

CHAPTER 38. COMPRESSORS

A COMPRESSOR is one of the four essential components of the basic vapor compression refrigeration system; the others are the condenser, evaporator, and expansion device. The compressor circulates refrigerant through the system and increases refrigerant vapor pressure to create the pressure differential between the condenser and evaporator. This chapter describes the design features of several categories of commercially available refrigerant compressors. There are two broad categories of compressors: positive displacement and dynamic. **Positive-displacement compressors** increase refrigerant vapor pressure by reducing the volume of the compression chamber through work applied to the compressor's mechanism. Positive-displacement compressors include many styles of compressors currently in use, such as reciprocating, rotary (rolling piston, rotary vane, single screw, twin screw), and orbital (scroll, trochoidal). **Dynamic compressors** increase refrigerant vapor pressure by continuous transfer of kinetic energy from the rotating member to the vapor, followed by conversion of this energy into a pressure rise. Centrifugal compressors function based on these principles. There are many reasons to consider each compressor style. Some compressors have physical size limitations that may limit their application to smaller equipment; some have associated noise concerns; and some have efficiency levels that make them more or less attractive. Each piece of equipment using a compressor has a certain set of design parameters (refrigerant, cost, performance, sound, capacity, etc.) that requires the designer to evaluate various compressor characteristics and choose the best compressor type for the application. [Figure 1](#) addresses volumetric flow rate of the compressor as a function of the differential pressure (discharge pressure minus suction pressure) against which the compressor is required to work. Three common compressor styles are represented on the chart. Positive-displacement compressors tend to maintain a relatively constant volumetric flow rate over a wide range of differential pressures, because this compressor draws a predetermined volume of vapor into its chamber and compresses it to a reduced volume mechanically, thereby increasing the pressure. This helps to keep the equipment operating near its design capacity regardless of the conditions. Centrifugal compressors dynamically compress the suction gas by converting velocity energy to pressure energy. Therefore, they do not have a fixed volumetric flow rate, and the capacity can vary over a range of pressure ratios. This tends to make centrifugal-based equipment much more application specific.

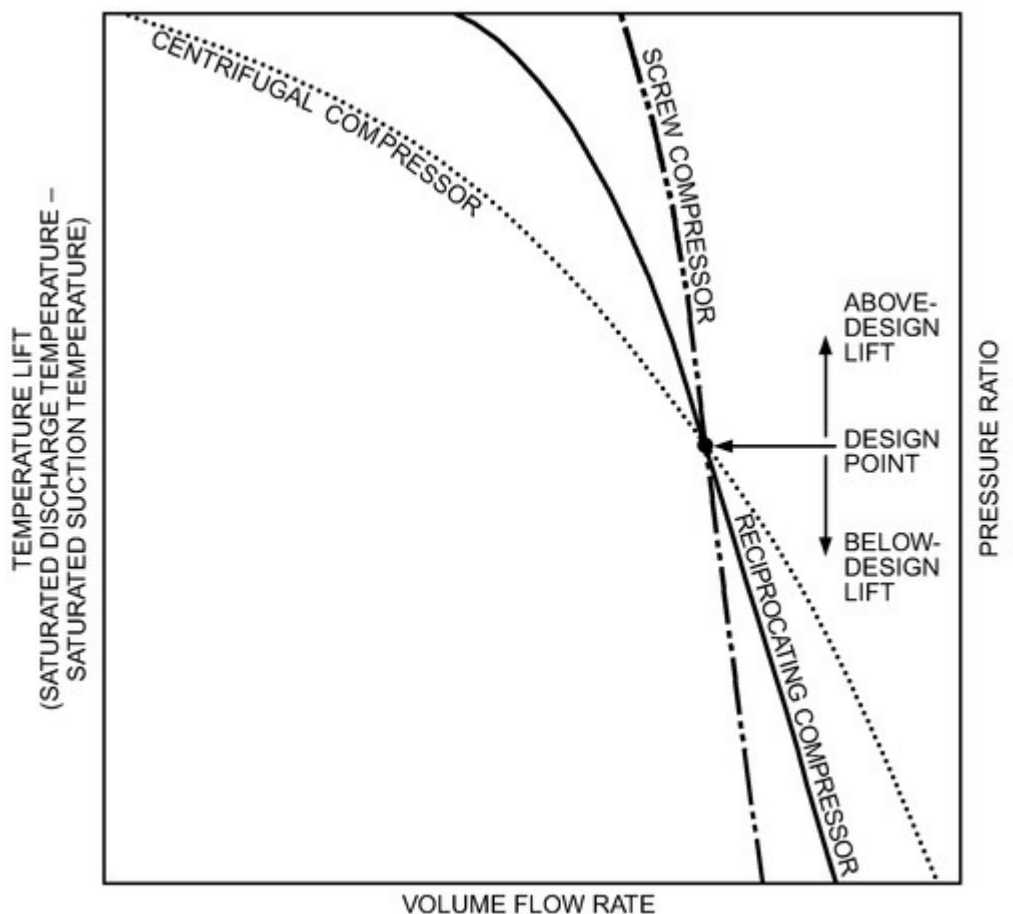


Figure 1. Comparison of Single-Stage Centrifugal, Reciprocating, and Screw Compressor Performance

1. POSITIVE-DISPLACEMENT COMPRESSORS

Major types of positive-displacement compressors classified by compression mechanism design are shown in [Figure 2](#). Compressors also can be further classified as single-stage or multistage, and by type of motor drive (electrical or mechanical), capacity control (single speed, variable speed, single speed with adjustable compression chamber volume), and drive enclosure (hermetic, semihermetic, open). **Open-drive compressors** are those in which the shaft or other moving part extends through a seal in the crankcase for an external drive. Ammonia compressors are predominantly manufactured only in the open design because of the incompatibility of the refrigerant and common hermetic motor materials. Most automotive compressors are also open-drive type.

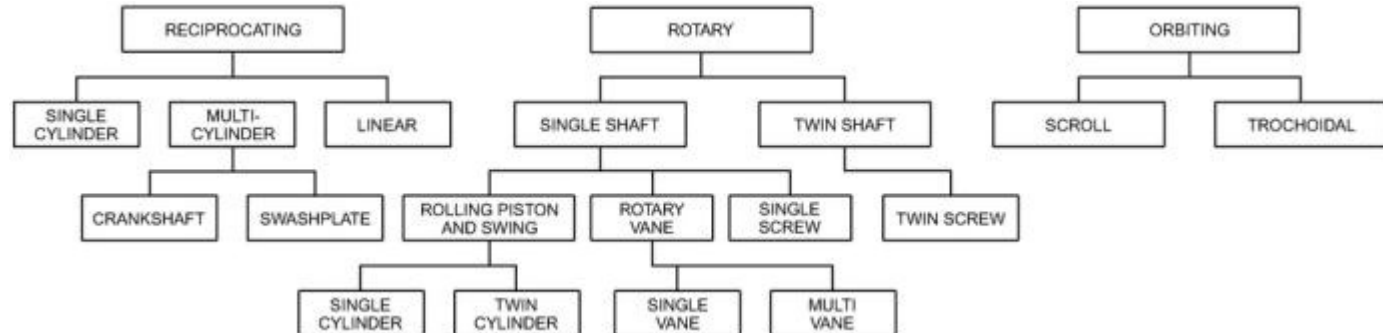


Figure 2. Types of Positive-Displacement Compressors (Classified by Compression Mechanism Design)

Hermetic compressors contain the motor and compressor in the same gastight housing, which is permanently sealed with no access for servicing internal parts in the field, with the motor shaft integral with the compressor crankshaft and the motor in contact with the refrigerant. Hermetic compressors normally have the motor-compressor pump assembly mounted inside a steel shell, which is sealed by welding. A **semihermetic compressor** (also called bolted, accessible, or serviceable) is a compressor of bolted construction that is amenable to field repair. The seal in the bolted joints is provided by O rings or gaskets.

1.1 PERFORMANCE

Compressor performance depends on an array of design compromises involving characteristics of the refrigerant, compression mechanism, and motor. The goal is to provide the following:

- Greatest trouble-free life expectancy
- Most refrigeration effect for least power input
- Lowest applied cost
- Wide range of operating conditions
- Acceptable vibration and sound level

The useful measure of compressor performance is the energy efficiency ratio (EER). The EER is the ratio of the compressor's refrigerating capacity to the input power. For a hermetic or semihermetic compressor, the EER includes the combined operating efficiencies of the motor and the compressor:

$$\text{EER (hermetic or semihermetic)} = \frac{\text{Capacity, Btu/h}}{\text{Input power to motor, W}}$$

The EER for an open compressor does not include motor efficiency:

$$\text{EER (open)} = \frac{\text{Capacity, Btu/h}}{\text{Input power to shaft, W}}$$

Because capacity and motor/shaft power vary with operating conditions, EER also varies with operating conditions. Power input per unit of refrigerating capacity (bhp/ton) is used to compare different compressors at the same operating conditions, primarily with open-drive industrial equipment.

$$\frac{\text{bhp}}{\text{ton}} = \frac{\text{Power input to shaft, bhp}}{\text{Compressor capacity, ton}}$$

Ideal Compressor
During operation, pressure and volume in the compression chamber vary as shown in [Figure 3](#). There are four sequential processes: first, gas is drawn into the compression chamber during the suction process (1–2); next is compression (2–3); and then higher-pressure gas is pushed out during the discharge process (3–4), followed by the next cycle.

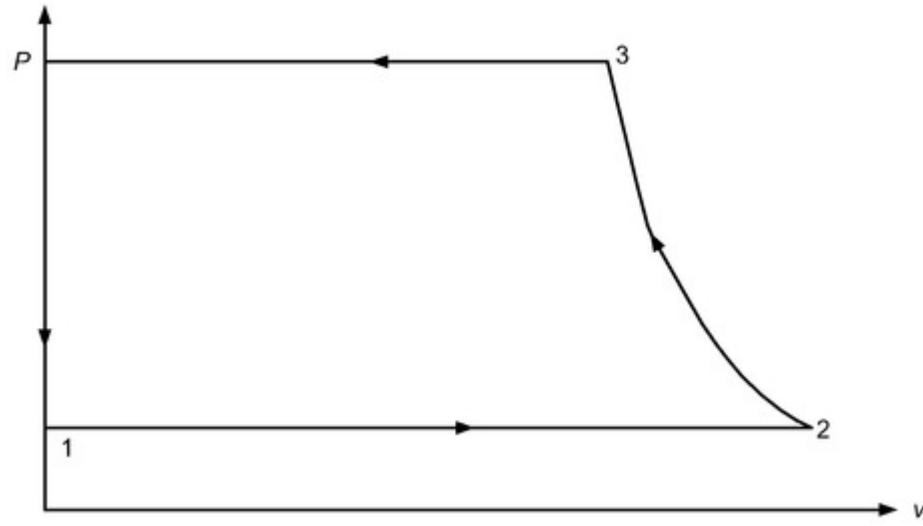


Figure 3. Ideal Compressor Cycle

The capacity of a compressor at a given operating condition is a function of the mass of gas compressed per unit time. Ideally, mass flow is equal to the product of the compressor displacement per unit time and the gas density, as shown in [Equation \(1\)](#):

$$m = \rho_v V_d \tag{1}$$

where
 m = ideal mass flow of compressed gas, lb/h
 ρ_v = density of gas entering compressor (at suction port), lb/ft³
 V_d = geometric displacement of compressor, ft³/h
The ideal refrigeration cycle, discussed in detail in [Chapter 2 of the 2013 ASHRAE Handbook—Fundamentals](#), consists of four processes, as shown in [Figure 4](#):
1–2: isentropic (reversible and adiabatic) compression
2–3: desuperheating, condensing, and subcooling at constant pressure
3–4: adiabatic expansion
4–1: boiling and superheating at constant pressure

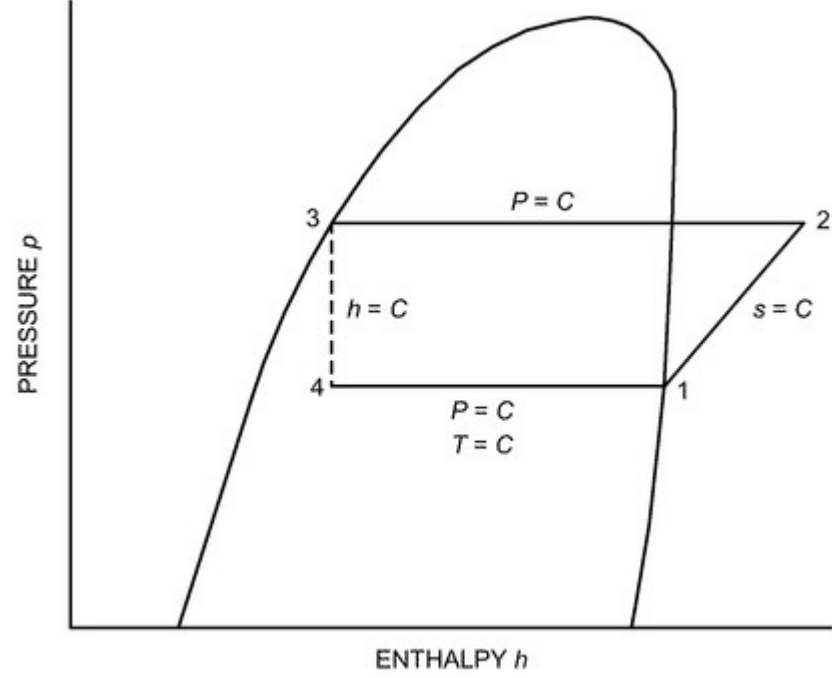


Figure 4. Pressure-Enthalpy Diagram for Ideal Refrigeration Cycle

The following quantities can be determined from the pressure-enthalpy diagram in [Figure 4](#) using m , the mass flow of gas from [Equation \(1\)](#).

$$Q_o = mQ_{\text{refrigeration effect}} = m(h_1 - h_4) \tag{2}$$

$$P_{oi} = mQ_{\text{work of compression}} = m(h_2 - h_1) = mw_{oi} \tag{3}$$

where
 w_o = specific work of isentropic compression, Btu/lb
 Q_o = ideal capacity, Btu/h
 P_{oi} = ideal power input, Btu/h

Actual Compressor
Ideal conditions never occur, so actual compressor performance differs from ideal performance. Various factors contribute to decreased capacity and increased power input. Depending on compressor type, some or all of the following factors can have a major effect on compressor performance.

- **Pressure drops in compressor**
 - Through shutoff valves
 - Through suction accumulator
 - Across suction strainer/filter
 - Across motor (hermetic compressor)
 - In manifolds (suction and discharge)
 - Through valves and valve ports (suction and discharge)
 - In internal muffler
 - Through internal lubricant separator
 - Across check valves
- **Heat gain by refrigerant from**
 - Cooling the hermetic motor
 - Internal heat exchange between compressor and suction gas
- **Power losses because of**
 - Friction
 - Lubricant pump power consumption
 - Motor losses
- **Valve inefficiencies** caused by imperfect mechanical action
- **Internal gas leakage**
- **Oil circulation**
- **Reexpansion** (clearance losses). The gas remaining in the compression chamber after discharge reexpands into the compression chamber during the suction cycle and limits the mass of fresh gas that can be brought into the compression chamber.
- **Over- and undercompression.** Overcompression occurs when pressure in the compression chamber reaches discharge pressure before finishing the compression process. Undercompression occurs when the compression chamber reaches the discharge pressure after finishing the compression process.
- **Deviation from isentropic compression.** In the actual compressor, the compression process deviates from isentropic compression primarily because of fluid and mechanical friction and heat transfer in the compression chamber. The actual compression process and work of compression must be determined from measurements.

Compressor Efficiency, Subcooling, and Superheating
Deviations from ideal performance are difficult to evaluate individually. They can, however, be grouped together and considered by category. Their effect on ideal compressor performance is characterized by the following efficiencies:
Volumetric efficiency η_v is the ratio of actual volumetric flow to ideal volumetric flow (i.e., the geometric compressor displacement).
Compression isentropic efficiency η_c considers only what occurs within the compression volume and is a measure of the deviation of actual compression from isentropic compression. It is defined as the ratio of work required for isentropic compression of the gas w_c to work delivered to the gas within the compression volume w_c .

$$\eta_{oi} = w_{oi} / w_d \tag{4}$$

For a multicylinder or multistage compressor, this equation applies only for each individual cylinder or stage.
Mechanical efficiency η_m is the ratio of work delivered to the gas (measured) to work input to the compressor shaft w_m .

$$\eta_m = w_d / w_m \tag{5}$$

Isentropic efficiency η_i is the ratio of work required for isentropic compression of the gas w_c to work input to the compressor shaft w_m .

$$\eta_i = w_{oi} / w_m = \eta_{oi} \eta_m \tag{6}$$

Motor efficiency η_e is the ratio of work input to the compressor shaft w_m to work input to the motor w_e .

$$\eta_e = w_m / w_e \tag{7}$$

Total compressor efficiency η_{com} is the ratio of work required for isentropic compression w_c to work input to the motor w_e .

$$\eta_{com} = w_{oi} / w_e = \eta_{oi} \eta_m \eta_e \tag{8}$$

Actual shaft compressor power is a function of the power input to the ideal compressor and the compression, mechanical, and motor efficiencies of the compressor, as shown in the following equation:

$$P_e = P_m / \eta_e = P_d / (\eta_m \eta_e) = P_{oi} / (\eta_{oi} \eta_m \eta_e) \tag{9}$$

where
 P_e = power input to motor
 P_o = power input to shaft

P_p = power input to the compression mechanism (pump)
 P_{is} = power required for isentropic compression
Actual capacity is a function of the ideal capacity and volumetric efficiency η_v of the compressor:

$$Q = Q_i \eta_v$$

(10)

Total heat rejection is the sum of refrigeration effect and heat equivalent of power input to the compressor. Heat radiation or using means for additional cooling may reduce this value. The quantity of heat rejection must be known in order to size condensers. Note that compressor capacity with a given refrigerant depends on saturation suction temperature (SST), saturation discharge temperature (SDT), superheating (SH), and subcooling (SC). **Saturation suction temperature (SST)** is the temperature of two-phase liquid/gas refrigerant at suction pressure. SST is often called evaporator temperature; however, in real systems, there is a difference because of pressure drop between evaporator and compressor. **Saturated discharge temperature (SDT)** is the temperature of two-phase liquid/gas refrigerant at discharge pressure. SDT is often called condensing temperature; however, in real systems, there is a difference because of the pressure drop between compressor and condenser. **Liquid subcooling** is not accomplished by the compressor. However, the effect of liquid subcooling is included in compressor ratings by some manufacturers. *Note:* Air-Conditioning, Heating, and Refrigeration Institute (AHRI) *Standard* 540 and European Committee for Standardization (CEN) *European Norm* (EN) 12900 do not include subcooling. **Suction Superheat.** In general, no liquid refrigerant should be present in suction gas entering the low-pressure side of the compressor (especially reciprocating and scroll), because it causes oil dilution and gas formation in the lubrication system. If liquid carryover is severe enough to reach the cylinders, excessive wear of valves, valve stops, pistons, and rings can occur; liquid slugging can break valves, pistons, and connecting rods. Measuring suction superheat can be difficult, and the indication of a small superheat (<10°F) does not necessarily mean that liquid is not present. An effective suction separator may be necessary to remove all liquid. In some cases, a small amount of liquid refrigerant in suction gas entering directly to the compression chamber (rotary) may help improve compressor reliability by effectively removing the heat of friction during boundary lubrication or reducing the discharge temperature. Some compressors are specifically designed to operate without suction superheat. In this case, special design features are introduced to keep liquid from reaching suction valves and cylinders; oil viscosity must also be adjusted to anticipate its dilution with refrigerant. High suction superheat may result in dangerously high discharge temperatures and, in hermetic compressors, high motor temperatures.

1.2 ABNORMAL OPERATING CONDITIONS, HAZARDS, AND PROTECTIVE DEVICES

To operate through the entire range of conditions for which the compressor was designed and to obtain the desired service life, it is important that mating components in the system be correctly designed and selected. Suction superheat must be controlled, lubricant must return to the compressor, and adequate protection must be provided against abnormal conditions. [Chapters 1, 2, 5, and 13](#) the *2014 ASHRAE Handbook—Refrigeration* provide more information on protection against abnormal conditions. Chapter 7 of that volume gives details of cleanup in the event of a hermetic motor burnout. Compressors are provided with one or more of the following devices for protection against abnormal conditions and to comply with various codes.

- **High-pressure protection** as required by Underwriters Laboratories and per ARI standards and ASHRAE *Standard* 15. This may include the following:
 - A high-pressure cutout [a pressure sensor that sends a signal to the switch to cut power to the compressor motor (drive)].
 - A high- to low-side internal relief valve, external relief valve, or rupture member to comply with ASHRAE *Standard* 15. The differential pressure setting depends on the refrigerant used and operating conditions. Care must be taken to ensure that the relief valve will not accidentally blow on a fast pull-down. Some welded hermetic compressors have an internal high- to low-pressure relief valve to limit maximum pressure in units not equipped with other high-pressure control devices.
- **High-temperature control** devices to protect against overheating and oil breakdown.
 - Motor overtemperature protective devices are addressed in the section on Integral Thermal Protection in [Chapter 45](#).
 - To protect against lubricant and refrigerant breakdown, a temperature sensor is sometimes used to stop the compressor when discharge temperature exceeds safe values. The switch may be placed internally (near the compression chamber) or externally (on the discharge line).
 - On larger compressors, lubricant temperature may be controlled by cooling with a heat exchanger or direct liquid injection, or the compressor may shut down on high lubricant temperature.
 - Where lubricant sump heaters are used to maintain a minimum lubricant sump temperature, a thermostat may be used to limit the maximum lubricant temperature.
- **Low-pressure protection** may be provided for
 - *Suction pressure.* Many compressors or systems are limited to a minimum suction pressure by a protective switch. Motor and compressor mechanism cooling, freeze-up, or pressure ratio usually determine the pressure setting.
 - *Compressor.* Forced-feed lubrication systems use lubricant-pressure, minimum-flow, or minimum-level protectors to prevent the compressor from operating with insufficient lubricant pressure.
- **Time delay or lockouts with manual resets** prevent damage to both compressor motor and contactors from repetitive rapid-starting cycles. Fixed-speed compressor motors experience a significant inrush electrical current during start-up. This current can reach the level of locked-rotor amps. If the motor restarts rapidly, without adequate cooling, overheating damage can occur. Time delays should be set for an appropriate interval to avoid this hazard.
- **Low-voltage and phase-loss or reversal protection** is used on some systems. Phase-reversal protection is used with multiphase devices to ensure proper direction of rotation.
- **Suction line strainer.** Some compressors are provided with a strainer at the suction inlet to remove any dirt that might be in suction line piping. Factory-assembled units with all parts cleaned at the time of assembly may not require the suction line strainer. A suction line strainer is normally required in all field-assembled systems.

Liquid Hazard
Liquid is essentially incompressible, so damage may occur when a compressor is handling liquid. This damage depends on the quantity of liquid, frequency with which it occurs, and type of compressor. Slugging, floodback, and flooded starts are three ways liquid can damage a compressor.

Slugging is the short-term pumping of a large quantity of liquid refrigerant and/or lubricant. It can occur just after start-up if refrigerant accumulated in the evaporator during shutdown returns to the compressor. It can also occur when system operating conditions change radically, such as during a defrost cycle. Slugging can also occur with quick changes in compressor loading.

Floodback is the continuous return of liquid refrigerant mixed with suction gas. It is a hazard to compressors that depend on maintaining a certain amount of lubricant for bearing surfaces. A properly sized suction accumulator can be used for protection.

Flooded start occurs when refrigerant is able to migrate to the compressor during shutdown. Compressors can be protected with crankcase heaters and automatic pumpdown cycles, where applicable.

Suction and Discharge Pulsations
Suction pulsation occurs because of the sudden flow and slight pressure drop in the suction line at the end of the reverse portion of the cycle and during suction. The frequency of this pulsation corresponds to the frequency of the compression cycle. Amplitude of the pulsation can reach 1.5 psi, especially if the compression chamber serves as a resonator. Suction pulsation may affect volumetric efficiency and a propagating compression wave may create structural problems caused by vibration of the suction line and evaporator. Suction pulsation has a negative effect on compressor sound and vibration levels. Specially designed suction mufflers can solve the problems induced by pressure pulsation. Also, a significant volume in the suction line, such as in a suction accumulator, or enclosed by the compressor shell, can reduce pulsation level. **Discharge pulsation** occurs because of the sudden flow and pressure fluctuation in the discharge line at the end of the compression portion of the cycle and during discharge. Over- or undercompression has a significant effect on these pulsations. The frequency of the discharge pulsation corresponds to the frequency of the compression cycle. Amplitude of the pulsation can reach more than 14.5 psi, especially if the compressor operates with CO₂. In many cases, discharge pulsation is a major contributor to compressor sound and vibration levels. Discharge pulsation can be very destructive (structural damage, damage to sensors, etc.). Discharge mufflers can alleviate the problems induced by pressure pulsation.

Noise
An acceptable sound level is a basic requirement of good design and application. The major contributors to compressor noise are internal turbulence, impact (valves), friction, and the electric motor. Using sound shields or blankets may be feasible in certain applications. [Chapter 48](#) of the *2011 ASHRAE Handbook—HVAC Applications* covers design criteria in more detail.

Vibration
Compressor vibration results from gas-pressure pulses and inertia associated with moving parts. If the compressor is considered as a cylindrical body, vibration can be differentiated as axial, radial, and torsional. With increase of pressure differential between discharge and suction, vibration increases, especially axial and radial components. At fixed suction and discharge conditions, lower compressor speed usually leads to higher vibration amplitude, especially in the torsional component. Vibration problems can be handled in the following ways.

Isolation. With this common method, the compressor is resiliently mounted in the unit by springs, synthetic rubber mounts, etc. In hermetic reciprocating compressors, the internal compressor assembly is usually spring-mounted within the welded shell, and the entire unit is externally isolated. Use of flexible suction and discharge tubes may be feasible in certain applications.

Amplitude Reduction. The amount of movement can be reduced by adding mass to the compressor. Mass is added either by rigidly attaching the compressor to a base, condenser, or chiller, or by providing a solid foundation. When structural transmission is a problem, particularly with large machines, the entire assembly is then resiliently mounted.

Balancing. Proper balancing of inertial forces is important in reducing vibration. Counterweights are often used in rotary and scroll compressors.

[Chapter 48](#) of the *2011 ASHRAE Handbook—HVAC Applications* has further information.

Shock
In designing for shock, three types of dynamic loads are recognized:

- Suddenly applied loads of short duration
- Suddenly applied loads of long duration
- Sustained periodic varying loads

Because the forces are primarily inertial, the basic approach is to maintain low equipment mass and make the strength of the carrying structure as great as possible. The degree to which this is done depends on the amount of shock loading.

Commercial Units. The major shock loading to these units occurs during shipment or when they operate on commercial carriers. Train service provides a severe test because of low forcing frequencies and high shock load. Shock loads as high as 10g have been recorded; 5g can be expected.

Trucking service results in higher forcing frequencies, but shock loads can be equal to, or greater than, those for rail transportation. Aircraft service forcing frequencies generally range from 20 to 60 Hz with shocks to 3g.

Military Units. Requirements are given in detail in specifications that exceed anything expected of commercial units. In severe applications, deformation of supporting members and shock isolators may be tolerated, provided that the unit performs its function.

Basically, the compressor components must be rigid enough to avoid misalignment or deformation during shock loading. Therefore, structures with low natural frequencies should be avoided.

Testing and Operating Requirements
Compressor tests are of two types: rating (performance) and reliability. **Standard rating conditions**, which are usually specified by compressor manufacturers, include the maximum possible compression ratio, maximum operating pressure differential, maximum permissible discharge pressure, and maximum inlet and discharge temperature. **Lubrication requirements**, which are prescribed by compressor manufacturers, include the type, viscosity, and other characteristics of the lubricants suitable for use with the many different operating levels and the specific refrigerant being used. **Power requirements** for compressor starting, pull-down, and operation vary, because unloading means differ in the many styles of compressors available. Manufacturers supply full information covering the various methods used. Testing for ratings must be in accordance with ASHRAE *Standard* 23. Manufacturers normally publish performance data at test conditions specified in AHRI *Standards* 520 or 540, or at other industry standard conditions. Additional compressor tests might address the following characteristics:

- Compressor performance over a range of conditions (performance curves)
- Sound level
- Durability or reliability
- Operational limits (operating envelope)
- Lubrication requirements (oil type, viscosity, amount, etc.)
- Electrical power requirements (start-up current draw, running current measurements, etc.)

Operating envelope shows compressor operating range as a function of saturation suction temperature or suction pressure (SST or SSP) and saturation discharge temperature or pressure (SDT or SDP). An example of the operating envelope is shown in [Figure 5](#).

This envelope is defined by the following extreme condition points:
High load (HL) is defined as intersection of maximum operating SDT or SDP and maximum operating SST or SSP. At this condition, the compressor experiences high power input, high average torque, and high average bearing load.

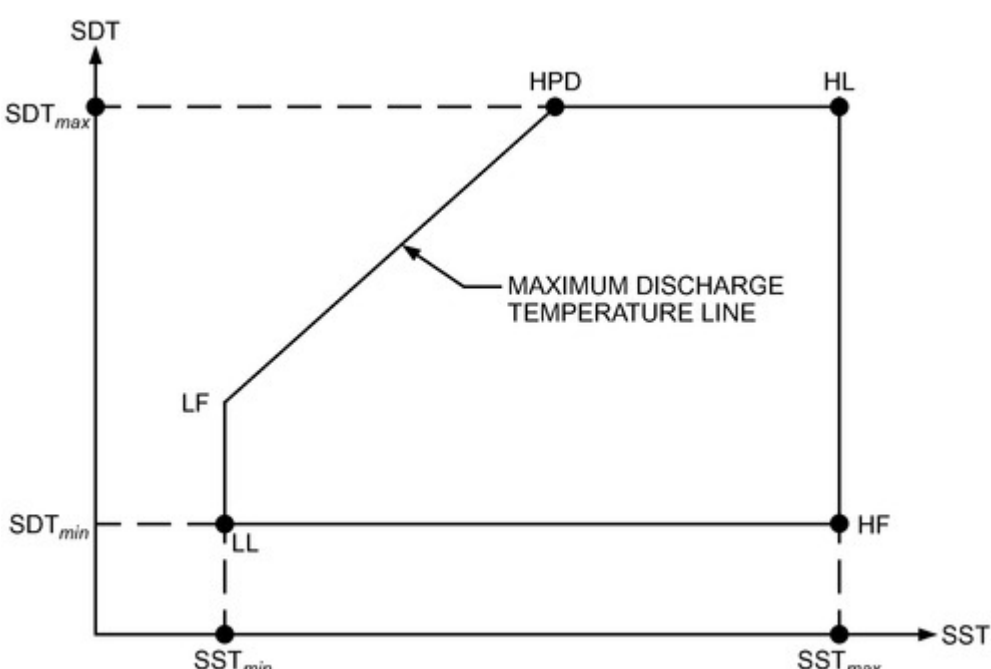


Figure 5. Example of Compressor Operating Envelope

High flow (HF) is defined as intersection of minimum operating SDT or SDP and maximum operating SST or SSP. At this condition, the compressor experiences high mass flow of the refrigerant and has high cooling capacity. **High pressure differential (HPD)** is defined as intersection of maximum operating SDT or SDP and the maximum discharge temperature line. At this condition, the compressor experiences high pressure differential between discharge and suction and the highest allowable discharge temperature. **Low flow (LF)** is defined as intersection of the maximum discharge temperature line and minimum operating SST or SSP. The combination of high pressure ratio and low refrigerant flow creates an extreme condition for compressor cooling and oil stability. Cooling capacity at this operating point is the lowest. **Low load (LL)** is defined as intersection of minimum operating SDT or SDP and minimum operating SST or SSP. At this point, the compressor experiences low power input and low average torque and bearing load. The common operating range for the air-conditioning SST is 15 to 60°F. The common operating range for the air-conditioning SDT is 75 to 160°F. There are compressor design options that can allow extension of these operating ranges. Note that compressor capacity (with given refrigerant) depends on SST, SDT, superheating, and subcooling.

