiency by weight
- *Tricon 3 stage separator:* gas with entrained liquid globules (mist), slugs, and solids (stable or unstable); removal to 3 μm at 97% efficiency

9.3.5 Evaluating the Proposed Configurations

Once the various bidders submit their offers, they must be evaluated using life-cycle cost and suitability criteria. An objective evaluation must keep in mind the following:

1. *Velocity.* Once the gas stream enters the vessel, there should be no internal configuration that would accelerate the gas back to the pipeline velocity. Causing the motion of gas to increase in velocity will only cause the liquid to shear into smaller and smaller globules.

2. *Pressure drop.* In no instance should a piece of separation equipment be designed with more than a 2-psi pressure drop from flange to flange when the vessel operating pressure exceeds 500 psig. At less than 500 psig, the flange-to-flange pressure drop should be limited to 1 psi or lower. Pressure drop consumes energy, and energy costs money. In no design of separation equipment should the pressure drop across an element arrangement be allowed to exceed 0.5 psi. As filter elements become wetted and 50% plugged, the pressure drop increases fourfold. If, say, the initial pressure drop is 0.5 psi and the elements become half-plugged, the pressure will increase to 2 psi. Once the elements become three-quarters plugged, the pressure will increase to 8 psi. This is 16 times the initial pressure drop, and a change of elements is now unavoidable. Keep the initial filter element pressure as far below 0.5 psi as possible to avoid frequent element change-out. Remember that the filter elements have to be disposed of, and this disposal can become expensive.

3. *Filter element cost.* Always ascertain the cost of replacement elements. Some vendors will practically give away vessels in order to generate spare parts sales. Find the inside diameter, the outside diameter, and the length of the proposed elements and how many of these make up the vessel internals. Using this information, calculate the surface area on the inside of the elements and the velocity of the gas entering the elements. Also from this information, determine the exit velocity leaving the elements. Note that this velocity should not exceed the reentrainment velocity of the liquid. Some of the reverse-flow coalescer offers you might receive will turn out to be "eggbeaters" that take whatever liquid enters the vessel and shears it into orders-of-magnitude amounts of smaller globules which are then reentrained in the gas stream. Liquid globules can be sheared so small that they cannot fall out again until they recoalesce downstream. But all the same, the liquid is there to do its damage to downstream equipment.

4. *Vessel Life.* Under ordinary circumstances, separation equipment should have a useful life of 20 to 25 years. Needless to say, corrosion problems, internal explosions, vibration or

pulsation, overloads, hydrate formation, lack of routine maintenance, incorrect or faulty maintenance practices, misapplication or use of equipment under unsuitable operating conditions, replacing elements with unsuitable or poor-quality substitutes, and various other forms of mistreatment can affect vessel life adversely.

5. *Reliability of the vendor.* If a piece of separation equipment is bought and put into service under conditions that deviate from the design intent, it may not live up to expectations. Such underperformance will usually manifest itself rather quickly. Yet, these unpleasant surprises can be avoided by selecting a reliable vendor as the source of supply. The person or team engaged in the selection and evaluation task should ask the following questions:

- Does the vendor have the facilities to manufacture the equipment, or is manufacturing "farmed out" to subvendors?
- Who builds the essential parts, such as the filter elements, the mist extractors, other internals, and the vessel itself?
- Who does the x-raying, hardness testing, ultrasonic examination, magnaflux examination, (both wet and dry, if required), stress relieving, hydrostatic testing, grit-blasting and painting, and final preparation for shipping?

6. *Value.* How important is proper performance of the separation equipment to the protection of downstream equipment? Certainly, a monetary value has to be placed on repair and maintenance of the downstream installation. To what extent would rotating equipment such as turbines, turbo-expanders, centrifugal or reciprocating compressors, internal combustion engines, dehydration, amine or molecular sieve units, refinery or petrochemical processes, meter runs, power plants, fired heaters, plant fuel, municipal fuel, and/or perhaps gas coming in from producing wells be affected by potential performance deficiencies of the separation equipment? What are prudent downtime risks, and what would be the cost of rectifying problems with downstream equipment caused by defective filtration equipment? A reliability-focused organization demands answers to these questions!

7. *Follow-up.* Who will ultimately make the determination if the goods specified and purchased are, in fact, the goods received? Will the responsibility change hands from selection to purchasing to operation with a relaxed regard for what was intended to happen and what is actually happening? In that case, only the very best and most conservatively designed piece of separation equipment should be purchased.

Contrary to conventional wisdom, there have been no "super breakthroughs" in the design of separation equipment in the past 30 years. On the other hand, considerable changes have been made in presentation and marketing methods over the past two or three decades. Some marketing claims as to how far the state of the art has advanced during the past several years, or even in recent months, are truly stretching the imagination. Beware, since they may simply be designed to sell spare parts and/or just stay alive in a highly competitive environment.

9.3.6 Life-Cycle-Cost Calculations

Life-cycle-cost calculations must be used to determine the wisest equipment choice. Life-cycle-based filter equipment cost is the total lifetime cost to purchase, install, operate, and maintain (including associated downtime), plus the downstream cost due to contamination from inadequately processed fluids or even the risk of damaging downstream equipment, and finally, the cost of ultimately disposing of a piece of equipment. A simplified mathematical expression might be

$$\mathrm{LCC} = C_{\mathrm{ic}} + C_{\mathrm{in}} + C_{\mathrm{e}} + C_{\mathrm{o}} + C_{\mathrm{m}} + C_{\mathrm{dt}} + C_{\mathrm{de}} + C_{\mathrm{env}} + C_{\mathrm{d}} \tag{9.11}$$

where LCC = life-cycle cost
C_{ic} = initial cost, purchase price (system, pipe, auxiliary services)
C_{in} = installation and commissioning cost
C_{e} = energy costs ("incremental Δp"-related)
C_{o} = operation costs, if applicable
C_{m} = maintenance and repair costs
C_{dt} = downtime costs
C_{de} = incremental repair cost, downstream equipment
C_{env} = environmental costs
C_{d} = decommissioning and/or disposal costs

Energy, maintenance, and downtime costs depend on the selection and design of the filtration equipment, system design and integration with the downstream equipment, design of the installation, and the way the system is operated. Matching the equipment carefully with the process unit's or production facility's requirements can ensure the lowest energy and maintenance costs, and yield maximum equipment life.

9.3.7 Conclusions

The initial investment costs go well beyond the initial purchase price for the equipment. Investment costs include engineering, bid process (*bid conditioning*), purchase order administration, testing, inspection, spare parts inventory, and training and auxiliary equipment. The purchase price of the filtration equipment is typically less than 15% of the total ownership cost. Installation and commissioning costs include the foundations, grouting, connecting of process piping, connecting electrical or instrument wiring, and (if provided) connecting auxiliary systems.

But suppose now that a team of engineers goes through the planning, bidding, procurement, installation, and evaluation stages of the separation equipment and finds that it matches the requirements exactly. Then comes the spare parts purchasing stage, and at that point, cheap incompatible sets of fiberglass pipe insulation elements are bought. Suppose further that these are to be installed, when dictated, by the best operating practice assigned to the installation.

Chances are that the element manufacturer will have made all kinds of promises and that a few dollars will have been saved, but what happens when these substitutes are installed? There is no question about it—the separation equipment can no longer live up to the job specifications, and bad things start to happen at that point. So, to the reliability-focused and risk-averse user, life-cycle costs are of immense importance. In contrast, repair-focused users are interested primarily in the initial purchase price. But there is consensus among best-in-class industrial and process plants that only truly reliability-focused facilities will be profitable a few years from now, and only they will survive.