1.3 MOTORS

Motors for positive-displacement compressors range from fractional to thousands of horsepower. When selecting a motor for driving a compressor, consider the following factors:

- Power and rotational speed.
- Voltage and number of phases. Smaller motors are usually 115 or 230 V and single phase. Large motors are usually three phase, 200 to 575 V.
- Starting and pull-up torques. Special attention should be paid to starting at extreme hot (highest system pressures) and cold (highest oil viscosity) temperatures. Consideration for system pressures at start-up (whether equalized or differential pressure exists across the compressor) is very important for single-phase compressors; start assist may be required with differential-pressure starting.
- Ambient and maximum temperature of the coolant. In hermetic and semihermetic compressors, the motor is usually cooled by refrigerant flow. This flow can be before suction (low-side compressors) or after discharge (high-side compressors). Note that there is no clear advantage or disadvantage in low- or high-side cooling. In low-side compressors, the motor is efficiently cooled by cold refrigerant returned from the evaporator. The cooler the motor, the higher its efficiency η_m . However, superheated refrigerant after the motor has lower density, which reduces the compressor capacity. In high-side compressors, it is an opposite effect: higher capacity and lower η_m (motor runs hotter). In both cases, the total effect on compressor performance is inconclusive and must be resolved by detailed thermal analysis or testing. Sometimes, selection of motor cooling is dictated by pure reliability or minimum size requirements (e.g., motors bigger than 5 hp usually require low-side cooling to be compact).
- Cost and availability.
- Insulation. In addition to electrical requirements (dielectric properties, high potential, etc.), motor insulation should be compatible with the refrigerant/oil mixture and should withstand a temperature up to 300°F without losing effectiveness.
- Efficiency and performance. The electrical motor performance curves (torque, efficiency, and power factor versus motor speed) must be analyzed against the compressor's operating requirements. For some applications, apparent efficiency (efficiency multiplied by power factor) is important because often the limiting factor is available kilovolt-amperes, not just power (watts).
- Locked-rotor amps. This important motor characteristic allows proper selection of motor protection devices or a current breaker. During the start of an ac motor without start assistance or a soft-start package, there is a significant inrush current. The amplitude of this current can be very close to locked-rotor amps. The duration of the inrush is also very important. It must be determined by testing, because it is a function of the individual compressor starting torque and operating speed.
- Type of protection required (fuse, internal or external current/temperature switch, etc.). See [Chapter 45](#) for more information.
- Single-speed, multispeed, variable-speed, or linear. Single-speed motors can be one, two, or three phase. Multispeed motors contain two or three sets of windings with different number of poles. Motors with two-pole windings provide speed equivalent to the electrical current (e.g., four-pole rotates at half the speed of two-pole). The following equation calculates motor synchronous speed based on electrical frequency and number of poles:

$$\text{Motor speed, rpm} = \frac{120 \times \text{Electrical frequency, Hz}}{\text{Number of poles}}$$

Multispeed motors require relays for switching the windings. These relays can be conventional or solid state (to avoid the effect of vibration).

Variable-speed motors can be ac, brushless dc (BLDC), or permanent magnet ac (PMAC) type. They require a special electrical controller/drive to operate. Any three-phase motor can be considered as an ac variable-speed motor. Control is comparatively simple, but motor efficiency usually does not exceed 90% because of slippage and rotor losses. BLDC and PMAC motors are much more efficient (up to 96%), but their control can be more involved. Permanent magnets are used in the rotors of BLDC and PMAC motors. Some BLDC and PMAC motors require Hall effect sensors or resolvers to identify rotor position, or sensorless technology can be used, which complicates the control. All BLDC and PMAC motors are synchronous, which means that their rotation is proportional to the frequency of the current (i.e., no slip).

Linear motors require a special controller and drive to operate.

- Location (high-pressure versus low-pressure side). High-side motors operate at much higher temperature than those on the low side, so their efficiency is noticeably lower. (See the discussion on ambient and maximum coolant temperature previously in this list.)

Although all motors can be started across-the-line, local utilities, local codes, or specification by the end user may require that motors be started at reduced power levels. These typically include part-winding, vye-delta, double-delta, autotransformer, and solid-state starting methods, all designed to limit inrush starting current. The chosen starting method must supply enough torque to accelerate the motor and overcome the torque required for compression.

Hermetic motors can be more highly loaded than comparably sized open motors because of the refrigerant/oil mixture used for cooling.

For effective hermetic motor application, the maximum design load should be as close as possible to the breakdown torque at the lowest voltage used. This approach yields a motor design that operates better at lighter loads and higher voltage. Overtorqued designs may increase discharge gas temperature at light loads and reduce compressor efficiency.

The single-phase motor presents more design problems than the polyphase, because the relationship between main and auxiliary windings becomes critical. Sometimes it is necessary to use starting equipment in this case.

The rate of temperature rise at the locked-rotor condition must be kept low enough to prevent excessive motor temperature with the motor protection available. The maximum temperature under these conditions should be held within the limits of the materials used. With better protection (and improved materials), a higher rate of rise can be tolerated, and a less expensive motor can be used.

Some types of hermetic motors commonly selected for various applications are as follows:

- **Small refrigeration compressors (single-phase)**
 - Low to medium torque—Split-phase or PSC (permanent split-capacitor)
 - High torque—CSCR (capacitor-start/capacitor-run) and CSIR (capacitor-start/induction-run)
- **Room air conditioner compressors (single-phase)**
 - PSC or CSCR
 - Two-speed (pole-switching) or variable-speed

Central air conditioning and commercial refrigeration

- Single-phase, PSC and CSCR to 4500 W
- Three-phase, 2 hp and above, across-the-line start
- 10 hp and above; part-winding; vye-delta, double-delta, and across-the-line start
- Two-speed (pole-switching) or variable-speed
- Electronically commutated dc motors (ECMs)

For further information on motors and motor protection, see [Chapter 45](#). Also see [Chapter 56](#) of the *2015 ASHRAE Handbook—HVAC Applications* for more on motor-starting effects.

2. RECIPROCATING COMPRESSORS

[Table 1](#) lists typical design features of reciprocating compressors. Most reciprocating compressors are single-acting, using pistons that are driven directly through a pin and connecting rod from the crankshaft. Double-acting compressors that use piston rods, crossheads, stuffing boxes, and oil injection are not used extensively and, therefore, are not covered here. [Figures 6](#) and [7](#) show the basic structure and pumping cycle for a typical reciprocating compressor piston. [Figures 8](#) and [9](#) show typical designs of multicylinder reciprocating compressors.

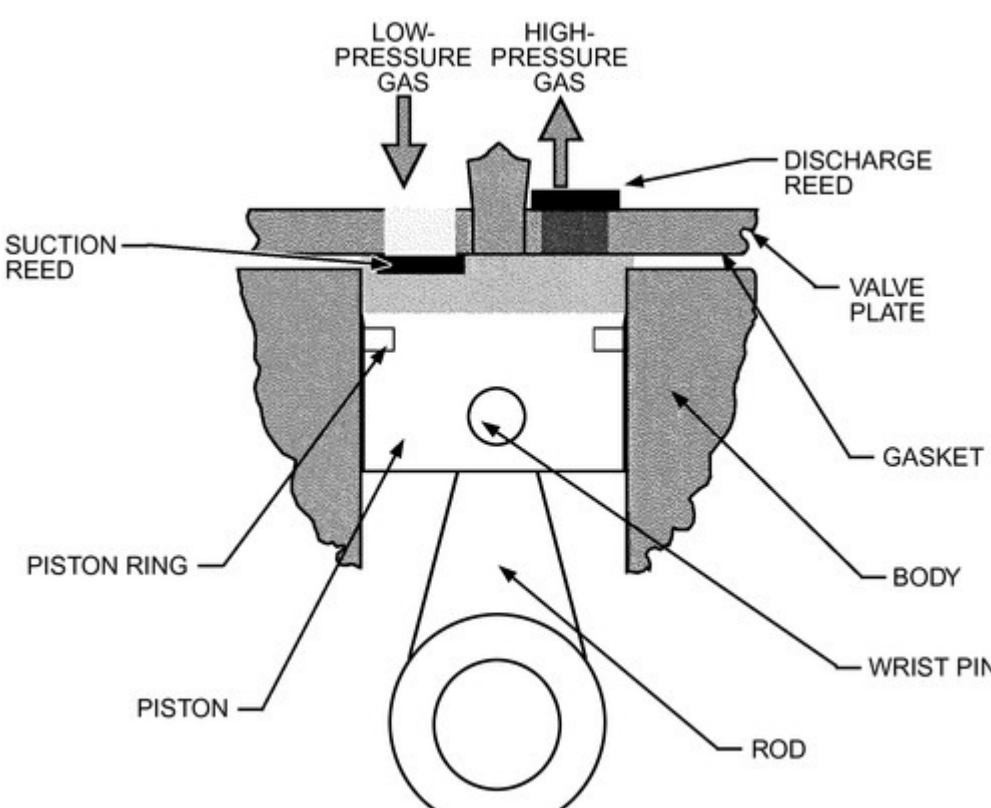


Figure 6. Basic Reciprocating Piston with Reed Valves

Single-stage compressors are primarily used for medium temperatures (0 to 30°F) and in air-conditioning applications, but can achieve temperatures below −30°F for refrigeration applications with suitable refrigerants. [Chapters 1](#) and [2](#) of the *2014 ASHRAE Handbook—Refrigeration* have information on other halocarbon and ammonia systems.

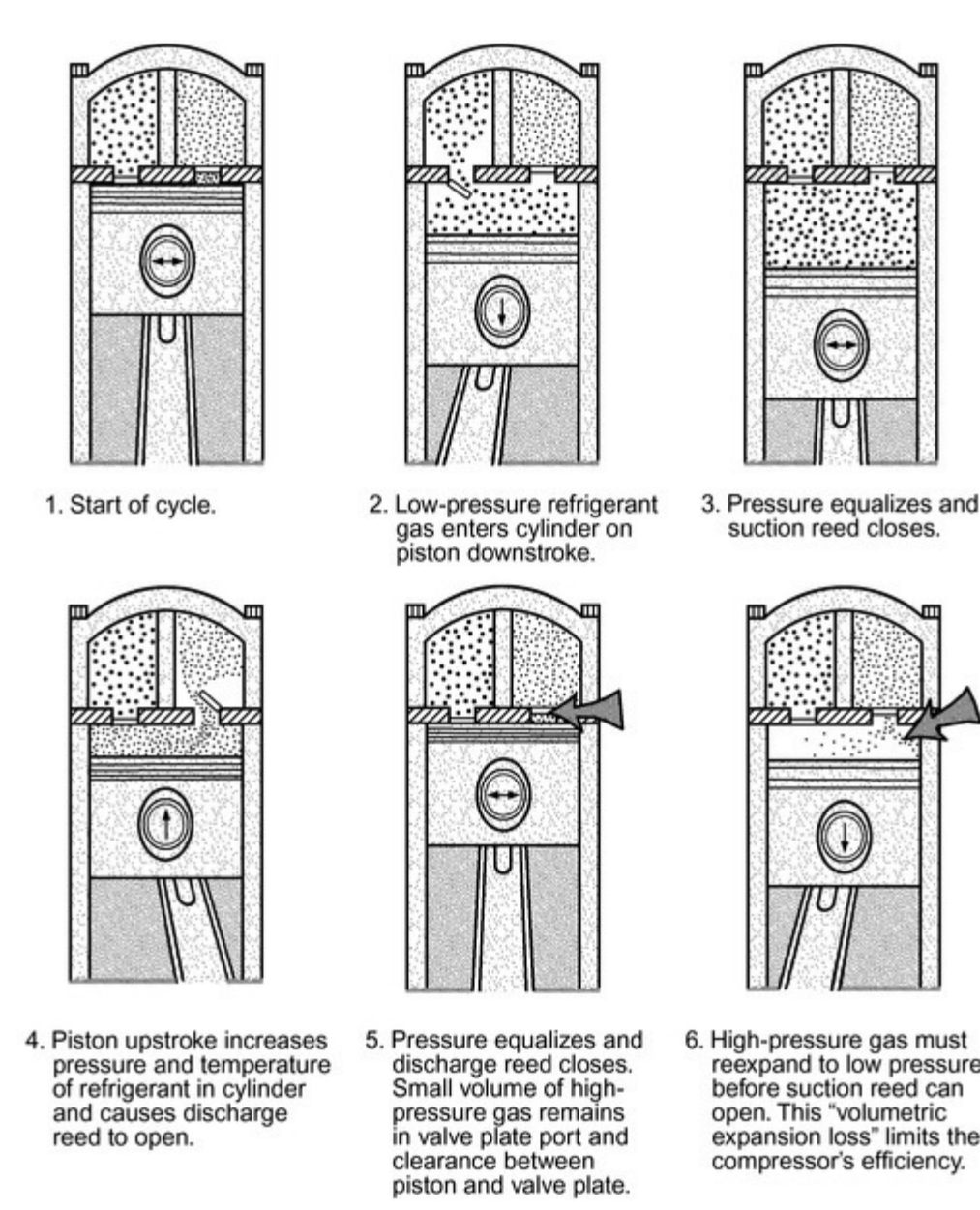


Figure 7. Pumping Cycle of Reciprocating Compressor

Booster compressors are typically used for low-temperature applications with R-22, R-404A/R-507, or ammonia. Saturated suction at –85°F can be achieved by using R-22 or R-404A/R-507, and –65°F saturated suction is possible using ammonia. The booster raises refrigerant pressure to a level where further compression can be achieved with a high-stage compressor, without exceeding the pressure-ratio limits of the respective machines. Because superheat is generated as a result of compression in the booster, intercooling is normally required to reduce the refrigerant stream temperature to the level required at the inlet to the high-stage unit. Intercooling methods include controlled liquid injection into the intermediate stream, mixing-type heat exchangers, and heat exchangers where no fluid mixing occurs. **Integral two-stage compressors** achieve low temperatures (–20 to –80°F) using appropriate refrigerants within the frame of a single compressor. Cylinders in the compressor are divided into respective groups so that the combination of volumetric flow and pressure ratios are balanced to achieve booster and high-stage performance effectively. Refrigerant connections between high-pressure suction and low-pressure discharge stages allow an interstage gas cooling system to be connected to remove superheat between stages. The intercooling in this case is similar to the methods used for individual high-stage and booster compressors.

Table 1. Typical Design Features of Reciprocating Compressors

Item	Refrigerant Type			
	Halo-, Fluoro-, or Hydrocarbon			Ammonia
	Open	Semihermetic	Welded Hermetic	Open
1. Number of cylinders: one to	16	12	6	16
2. Power range	0.17 hp and up	0.5 to 150 hp	0.17 to 25 hp	10 hp and up
3. Cylinder arrangement				
a. Vertical, V or W, radial	X	X		
b. Radial, horizontal opposed			X	
c. Horizontal, vertical V or W		X		X
4. Drive				
a. Electric motor				
b. Direct drive, (coupling), V belt chain, gear, by electric motor or engine	X	X	X	X
5. Lubrication: splash or force feed, flooded	X	X	X	X
6. Suction and discharge valves: ring plate or ring or reed flexing	X	X	X	X
7. Suction and discharge valve arrangement				
a. Suction and discharge valves in head	X	X	X	X
b. Uniflow: suction valves in top of piston, suction gas entering through cylinder walls; discharge valves in head	X			X
8. Cylinder cooling				
a. Suction-gas-cooled	X	X	X	X
b. Water jacket cylinder wall, head, or cylinder wall and head	X			X
c. Air-cooled	X	X	X	X
d. Refrigerant-cooled heads	X			X
9. Cylinder head				
a. Spring-loaded	X	X	X	X
b. Bolted	X	X	X	X
10. Bearings				
a. Sleeve, antifriction	X	X	X	X
b. Tapered roller	X			X
11. Capacity control, if provided: manual or automatic				
a. Suction valve lifting	X	X	X	X
b. Bypass-cylinder heads to suction	X	X	X	X
c. Closing inlet	X	X		X
d. Adjustable clearance	X	X		X
e. Variable-speed	X	X	X	X
12. Materials				
Motor insulations and rubber materials must be compatible with refrigerant and lubricant mixtures; otherwise, no restrictions		X	X	
No copper or brass				X
13. Lubricant return				
a. Crankcase separated from suction manifolds, oil return check valves, equalizers, spinners, foam breakers	X	X		X
b. Crankcase common with suction manifold			X	
14. Synchronous fixed speeds, rpm	250 to 3600	1500 to 3600	1500 to 3600	250 to 1500
15. Pistons				
a. Aluminum or cast iron	X	X	X	X
b. Ringless	X	X	X	X
c. Compression and oil-control rings	X	X	X	X
16. Connecting rod				
Split rod with removable cap or solid eccentric strap	X	X	X	X
17. Mounting				
Internal spring mount		X	X	
External spring mount		X	X	
Rigidly mounted on base	X	X		X

Capacity reduction with reciprocating compressors may be achieved by cylinder unloading, as in the case of single-stage compressors. Special consideration must be given to maintaining the correct relationship between high- and low-pressure stages.

Performance Data

Figure 10 shows a typical set of capacity and power curves for a four-cylinder semihermetic compressor, 2.38 in. bore, 1.75 in. stroke, 1740 rpm, operating with R-22. Compressor curves should contain the following information:

- Compressor identification
- Degrees of subcooling and correction factors for zero or other subcooling temperatures
- Degrees of superheat
- Compressor speed
- Refrigerant
- Suction gas superheat and correction factors
- Compressor ambient
- External cooling requirements (if any)
- Maximum power or maximum operating conditions
- Minimum operating conditions at fully loaded and fully unloaded operation

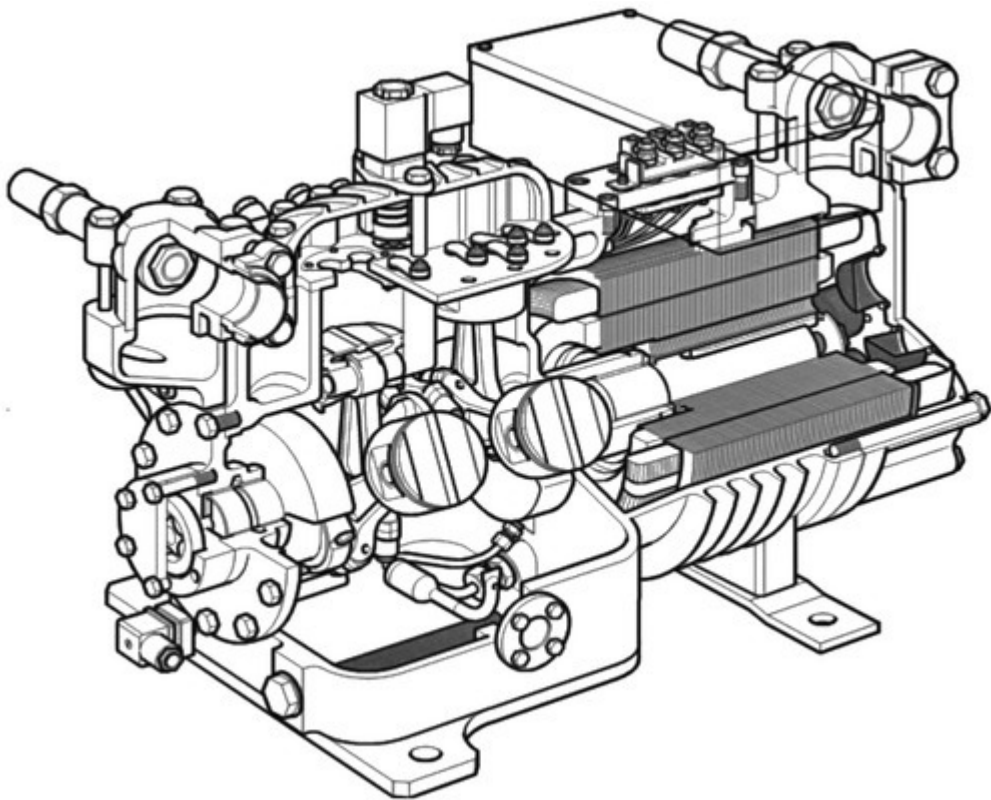


Figure 8. Six-Cylinder Semihermetic Compressor

Motor Performance

Motor efficiency is usually a compromise between cost and size. Generally, the physically larger a motor is for a given rating, the more efficient it can be. The accepted efficiency range for ac motors is 85 to 95%. Uneven loading has a marked effect on motor efficiency. It is important that cylinders be spaced evenly. Also, the more cylinders there are, the smaller the impulses become. Greater moments of inertia of moving parts and higher speeds reduce the impulse effect. Small, evenly spaced impulses also help reduce noise and vibration.

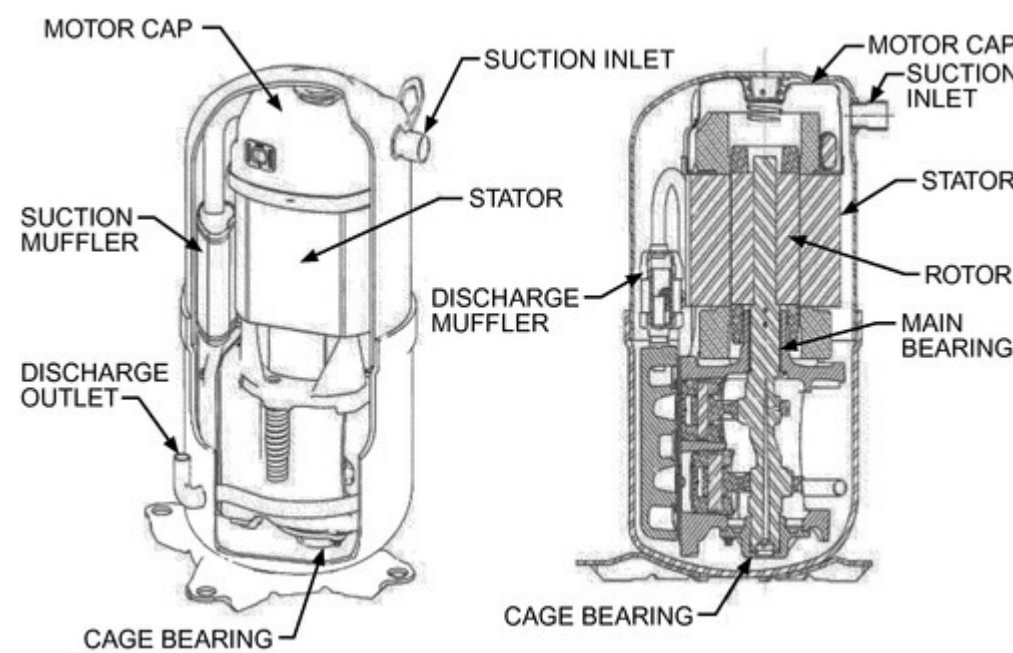


Figure 9. Hermetic Reciprocating Compressor

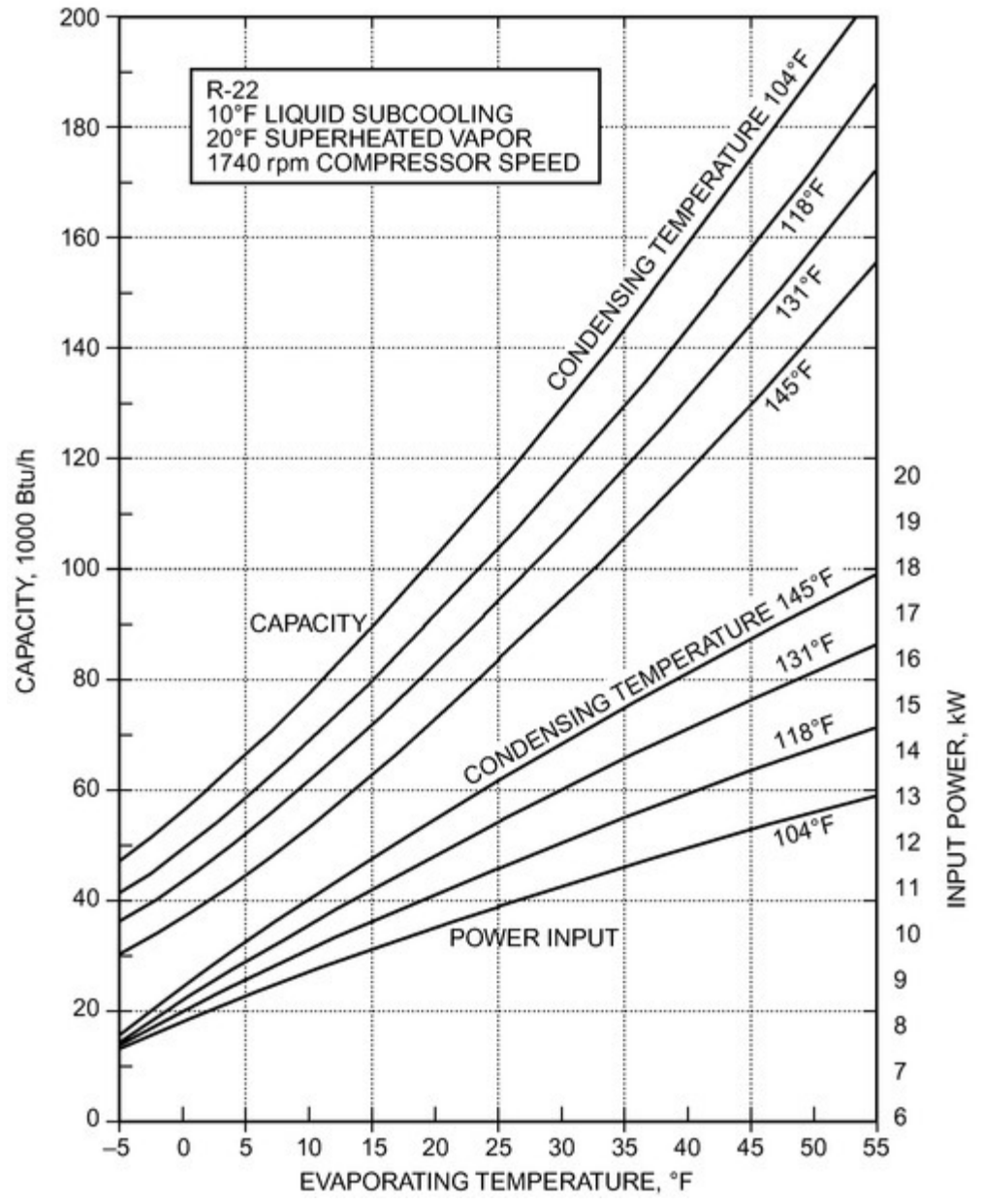


Figure 10. Capacity and Power-Input Curves for Typical Semihermetic Reciprocating Compressor

Because many compressors start against load, it is desirable to estimate starting torque. The following equation is for a single-cylinder compressor. It neglects friction, valve losses, leakage, and the fact that tangential force at the crankpin is not always equal to normal force at the piston. This equation also assumes considerable gas leakage at the discharge valves but little or no leakage past the piston rings or suction valves. It gives only a preliminary estimate.

$$T_s = \frac{(p_2 - p_1)As}{2N_2/N_1}$$

(11)

where

- T_s = starting torque, lb·in.
- p_2 = discharge pressure, psi
- p_1 = pressure on other side of piston, psi
- A = area of cylinder, in²
- s = stroke of compressor, in.
- N_2 = motor speed, rpm
- N_1 = compressor speed, rpm

Table 2. Motor-Starting Torques

No. Cylinders	Arrangement of Crank Throws	Angle Between Cylinders	Approximate Torque from Equation (11)
1	Single	N/A	T_s
2	Single	90°	1.025 T_s
2	180° apart	0° or 180°	T_s
3	Single	60°	1.225 T_s
3	120° apart	120°	T_s
4	180°, 2 rods/crank	90°	1.025 T_s
6	180°, 3 rods/crank	60°	1.23 T_s

Equation (11) shows that when pressures are balanced or almost equal ($p_2 = p_1$), torque requirements are considerably reduced. Thus, a pressure-balancing device on an expansion valve or a capillary tube that equalizes pressures at shutdown allows the compressor to be started without excessive effort. For multicylinder compressors, both the number of cylinders that might be on a compression stroke and the position of the rods at start must be analyzed. Because the force needed to push the piston to the top dead center is a function of how far the rod is from the cylinder centerline, the worst possible angles these might assume can be graphically determined by torque diagrams. The torques for some arrangements are shown in Table 2.

Main Bearings. Both the crank and drive means may be overhung with bearings in between; however, usual practice places the cylinders between main bearings. Main bearings are made of steel-backed babbitt, steel-backed or solid bronze, or aluminum. Bearings are usually integral to an aluminum crankcase. By automotive standards, unit loadings are low. The oil/refrigerant mixture frequently provides only marginal lubrication, but 8000 h/year operation in commercial refrigeration service is possible. For conventional shaft diameters and speeds, 600 psi main bearing loading based on projected area is not unusual. Running clearances average 0.001 in. per inch of diameter with steel-backed babbitt bearings and a steel or iron shaft. Bearing oil grooves placed in the unloaded area are usual. Feeding oil to the bearing is only one requirement; another is venting evolved refrigerant gas and providing the lubricant escape from the bearing to carry away heat.

In most compressors, crankshaft thrust surfaces (with or without thrust washers) must be provided in addition to main bearings. Thrust washers may be steel-backed babbitt, bronze, aluminum, hardened steel, or polymer and are usually stationary. Oil grooves are often included in the thrust face. Connecting rods have the large end split and a bolted cap for assembly. Unsplit eccentric straps require the crankshaft to be passed through the big bore at assembly. Rods or straps are of steel, aluminum, bronze, nodular iron, or gray iron. Steel or iron rods often require inserts of bearing material such as steel-backed babbitt or bronze, whereas aluminum and bronze rods can bear directly on the crankpin and piston pin. Refrigerant compressor service limits unit loading to 3000 psi based on projected area with a bronze bushing in the rod small bore and a hardened steel piston pin. An aluminum rod load at the piston pin of 2000 psi has been used. Large end-unit loads are usually under 1000 psi.

The Scotch yoke type of piston-rod assembly has also been used. In small compressors, it has been fabricated by hydrogen-brazing steel components. Machined aluminum components have been used in large hermetic designs. Piston, Piston Ring, and Piston Pin. Pistons are usually made of cast iron or aluminum. Cast-iron pistons with a running clearance of 0.0004 in. per inch of diameter in the cylinder will seal adequately without piston rings. With aluminum pistons, rings are required because a running clearance in the cylinder of 0.002 in. per inch or more of diameter may be necessary, as determined by tests at extreme conditions. A second or third compression ring may add to power consumption with little increase in capacity; however, it may help oil control, particularly if drained. Oil-scraping rings with vented grooves may also be used. Cylinder finishes are usually obtained by honing, and a 12 to 40 μ m range will give good ring sealing. An effective oil scraper can often be obtained with a sharp corner on the piston skirt.

Minimum piston length is determined by the side thrust and is also a function of running clearance. Where clearance is large, pistons should be longer to prevent slap. An aluminum piston (with ring) having a length equal to 0.75 times the diameter, with a running clearance of 0.002 in. per inch of diameter, and a rod-length-to-crank-arm ratio of 4.5 has been used successfully. Piston pins are steel, case-hardened to Rockwell C 50 to 60 and ground to a 8 μ m finish or better. Pins can be restrained against rotation in either the piston bosses or the rod small end, be free in both, or be full-floating, which is usually the case with aluminum pistons and rods. Retaining rings prevent the pin from moving endwise and abrading the cylinder wall.

There is no well-defined limit to piston speed; an average velocity of 1200 fpm, determined by multiplying twice the stroke in feet by the rpm, has been used successfully. Suction and Discharge Valves. These valves are important components of a reciprocating compressor. Successful designs provide high volumetric efficiency and low pressure loss. Improper timing of opening/closing and excessive leakage significantly affect volumetric efficiency. Excessive pressure loss across the valve results from high gas velocities, poor mechanical action, or both. Note that the valve flow area gradually increases during opening, which creates conditions for an overpressure pulse in discharge valves and underpressure pulse for suction valves. These pulses significantly affect compressor performance, noise, and vibration. Minimizing such pulses usually leads to better overall compressor performance.

A valve should meet the following requirements:

- Sufficient flow areas with shortest possible path
- Straight gas flow path, minimum directional changes
- Valve mass and lift should satisfy timing requirements
- Symmetry of design with minimum pressure imbalance
- Minimum clearance volume
- Durability
- Low cost
- Tight sealing at ports (low leakage)
- Minimum valve flutter

Most valves in use today fall in one of the following groups:

- **Free-floating reed valve,** with backing to limit movement, seats against a flat surface with circular or elongated ports. It is simple, and stresses can be readily determined, but it is limited to relatively small ports; therefore, multiples are often used. Totally backed with a curved stop, it can stand considerable abuse.
- **Reed, clamped at one end,** with full backstop support or a stop at the tip to limit movement, has a more complex motion than a free-floating reed; it may have multiple modes of deformation. Considerable care must be taken in design to ensure reliability.
- **Ring valve** usually has a spring return. A free-floating ring is seldom used because of its high leakage loss. Improved performance is obtained by using spring return, in the form of coil springs or flexing backup springs, with each valve. Ring-type valves are particularly adaptable to compressors using cylinder sleeves.
- **Valve formed as a ring** has part of the valve structure clamped. Generally, full rings are used with one or more sets of slots arranged in circles. By clamping the center, alignment is ensured and a force is obtained that closes the valve. To limit stresses, the valve proportions, valve stops, and supports are designed to control and limit valve motion.

Lubrication. Lubrication systems range from a simple splash system to the elaborate forced-feed systems with filters, vents, and equalizers. The type of lubrication required depends largely on bearing load and application. For low to medium bearing loads and factory-assembled systems where cleanliness can be controlled, the splash system gives excellent service. Bearing clearances must be large, however; otherwise, oil does not enter the bearing readily. Thus, the splashing effect of the dippers in the oil and the freer bearings cause the compressor to operate somewhat noisily. Furthermore, the splash at high speed encourages frothing (foaming) and oil pumping; this is not a problem in packaged equipment but may be in remote systems where gas lines are long. Foaming can also help reduce compressor noise, and sometimes foaming agents can be added to the oil just for this purpose.

Lubrication for a **flooded system** includes disks, screws, grooves, oil-ring gears, or other devices that lift the oil to the shaft or bearing level. These devices flood the bearing and are not much better than splash systems, except that the oil is not agitated as violently, so operation is quieter. Because little or no pressure is developed by this method, it is not considered forced-feed.

In **forced-feed lubrication,** a pump gear, vane, or plunger develops pressure, which forces oil into the bearing. Smaller bearing clearances can be used because adequate pressure feeds oil in sufficient quantity for proper bearing cooling. As a result, compressor operation may be quieter. Gear pumps are common. Spur gears are simple but tend to promote flashing of refrigerant dissolved in the oil because of the sudden opening of the tooth volume as two teeth disengage. This disadvantage is not apparent in internal eccentric gear or vane pumps where the suction volume gradually opens. The eccentric gear pump, vane pump, or piston pump therefore give better performance than simple gear pumps when the pump is not submerged in the oil.

Oil pumps must be made with proper clearances to pump a mixture of gas and oil. The pump discharge should have provision to bleed a small quantity of oil into the crankcase. A bleed vents the pumps, prevents excess pressure, and ensures faster priming. A strainer should be inserted in the suction line to keep foreign substances from the pump and bearings. If large quantities of the particles are present and bearing load is high, it may be necessary to add an oil filter to the discharge side of the pump.

Oil must return from suction gas into the compressor crankcase. A flow of gas from piston leakage opposes this oil flow, so leakage gas velocity must be low to permit oil to separate from the gas. A separating chamber may be built as part of the compressor to help separate oil from the gas. In many designs, a check valve is inserted at the bottom of the oil return port to prevent a surge of crankcase oil from entering the suction. This check valve must have a bypass, which is always open, to allow the check valve to open wide after the oil surge has passed. When a separating chamber is used, the oil surge is trapped before it can enter the suction port, thus making a check valve less essential.

Seals. Stationary and rotary seals have been used extensively on open reciprocating compressors. Older stationary seals usually used metallic bellows and a hardened shaft for a wearing surface. Their use has diminished because of high cost. The rotary seal costs less and is more reliable. A synthetic seal tightly fitted to the shaft prevents leakage and seals against the back face of the stationary member of the seal. The front face of this carbon nose seals against a stationary cover plate. This design has been used on shafts up to 4 in. in diameter. The rotary seal should be designed so that the carbon nose is never subjected to the full thrust of the shaft, the spring should be designed for minimum cocking force, and materials should be selected to minimize swelling and shrinking.

Special Devices

Capacity Control. Capacity control may be obtained by (1) controlling suction pressure by throttling, (2) controlling discharge pressure, (3) returning discharge gas to suction, (4) adding reexpansion volume, (5) changing the stroke, (6) opening a cylinder discharge port to suction while closing the port to discharge manifold, (7) changing compressor speed, (8) closing off cylinder inlet, and (9) holding the suction valve open. The most commonly used methods are opening the suction valves by some external force (9), gas bypassing within the compressor (6), suction shutoff, gas bypassing outside the compressor (3), and variable speed (7).

When capacity control compressors are used, system design becomes more important, and the following must be considered:

- Possible increase in compressor vibration (lower capacity), discharge gas pulsations, and sound level (higher capacity)
- Minimum operating conditions as limited by discharge or motor temperatures (or both) at part-load conditions
- Good oil return at lowest capacity
- Rapid cycling of unloaders
- Refrigerant metering device (TXV or capillary tube) capable of controlling at minimum capacity

Crankcase Heaters. When it is possible that refrigerant can accumulate in the compressor crankcase (cold start, gravitation, etc.), dilute the oil excessively, and result in flooded starts, a crankcase heater should be used. The heater should maintain the oil at least 20°F above the rest of the system at shutdown and well below the breakdown temperature of the oil at any time.

Internal Centrifugal Separators. Some compressors are equipped with antislug devices in the gas path to the cylinders. This device centrifugally separates oil and liquid refrigerant from the flow of foam during a flooded start and thus protects the cylinders. It does not eliminate other hazards caused by liquid refrigerant in gas compressors.

Automatic Oil Separators. Oil separators are used most often to reduce the amount of oil discharged into the system by the compressor and to return oil to the crankcase. They are recommended for all field-erected systems and on packaged equipment where lubricant contamination has a negative effect on evaporator capacity and/or where lubricant return at reduced capacity is marginal.

Application

Parallel Operation. Where multiple compressors are used, the trend is toward completely independent refrigerant circuits. This has an obvious advantage in the case of hermetic motor burnout and with lubricant equalization.

Parallel operation of compressors in a single system has some operational advantage at part load. The inherent higher system efficiency when operating only one compressor is a desirable advantage. However, two smaller compressors that are manufactured in very high volume are also likely to cost less than a single commercial compressor of equal capacity. The displacement of each compressor can also be varied to provide even more capacity control options.

Careful attention must be given to apportioning returned oil to the multiple compressors so that each always has an adequate quantity. Figure 11 shows the method most widely used. Line A connects the tops of the crankcases and tends to equalize the pressure above the oil; line B permits oil equalization at the normal level. Lines of generous diameter must be used. Generally, line A is a large diameter, and line B is a small diameter, which limits possible blowing of oil from one crankcase to the other.

A central reservoir for returned oil may also be used with means (such as crankcase float valves) for maintaining the proper levels in the various compressors. With staged systems, the low-stage compressor oil pump can sometimes deliver a measured amount of oil to the high-stage crankcase. The high-stage oil return is then sized and located to return a slightly greater quantity of oil to the low-stage crankcase. Where compressors are at different elevations and/or staged, pumps in each oil line are necessary to maintain adequate crankcase oil level. In both cases, proper gas equalization must be provided.

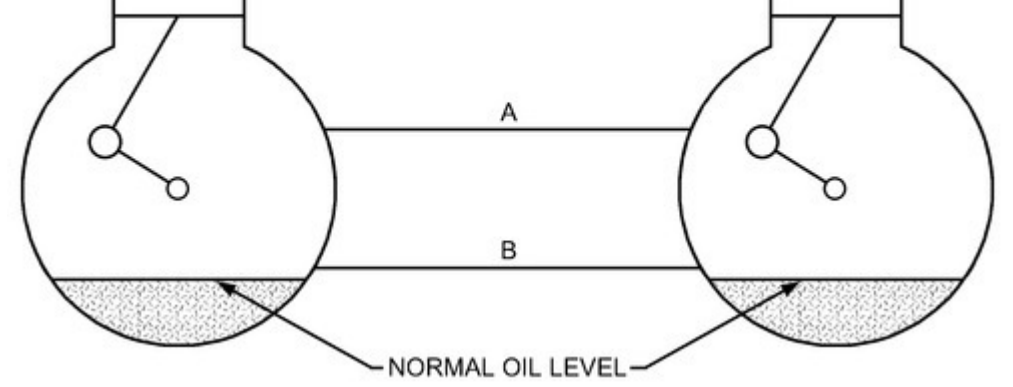


Figure 11. Modified Oil-Equalizing System

When fitted with check valves as noted in Figure 12, the compressors may be staged on to reduce power inrush, and/or each compressor can operate independently to provide refrigeration capacity modulation. For vibration control and optimum interconnecting tubing life, the compressors are usually hard mounted to a set of rails as shown, and the assembly is then isolated from the system base pan using rubber mounting grommets.

Operation at Low Suction Pressures. Because reciprocating compressors do not rely on refrigerant flow to deliver oil and cool the surfaces with boundary lubrication, they are uniquely suitable for low-suction-pressure applications, such as medium- and low-temperature refrigeration. Efficiency of these compressors is lower than more advanced designs, but reliability is extremely high.

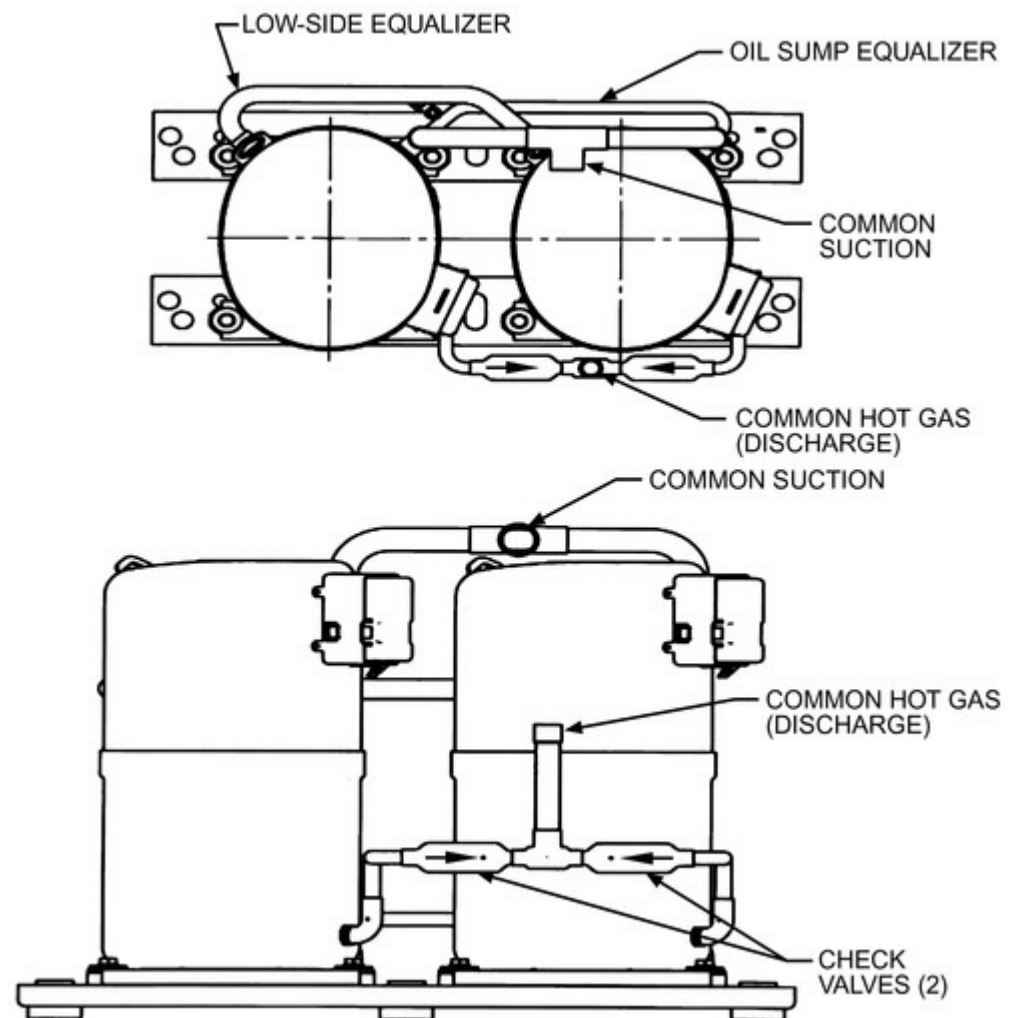
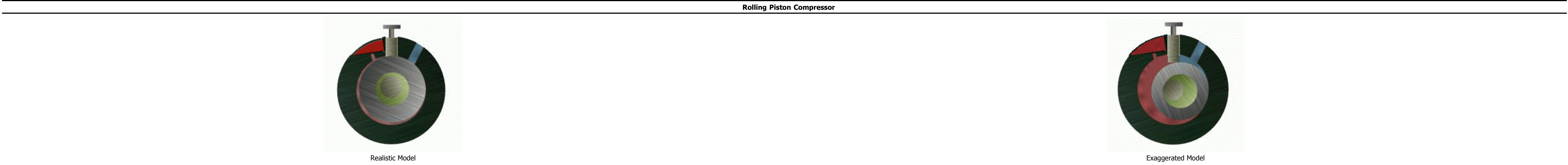


Figure 12. Tandem Compressors

3. ROTARY COMPRESSORS

3.1 ROLLING-PISTON COMPRESSORS

Rolling-piston, or fixed-vane, rotary compressors are used in air-conditioning units in capacities up to about 30 kBu/h and refrigerators up to 2.5 hp (Figure 13). This type of compressor uses a roller mounted on the eccentric of a shaft with a single vane or blade suitably positioned in the nonrotating cylindrical housing, generally called the cylinder block. The blade reciprocates in a slot machined in the cylinder block. This reciprocating motion is caused by the eccentrically moving roller.



1. Modeling and Animations using 3D Studio Max.
2. Air flow animation using Photoshop.
3. Compositing Frames using Compozer.
4. GIF animation using gifmerge.

Figure 13. Fixed-Vane, Rolling-Piston Rotary Compressor

Geometrical displacement for this compressor can be calculated from

$$V_d = \pi H(D^2 - d^2)/4$$

(12)

where

V_d = displacement

H = cylinder block height

D = cylinder diameter

d = roller diameter

The drive motor stator and compressor are solidly mounted in the compressor housing. This design feature can lead to significant torsional vibration, and special measures should be implemented to avoid damage to suction and discharge tubes. A discharge tube attached to the compressor should be in the form of "C" coil. If a suction accumulator is used, it must be attached to the compressor shell by supporting brackets of sufficient strength. Special grommets should be used to avoid transmitting the vibration to the compressor support. Using flexible suction and discharge tubes can significantly reduce vibration transmission and noise.

Because the amplitude of the torsional vibration increases at lower compressor speed, it may be a problem to operate a single-cylinder rotary compressor below 30 revolutions per second. Twin-cylinder rotary compressors with eccentrics on the same shaft in opposite directions can produce less torsional vibration and are suitable to operate at as low as 10 revolutions per second.

Rotary types are usually high-side compressors. Suction gas is piped directly into the suction port of the compressor, and compressed gas is discharged into the compressor housing shell. This high-side shell design is used because, in this case, the vane does not require a strong spring (after discharge pressure is built up, it creates enough force to maintain engagement between the roller and vane without the spring force) and lubrication of the vane in the slot is ensured by the pressure differential between the oil sump (high pressure) and suction chamber.

To avoid significant discharge pulsations inside the shell and the consequent high noise level, a special discharge muffler is installed to cover the discharge valve. Discharge mufflers can be made from different materials, including plastics.

Maintaining the proper oil level in rotary compressors is extremely important. The oil level should be high enough to cover the vane, providing adequate lubrication and leakage reduction, but not higher than the discharge port, because excessive oil will be pushed out of the compressor by discharge gas and may contaminate the heat exchangers. In some designs, the electrical rotor is used as an oil separator (special blades are installed on the rotor top), or a special plate is installed to shield the discharge tube entry and force separation of refrigerant and oil by inertia.

Internal leakage is controlled through hydrodynamic sealing and selection of mating parts for optimum clearance. Hydrodynamic sealing depends on clearance, surface speed, surface finish, and oil viscosity. Close tolerance and low-surface-finish machining is necessary to support hydrodynamic sealing and to reduce gas leakage.

Performance

Rotary compressors have a high volumetric efficiency because of the small clearance volume and correspondingly low reexpansion losses inherent in their design. Figure 14 and Table 3 show performance of a typical rolling-piston rotary compressor, commercially available for room air conditioning and small, packaged heat pump applications.

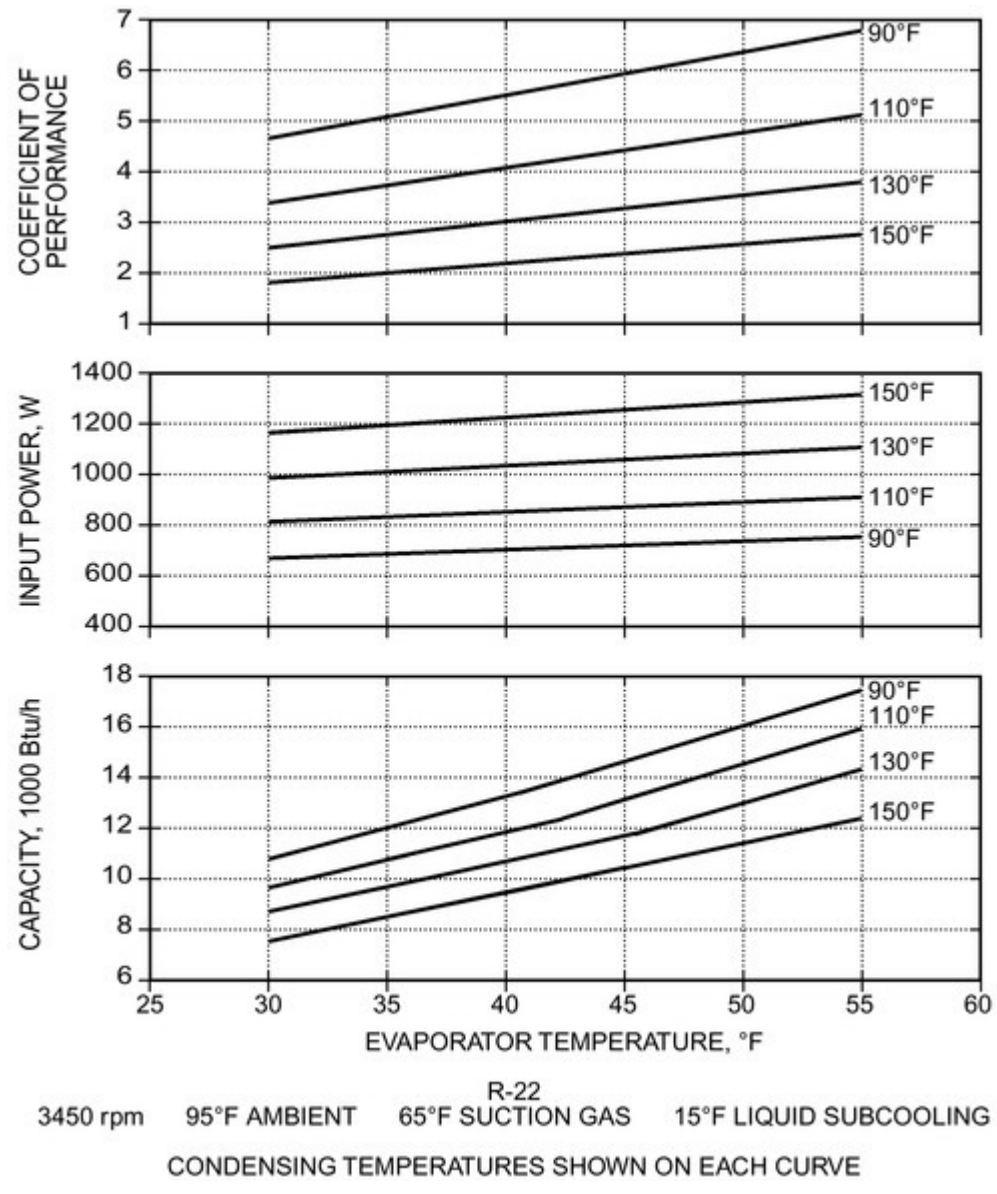


Figure 14. Performance Curves for Typical Rolling-Piston Compressor

An acceptable sound level is important in the design of any small compressor. Figure 15 shows the influence of a compressor on home refrigerator noise. Because gas flow is continuous and no suction valve is required, rotary compressors can be relatively quiet. The sound spectrum (sound "signature") of the rotary compressors is quite distinctive. At 60 revolutions per second, sound level elevation between 600 and 900 Hz can be explained by gas flow and pulsations, from 1500 to 2500 Hz by valve impact, and above 3500 Hz mostly by friction.

Features

Shafts and Journals. Shaft deflection under load is caused by compression gas loading of the roller and the torsional and side-pull loading of the motor rotor. Design criteria must require minimum oil film under the maximum run and starting loads. The motor rotor should have minimal deflection, to eliminate starting problems under extreme conditions of torque; the air gap between rotor and stator should be maintained as constant as possible around the circumference.

Shafts are generally made from steel forging and ductile iron. Journals are ground round to high precision and polished to a finish of 10 μin or better. The lower portion of the shaft (eccentric and lower journals) is hollow inside and serves as an oil pump using centrifugal force to deliver oil to the thrust surfaces and journals through labyrinth holes. Vertical or angular grooves are made in the journals or bearings, to aid in distribution of the oil.

Bearings. The bearing must support the rotating member under all conditions. Powdered metal has been extensively used for these components, because its porous properties help in lubrication. This material can also be formed into complex bearing shapes with little machining required. Cast iron is also widely used, especially for the lower bearing.

Vanes. Vanes are designed for reliability by the choice of materials and lubrication. The vanes are hardened, ground, and polished to the best finish obtainable. Steel, powdered metal, and aluminum alloys have been used; however, the best results are achieved by using M2 tool steel. Special coatings and excessive hardness (above 70 Rockwell C) have not shown meaningful improvement in reliability or long-term performance. Aluminum is impractical because of its significant difference from iron in thermal expansion, which requires higher clearance between the vane and vane slot and, consequently, leads to higher leakage and inconsistent lubrication.

Table 3. Typical Rolling-Piston Compressor Performance	
Compressor speed	3450 rpm
Refrigerant	R-22
Condensing temperature	130°F
Liquid refrigerant temperature	115°F
Evaporator temperature	45°F
Suction pressure	90.7 psia
Suction gas temperature	65°F
Evaporator capacity	12,000 Btu/h
Energy efficiency ratio	11.0 Btu/W · h
Coefficient of performance	3.22
Input power	1090 W

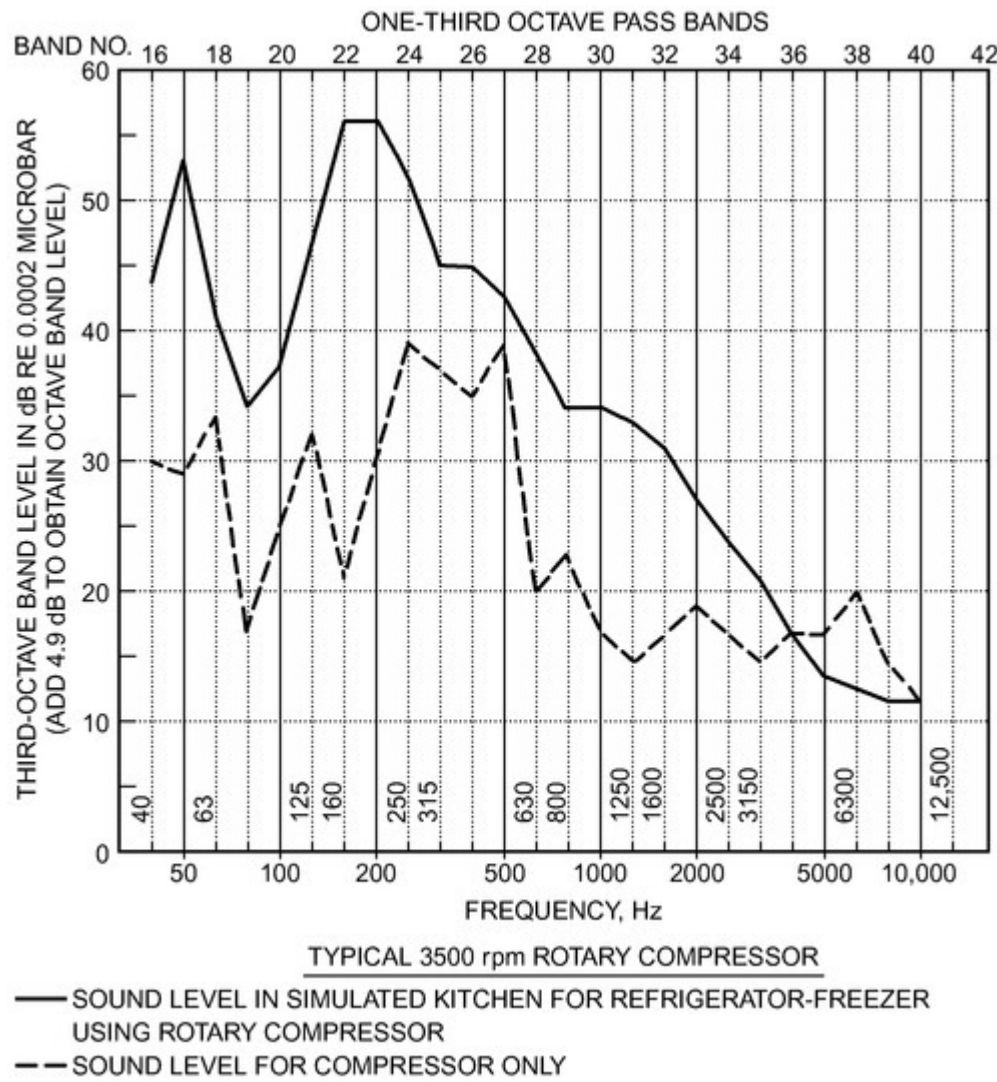


Figure 15. Sound Level of Combination Refrigerator-Freezer with Typical Rotary Compressor

The vane tip radius should be selected to minimize Hertz stresses between the roller and the vane, but the sharp edge of the vane should never be in contact with the roller.

Vane Springs. Vane springs force the vane to stay in contact with the roller during start-up. The spring rate depends on the inertia of the vane. Two types of springs are widely used: C-type and helical.

Valves. Only discharge valves are required by rolling-piston rotary compressors. They are usually simple reed valves made of high-grade steel. Proper design of the valve stop is important for valve reliability and noise reduction.

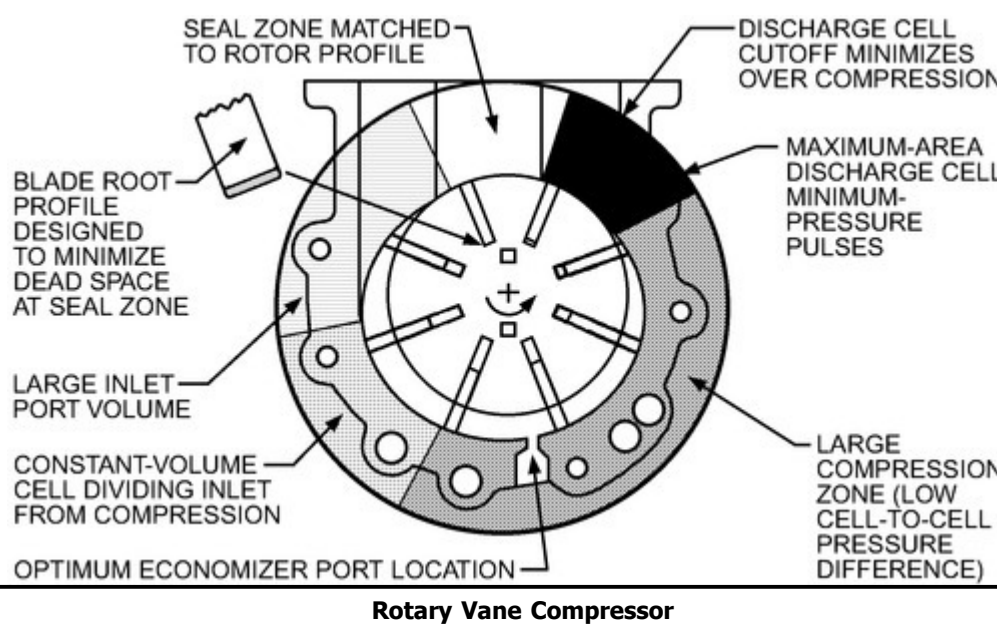
Lubrication. A properly designed lubrication system circulates an ample supply of clean oil to all working surfaces, bearings, vanes, vane slots, and seal faces. High-side pressure in the housing shell ensures a sufficient pressure differential across the passageways that distribute oil to the bearing surfaces.

Mechanical Efficiency. High mechanical efficiency depends on minimizing friction losses. Friction losses occur in the bearings and between the vane and slot wall, vane tip, and roller wall, and roller and bearing faces. The amount and distribution of these losses vary based on the geometry of the compressor.

Motor Selection. Breakdown torque requirements depend on the displacement of the compressor, the refrigerant, and the operating range. Domestic refrigerator compressors typically require a breakdown torque of about 36 to 38 oz · ft per cubic inch of compressor displacement per revolution. Similarly, larger compressors using R-22 for window air conditioners require about 67 to 69 oz · ft breakdown torque per cubic inch of compressor displacement per revolution. Rotary machines do not usually require complete unloading for successful starting. Pressure differentials up to 14.5 psi can be tolerated. The starting torque of standard split-phase motors is ample for small compressors. Permanent split-capacitor motors for air conditioners of various sizes provide sufficient starting and improve the power factor to the required range.

3.2 ROTARY-VANE COMPRESSORS

Rotary-vane compressors have a low weight-to-displacement ratio, which, in combination with compact size, makes them suitable for transport application. Small compressors in the 3 to 50 hp range are single-staged for a saturated suction temperature range of −13 to 59°F at saturated condensing temperatures up to 167°F. Currently, R-22, R-134a, R-404A, R-407, R-410A, R-507, and R-717 are the refrigerants used for these compressor applications. Figure 16 is a cross-sectional view of an eight-bladed compressor. The eight discrete volumes are referred to as cells. A single shaft rotation produces eight distinct compression strokes. Although conventional valves are not required for this compressor, suction and discharge check valves are recommended to prevent reverse rotation and oil logging during shutdown. The design of the compressor results in a fixed, built-in compression ratio. Compression ratio is determined by the relationship between the volume of the cell as it is closed off from the suction port to its volume before it opens to the discharge port.



Rotary Vane Compressor



1. Modeling and Animations using 3D Studio Max.
2. Air flow animation using Photoshop.
3. Compositing Frames using Compozer.
4. GIF animation using gifmerge.

Figure 16. Rotary-Vane Compressor

The compressors currently available are of an oil-flooded, open-drive design, which requires an oil separator. Single-stage separators are used in close-coupled systems with high saturation suction temperature (SST), where oil return is not a problem. Two-stage separators with a coalescing second stage are used in low-SST systems, in ammonia systems, and in flooded evaporators likely to trap oil. Rotary-vane compressors are more complicated (more parts) and, consequently, more expensive than rolling-piston or scroll compressors. Furthermore, the efficiency of rotary-vane compressors is comparatively low. Therefore, many of these compressors are being replaced with more advanced scroll or rotary types. However, the reliability of well-established rotary-vane designs is on par with other types.

3.3 SCREW COMPRESSORS

Screw compressors for refrigeration and air-conditioning applications are of two distinct types: single-screw and twin-screw. Both are conventionally used with fluid injection where sufficient fluid cools and seals the compressor. Screw compressors have the capability to operate at pressure ratios above 20:1 single stage.

Single-Screw Compressors

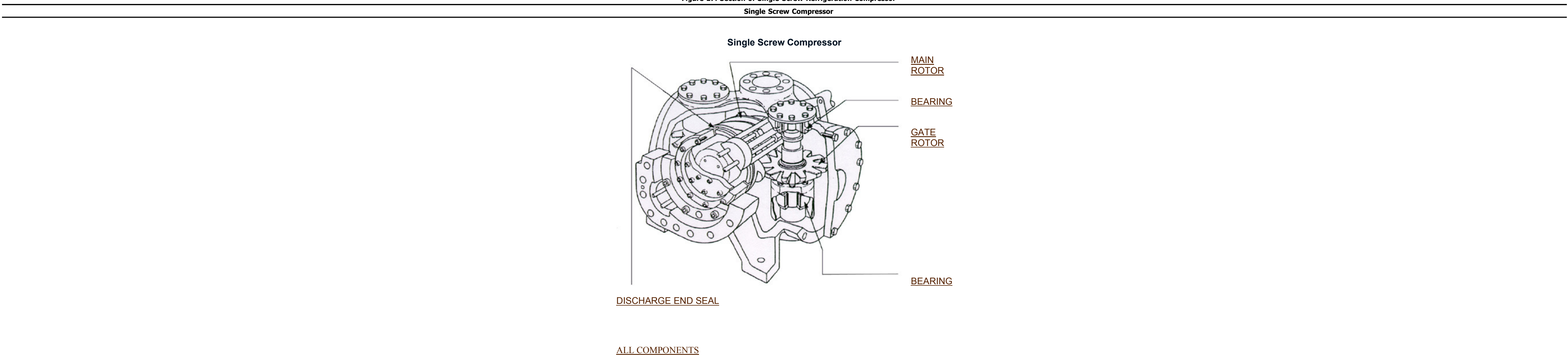
Description. A single-screw compressor consists of a single cylindrical main rotor that works with one or a pair of gate rotors. Both the main rotor and gate rotor(s) can vary widely in terms of form and mutual geometry. [Figure 17](#) shows the design normally encountered in refrigeration. The main rotor has helical grooves, with a cylindrical periphery and a globoid (or hourglass) root profile. The two identical gate rotors are located on opposite sides of the main rotor. The casing enclosing the main rotor has two slots, which allow the teeth of the gate rotors to pass through them.

Compression Process.

The operation of the single-screw compressor can be divided into three distinct phases: suction, compression, and discharge. The process is shown in [Figure 18](#).

Mechanical Features.

Rotors. The screw rotor is normally made of cast or ductile iron, and the mating gate rotors are made from an engineered plastic. The inherent lubricating quality of the plastic and its compliant nature allow the single-screw compressor to achieve close clearances with conventional manufacturing practice.



The compressor is driven through the main rotor shaft, and the gate rotors follow by direct meshing action with the main rotor. The geometry of the single-screw compressor is such that gas compression power is transferred directly from the main rotor to the gas. No power (other than small frictional losses) is transferred across the meshing points to the gate rotors.

Compression Process. The operation of the single-screw compressor can be divided into three distinct phases: suction, compression, and discharge. The process is shown in [Figure 18](#).

Mechanical Features.

Rotors. The screw rotor is normally made of cast or ductile iron, and the mating gate rotors are made from an engineered plastic. The inherent lubricating quality of the plastic and its compliant nature allow the single-screw compressor to achieve close clearances with conventional manufacturing practice.



The gate rotors are mounted on a metal support designed to carry the differential pressure between discharge pressure and suction pressure. The gate rotor function is equivalent to that of a piston in that it sweeps the groove and causes compression to occur. Furthermore, the gate rotor is in direct contact with the screw groove flanks and thus also acts as a seal. Each gate rotor is attached to its support by a simple spring and dashpot mechanism, allowing the gate rotor, with a low moment of inertia, to have an angular degree of freedom from the larger mass of the support. This method of attachment allows the gate rotor assemblies to pass a significant amount of liquid slug during transient operation without damage or wear.

Bearings. In a typical open or semihermetic single-screw compressor, the main rotor shaft contains one pair of angular contact ball bearings (an additional angular contact or roller bearing is used for some heat pump semihermetic compressors). On the opposite side of the screw, one roller bearing is used.

Note that compression takes place simultaneously on each side of the main rotor of the single-screw compressor. This balanced gas pressure results in virtually no load on the rotor bearing during full load and while symmetrically unloaded as shown in [Figure 19](#). Should the compressor be unloaded asymmetrically (see economizer operation below 50% capacity), the designer is not restricted by the rotor geometry and can easily add bearings with a long design life to handle the load. Axial loads are also low because the grooves terminate on the outer cylindrical surface of the rotor and suction pressure is vented to both ends of the rotor ([Figure 19](#)).

The gate rotor bearing must overcome a small moment force caused by the gas acting on the compression surface of the gate rotor. Each gate rotor shaft has at least one bearing for axial positioning (usually a single angular contact ball bearing can perform the axial positioning and carry the small radial load at one end), and one roller or needle bearing at the other end of the support shaft also carries the radial load.



Cooling, Sealing, and Bearing Lubrication. A major function of injecting a fluid into the compression area is removing heat of compression. Also, because a single-screw compressor has fixed leakage areas, the fluid helps seal leakage paths. Fluid is normally injected into a closed groove through ports in the casing or in the moving capacity control slide. Most single-screw compressors can use many different injection fluids, oil being the most common, to suit the nature of the gas being compressed.

Oil-Injected Compressors. Oil is used in single-screw compressors to seal, cool, lubricate, and actuate capacity control. It gives a flat efficiency curve over a wide compression ratio and speed range, thus decreasing discharge temperature and noise.

Oil-injected single-screw compressors operate at high head pressures using common high-pressure refrigerants such as R-12, R-134a, and R-717. They also operate effectively at high pressure ratios because the injected oil cools the compression process.

Oil injection requires an oil separator to remove the oil from the high-pressure refrigerant ([Figure 20](#)). For applications with exacting demands for low oil carryover, separation equipment is available to leave less than 5 ppm oil in the circulated refrigerant.

With most compressors, oil can be injected automatically without a pump because of the pressure difference between the oil reservoir (discharge pressure) and the reduced pressure in a flute or bearing assembly during compression. However, this injection should be done after the compression chamber is closed (to avoid capacity loss) but before the pressure is higher than discharge (in case of overcompression), which may be very difficult to design. To avoid this limitation, a continuously running oil pump is used in some compressors to generate oil pressure 30 to 45 psi over compressor discharge pressure. This pump requires 0.3 to 1.0% of the compressor's motor power.



External oil cooling between the oil reservoir and the point of injection is possible. Various heat exchangers are available to cool the oil: (1) separate water supply, (2) chiller water on a packaged unit, (3) condenser water on a packaged unit, (4) water from an evaporative condenser sump, (5) forced air-cooled oil cooler, and (6) high-pressure liquid recirculation (thermosiphon). Heat added to the oil during compression is the amount usually removed in the oil cooler.

Oil-Injection-Free Compressors. Although single-screw compressors operate well with oil injection, they also operate with good efficiency in an oil-injection-free (OIF) mode with many common halocarbon refrigerants. This means that fluid injected into the compression chamber is the condensate of the fluid being compressed. For air conditioning and refrigeration, where pressure ratios are in the range of 2 to 8, the oil normally injected into the casing may be replaced by liquid refrigerant. Not much lubrication is required because the only power transmitted from the screw to the gate rotors is that needed to overcome small frictional losses. Thus, the refrigerant need only cool and seal the compressor. The liquid refrigerant may still contain a small amount of oil to lubricate the bearings (0.1 to 1%, or higher, depending on compressor design). A typical OIF circuit is depicted in [Figure 21](#).



This method is used when an oil separator is not available. In this case, the liquid refrigerant carries a significant amount of oil with it. The methods of refrigerant injection include the following:

- **Direct injection of liquid refrigerant** into the compression process. Injection is controlled directly from the compressor discharge temperature, and loss of compressor capacity is minimized as injection takes place in a closed flute just before discharge occurs. This method requires very little power (typically less than 5% of compressor power).
 - A **small refrigerant pump** draws liquid from the receiver and injects it directly into the compressor discharge line. The injection rate is controlled by sensing discharge temperature and modulating the pump motor speed. The power penalty in this method is the pump power (about 1 hp for compressors up to 1000 hp), which can result in energy savings over refrigerant injected into the compression chamber.
- Oil-injection-free operation has the following advantages:
- It requires no discharge oil separator, unless an oil separator is needed to reduce the oil circulation rate in the system.
 - Compressors require no oil or refrigerant pumps.
 - External coolers are not required.

Economizers. Screw compressors are available with a secondary suction port between the primary compressor suction and discharge port. This port, when used with an economizer, provides the means to increase compressor capacity efficiency.

In operation, gas is drawn into the rotor grooves in the normal way from the suction line. The grooves are then sealed off in sequence, and compression begins. An additional charge is added to the closed flute through a suitably placed port in the casing by an intermediate gas source at a slightly higher pressure than that reached in the compression process at that time. The original and additional charges are then compressed together to discharge conditions. The compressor's pumping capacity at suction conditions is not even slightly affected by this additional flow through the economizer port.

When the port is used with an economizer, the effective refrigerating capacity of the economized compressor is increased over the noneconomized compressor by the increased heat absorption capability of the liquid entering the evaporator. Furthermore, the only additional mass flow the compressor must handle is flash gas entering a closed flute, which is above suction pressure. Thus, under most conditions, the capacity improvement also improves efficiency. Economizers become effective when the pressure ratio is 3.5 and above.

[Figure 22](#) shows a pressure-enthalpy diagram for a flash tank economizer. In it, high-pressure liquid passes through an expansion device and enters a tank at an intermediate pressure between suction and discharge. This pressure is maintained by pressure in the compressor's closed flute (closed from suction). Gas generated from the expansion enters the compressor through the economizer port. When passed to the evaporator, the liquid (which is now saturated at the intermediate pressure) gives a larger refrigeration capacity per pound. In addition, the percentage increase in power input is lower than the percentage capacity increase.

As a screw compressor is unloaded, economizer pressure falls toward suction pressure. As a result, the additional capacity and improved efficiency of the economizer fall to zero at 70 to 80% of full-load capacity.



The single-screw compressor has two compression chambers, each with its own slide valve. Each slide valve can be operated independently, which allows economizer gas to be introduced into one side of the compressor. By operating the slide independently, the chamber without the economizer gas can be unloaded to 0% capacity (50% capacity of the compressor). The other chamber remains at full capacity and retains the full economizer effect, making the economizer effective below 50% compressor capacity.

The secondary suction port may also be used for (1) a system-side load or (2) a second evaporator that operates at a temperature above that of the primary evaporator.

Some single-screw compressor designs use a patented **centrifugal economizer** that replaces the force of gravity in a flash economizer with centrifugal force to separate flash gas generated at an intermediate pressure from liquid refrigerant before liquid enters the evaporator. The centrifugal economizer thereby uses a much smaller pressure vessel and, in some designs, the economizer fits within the envelope of a standard motor housing without having to increase its size.

Separation is achieved by a centrifugal impeller mounted on the compressor shaft (see [Figure 20](#)); a special valve maintains a uniformly thick liquid ring around the circumference of the impeller, ensuring that no gas leaves with the liquid going to the evaporator. Flash gas is then ducted to closed grooves in the compressor screws. Some designs use flash gas with a similar liquid refrigerant to cool the motor before the gas enters the closed compression groove.

Volume (Compression) Ratio. The degree of compression in the rotor grooves is predetermined for a particular port configuration on screw compressors having fixed suction and discharge ports. A characteristic of the compressor is the volume (compression) ratio V_r , which is defined as the ratio of the volume of the groove at the start of compression to the volume of the same groove when it first begins to open to the discharge port. Hence, the volume ratio is determined by the size and shape of the discharge port.

For maximum efficiency, pressure generated within the grooves during compression should exactly equal the pressure in the discharge line at the moment when the groove opens to it. If this is not the case, either over- or undercompression occurs, both resulting in internal losses. Such losses increase power consumption and noise and reduce efficiency.

Volume ratio selection should be made according to operating conditions.

Compressors equipped with slide valves (for capacity modulation) usually locate the discharge port at the discharge end of the slide valve. Alternative port configurations yielding the required volume ratios are then designed into the capacity control components, thus providing easy interchangeability both during construction and after installation (although partial disassembly is required).

Single-screw compressors in refrigeration and process applications are equipped with a simple slide valve to vary compressor volume ratio while the compressor is running. The slide valve advances or delays the discharge port opening ([Figure 23](#)). Note that a separate slide has been designed to modulate the capacity independently of the volume ratio slide (see [Figures 24](#) to [26](#)). Having independent modulation of volume ratio (through discharge port control) and capacity modulation (through a completely independent slide that only varies the position where compression begins) allows the single-screw compressor to achieve efficient volume ratio control when capacity is less than full load.

Capacity Control. As with all positive-displacement compressors, both speed modulation and suction throttling can be used. Ideal capacity modulation for any compressor includes (1) continuous modulation from 100% to less than 10%, (2) good part-load efficiency, (3) unloaded starting, and (4) unchanged reliability.

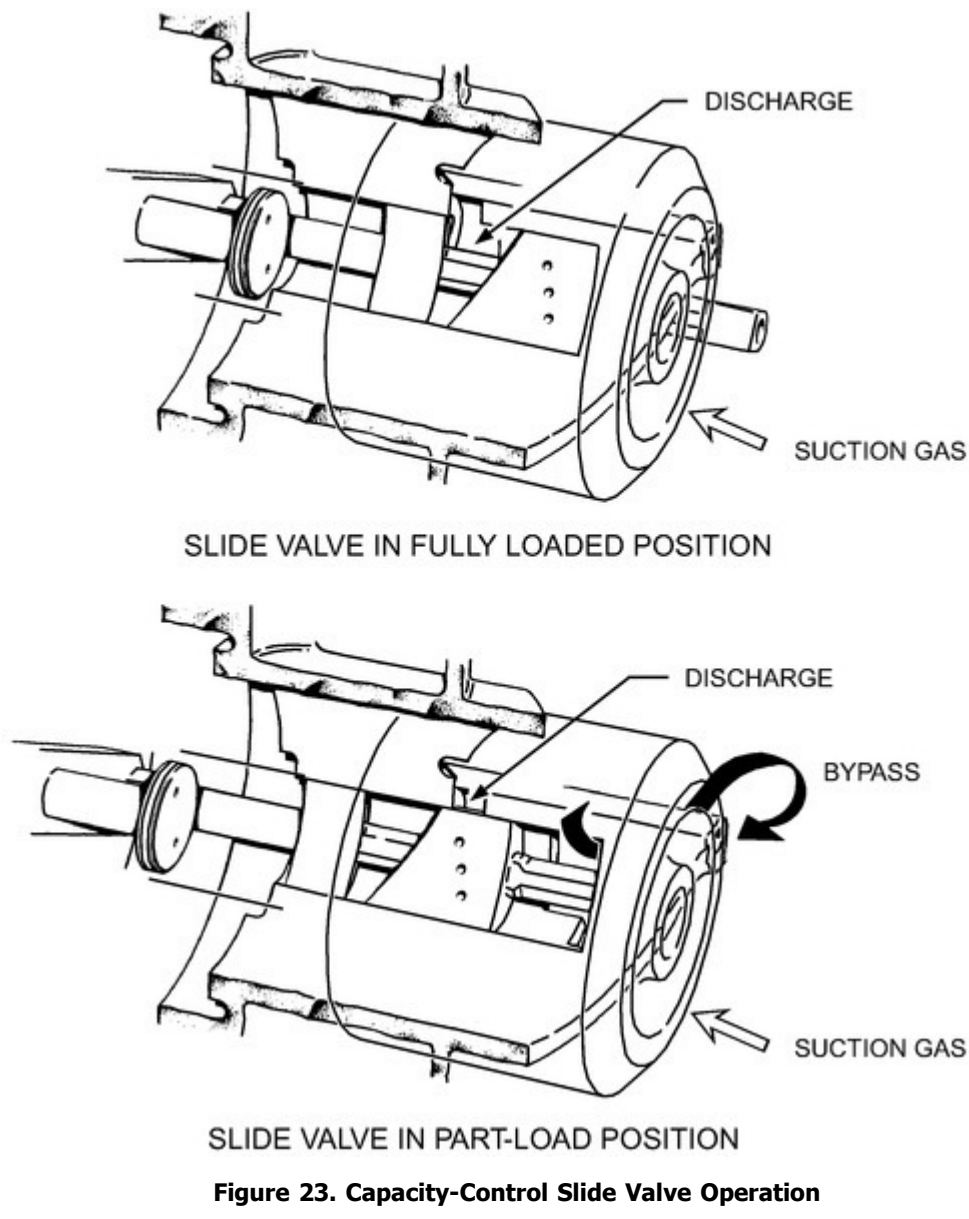


Figure 23. Capacity-Control Slide Valve Operation

Variable compressor displacement, the most common means for meeting these criteria, usually takes the form of two movable slide valves in the compressor casing (the single-screw compressor has two gate rotors forming two compression areas). At part load, each slide valve produces a slot that delays the point at which compression begins. This reduces groove volume, and hence compressor throughput. Because suction volume is displaced before compression takes place, little or no thermodynamic loss occurs. However, if no other steps were taken, this mechanism would result in an undesirable drop in the effective volume ratio, in undercompression and inefficient part-load operation.

This problem is avoided either by arranging that the capacity modulation valve reduces the discharge port area as the bypass slot is created (Figure 23) or having one valve control capacity only and a second valve independently modulate volume ratio (Figures 24 to 26). A full modulating mechanism is provided in most large single-screw compressors, whereas two-position slide valves are used where requirements allow. The specific part-load performance is affected by a compressor's built-in volume (compression) ratio, evaporator temperature, and condenser temperature, and whether the slide valves are symmetrically or asymmetrically controlled.

Detailed design of the valve mechanism differs between makes of compressors but usually consists of an axial sliding valve along each side of the rotor casing (Figure 23). This mechanism is usually operated by a hydraulic or gas piston and cylinder assembly in the compressor itself or by a positioning motor. The piston is actuated by oil, discharge gas, or high-pressure liquid refrigerant at discharge pressure driven in either direction according to the operation of a four-way solenoid valve.

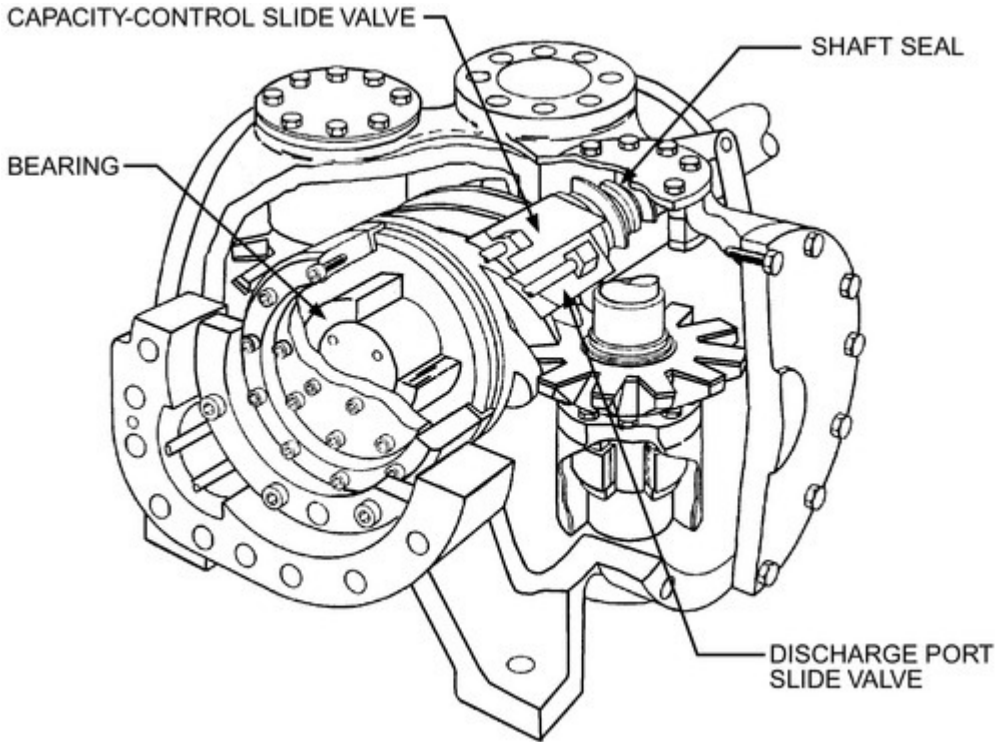


Figure 24. Refrigeration Compressor Equipped with Variable-Capacity Slide Valve and Variable-Volume-Ratio Slide Valve

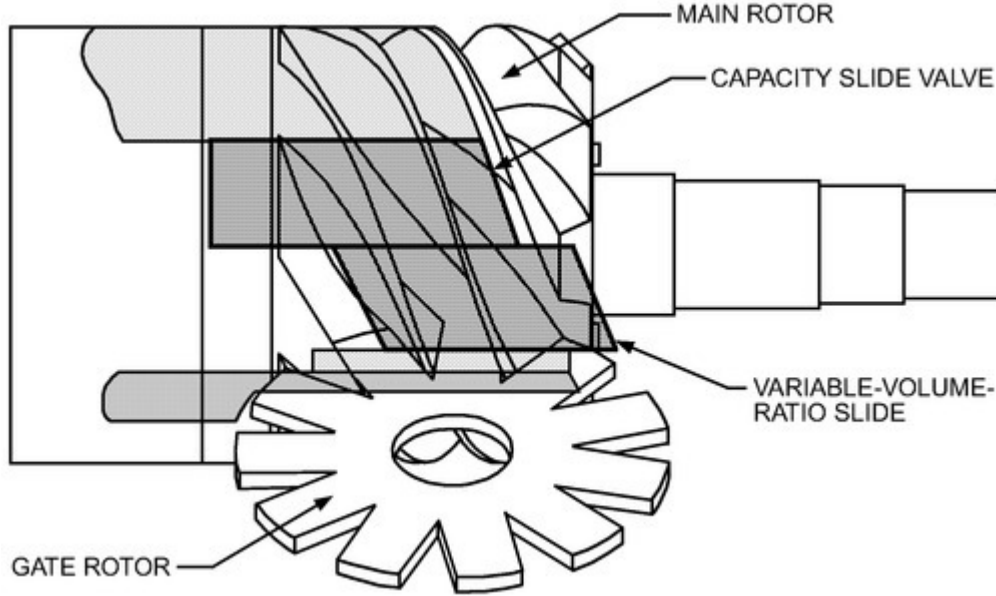


Figure 25. Capacity Slide in Full-Load Position and Volume Ratio Slide in Intermediate Position

Figure 25 shows a capacity slide valve (top) and a variable-volume-ratio slide (bottom). The capacity slide is in the full-load position, and the volume ratio slide is at a moderate volume ratio. Figure 26 depicts the same system as shown in Figure 25, except that the capacity slide is in a partially loaded position, and the volume ratio slide has moved to a position to match the new conditions.

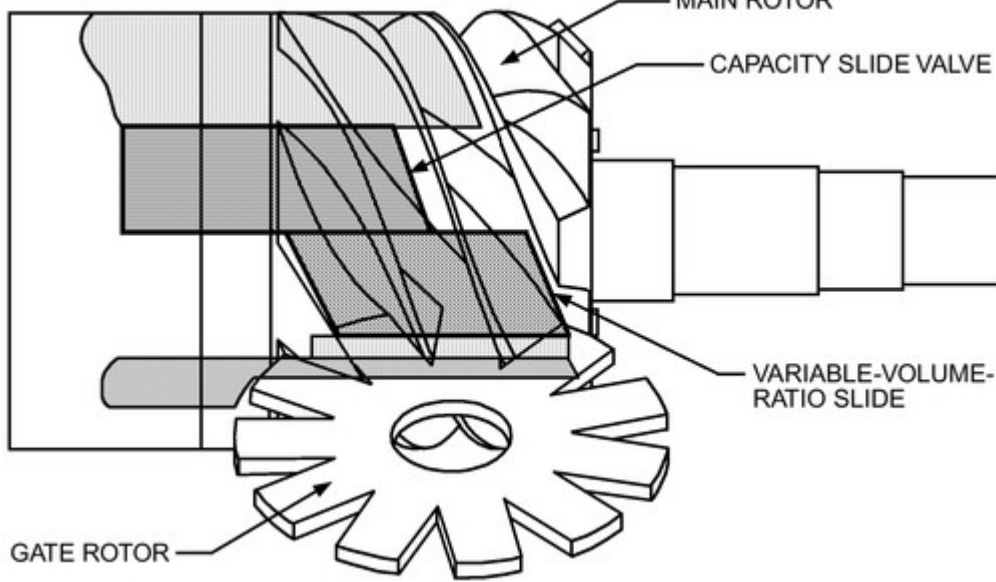


Figure 26. Capacity Slide in Part-Load Position and Volume Ratio Slide Positioned to Maintain System Volume Ratio

The single-screw compressor's two compression chambers, each with its own capacity slide valve that can be operated independently, permits one slide valve to be unloaded to 0% capacity (50% compressor capacity) while the other slide valve remains at full capacity. This asymmetrical operation improves part-load efficiency below 50% capacity, and further part-load efficiency gains are realized when the economizer gas is only entered into a closed groove on the side that is unloaded second (see explanation in the section on Economizers). Figure 27 demonstrates the effect of asymmetrical capacity control of a single-screw compressor.

Performance: Figures 28 and 29 show typical efficiencies of all single-screw compressor designs. High isentropic and volumetric efficiencies result from internal compression, the absence of suction or discharge valves and their losses, and extremely small clearance volumes. The curves show the importance of selecting the correct volume ratio in fixed-volume-ratio compressors.

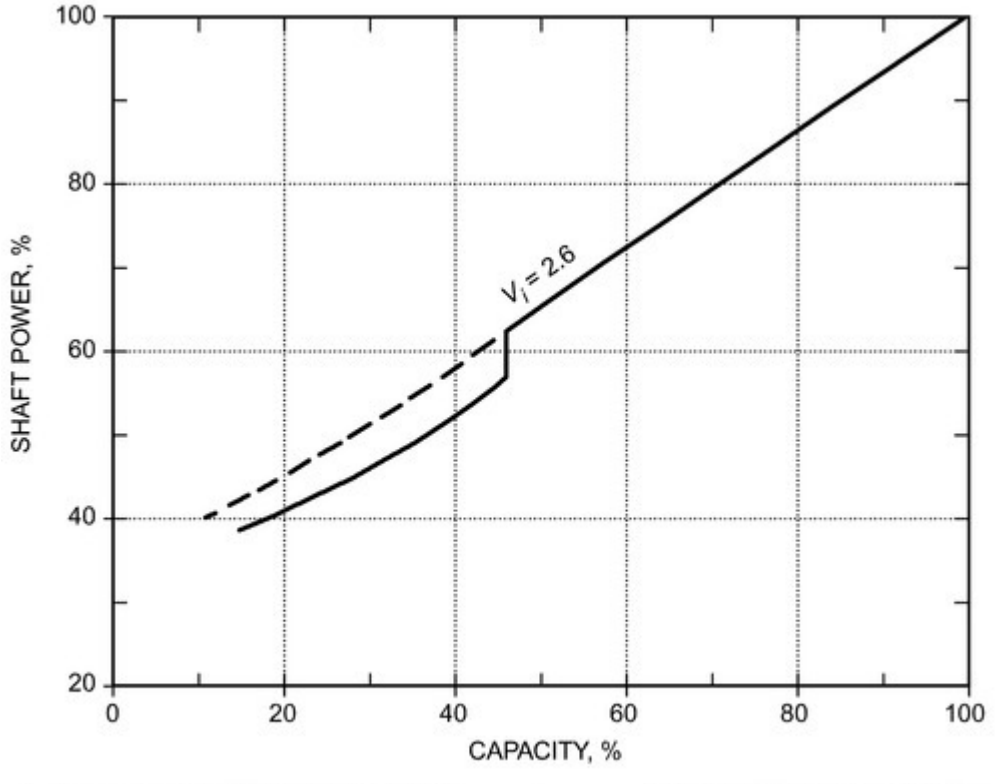


Figure 27. Part-Load Effect of Symmetrical and Asymmetrical Capacity Control

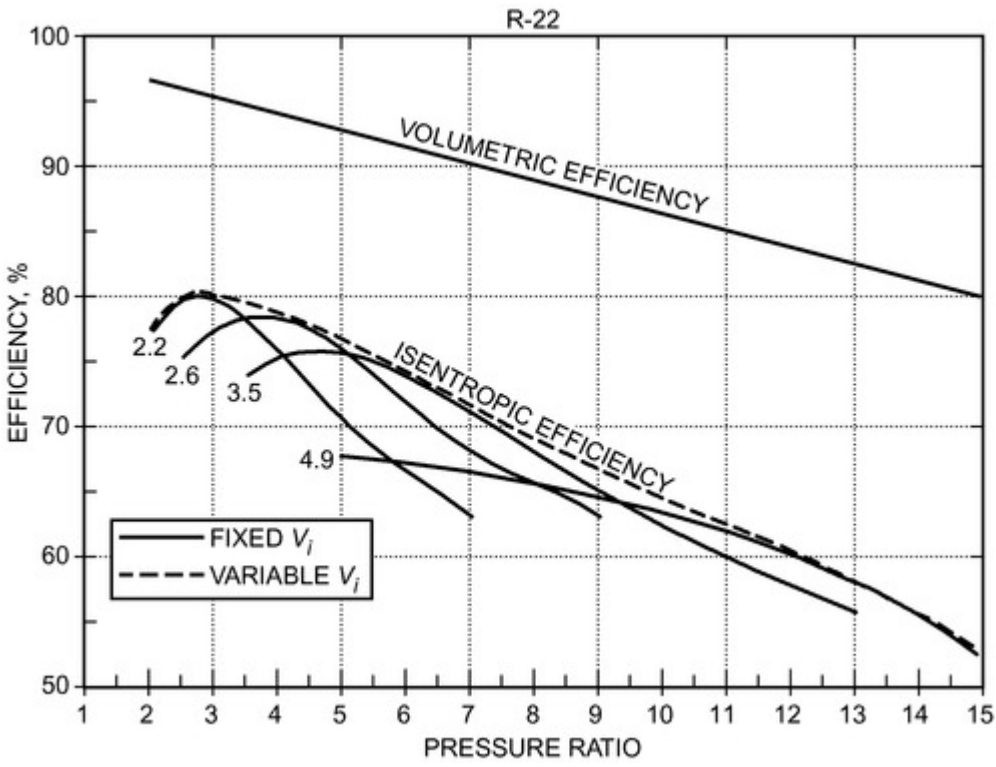


Figure 28. Typical Open-Compressor Performance on R-22

Manufacturers' data for operating conditions versus speed should not be extrapolated. Screw compressor performance at reduced speed is usually significantly different from that specified at the normally rated point because of the significant effect of leakage. Performance data normally include information about the degree of liquid subcooling and suction superheating assumed in data.

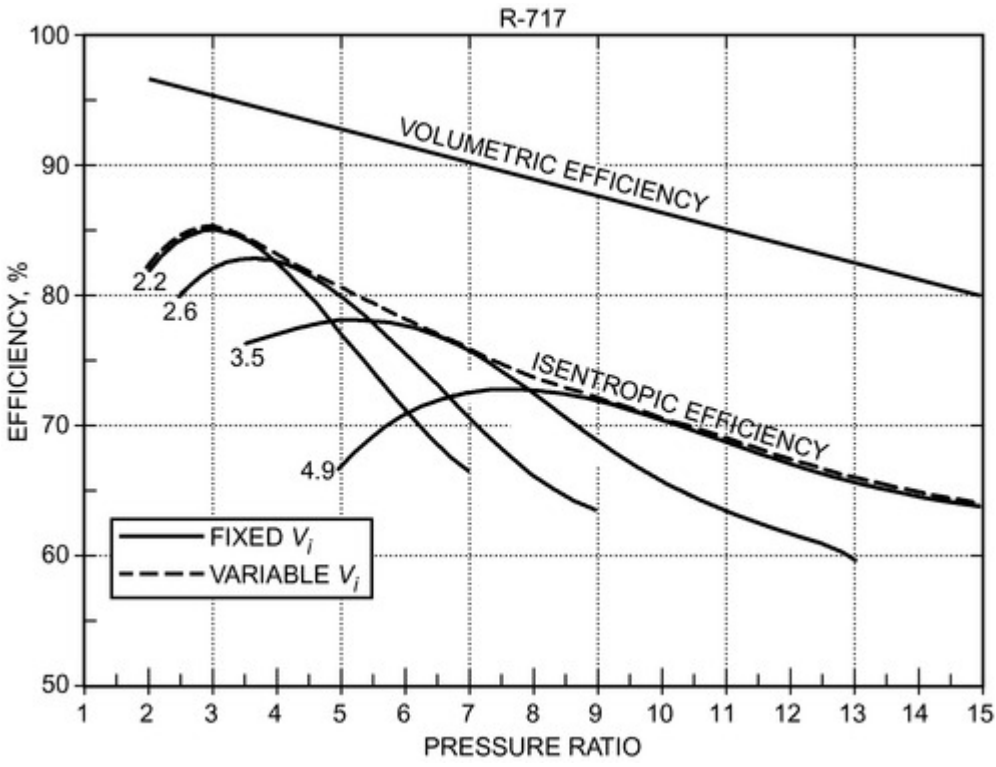


Figure 29. Typical Compressor Performance on R-717 (Ammonia)

Applications: Single-screw compressors are widely used as refrigeration compressors, using halocarbon refrigerants, ammonia, and hydrocarbon refrigerants. A single gate rotor semihermetic version is being used in large supermarkets. Oil-injected and oil-injection-free (OIF) semihermetic compressors are widely used for air-conditioning and heat pump service, with compressor sizes ranging from 40 to 500 tons.

Semihermetic Design: Figure 30 shows a semihermetic single-screw compressor. Figure 31 shows a semihermetic single-screw compressor using only one gate rotor. This design has been used in large supermarket rack systems. The single-gate-rotor compressor exhibits high efficiency and has been designed for long bearing life, which compensates for the unbalanced load on the screw rotor shaft with increasing bearing size.

Noise and Vibration. The inherently low noise and vibration of single-screw compressors are due to small torque fluctuation and no valves required in the compression chamber. In particular, OIF technology eliminates the need for oil separators, which have traditionally created noise.

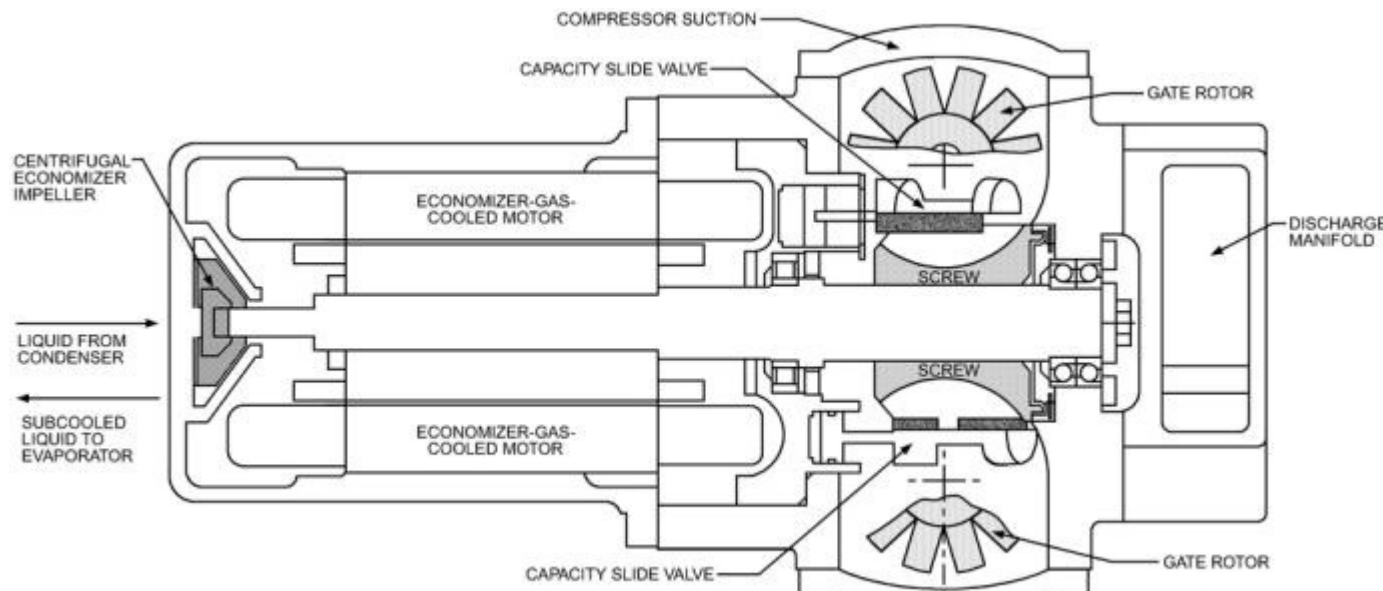


Figure 30. Typical Semihermetic Single-Screw Compressor

Twin-Screw Compressors

Twin screw is the common designation for double helical rotary screw compressors. A twin-screw compressor consists of two mating helically grooved rotors: male (lobes) and female (flutes or gullies) in a stationary housing with inlet and outlet gas ports (Figure 32). Gas flow in the rotors is mainly in an axial direction. Frequently used lobe combinations are 4 + 6, 5 + 6, and 5 + 7 (male + female). For instance, with a four-lobe male rotor, the driver rotates at 3600 rpm; the six-lobe female rotor follows at 2400 rpm. The female rotor can be driven through synchronized timing gears or directly driven by the male rotor on a light oil film. In some applications, it is practical to drive the female rotor, which results in a 50% speed and displacement increase over the male-driven compressor, assuming a 4 + 6 lobe combination. Geared speed increasers are also used on some applications to increase the capacity delivered by a particular compressor size.

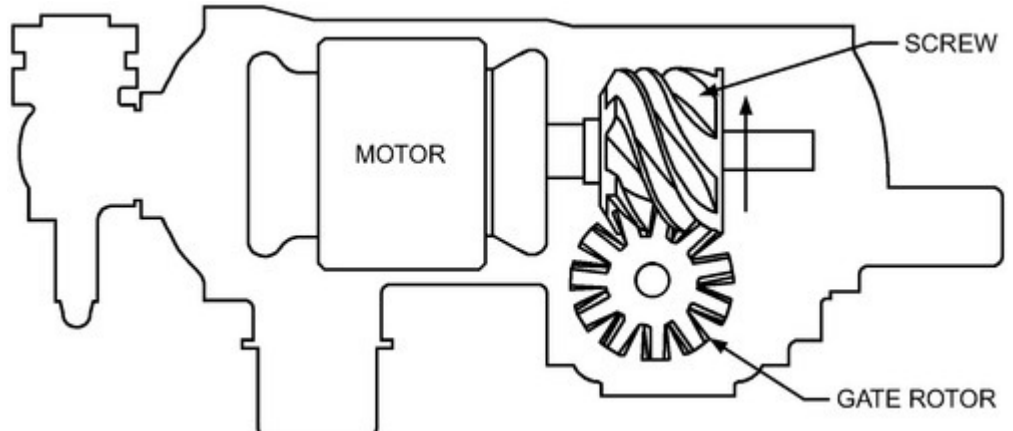


Figure 31. Single-Gate-Rotor Semihermetic Single-Screw Compressor

Twin helical screws find application in many air-conditioning, refrigeration, and heat pump applications, typically in the industrial and commercial market. Machines can be designed to operate at high or low pressure and are sometimes applied below 2:1 and above 20:1 compression ratios single-stage. Commercially available compressors are suitable for application on the majority of refrigerants.

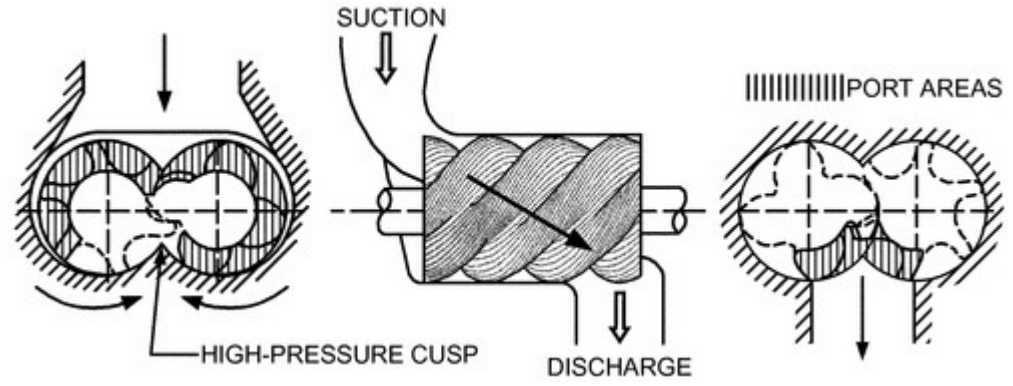
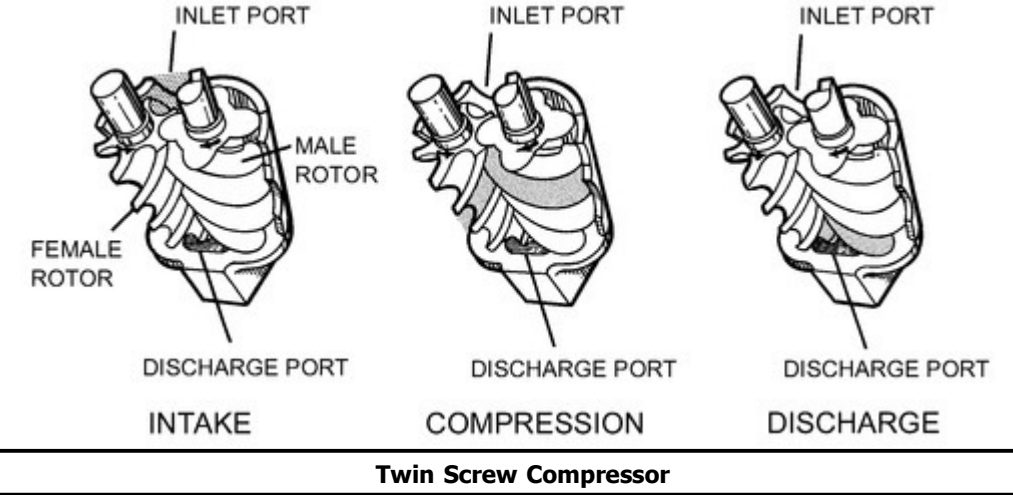


Figure 32. Twin-Screw Compressor

Compression Process. Compression is obtained by direct-volume reduction with pure rotary motion. For clarity, the following description of the three basic compression phases is limited to one male rotor lobe and one female rotor interlobe space (Figure 33). **Suction.** As the rotors begin to unmesh, a void is created on both the male side (male thread) and the female side (female thread), and gas is drawn in through the inlet port. As the rotors continue to turn, the interlobe space increases, and gas flows continuously into the compressor. Just before the point at which the interlobe space leaves the inlet port, the entire length of the interlobe space is completely filled with gas.



Twin Screw Compressor

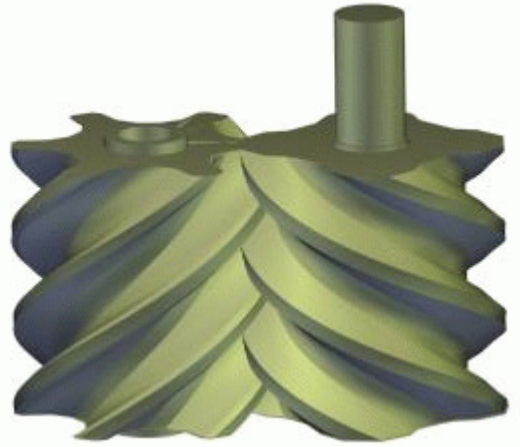


Figure 33. Twin-Screw Compression Process

Compression. Further rotation starts meshing another male lobe with another female interlobe space on the suction end and progressively compresses the gas in the direction of the discharge port. Thus, the occupied volume of the trapped gas within the interlobe space decreases and the gas pressure consequently increases. **Discharge.** At a point determined by the designed built-in volume ratio, the discharge port is uncovered and the compressed gas is discharged by further meshing of the lobe and interlobe space. During the remeshing period of compression and discharge, a fresh charge is drawn through the inlet on the opposite side of the meshing point. With four male lobes rotating at 3600 rpm, four interlobe volumes are filled and give 14,400 discharges per minute. Because the intake and discharge cycles overlap effectively, gas flow is smooth and continuous. **Mechanical Features.** **Rotor Profiles.** Helical rotor design started with an asymmetrical point-generated rotor profile. This profile was only used in compressors with timing gears. The symmetrical, circular rotor profile was introduced because it was easier to manufacture than the preceding profile, and it could be used with or without timing gears. Current rotor profiles are normally asymmetrical line-generated profiles, giving higher performance because of better rotor dynamics and decreased leakage area. This design allows female rotor drive, as well as conventional male drive. Rotor profile, blowhole, length of sealing line, quality of sealing line, torque transmission between rotors, rotor-housing clearances, interlobe clearances, and lobe combinations are optimized for specific pressure, temperature, speed, and wet or dry operation. Optimal rotor tip speed is 3000 to 8000 fpm for wet operation (oil-flooded) and 12,000 to 24,000 fpm for dry operation. **Rotor Contact and Loading.** Contact between the male and female rotors is mainly rolling, primarily at a contact band on each rotor's pitch circle. Rolling at this contact band means that very little wear occurs. However, even a minor sliding motion can generate significant local heat from friction, and if there is not enough gas or lubricant flow at this area, a local scoring may occur that leads to increased internal leakage, higher friction losses, and sometimes mechanical failure because of locked rotors. **Gas Forces.** On the driven rotor, the internal gas force always creates a torque in a direction opposite to the direction of rotation. This is known as positive or braking torque. For the undriven rotor, the design can be such that the torque is positive, negative, or zero, except on female-drive designs, where zero or negative torque does not occur. Negative torque occurs when internal gas force tends to drive the rotor. If the average torque on the undriven rotor is near zero, this rotor is subjected to torque reversal as it goes through its phase angles. Under certain conditions, this can cause instability. Torque transmitted between the rotors does not create problems because the rotors are mainly in rolling contact. The transmitted torque from male rotor to female rotor is normally 5 to 25% of input torque, compared to the transmitted torque from female rotor to male rotor, which is normally 50 to 60% of input torque. **Rotors in an operating compressor** are subjected to radial, axial, and tilting loads. Tilting loads are radial loads caused by axial loads outside of the rotor center line. Axial load is normally balanced with a balancing piston for larger high-pressure machines (rotor diameter above 4 in. and discharge pressure above 160 ps). Balancing pistons are typically close-tolerance, labyrinth-type devices with high-pressure oil or gas on one side and low pressure on the other. They are used to produce a thrust load to offset some of the primary gas loading on the rotors, thus reducing the amount of thrust load the bearings support. **Bearings.** Twin-screw compressors normally have either four or six bearings, depending on whether one or two bearings are used for the radial and axial loads. Some designs incorporate multiple rows of smaller bearings per shaft to share loads. Sleeve bearings were used historically to support radial loads in machines with male rotor diameters larger than 6 in.; antifriction bearings were generally applied to smaller machines. However, improvements in antifriction designs and materials have led to compressors with up to 14 in. rotor diameter with full antifriction bearing designs. Cylindrical and tapered roller bearings and various types of ball bearings are used in screw compressors for carrying radial loads; this may lead to an increase in the number of bearings. The most common thrust or axial load-carrying bearings are angular-contact ball bearings, although tapered rollers or tilting pad bearings are used in some machines. Use of moldable polymeric materials with higher temperature limits for bearing cages allows high-speed operation of the compressor at higher discharge temperatures. **General Design.** Screw compressors are often designed for particular pressure ranges. Low-pressure compressors have long, high displacement rotors and adequate space to accommodate bearings to handle the relatively light loads. They are frequently designed without thrust balance pistons, because the bearings alone can handle the low thrust loads and still maintain good life. High-pressure compressors have short, strong rotors (shallow grooves) and therefore have space for large bearings. Larger compressors are normally designed with balancing pistons for high thrust bearing life. **Rotor Materials.** Rotors are normally made of steel, but aluminum, cast iron, and nodular iron are used in some applications. Special surface treatment or coatings can be used to reduce wear and improve oil adhesion. **Capacity Control.** As with all positive-displacement compressors, both speed modulation and suction throttling can reduce the volume of gas drawn into a screw compressor. Ideal capacity modulation for any compressor would be (1) continuous modulation from 100% to less than 10%, (2) good part-load efficiency, (3) unloaded starting, and (4) high reliability throughout the operational range. However, not all applications need ideal capacity modulation. Variable compressor displacement and variable speed are the best means for meeting these criteria. Various mechanisms achieve variable displacement, depending on the requirements of a particular application. **Capacity-Slide Valve.** A slide valve for capacity control is a valve with sliding action parallel to the rotor bores, within or close to the high-pressure region. It reduces the active length of the housing profile, thus controlling compressor displacement and capacity. There are two types of capacity slide valves:

- Capacity slide valves regulating the discharge port are located in the high-pressure cusp region. They control capacity as well as the location of the radial discharge port at part load. The axial discharge port is designed for a volume ratio giving good part-load performance without losing full-load performance. Figure 34 shows a schematic of the most common arrangement.
- Capacity slide valves *not* regulating the discharge port outside the high-pressure cusp region control only capacity.

The first type is most common. It is generally the most efficient capacity reduction method, because of its indirect correction of built-in volume (compression) ratio at part load and its ability to give large volume reductions without large movement of the slide valve.

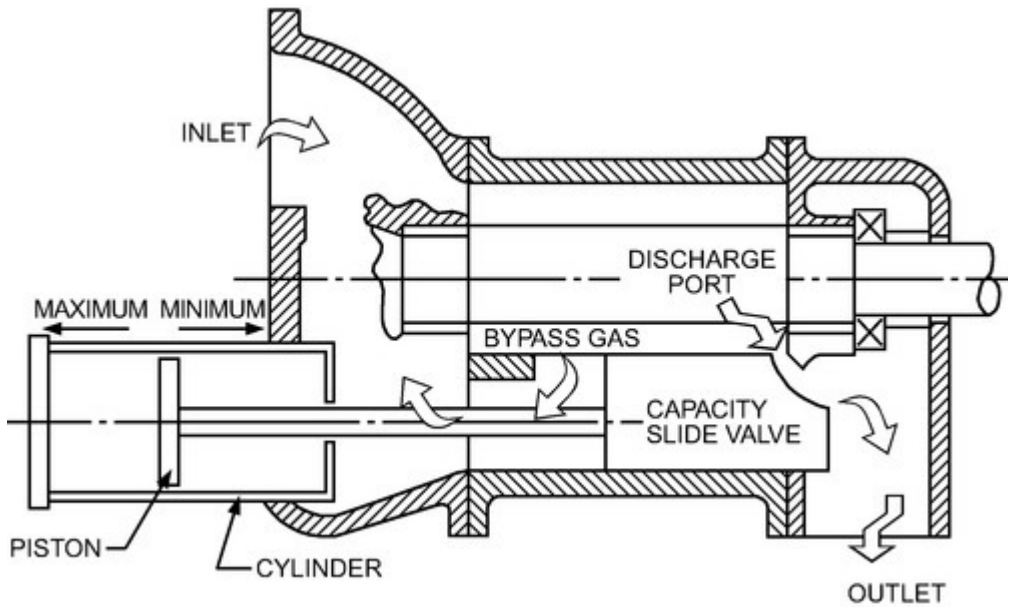


Figure 34. Slide Valve Unloading Mechanism

Capacity Slot Valve. A capacity slot valve consists of a number of slots that follow the rotor helix and face one or both rotor bores. The slots are gradually opened or closed with a plunger or turn valve. These recesses in the casing wall increase the volume of compression space and also create leakage paths over the lobe tips. The result is somewhat lower full-load performance compared to a design without slots. **Capacity Lift Valve.** Capacity lift valves or plug valves are movable plugs in one or both rotor bores (with radial or axial lifting action) that regulate the actual start of compression. These valves control capacity in a finite number of steps, rather than by the infinite control of a conventional slide valve (Figure 35). Neither slot nor lift valves offer quite as good efficiency at part load as a slide valve, because they do not relocate the radial discharge port. Thus, undercompression losses at part load can be expected if machines have the correct volume (compression) ratio for full-load operation and the pressure ratio at part load does not reduce.

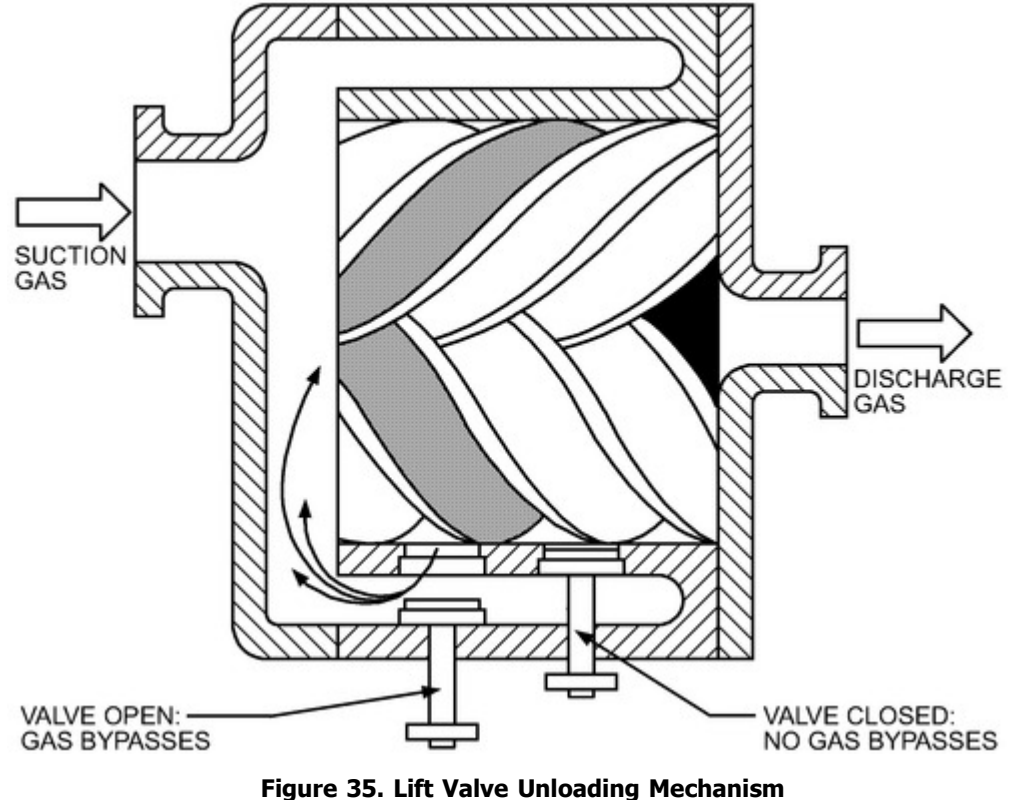


Figure 35. Lift Valve Unloading Mechanism

Volume (Compression) Ratio. In all positive-displacement rotary compressors with fixed port location, the degree of compression in the rotor thread is determined by the location of suction and discharge ports. The built-in volume (compression) ratio of screw compressors is defined as the ratio of volume of the thread at the start of compression to the volume of the same thread when it begins to open to the discharge port. The suction port must be located to trap the maximum suction charge; hence, the compression ratio is determined by the location of the discharge port. Only suction pressure and compression volume ratio determine the internal pressure achieved before opening to discharge. However, condensing and evaporating temperatures determine discharge pressure and compression ratio in piping that leads to the compressor. Any mismatch between internal and system discharge pressures results in under- or overcompression loss and lower efficiency. If the operating conditions of the system seldom change, it is possible to specify a fixed-volume-ratio compressor that will give good efficiency. Compressor manufacturers normally make compressors with three or four possible discharge port sizes that correspond to system conditions encountered frequently. Generally, the designer is responsible for specifying a compressor that most closely matches expected pressure conditions. The required compression ratio for a particular application can be determined as follows:

$$CR = PR^{1/k} \tag{13}$$

where
CR = compression ratio
PR = pressure ratio = p_d/p_s
 p_d = expected discharge pressure (absolute)
 p_s = expected suction pressure (absolute)
 k = isentropic coefficient for refrigerant used, from refrigerant tables [e.g., Lemmon et al. (2002)]

Usually, in slide-valve-equipped compressors, the radial discharge port is located in the discharge end of the slide valve. For a given ratio L/D of rotor length to rotor diameter and a given stop position, a short slide valve gives a low volume ratio, and a long slide valve gives a higher compression ratio. The difference in length basically locates the discharge port earlier or later in the compression process. Different-length slide valves allow changing the compression ratio of a given compressor, although disassembly is required. **Variable Volume (Compression) Ratio.** While operating, some twin-screw compressors adjust the compressor volume ratio to the most efficient ratio for whatever pressures are encountered. In fixed-volume-ratio compressors, the slide valve motion toward the inlet end of the machine is stopped when it comes in contact with the rotor housing in that area. In most common variable-volume-ratio machines, this portion of the rotor housing has been replaced with a second slide, the movable slide stop, which can be actuated to different locations in the slide valve bore (Figure 36).

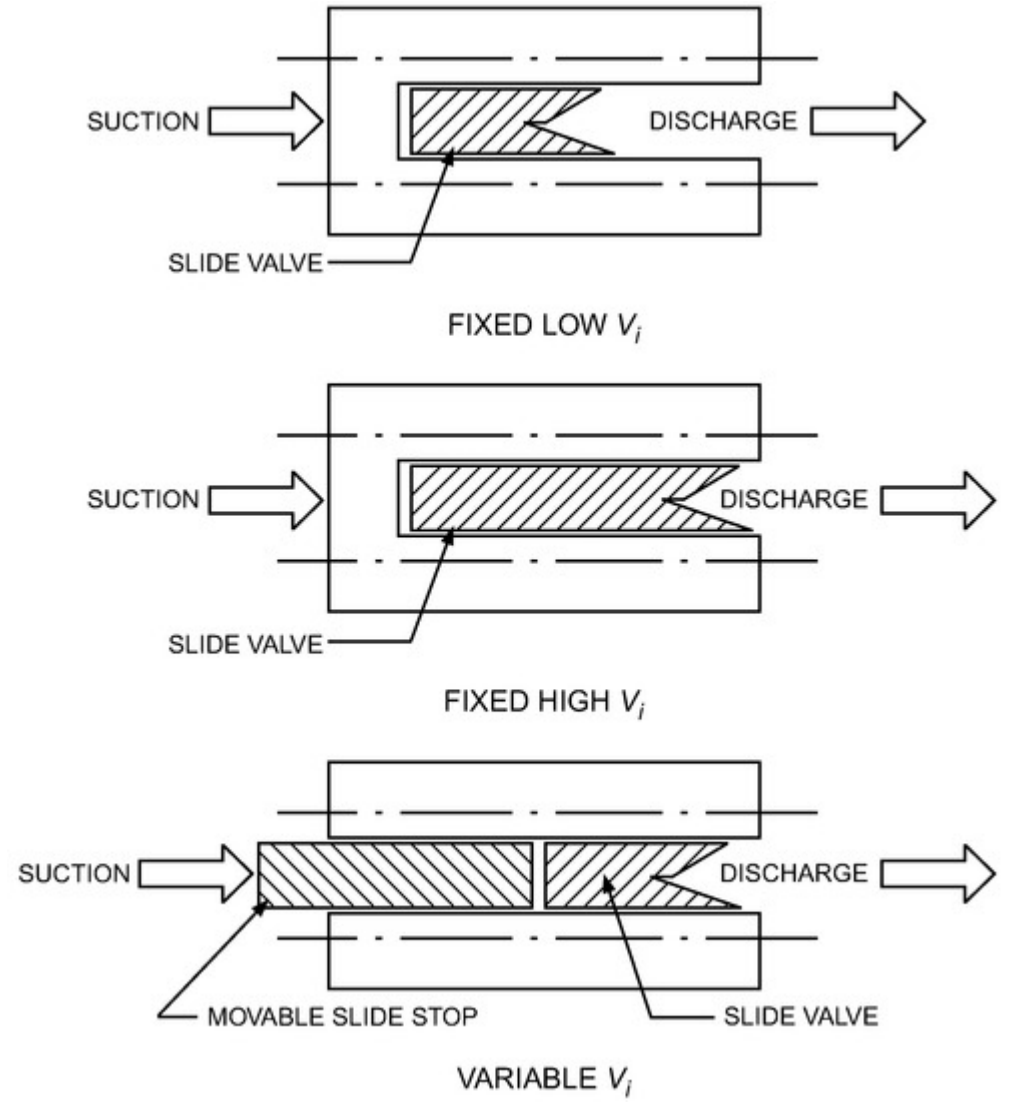


Figure 36. View of Fixed- and Variable-Volume-Ratio (V_1) Slide Valves from Above

By moving the slides back and forth, the radial discharge port can be relocated during operation to match the compressor volume ratio to the optimum. This added flexibility allows operation at different suction and discharge pressure while maintaining maximum efficiency. Comparative efficiencies of fixed- and variable-volume-ratio screw compressors are shown in Figure 37 for full-load operation on ammonia and R-22 refrigerants. The figure shows that a variable-volume-ratio compressor efficiency curve encompasses the peak efficiencies of compressors with fixed volume ratio over a wide range of pressure ratio. Following are other secondary effects of a variable volume ratio:

- Less oil foam in oil separator (no overcompression)
- Less oil carried over into the refrigeration system (because of less oil foam in oil separator)
- Extended bearing life; minimized load on bearings
- Extended efficient operating range with economizer discharge port corrected for flash gas from economizer, as well as gas from suction
- Less noise
- Lower discharge temperatures and oil cooler heat rejection

The greater the changes in either suction or condensing pressure, the more benefits are possible with a variable-volume ratio. Efficiency improvement as high as 30% is possible, depending on the application, refrigerant, and operating range.

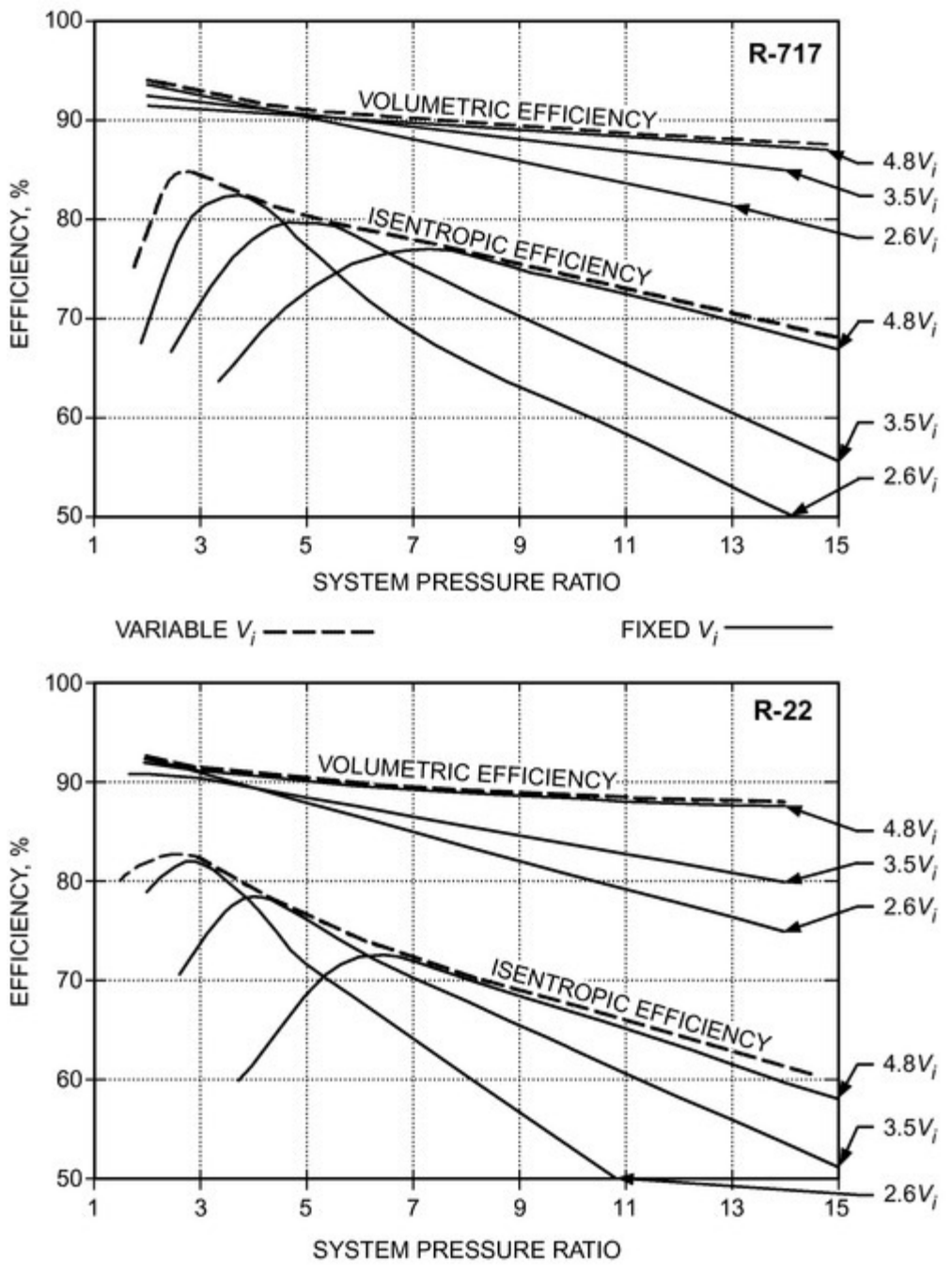


Figure 37. Twin-Screw Compressor Efficiency Curves

Performance Characteristics. Figure 37 shows the full-load efficiency of a modern twin-screw compressor. Values for both fixed- and variable-volume-ratio compressors without economizers are shown. High isentropic and volumetric efficiencies result from internal compression, the absence of suction or discharge valves, and small clearance volume. The curves show that, although volumetric efficiency depends little on the choice of volume ratio, isentropic efficiency depends strongly on it. Performance data usually note the degree of liquid subcooling and suction superheating assumed. If an economizer is used, the pressure drop and temperature of liquid approaching the economizer should be specified.

Noise. The most significant sources of noise in screw compressors are rotor contact and discharge pulsations. Adequate lubrication and discharge port design can significantly alleviate this issue. If screws are driven by gears, gear noise may become dominant.

Oil Injection. Two primary types of compressor lubrication systems are used in twin-screw compressors: dry and oil-flooded.

Dry Operation (No Rotor Contact). Because the two rotors in twin-screw compressors are parallel, timing gears are a practical means of synchronizing the rotors so that they do not touch each other. Eliminating rotor contact eliminates the need for lubrication in the compression area; however, a small amount of oil may be needed to provide sealing. Initial screw compressor designs were based on this approach, and dry screws are still used in the gas process industry and in some transportation applications.

Synchronized twin-screw compressors once required high rotor tip speed to minimize leakage, and thus were noisy. However, with current profile technology, the synchronized compressor can run at a lower tip speed and higher pressure ratio, giving quieter operation. The added cost of timing gears and internal seals generally make the dry screw more expensive than an oil-flooded screw for normal refrigeration or air conditioning.

Oil-Flooded Operation. The oil-flooded twin-screw compressor is the most common type of screw compressor used in refrigeration and air conditioning. Oil-flooded compressors typically have oil supplied to the compression area at a volume rate of about 0.5% of the displacement volume. Part of this oil is used for lubricating bearings and shaft seal. Different oils are used, as required by the refrigerant or gas application. In the case of oil injection, oil is normally injected into a closed thread through ports in the moving slide valve and/or through stationary ports in the casing.

The oil fulfills three primary purposes: sealing, cooling, and lubrication. It tends to fill any leakage paths between and around the rotors. This keeps volumetric efficiency high, even at high compression ratios. Often, compressor volumetric efficiency exceeds 85%, even at 25:1 single stage (ammonia, 7.6 in. rotor diameter). High internal oil circulation reduces the influence of speed on compressor performance and lessens operational noise. Oil transfers much of the heat of compression from the gas to the oil, keeping the typical discharge temperature below 250°F, which allows high compression ratios without the danger of breaking down the refrigerant or oil. Lubrication by the oil protects bearings, seals, and rotor contact areas.

The ability of a screw compressor to tolerate oil also permits the compressor to handle a certain amount of liquid, as long as the liquid quantity is not large enough to lock the rotors hydraulically.

Oil Separation and Cooling. Oil injection requires an oil separator to remove oil from the high-pressure refrigerant. Collecting separation equipment routinely gives less than 5 ppm oil in the circulated refrigerant. Compressors used in direct-expansion (DX) systems and/or on packaged units have less-efficient separation capability.

Oil injection is normally achieved by one of two methods: (1) with a continuously running oil pump capable of generating an oil pressure of 30 to 45 psi over compressor discharge pressure, representing 0.3 to 1.0% of compressor motor power; or (2) with some compressors, oil can be injected automatically without a pump because of the pressure difference between the oil reservoir (discharge pressure) and the reduced pressure in an intermediate compression chamber.

Depending on the refrigerant and operating conditions, screw compressors can operate with or without oil cooling. There are performance advantages in maintaining low discharge-gas temperature by oil cooling. One cooling method is by direct injection of liquid refrigerant into the compression process. The amount of liquid injected is normally controlled by sensing the discharge temperature and injecting enough liquid to maintain a constant temperature. Some injected liquid mixes with the oil and leaks to lower-pressure threads, where it tends to raise pressure and reduce the amount of gas the compressor can draw in. Also, any liquid that has time to absorb heat and expand to vapor must be recompressed, which tends to raise absorbed power levels. Compressors are designed with liquid injection ports that inject liquid as late as possible in the compression to minimize capacity and power penalties. Typical penalties for liquid injection are in the 1 to 10% range, depending on the compression ratio. Because of this, and the danger of excessive lubricant dilution, direct liquid injection has more stringent limits than other oil-cooling methods.

Another method of oil cooling draws liquid from the receiver with a small refrigerant pump and injects it directly into the compressor discharge line. The power penalty in this method is the pump power (about 1 hp for compressors up to 1000 hp).

In the third method, oil is cooled outside the compressor between the oil reservoir and the point of injection. Various configurations of heat exchangers are available for this purpose, and oil cooler heat rejection can be accomplished by (1) separate water supply, (2) chiller water on a packaged unit, (3) condenser water on a packaged unit, (4) water from an evaporative condenser sump, (5) forced air-cooled oil cooler, (6) liquid refrigerant, and (7) high-pressure liquid recirculation (thermosiphon).

External oil coolers using water or other means from a source independent of the condenser allow condenser size to be reduced by an amount corresponding to the oil cooler capacity. Where oil is cooled within the refrigerant system by means such as (1) direct injection of liquid refrigerant into the compression process or the discharge line, (2) direct expansion of fluid in an external heat exchanger, (3) using chiller water on a packaged unit, (4) recirculating high-pressure liquid from the condenser, or (5) water from an evaporative condenser sump, the condenser must be sized for the total heat rejection (i.e., evaporator load plus shaft power for open compressors, and input power for hermetic compressors).

With an external oil cooler, the mass flow rate of oil injected into the compressor is usually determined by the desired discharge temperature rather than by the compressor sealing requirements, because oil acts predominantly as a heat transfer medium. Conversely, with direct liquid injection cooling, the oil requirement is dictated by the compressor lubrication and sealing needs.

Economizers. Twin-screw compressors are available with a secondary suction port between the primary compressor suction and discharge ports. This port can accept a second suction load at a pressure above the primary evaporator, or flash gas from a liquid subcooler vessel, known as an economizer.

In operation, gas is drawn into the rotor throat from the suction line. The thread is then sealed in sequence and compression begins. An additional charge may be added to the closed thread through a suitably placed port in the casing or sliding valve. The port is connected to an intermediate gas source at a pressure slightly higher than that reached in the compression process at that time. Both original and additional charges are then compressed to discharge conditions.

When the port is used as an economizer, some high-pressure liquid is vaporized at the side port pressure and subcools the remaining high-pressure liquid nearly to the saturation temperature at operating-side port pressure. Because this has little effect on compressor suction capacity, the effective refrigerating capacity of the compressor is increased by the increased heat absorption capacity of the liquid entering the evaporator. Furthermore, the only additional mass flow the compressor must handle is the flash gas entering a closed thread, which is above suction pressure. Thus, under most conditions, the capacity improvement is accompanied by an efficiency improvement.

Economizers become effective when the pressure ratio is about two and above (depending on volume ratio). Subcooling can be made with a direct-expansion shell-and-tube or plate heat exchanger, flash tank, or shell-and-coil intercooler.

As twin-screw compressors are unloaded, economizer pressure with a fixed port falls toward suction pressure. The additional capacity and improved efficiency of the economizer system is no longer available below a certain percentage of capacity, depending on design. Some compressors have the economizer port in the slide valve. This allows the economizer to be active down to the lowest percentage of capacity.

Hermetic and Semihermetic Compressors. Hermetic and semihermetic screw compressors are commercially available through 200 tons of refrigeration effect using R-22 or equivalent HFCs. Hermetic motors can operate under discharge, suction, or intermediate pressure. Motor cooling can be with gas, oil, and/or liquid refrigerant. Oil separation for these types of compressors may be accomplished with either an integrated oil separator or a separately mounted oil separator in the system. Figures 38 to 41 show four types of fixed-speed twin-screw compressors with slide valve unloading systems. For the lower capacity range (7 to 17 tons), welded-shell horizontal hermetic compressors are also available. Because of their smooth running characteristics, small size, and good capability for frequency inverter drive (1200 to 5220 rpm), they are preferable for railway and other transportation air-conditioning systems.

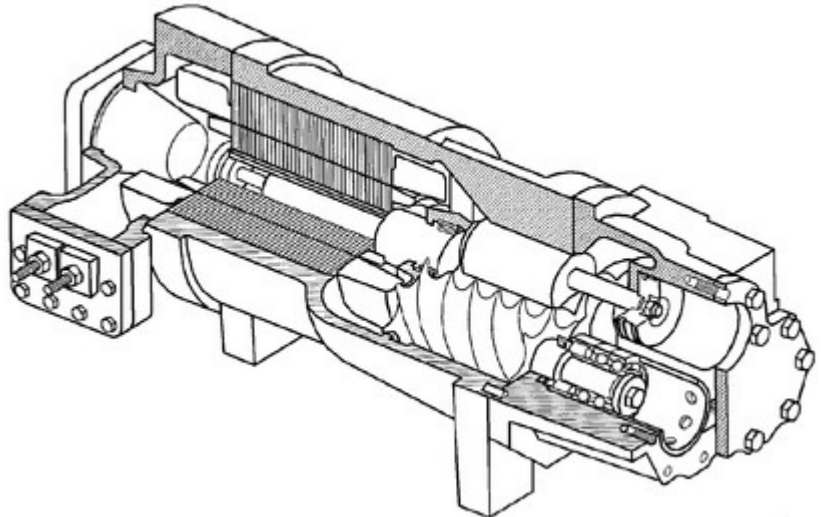


Figure 38. Semihermetic Twin-Screw Compressor with Suction-Gas-Cooled Motor and Slide Valve Unloading Mechanism

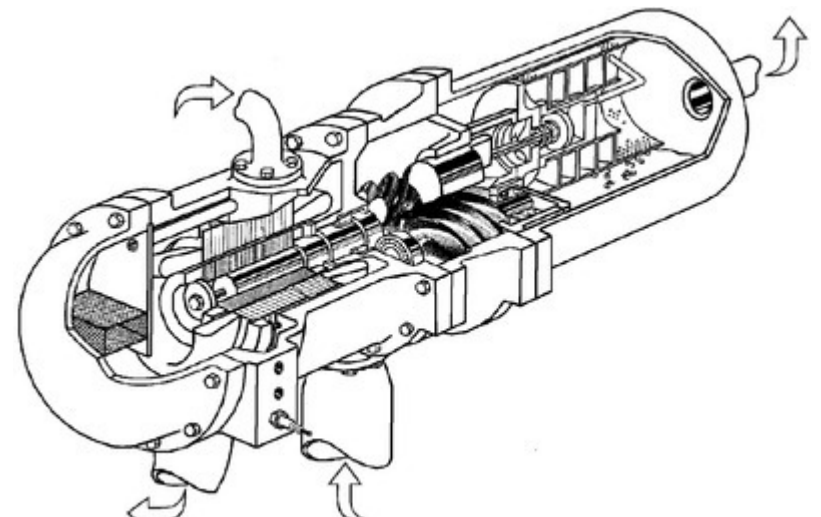


Figure 39. Semihermetic Twin-Screw Compressor with Motor Housing Used as Economizer; Built-In Oil Separator, and Slide Valve Unloading Mechanism

The increasing application of frequency-inverter drives for highly efficient capacity modulation by speed variation resulted in numerous further developments. Twin-screw technology in general shows very good potential for increasing operational speed above synchronous levels (3000 rpm at 50 Hz, 3600 rpm at 60 Hz), thus providing higher cooling capacity at full-load conditions as well as a wider speed range for capacity modulation. Consequently, the nominal displacement and physical size of the compressor can be reduced, which can reduce cost to partly compensate for the extra cost of the inverter drive.

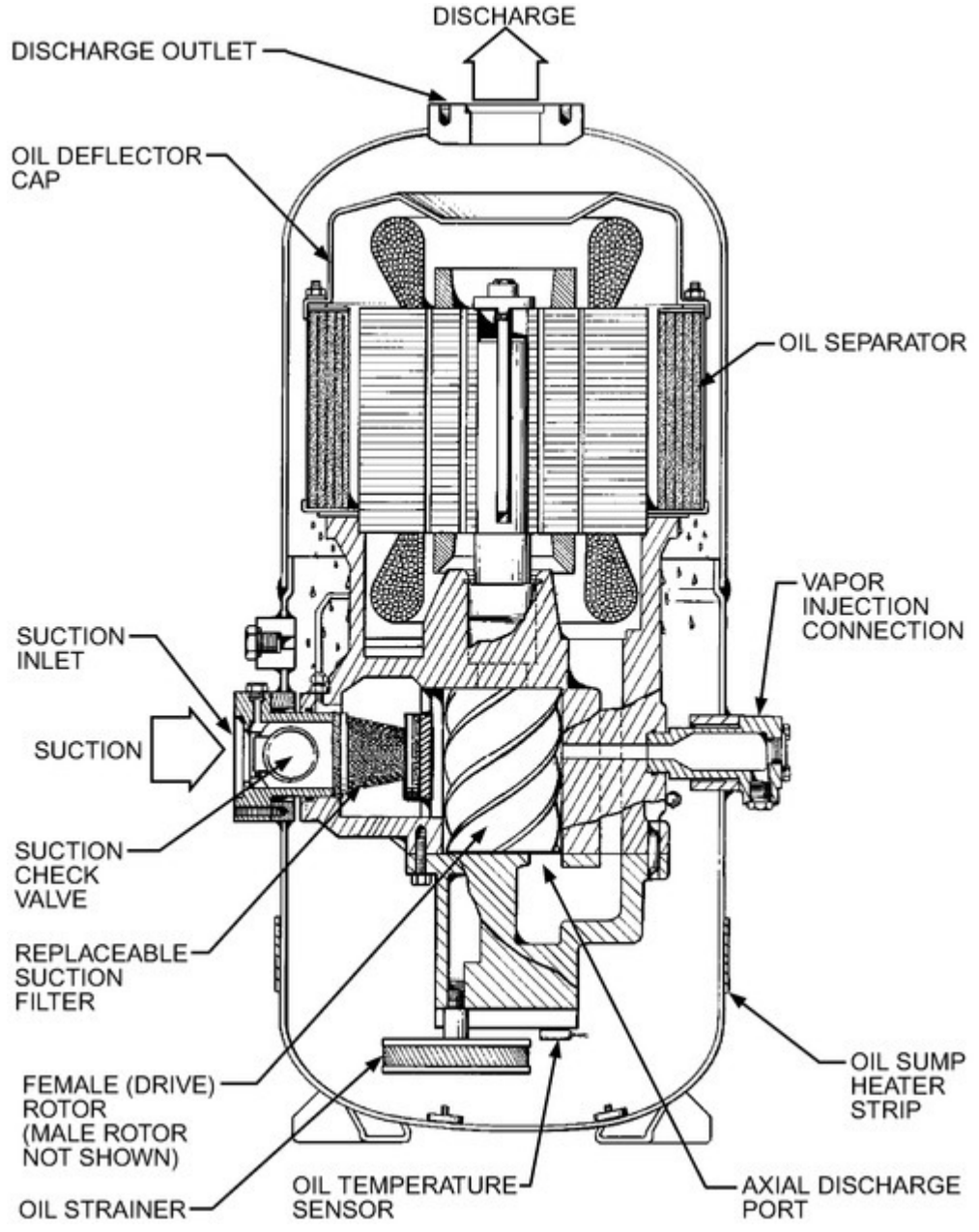


Figure 40. Vertical, Discharge-Cooled, Hermetic Twin-Screw Compressor

Besides individually packaging the compressor with an external frequency inverter (air-, water-, or refrigerant-cooled), some recent designs include an integrated, refrigerant-cooled drive. Because of the very intensive inverter cooling, this solution can be very compact. There is also no need for maintenance, and ambient temperature conditions do not influence performance and life span. For the lowest possible efficiency losses, the evaporated refrigerant leaving the cold plate (also the base frame for the inverter assembly) is sucked into a closed pocket of the rotor profile. Figure 42 shows a sectional drawing of a compressor with an integrated refrigerant-cooled inverter.

3.4 SCROLL COMPRESSORS

Scroll compressors are orbital motion, positive-displacement machines that compress gas using two interfitting, spiral-shaped scroll members (Figure 43). They are currently used in residential and commercial air-conditioning, refrigeration, and heat pump applications as well as in automotive air conditioning. In addition, scroll compressors are being applied in emerging markets requiring the compression of carbon dioxide, helium, and natural gas. The size and application range of scroll designs indicate scroll compressors' potential for new applications.

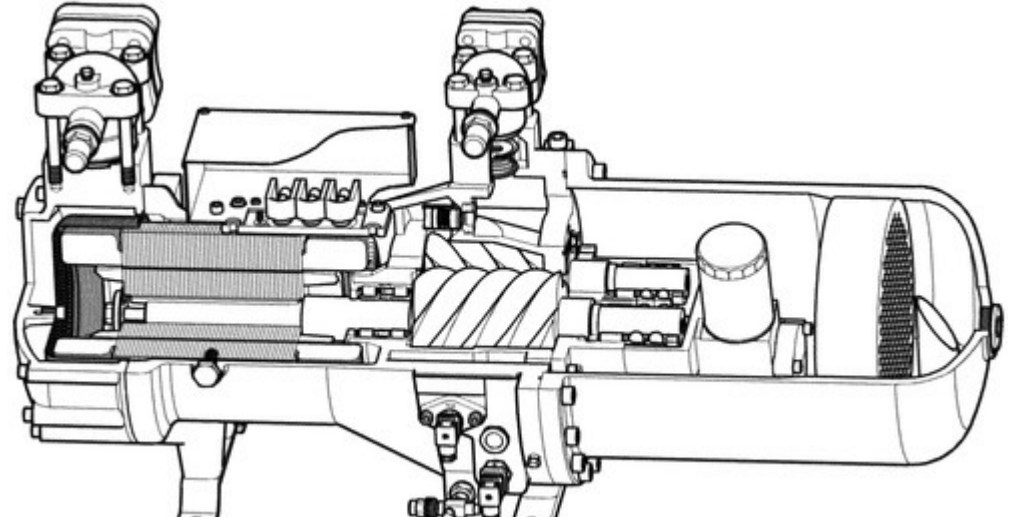


Figure 41. Semihermetic Twin-Screw Compressor with Suction-Gas-Cooled Motor, Slide Valve Unloading Mechanism, Economizer Port in Slide Valve, and Built-In Oil Separator

To function effectively with both high efficiency and low noise, a scroll compressor requires close-tolerance machining of the scroll members, as is now available with computer numerical control (CNC) manufacturing technology. Design features such as axial and radial compliance have also been proven necessary for most designs to meet the high-performance and low-variability requirements of high-volume manufacturing environments.

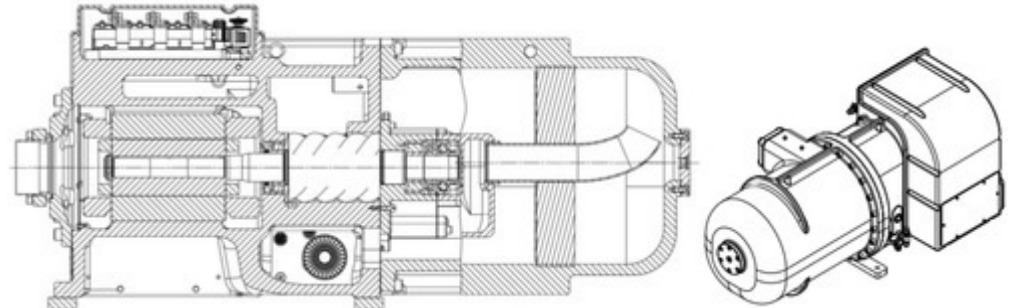


Figure 42. Semihermetic Twin-Screw Compressor with Suction-Gas-Cooled Motor, Refrigerant-Cooled Frequency Inverter, Means for V_1 Control, and Built-In Oil Separator

Mechanical Features

Scroll Members and Compression. In a vertical scroll compressor as shown in Figure 44, the scroll components consist of a lower (orbiting) scroll and an upper nonrotating scroll, often referred to as a "fixed" scroll, although it can by design move axially to support the gas sealing process. Gas sealing is critical to the performance advantage of scroll compressors and must be accomplished at both the vane tips and bases of the intermeshed scroll pair and at the flank contact locations. Tip/base sealing is generally considered more critical than flank sealing, with leakage control requiring significantly smaller tolerances.

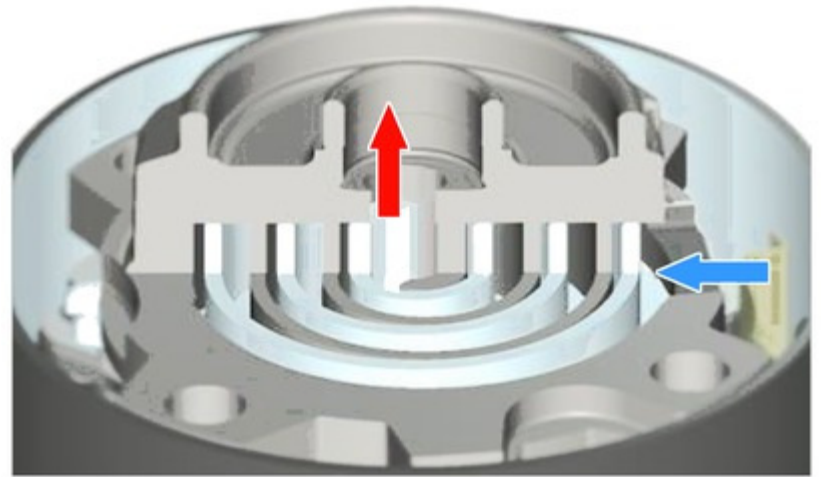


Figure 43. Interfitted Scroll Members (Purvis 1987)

Scroll members are typically a geometrically matched pair, assembled 180° out of phase. Each scroll member is open on one end of the vane and bound by a base plate on the other. The two scrolls are fitted to form pockets between their respective base plates and various lines of contact between their vane walls. One scroll is held fixed, while the other moves in an orbital path with respect to the first. The flanks of the scrolls remain in contact, although the contact locations move progressively inward. Relative rotation between the pair is typically prevented by an interconnecting Oldham coupling, which physically connects the scrolls and permits all planar motion between them except relative rotation.

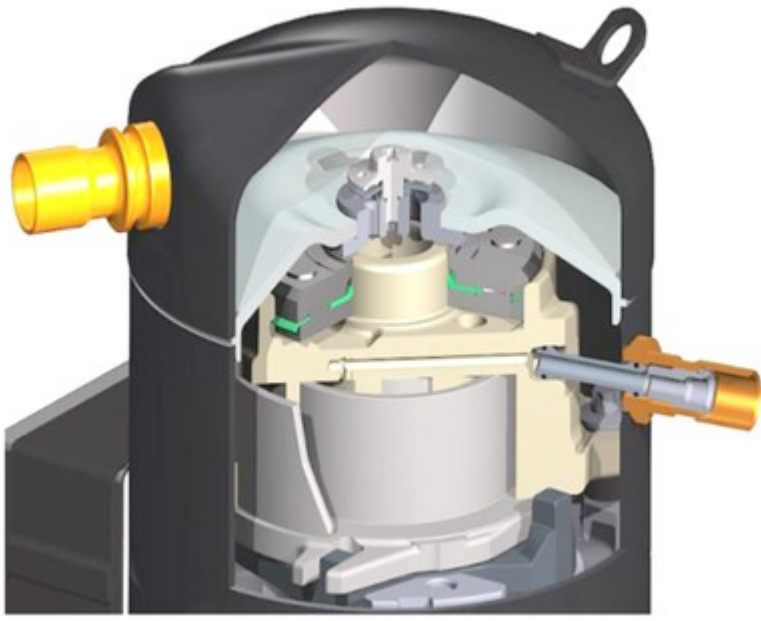


Figure 48. Scroll with Vapor Injection (Elson et al. 2008)

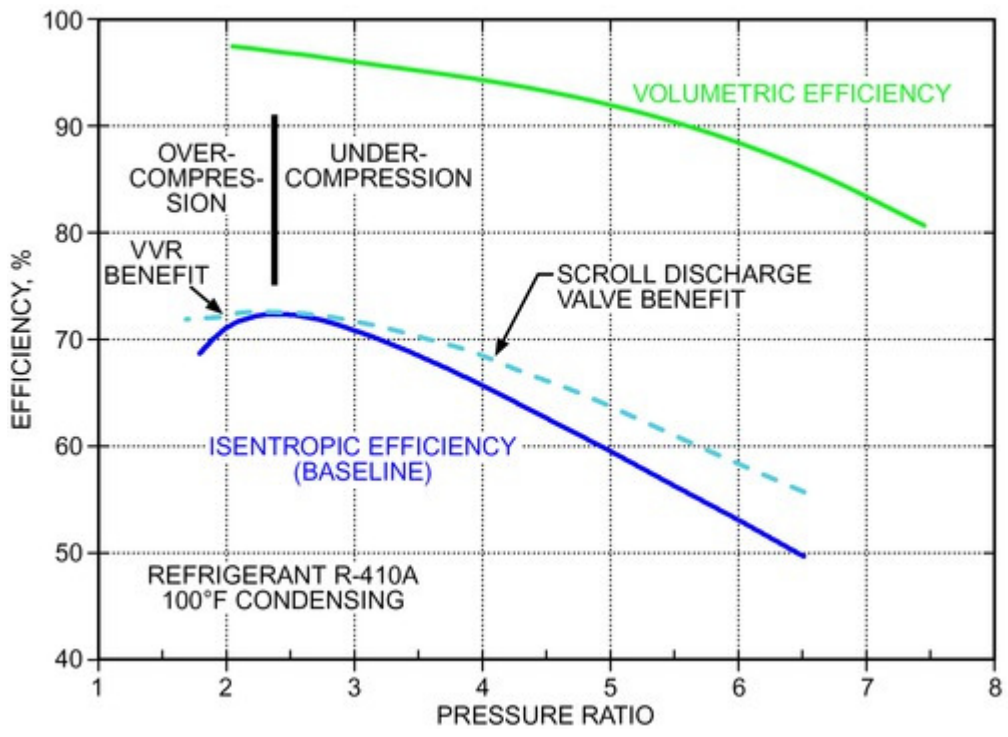


Figure 49. Volumetric and Isentropic Efficiency for Scroll Compressors

Depending on the application and refrigerant choice, the design pressure ratio of the scroll elements is generally selected to achieve optimal performance at system rating conditions. This basic design approach can also be further optimized for higher-pressure-ratio refrigeration and heat pump applications (under compression) by adding a dynamic discharge valve at the discharge port of the fixed scroll. This discharge valve is particularly important for pressure ratios of 6 to 8 or higher, when the scroll design pressure ratio is significantly below the compressor operating pressure ratio such that gas backflow and recompression can result in excessive compression work and inefficiency.

To minimize overcompression losses associated with low operating pressure ratios, additional discharge ports can be located in the pockets between the initial suction pockets and the main discharge port. This allows the scroll to operate at a variable volume ratio (VVR) determined by both the location of the primary discharge port and the additional ports shown in Figure 50.

Scroll compressors offer a flatter capacity versus the outdoor ambient curve than compressors with built-in clearance volume, which means they can more closely approach indoor cooling or heating requirements at high-demand conditions. As a result, the heat pump mode requires less supplemental heating, and the cooling mode is more comfortable, because cycling decreases as demand decreases (Figure 51). Scroll compressors available for the residential North American market are typically specified as producing AHRI operating efficiencies [coefficients of performance (COPs)] in the range of 2.8 to 3.1 using R-410A refrigerant.

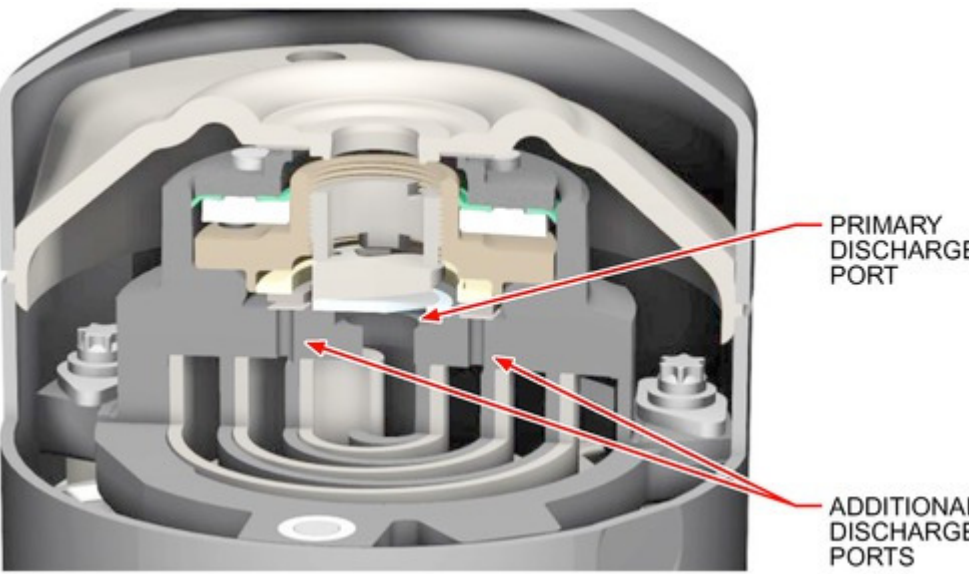


Figure 50. Variable Volume Ratio (VVR)

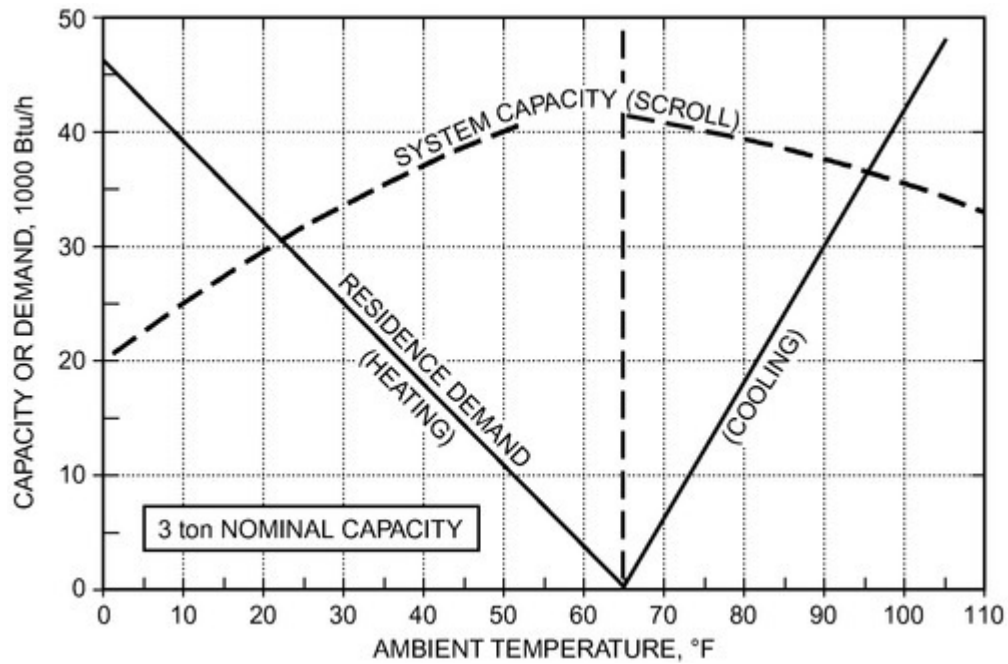


Figure 51. Scroll Capacity Versus Residence Demand (Purvis 1987)

Noise and Vibration

Scroll compressors have an inherent potential for low sound and vibration. They include minimal moving parts compared to other compressor technologies. Their continuous suction-compression-discharge process and low-gas-pressure pulsation help keep system noise and vibration low. A near-perfect dynamic balancing of the orbiting scroll inertia with counterweights eliminates possible mechanical vibration caused by rotating parts. Also, smooth surface finish and accurate machining of the vane profiles and base plates of both scroll members (a requirement for low gas leakage) aid in minimal impact noise from vane contact. A typical residential scroll compressor sound spectrum (Figure 52) shows a relatively uniform sound characteristic with minimal low-frequency content, which helps reduce sound transmission to the residential environment.

Operation and Maintenance

Most scroll compressors used today are hermetic, requiring virtually no maintenance or special operation procedures. However, for proper application and operation, the compressor manufacturer's manual should be followed.

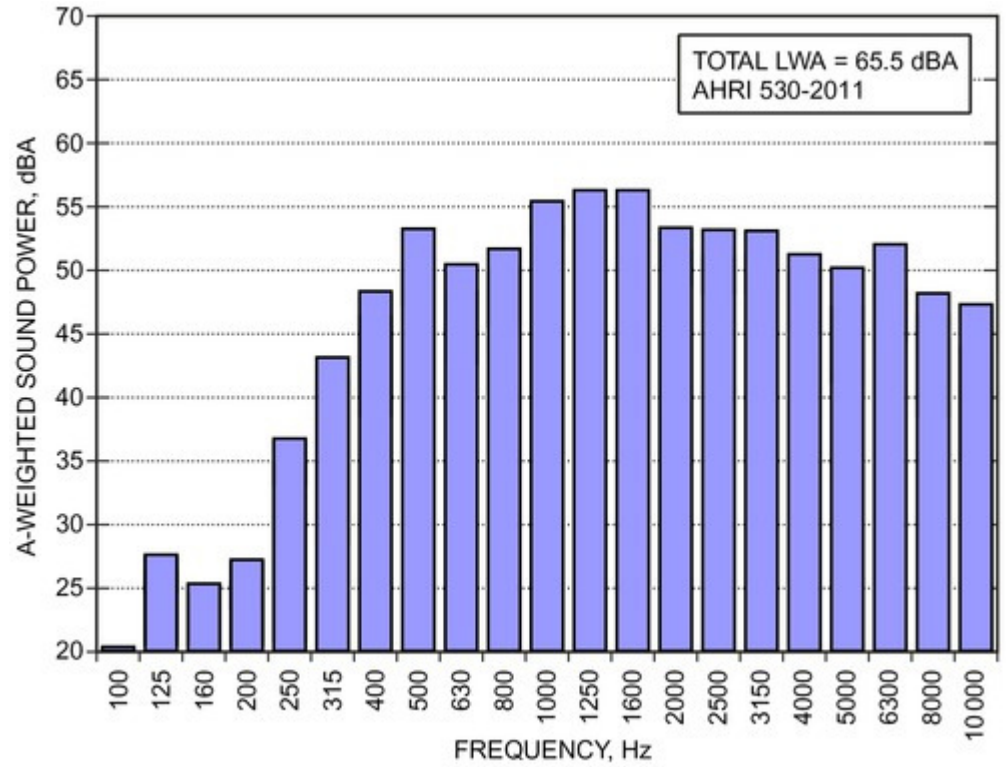


Figure 52. Typical Residential Scroll Sound Spectrum

3.5 TROCHOIDAL COMPRESSORS

Trochoidal compressors are small, rotary, positive-displacement compressors that can run at high speed up to 9000 rpm. They are manufactured in various configurations. Trochoidal curvatures can be produced by the rolling motion of one circle outside or inside the circumference of a basic circle, producing either epitrochoids or hypotrochoids, respectively. Both types of trochoids can be used either as a cylinder or piston form, so that four types of trochoidal machines can be designed (Figure 53).

In each case, the counterpart of the trochoid member always has one apex more than the trochoid itself. In the case of a trochoidal cylinder, the apexes of the piston show slipping along the inner cylinder surface; for trochoidal piston design, the piston shows a gear-like motion. As seen in Figure 53, a built-in compression pressure ratio disqualifies many configurations as valid concepts for refrigeration compressor design. Because of additional valve ports, clearances, etc., and the resulting decrease in the built-in maximum theoretical compression ratio, only the first two types with epitrochoidal cylinders, and all candidates with epitrochoidal pistons, can be used for compressor technology. The latter, however, require sealing elements on the cylinder as well as on the side plates, which does not allow the design of a closed sealing borderline.

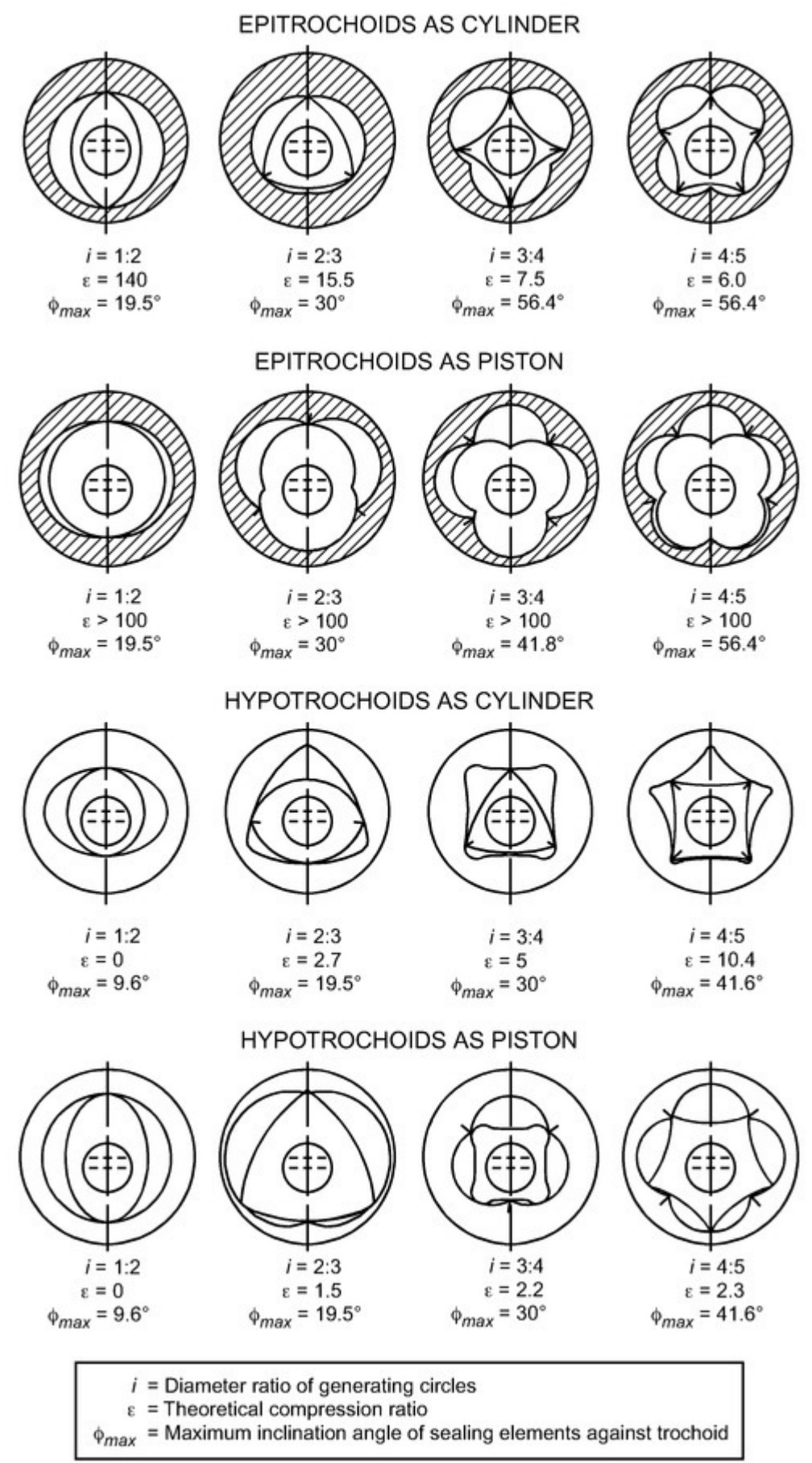


Figure 53. Possible Versions of Epitrochoidal and Hypotrochoidal Machines

In the past, trochoidal machines were designed much like those of today. However, like other positive-displacement rotary concepts that could not tolerate high internal oil circulation, early trochoidal compressors failed because of sealing problems. The invention of a closed sealing border by Wankel changed this (Figure 54). Today, Wankel trochoidal compressors with a three-sided epitrochoidal piston (motor) and two-envelope cylinder (casing) are built in capacities of up to 2 tons.

Description and Performance

Compared to other compressors of similar capacity, trochoidal compressors have many advantages typical of reciprocating compressors. Because of the closed sealing border of the compression space, these compressors do not require extremely small, expensive manufacturing tolerances; neither do they need oil for sealing, keeping them at low-pressure side with the advantage of low solubility and high viscosity of the oil-refrigerant mixture. Valves are usually used on a high-pressure side while suction is ported. A valveless trochoidal compressor can also be built.

Wankel compressor performance compares favorably with the reciprocating piston compressors at a higher speed and moderate pressure ratio range. A smaller number of moving parts, less friction, and resulting higher mechanical efficiency improve overall isentropic efficiency. This can be observed at higher speed when sealing is better, and in the moderate pressure ratio range when the influence of the clearance volume is limited.

4. CENTRIFUGAL COMPRESSORS

Centrifugal compressors, sometimes called turbocompressors, belong to a family of turbomachinery that includes fans, propellers, and turbines. These are classified as "dynamic" machines because they continuously exchange angular momentum between a rotating mechanical element and a steadily flowing fluid. For effective momentum exchange, their rotating speeds must be higher, but little vibration or wear results because of the steady motion and the absence of contacting parts such as pistons or vanes. Because their flows are continuous, turbomachines have greater volumetric capacities, size for size, than do positive-displacement devices.

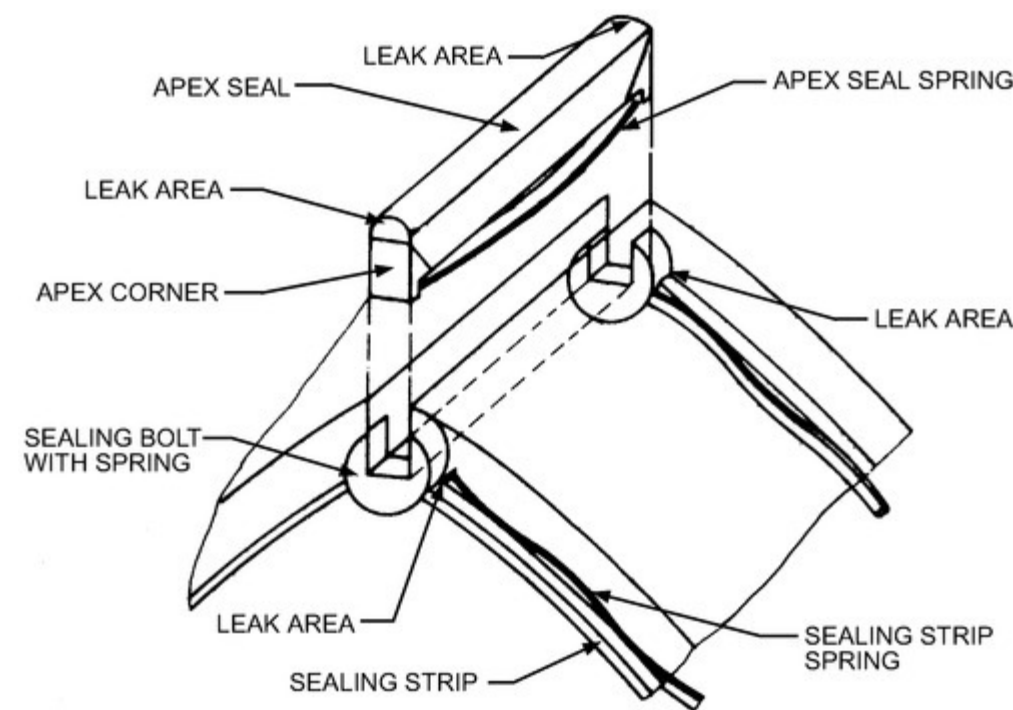
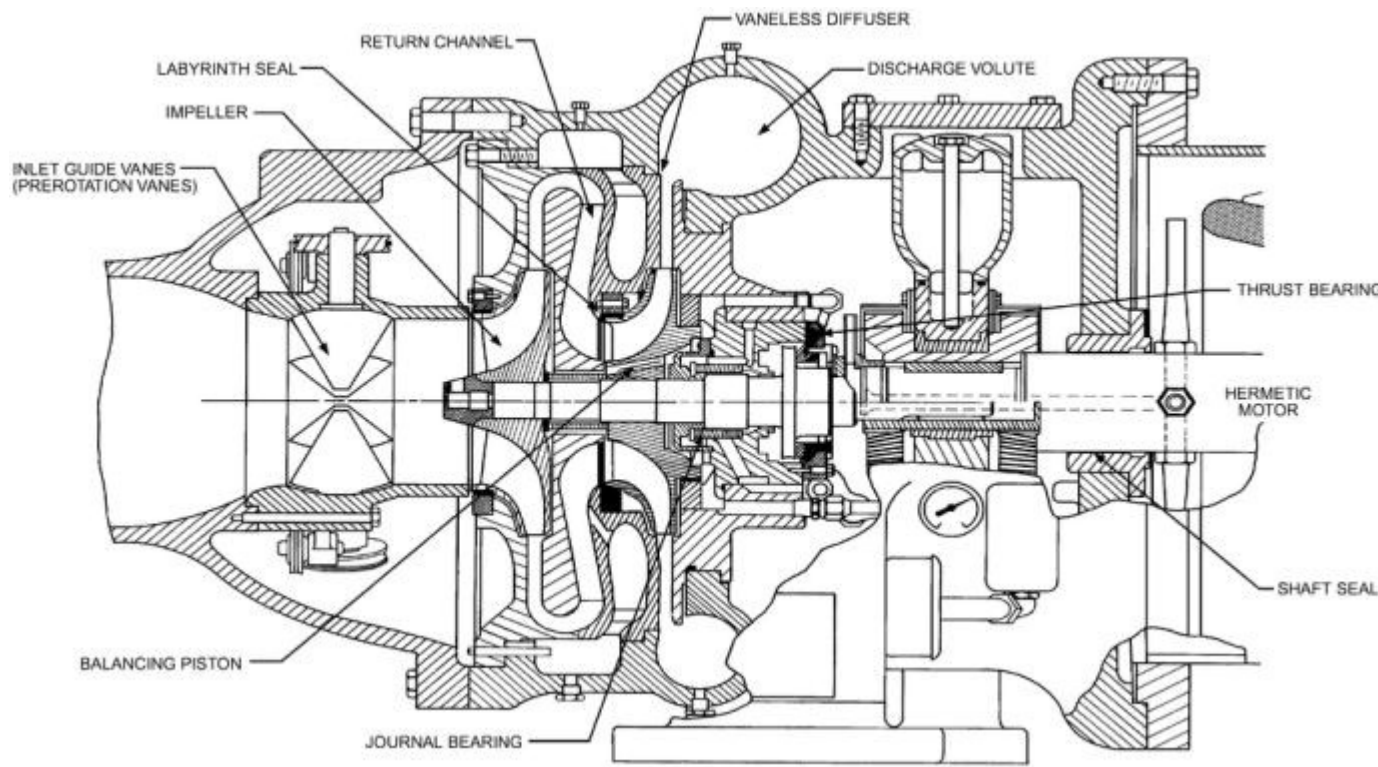


Figure 54. Wankel Sealing System for Trochoidal Compressors

Centrifugal compressors are used in a variety of refrigeration and air-conditioning installations. Suction flow ranges between 60 and 30,000 cfm, with rotational speeds between 1800 and 90,000 rpm. However, the high angular velocity associated with a low volumetric flow establishes a minimum practical capacity for most centrifugal applications. The upper capacity limit is determined by physical size, a 30,000 cfm compressor having a diameter of 6 or 7 ft. Centrifugal compressors are well-suited for air-conditioning and refrigeration applications because of their ability to produce a high pressure ratio. Suction flow enters the rotating element (impeller) axially, and is discharged radially at a higher velocity. The change in diameter through the impeller increases the gas velocity. This velocity (dynamic) pressure is then converted to static pressure through diffusion, which generally begins within the impeller and ends in a radial diffuser and volute outboard of the impeller. Suction gas generally passes through a set of adjustable inlet guide vanes or an external suction damper before entering the impeller. These devices are used for capacity control. High-velocity gas discharging from the impeller enters the radial diffuser, which can be vaned or vaneless. Vaned diffusers are typically used in compressors designed to produce high pressure. These vanes are generally fixed but can be adjustable. Adjustable diffuser vanes can be used for capacity modulation either in lieu of or in conjunction with the inlet guide vanes. A centrifugal compressor can be single-stage, with only one impeller, or it can be multistage, with two or more impellers mounted in the same casing, as shown in Figure 55. For process refrigeration, a compressor can have as many as 10 stages. In multistage compressors, gas discharged from the first stage is directed to the inlet of the second stage through a return channel. The return channel can contain a set of fixed-flow straightening vanes or an additional set of adjustable inlet guide vanes. Once the gas reaches the last stage, it is discharged from the impeller into a volute or collector chamber. From there, the high-pressure gas passes through the compressor discharge connection.



Centrifugal compressor flow animation



Centrifugal compressor parts

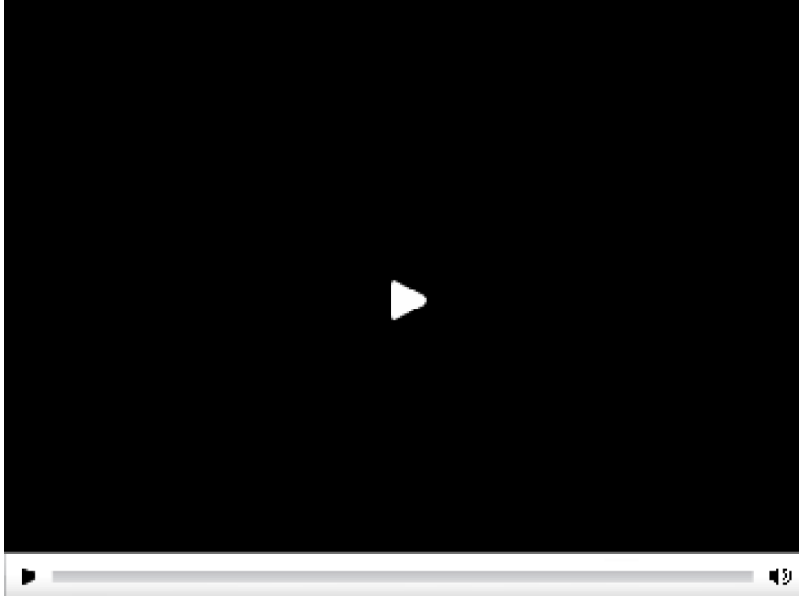


Figure 55. Centrifugal Refrigeration Unit Cross Section

When multistage compressors are used, interstage gas flows can be introduced between stages so that one compressor performs several functions at several temperatures.

Refrigeration Cycle

Typical applications might involve a single-, two-, or three-stage halocarbon compressor or a seven-stage ammonia compressor. Figure 56 illustrates a simple vapor compression cycle in which a centrifugal compressor operates between states 1 and 2. Figure 57 shows a more complex cycle, with two stages of compression and interstage liquid flash cooling. This cycle has a higher coefficient of performance than the simple cycle and is frequently used with two- through four-stage halocarbon and hydrocarbon compressors.

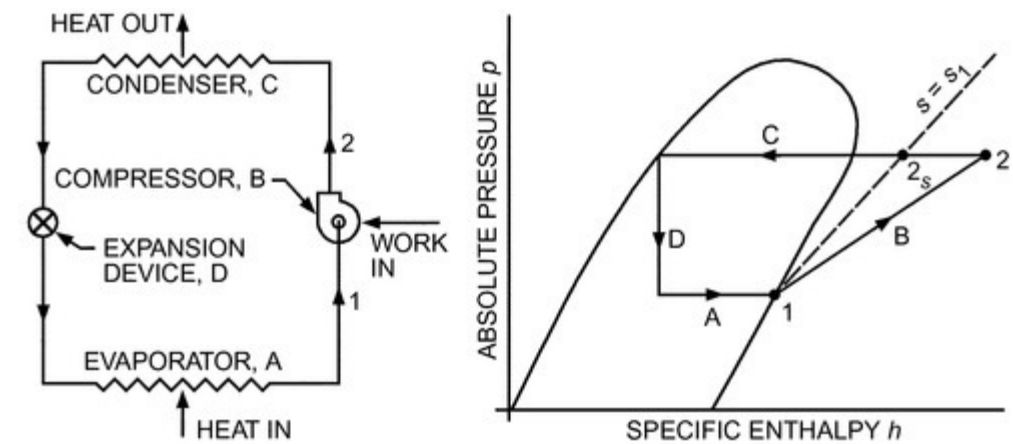


Figure 56. Simple Vapor Compression Cycle

Figure 58 shows a vapor compression cycle in which the expansion device is replaced by a power-recovering two-phase-flow turbine. The power recovered by the turbine is used to reduce the required compressor input work (Bras 1995). Although not commonly applied on commercial centrifugals, power recovery during expansion reduces the enthalpy of the two-phase mixture, thus increasing the refrigeration effect of this cycle.

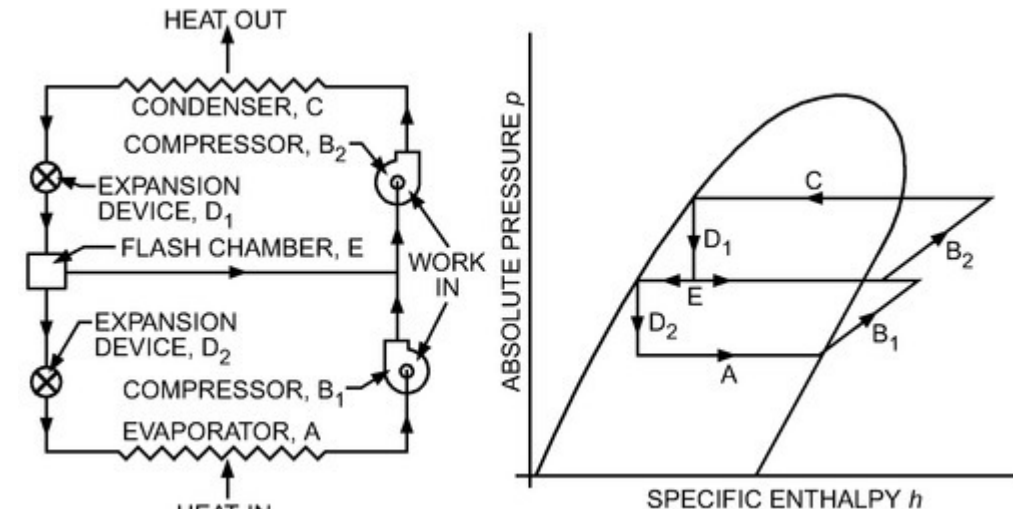


Figure 57. Compression Cycle with Flash Cooling

More than one stage of flash cooling can be applied to compressors with more than two impellers. Liquid subcooling and interstage desuperheating can also be advantageously used. For more information on refrigeration cycles, see Chapter 2 of the 2013 ASHRAE Handbook—Fundamentals.

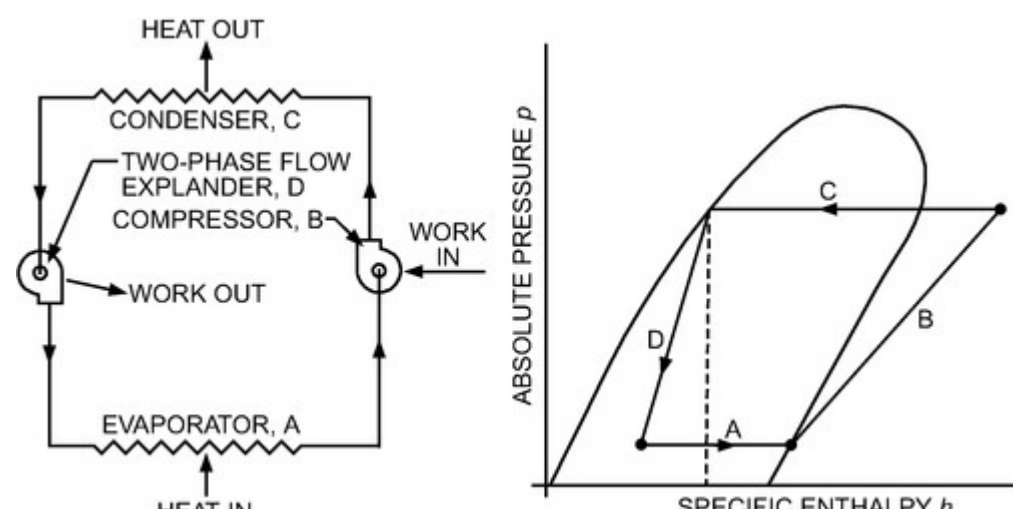


Figure 58. Compression Cycle with Power Recovery Expander

Angular Momentum

The momentum exchange, or energy transfer, between a centrifugal impeller and a flowing refrigerant is expressed by

$$W_i = u_i c_u / g_c \tag{14}$$

where
 W_i = impeller work input per unit mass of refrigerant, ft · lb_m / lb_m
 u_i = impeller blade tip speed, fps
 c_u = tangential component of refrigerant velocity leaving impeller blades, fps
 g_c = gravitational constant = 32.17 lb_m · ft / lb_f · s²

These velocities are shown in Figure 59, where refrigerant flows out from between the impeller blades with relative velocity b and absolute velocity c . The relative velocity angle β is a few degrees less than the blade angle because of a phenomenon known as slip. Equation (14) assumes that refrigerant enters the impeller without any tangential velocity component or swirl. This is generally the case at design flow conditions. If the incoming refrigerant were already swirling in the direction of rotation, the impeller's ability to impart angular momentum to the flow would be reduced. A subtractive term would then be required in the equation. Likewise, flow swirling in the direction opposite rotation would theoretically yield a positive effect on the angular momentum imparted. Some of the work done by the impeller increases refrigerant pressure; the remainder only increases its kinetic energy. The ratio of pressure-producing work to total work is known as the impeller reaction. Because this varies from about 0.4 to about 0.7, an appreciable amount of kinetic energy leaves the impeller with magnitude $c^2/2g_c$. To convert this kinetic energy into additional pressure, a diffuser is located after the impeller. Radial vaneless diffusers are most common, but vaned, volute, scroll, and conical diffusers are also used.

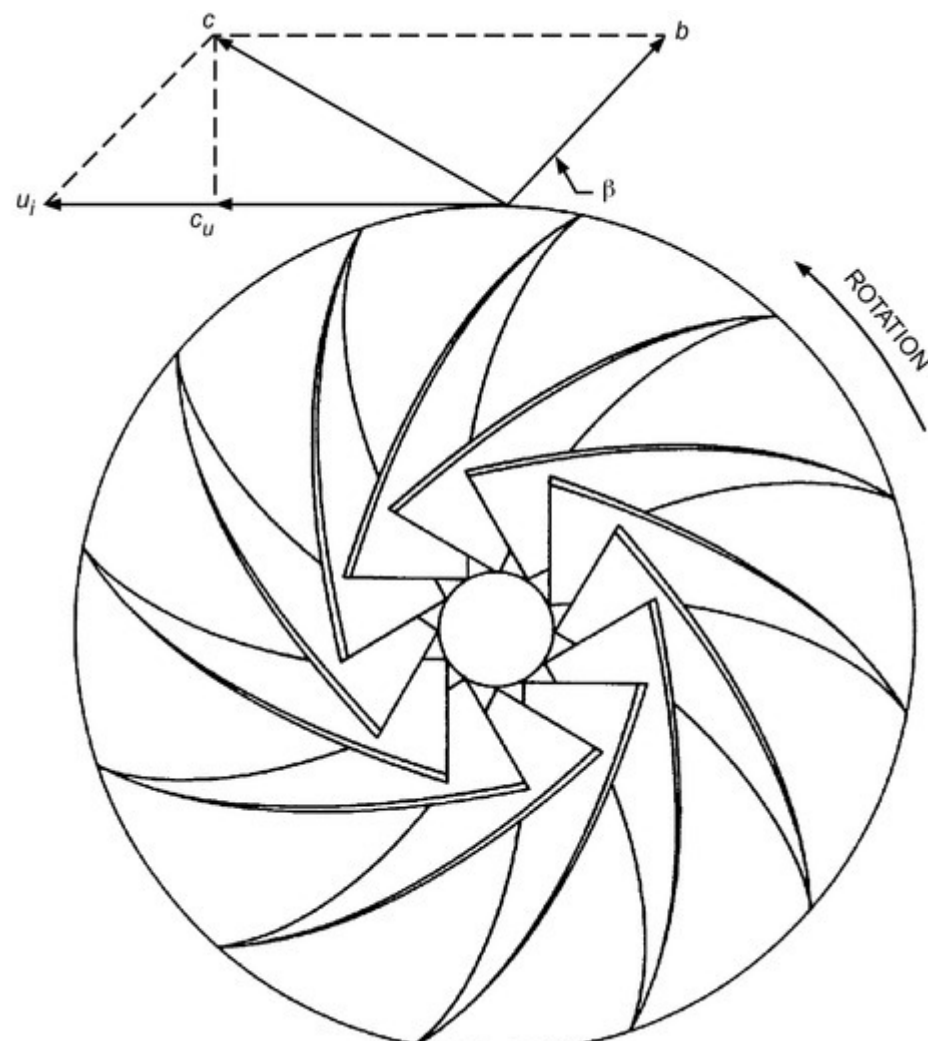


Figure 59. Impeller Exit Velocity Diagram

In a multistage compressor, flow leaving the first diffuser is guided to the inlet of the second impeller and so on through the machine, as shown in Figure 59. The total compression work input per unit mass of refrigerant is the sum of the individual stage inputs:

$$W = \sum W_i \tag{15}$$

provided that mass flow rate is constant throughout the compressor.

Specific Speed. This nondimensional index of optimum performance characteristic of geometrically similar stages is defined by

$$N_s = N \sqrt{Q_i} / W_{pi}^{0.75} = (1/\pi^3 k_i^{0.75}) \sqrt{Q_i} / ND_i^3 \tag{16}$$

The highest efficiencies are generally attained in stages with specific speeds between 600 and 850.

Mach Number

Two different Mach numbers are used. The flow Mach number M is the ratio of flow velocity c to acoustic velocity a at a particular point in the fluid stream:

$$M = c/a \tag{17}$$

where

$$a = v \sqrt{-g(\partial p / \partial v)_s} = \sqrt{\eta_s g p v}$$

(18)

The flow Mach number in a typical compressor varies from about 0.3 at the stage inlet and outlet to about 1.0 at the impeller exit. With increasing flow Mach number, the losses increase because of separation, secondary flow, and shock waves.

The impeller Mach number M_i , which is a pseudo Mach number, is the ratio of impeller tip speed to acoustic velocity a at the stage inlet:

$$M_i = u_i / a_i$$

(19)

Performance

From an applications standpoint, more useful parameters than μ and $(Q/\Delta P^2)$ are Ω and Θ (Sheets 1952):

$$\Omega = g W_p / a_i^2 = \mu (\sum u_i^2 / a_i^2)$$

(20)

$$\Theta = Q_1 a_1 D_1^2 = (M_1 / \pi) (Q_1 / N D_1^3)$$

(21)

Performance is typically represented with compressor maps providing nondimensional flow on the x -axis and nondimensional pressure rise on the y -axis. Typical compressor maps for fixed- and variable-speed compressors are shown in [Figures 60 and 61](#).

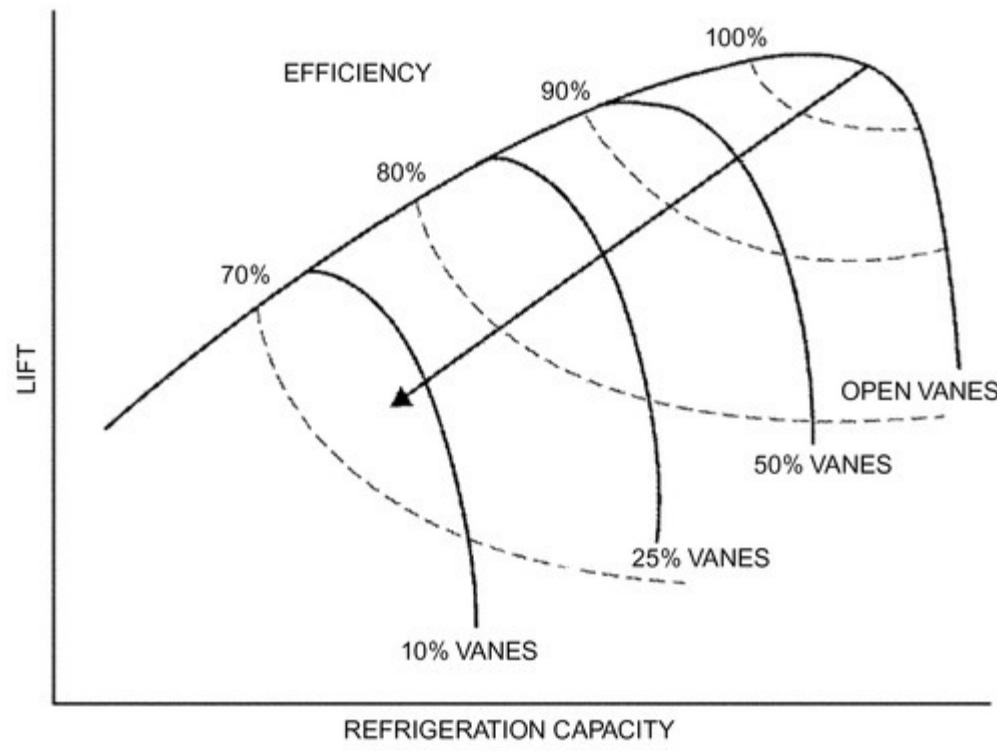


Figure 60. Typical Constant-Speed Centrifugal Compressor Performance

A typical compressor performance map is shown in [Figure 62](#), where percent of rated work is plotted with efficiency contours against percent of rated volumetric flow at various speeds. Point A is the design point at which the compressor operates with maximum efficiency. Point B is the selection or rating point at which the compressor is being applied to a particular system. From the application or user's point of view, Ω and Θ have their 100% values at point B.

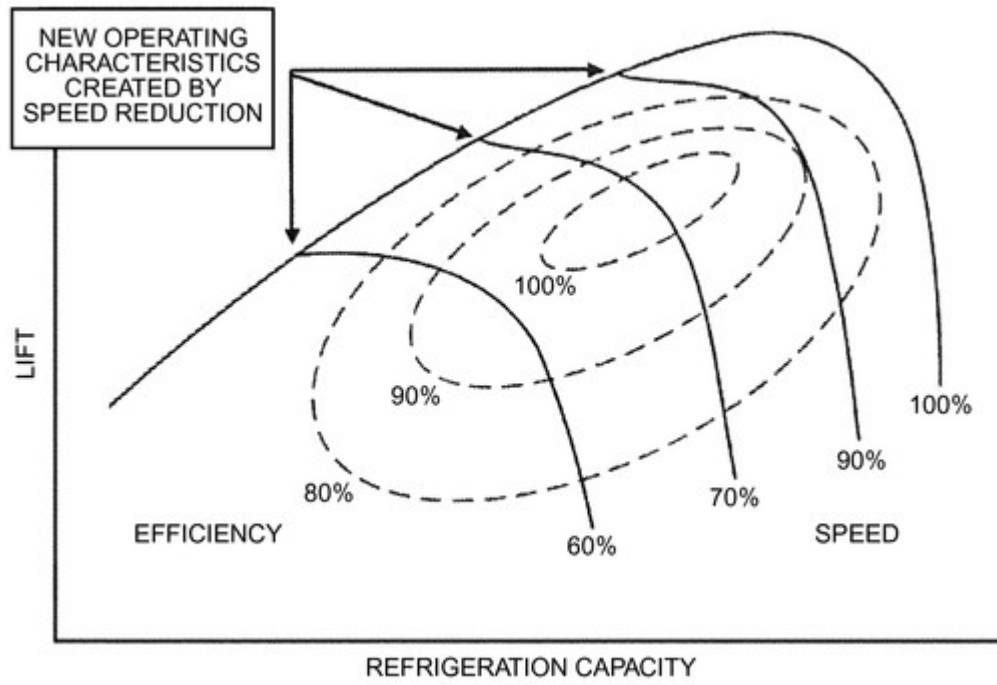


Figure 61. Typical Variable-Speed Centrifugal Compressor Performance

Surging

Part-load range is limited (on the left side of the performance map) by a **surge envelope**. Satisfactory compressor operation to the left of this line is prevented by unstable **surging** or **hunting**, in which refrigerant alternately flows backward and forward through the compressor, accompanied by increased noise, vibration, and heat. Prolonged operation under these conditions can damage the compressor.

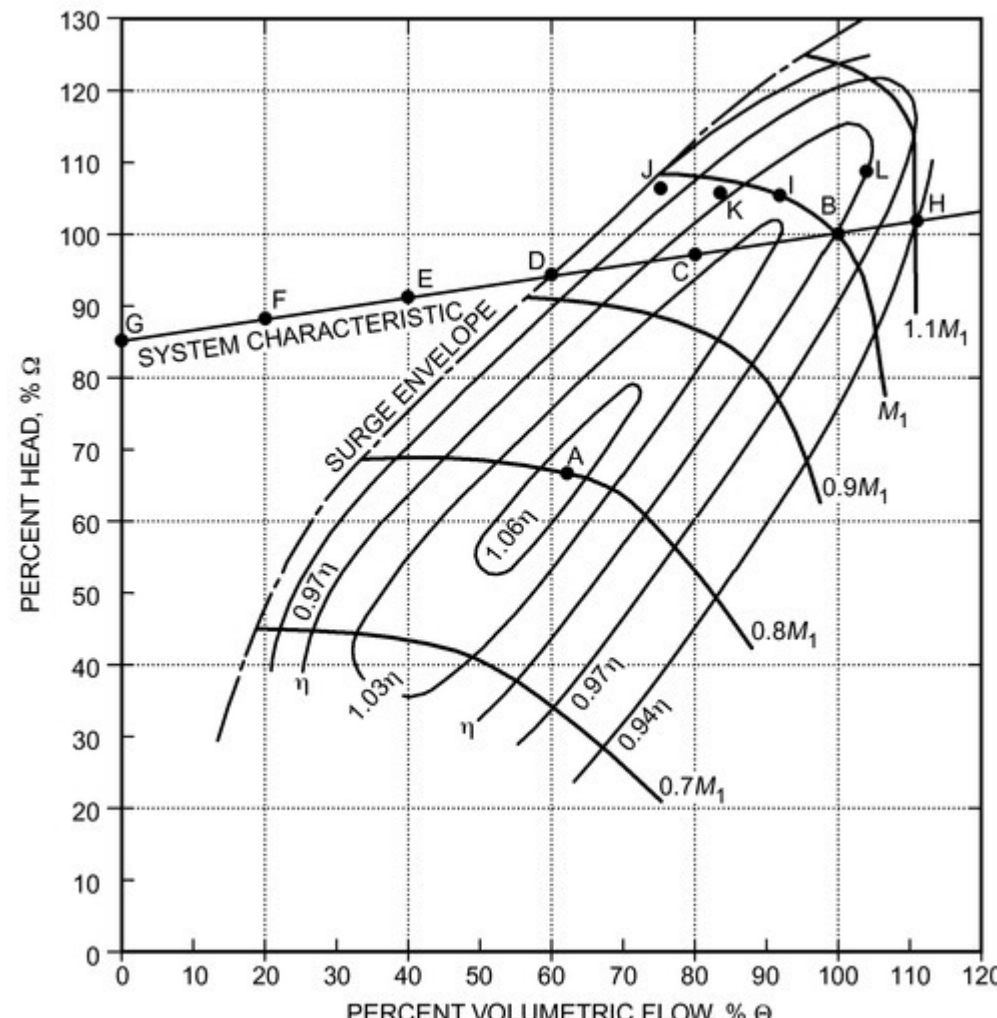


Figure 62. Typical Compressor Performance Curves

Flow reverses during surging about once every 2 s. Small systems surge at higher frequencies and large systems at lower. Surging can be distinguished from other kinds of noise and vibration by the fact that its flow reversals alternately unload and load the driver. Motor current varies markedly during surging, and turbines alternately speed up and slow down. Another kind of instability, **rotating stall**, may occur slightly to the right of the true surge envelope. This phenomenon forms rotating stall pockets or cells in the diffuser. It produces a roaring noise at a frequency determined by the number of cells formed and the impeller running speed. Driver load is steady during rotating stall, which is harmless to the compressor, but may vibrate components excessively.

System Balance and Capacity Control

In a centrifugal compressor, system balance is achieved through compressor capacity control. The method of capacity control selected depends on the intended refrigeration system characteristic for the application and the associated economics of the various control strategies. Capacity control methods affect compressor head capability and efficiency, which must be considered when selecting the appropriate method for an application.

Refrigeration and volumetric capacity are directly related to compressor speed, but the compressor's ability to produce pressure is a function of the square of a change in compressor speed. For example, operating at half compressor speed results in half the volumetric capacity and one-quarter the available pressure ratio. Variable-speed control requires a system characteristic of decreasing isentropic or polytropic work (pressure difference) with decreasing flow (capacity) to perform more efficiently. Reducing condensing pressure generally decreases isentropic or polytropic work. Impeller tip speed must remain constant if lift requirements do not change.

Methods of capacity control include speed variation, prerotation vanes, suction throttling, adjustable diffuser vanes, movable diffuser walls, impeller throttling sleeves, and combinations of these, such as prerotation vanes with variable speed. Each method has advantages and disadvantages in terms of performance, complexity, and cost that should be carefully considered before deciding on a capacity control strategy. The most common capacity control methods use speed variation and prerotation vanes.

Speed variation modulates capacity by adjusting the compressor drive speed to match the system characteristic. Speed variation is typically done by a variable-speed motor drive package, a turbine drive, or a generator-driven motor.

[Figure 62](#) shows a centrifugal compressor performance map using speed variation to modulate capacity without prerotation vanes. In addition to the head and flow characteristic, it shows the speed and efficiency at which the compressor operates in that particular application. A refrigeration system characteristic for a typical brine cooling system curve has been overlaid on the map, passing through points B, C, D, E, F, G, and H. With increased speed, the compressor at point H produces more than its rated capacity; with decreased speeds at points C and D, it produces less. Because of surging, the compressor cannot be operated satisfactorily at points E, F, or G. The system can be operated at these capacities, however, by using hot-gas bypass. Volume flow at the compressor suction must be at least that for point D in [Figure 62](#); this volume flow is reached by adding hot gas from the compressor discharge to the evaporator, or compressor suction piping. When hot-gas bypass is used, no further power reduction occurs as load decreases. The compressor is artificially loaded to stay out of the surge envelope. The increased volume caused by hot-gas recirculation performs no useful refrigeration.

Prerotation vanes (see [Figure 55](#)), also known as **inlet guide vanes**, modulate capacity by altering the direction of the fluid flow entering the impeller relative to the impeller blade leading edge. Setting the vanes to swirl flow in the direction of rotation produces a new compressor performance curve without any change in speed. Controlled positioning of the vanes can be done by pneumatically, electrically, or hydraulically.

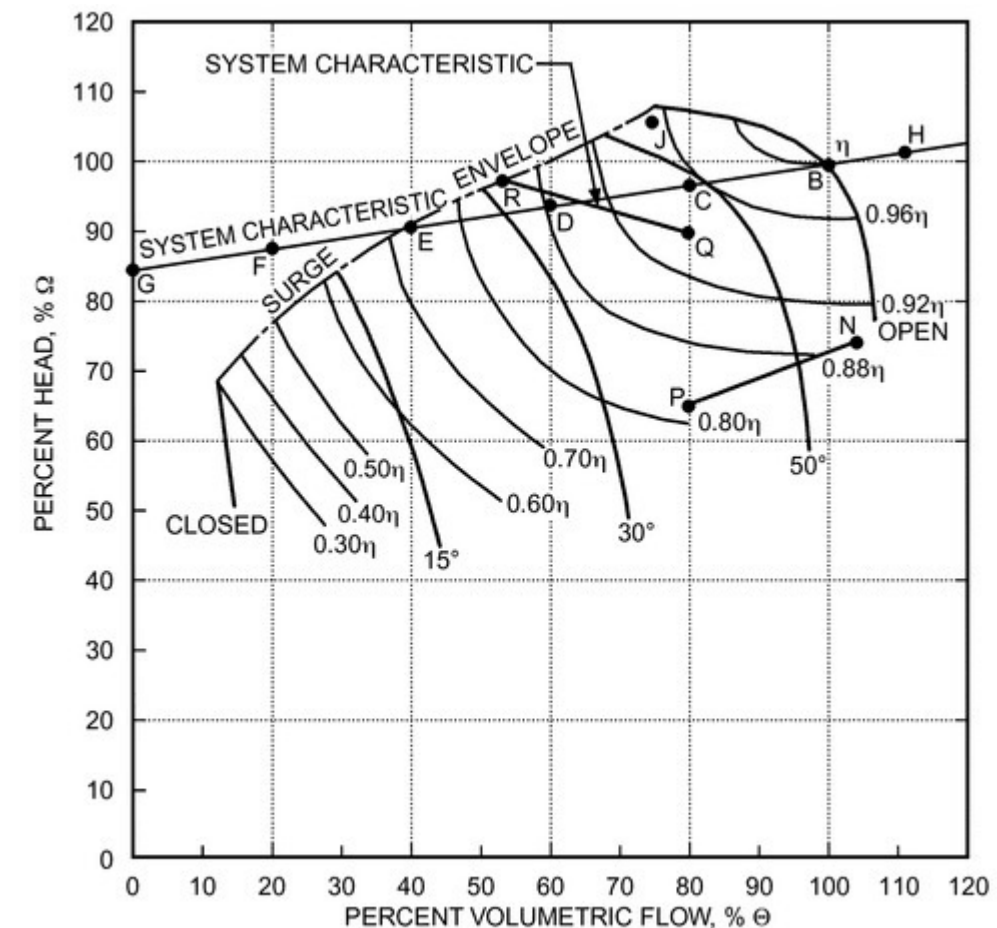


Figure 63. Typical Compressor Performance with Various Prerotation Vane Settings

[Figure 63](#) shows a centrifugal compressor performance map at constant driver speed, using prerotation vanes to modulate capacity. Typical curves for five different vane positions are shown in [Figure 63](#) for the compressor in [Figure 62](#) at the constant speed M_1 . In addition to the head and flow characteristic, it shows the prerotation vane position and efficiency at which the compressor operates in that particular application. With prerotation vanes wide open, the performance curve is identical to the M_1 curve in [Figure 62](#). The other curves are different, as are the efficiency contours and the surge envelope. The same system characteristic has been superimposed on this performance map, as in [Figure 62](#), to provide a comparison of these two modes of operation. In [Figure 63](#), point E can be reached with prerotation vanes; point H cannot. Theoretically, turning the vanes against rotation would produce a performance line passing through point H, but sonic relative inlet velocities prevent this unless operating at low Mach numbers. Hot-gas bypass is still necessary at points F and G with prerotation vane control, but to a lesser extent than with variable speed.

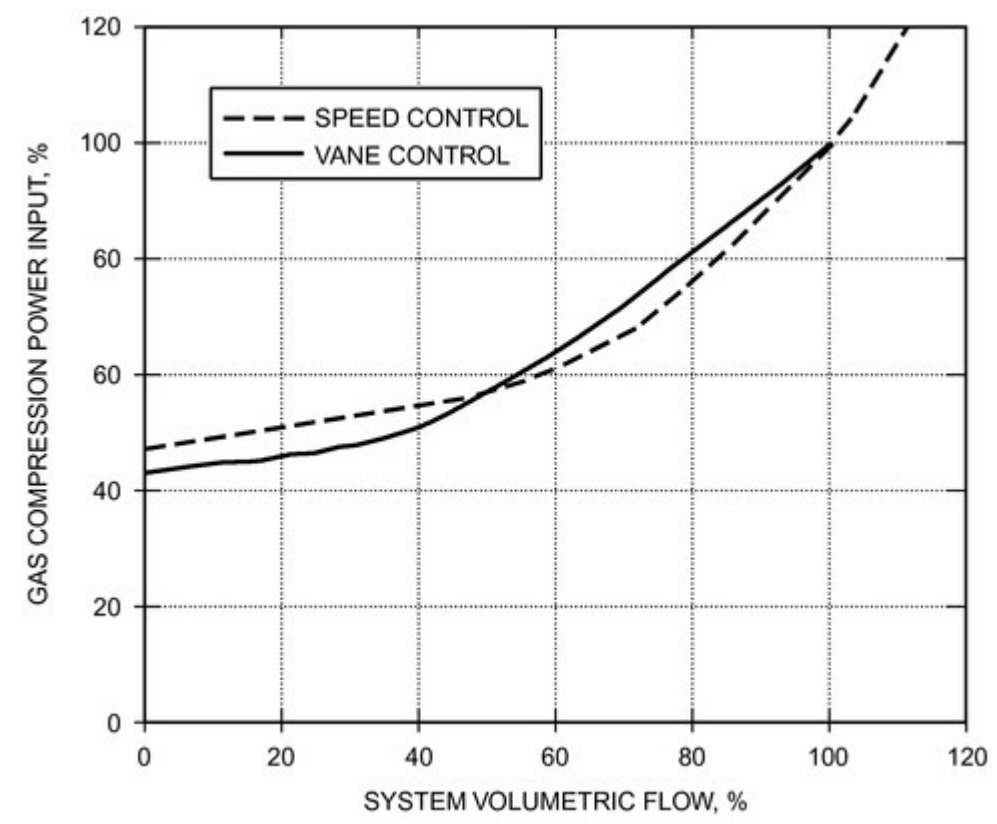


Figure 64. Typical Part-Load Gas Compression Power Input for Speed and Vane Capacity Controls

Gas compression powers for both control methods are depicted in [Figure 64](#). For the compressor and system assumed in this example, [Figure 64](#) shows that speed control requires less gas compression power down to about 55% of rated capacity. Prerotation vane control requires less power below 55%. Complete analysis must also consider friction loss and driver efficiency. Typical losses in a variable-speed drive and motor combination increase the full-speed power consumption of the system by 2.0 to 3.5%.

In applications where pressure requirements do not vary significantly at part load, prerotation vanes alone are typically suitable. When pressure requirements vary at part load, variable speed can provide distinct operational and economic advantages. In practice, variable speed is typically used in combination with prerotation vanes. In either case, a thorough energy analysis should be performed for the specific application to ensure that the selected capacity control method and system balance most economically and efficiently meet the application requirements.

4.1 APPLICATION

Vibration

Excessive vibration of a centrifugal compressor indicates malfunction, which may lead to failure. Periodic checking or continuous monitoring of the vibration spectrum at suitable locations is, therefore, useful in ascertaining the operational health of the machine. The relationship between internal displacements and stresses and external vibration is different for each compressor design. In a given design, this relationship also differs for the various causes of internal displacements and stresses, such as imbalance of rotating parts (either inherent or caused by deposits, erosion, corrosion, looseness, or thermal distortion), bearing instability, misalignment, distortion because of piping loads, broken motor rotor bars, or cracked impeller blades. It is, therefore, impossible to establish universal rules for the level of vibration considered excessive.

To establish meaningful criteria for a given machine or design, it is necessary to have baseline data indicative of proper operation. Significant increases of any frequency component of the vibration spectrum above the baseline then indicate a deterioration in the machine's operation; the frequency component for which this increase occurs is a good indication of the part of the machine deteriorating. Increases in the component at the fundamental running frequency, for instance, are usually because of deterioration of balance. Increases at approximately one-half the fundamental running frequency are caused by fluid-film bearing instability, and increases at twice the running frequency usually result from deterioration of alignment, particularly coupling alignment. Electrically induced vibration is typically at twice the fundamental frequency (e.g., 120 Hz for a 60 Hz line frequency).

As a general guide to establishing satisfactory vibration, a constant velocity criterion is sometimes used when the operating speed is between 600 and 60,000 rpm. Below 600 rpm, displacement is typically used as the criterion; above 60,000 rpm, acceleration is typically the criterion. In most cases, a velocity amplitude of 0.2 in/s is a reasonable criterion for vibration measured on the bearing housing.

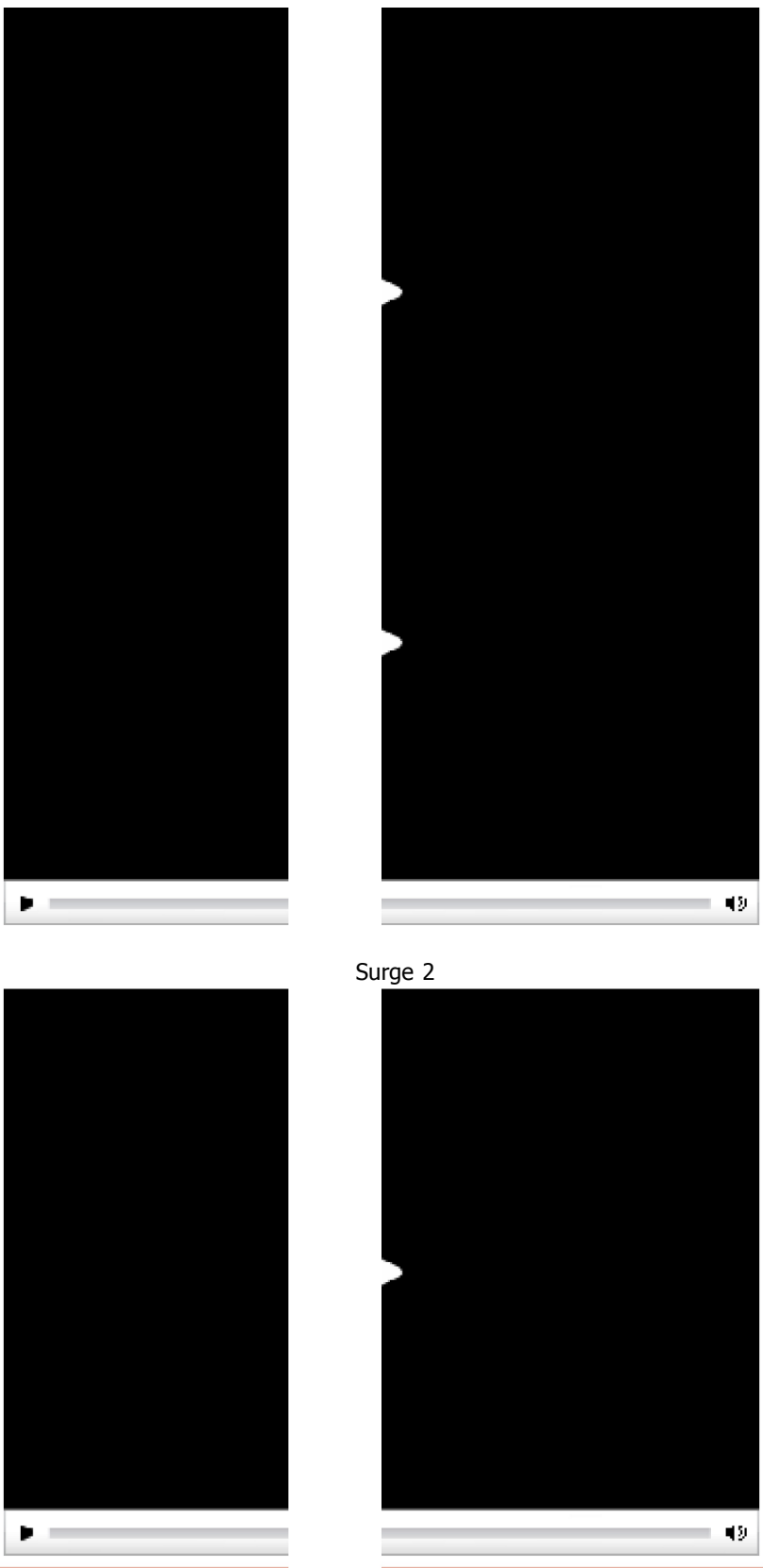
Although measuring vibration amplitude on the bearing housing is convenient, the value of such measurements is limited because the stiffness of the bearing housing in typical centrifugal compressors is generally considerably larger than that of the oil film. Thus, vibration monitoring systems often use noncontacting sensors, which measure displacement of the shaft relative to the bearing housing, either instead of, or in addition to, monitoring bearing housing vibration (Mitchell 1977). Such sensors are also useful for monitoring axial displacement of the shaft relative to the thrust bearing.

In some applications, compressor vibration, which is perfectly acceptable from a reliability standpoint, can cause noise problems if the machine is not properly isolated from the building. Vibration tests of the installed machine under operating conditions give a base comparison for future reference.

Chiller Sounds

Stall

Steady



Noise

Satisfactory application of centrifugal compressors requires careful consideration of noise control, especially if compressors are located near a noise-sensitive area of a building. The noise of centrifugal compressors is primarily of aerodynamic origin, principally gas pulsations associated with the impeller frequency and gas flow noise. Most predominant noise sources are of a sufficiently high frequency (above 1000 Hz) so that noise can be significantly reduced by carefully designed acoustical and structural isolation of the machine. Although the noise originates within the compressor proper, most is usually radiated from the discharge line and condenser shell. Equipment room noise can be reduced by up to 10 dB by covering the discharge line and condenser shell with acoustical insulation. In geared compressors, gear-mesh noise may also contribute to high-frequency noise; however, these frequencies are often above the audible range. This noise can be reduced by applying sound insulation material to the gear housing.

There are two important aspects in noise considerations for centrifugal compressors applications. In the equipment room, OSHA regulations specify employer responsibilities with regard to exposure to high sound levels. Increasing liability concerns are making designers more aware of compressor sound level considerations. Another important consideration is noise travel beyond the immediate equipment room.

Noise problems with centrifugal refrigeration equipment can occur in noise-sensitive parts of the building, such as a nearby office or conference room. The cost of controlling compressor noise transmission to such areas should be considered in building layout and weighed against cost factors for alternative locations of the equipment in the building.

If the equipment room is close to noise-sensitive building areas, it is usually cost-effective to have noise and vibration isolation designed by an experienced acoustical consultant, because small errors in design or execution can make the results unsatisfactory (Hoover 1960).

Blazier (1972) covers general information on typical noise levels near centrifugal refrigeration machines. Schaffer (2005) is another available source. Data on the noise output of a specific machine should be obtained from the manufacturer; the request should specify that measurements be in accordance with the current edition of ARI *Standard* 575.

Drivers

Centrifugal compressors are driven by almost any prime mover: a motor, turbine, or engine. Power requirements range from 33 to 12,000 hp, and driver selection must be compatible with compressor design speed. The driver is coupled to the compressor either directly or through a gear set to achieve the desired speed. Higher speeds can be achieved with a variable-speed drive. Flexible couplings are required to accommodate the angular, axial, and lateral misalignments that may arise in a drive train. Additional information on prime movers may be found in [Chapters 7](#) and [45](#).

Centrifugal refrigeration compressors are used in many special applications. These units use single-, two-, and three-stage compressors driven by open and hermetic motors. These designs have internal gears and direct drives, both of which are quieter, less costly, and more compact than external gearboxes. Internal gears or high-speed, variable-frequency-driven motors may be used when compressors operate at speeds higher than two-pole motor synchronous speed. [Chapter 43](#) discusses centrifugal water-chilling systems in greater detail.

A hermetic compressor absorbs motor heat because the motor is cooled by the refrigerant. An open motor is cooled by air in the equipment room; heat rejected by a hermetic motor must be considered in the refrigeration system design. Heat from an open motor must usually be removed from the equipment room, generally by mechanical ventilation. Because they operate at a lower temperature, hermetic motors are usually smaller than open motors for a given power rating. If a motor burns out, a hermetic system will require thorough cleaning, whereas an open motor will not. When serviced or replaced, an open motor must be carefully aligned to ensure reliable performance.

Starting torque must be considered in selecting a driver, particularly a motor or single-shaft gas turbine. Compressor torque is roughly proportional both to speed squared and to refrigerant density. The latter is often much higher at start-up than at rated operating conditions. If prerotation vanes or suction throttling cannot provide sufficient torque reduction for starting, the standby pressure must be lowered by some auxiliary means.

In certain applications, a centrifugal compressor drives its prime mover backward at shutdown. The compressor is driven backward by refrigerant equalizing through the machine. The extent to which reverse rotation occurs depends on the kinetic energy of the drive train relative to the expansive energy in the system. Large installations with dense refrigerants are most susceptible to running backward, a modest amount of which is harmless if suitable provisions have been made. Reverse rotation can be minimized or eliminated by closing discharge valves, side-load valves, and prerotation vanes at shutdown and opening hot-gas bypass valves and liquid refrigerant drains.

Paralleling

Problems associated with paralleling turbine-driven centrifugal compressors at reduced load are illustrated by points I and J in [Figure 62](#). These represent two identical compressors connected to common suction and discharge headers and driven by identical turbines. A single controller sends a common signal to both turbine governors so that both compressors should be operating at part-load point K (full load is at point L). The I machine runs 1% faster than its twin because of their respective governor adjustments, whereas the J compressor works against 1% more pressure difference because of the piping arrangement. The result is a 20% discrepancy between the two compressor loads.

One remedy is to readjust the turbine governors so that the J compressor runs 0.5% faster than the other unit. A more permanent solution, however, is to eliminate one of the common headers and to provide either separate evaporators or separate condensers. This increases the compression ratio of whichever machine has the greater capacity, decreases the compression ratio of the other, and shifts both toward point K.

The problem of imbalance, associated with turbine-driven centrifugal compressors, is minimal in fixed-speed compressors with vane controls. A loading discrepancy comparable to this example would require a 25% difference in vane positions.

Paralleling centrifugal compressors offers advantages in redundancy and improved part-load operation. This arrangement provides the capability of efficiently unloading to a lower percentage of total load. When the unit requirement reduces to 50%, one compressor can carry the complete load and operates at a higher percent volumetric flow and efficiency than a single large compressor.

Means must be provided to prevent refrigeration flow through the idle compressor to prevent inadvertent flow of hot-gas bypass through the compressor. In addition, isolation valves should be provided on each compressor to allow removal or repair of either compressor.

Other Specialized Applications

Centrifugal compressors are used in petroleum refineries, marine refrigeration, and in the chemical industry, as covered in [Chapters 26](#) and [46](#) of the [2014 ASHRAE Handbook—Refrigeration](#). Marine requirements are also detailed in ASHRAE *Standard* 26.

4.2 MECHANICAL DESIGN

Impellers

Impellers without covers, such as the one shown in [Figure 51](#), are known as open or unshrouded designs. Those with covered blades (see [Figure 53](#)) are known as shrouded impellers. Open models must operate close to contoured stationary surfaces to avoid excessive leakage around their vanes. Shrouded designs must be fitted with labyrinth seals around their inlets for a similar purpose. Labyrinth seals behind each stage are required in multistage compressors.

Impellers must be shrunk, clamped, keyed, or bolted to their shafts to prevent loosening caused by thermal and centrifugal expansions. Generally, they are made of cast or brazed aluminum or of cast, brazed, riveted, or welded steel. Aluminum has a higher strength-weight ratio than steel, up to about 300°F, which allows higher rotating speeds with lighter rotors. Steel impellers retain their strength at higher temperatures and are more resistant to erosion. Alternative materials may be required in corrosive applications.

Casings

Centrifugal compressor casings are about twice as large as their largest impellers, with suction and discharge connections sized for flow Mach numbers between 0.1 and 0.3. They are designed for the pressure requirements of ASHRAE *Standard* 15. A hydrostatic test pressure 50% greater than the maximum design working pressure is customary. If the casing is listed by a nationally recognized testing laboratory, a hydrostatic test pressure three times the working pressure is required. Cast iron is the most common casing material, used for temperatures as low as −150°F and pressures as high as 300 psia. Nodular iron and cast or fabricated steel are also used for low temperatures, high pressures, high speeds, and hazardous applications. Multistage casings are usually split horizontally, although unsplit barrel designs can also be used.

Rotor Dynamics

Centrifugal compressors are designed so that the first lateral critical speed is either well above or well below the operating speed. Operation at a speed between 0.8 and 1.1 times the first lateral speed is generally unacceptable from a reliability standpoint. The second lateral critical speed should be at least 25% above the operating speed of the machine.

Manufacturers have full responsibility for making sure critical speeds are not too close to operating speeds. Operating speed depends on the required flow of the application. Thus, the designer must ensure that the critical speed is sufficiently far away from the operating speed.

In applying open-drive machines, it is also necessary to consider torsional critical speed, which is a function of the designs of the compressor, drive turbine or motor, and coupling(s). In geared systems, gearbox design is also involved. Manufacturers of centrifugal compressors use computer programs to calculate torsional natural frequencies of the entire system, including the driver, coupling(s), and gears, if any. Responsibility for performing this calculation and ascertaining that the torsional natural frequencies are sufficiently far away from torsional exciting frequencies should be shared between the compressor manufacturer and the designer.

For engine drives, it may be desirable to use a fluid coupling to isolate the compressor (and gear set) from engine torque pulsations. Depending on compressor bearing design, there may be other speed ranges that should be avoided to prevent the nonsynchronous shaft vibration commonly called **oil whip** or **oil whirl**.

Bearings

Centrifugal compressors use hydrodynamic, rolling element, foil, and magnetic bearings to support radial and thrust loads. Radial loads are a result of static weight of the rotating assembly, gear mesh separation forces (if so configured), and, to a much lesser extent, aerodynamic loads. Thrust loads are primarily the result of the pressure field behind an impeller exceeding the combined pressure and momentum forces acting on the impeller inlet. In multistage designs, each impeller adds to the total thrust, unless some are mounted in the opposite direction to oppose the thrust. In some designs, a balancing piston is used behind the last stage impeller to reduce the overall thrust loads (see [Figure 55](#)). To avoid axial rotor vibration, some net axial load must be retained on the rotating assembly. This can be achieved using preload bearings or careful consideration of the thrust characteristics over the machine operating range. Regardless of the bearing system chosen, the bearings' dynamic stiffness and damping characteristics must be considered when determining compressor critical speeds, to ensure stable turbomachinery operation over the operating range.

Lubrication

Like motors and gears, bearings and lubrication systems of centrifugal compressors can be internal or external, depending on whether they operate in refrigerant atmospheres. For simplicity, size, and cost, most air-conditioning and refrigeration compressors have internal bearings, as shown in [Figure 55](#). In addition, they often have internal oil pumps, driven either by an internal motor or the compressor shaft; the latter arrangement is typically used with an auxiliary oil pump for starting and/or back-up service. Bearings may be lubricated with oil or refrigerant.

Most refrigerants are soluble in lubricating oils, the extent increasing with refrigerant pressure and decreasing with oil temperature. A compressor's oil may typically contain 20% refrigerant (by mass) during idle periods of high pressure and 5% during normal operation. Thus, refrigerant will come out of solution and foam the oil when such a compressor is started.

To prevent excessive foaming from cavitating the oil pump and starving the bearings, oil heaters minimize refrigerant solubility during idle periods. Standby oil temperatures between 130 and 150°F are required, depending on pressure. Once a compressor starts, its oil should be cooled to increase oil viscosity and maximize refrigerant retention during pulldown.

A sharp reduction in pressure before starting tends to supersaturate the oil. This produces more foaming at start-up than would the same pressure reduction after the compressor has started. Machines designed for a pressure ratio of 20 or more may reduce pressure so rapidly that excessive oil foaming cannot be avoided, except by maintaining a low standby pressure. Additional information refrigerant solubility in oil can be found in [Chapters 1, 4, and 12](#) of the [2014 ASHRAE Handbook—Refrigeration](#).

In a foil bearing, once the shaft is spinning fast enough, the working fluid is pulled into the bearing by viscous effects. Accordingly, foil bearings require an adequate supply of lubricant to pressurize and cool the bearings, but they do not require an external, pressurized system to supply the working fluid. Once pressurized, the gas pushes the foil away from the shaft so there is no contact. A relatively high-speed shaft is required to initiate the (air) gap; once this has been achieved, no wear occurs.

External bearings avoid the complications of refrigerant-oil solubility at the expense of some oil recovery problems. Any nonhermetic compressor must have at least one shaft seal. Mechanical seals are commonly used in refrigeration machines because they are leak-tight during idle periods. These seals require some lubricating oil leakage when operating, however. Shaft seals leak oil out of compressors with internal bearings and into compressors with external bearings. Means for recovering seal oil leakage with minimal refrigerant loss must be provided in external bearing systems.

Oil-Free Centrifugal Compressors

Oil-free chillers are enabled by various bearing technologies. Currently, these include foil, hybrid ceramic, and magnetic bearings.

Foil Bearings. Foil bearings are self-acting, compliant-surface (spring loaded) hydrodynamic bearings that use ambient air or process gas as the lubricant. They can be designed to carry radial or axial loads. Typical bearing applications include turbochargers, generators, air cycle machines, and aircraft and automobile engines.

A foil bearing consists of a top foil and "bump" foil. The top foil provides a scuff-resistant start/stop surface that captures gas and develops the film that supports the journal. The bump foil supports the top foil and can be tailored for the required bearing stiffness.

In a foil bearing, once the shaft is spinning fast enough, the working fluid is pulled into the bearing by viscous effects. Accordingly, foil bearings require an adequate supply of lubricant to pressurize and cool the bearings, but they do not require an external, pressurized system to supply the working fluid. Once pressurized, the gas pushes the foil away from the shaft so there is no contact. A relatively high-speed shaft is required to initiate the (air) gap; once this has been achieved, no wear occurs.

Hybrid Ceramic Bearings. Hybrid ceramic rolling-element bearings are similar to steel rolling-element bearings with the exception of using ceramic elements instead of steel. Typical construction consists of high-quality stainless steel races and silicon nitride ceramic rollers. In application, they can be designed to carry radial or axial loads using ball and/or cylindrical rollers.

Compared to steel, ceramic rollers are less dense and have higher modulus of elasticity and lower thermal expansion and friction coefficients. The lower density leads to lower centrifugal forces, allowing higher operating speeds. The higher modulus of elasticity results in higher bearing stiffness and lower contact areas, which, coupled with lower friction coefficients, lead to lower losses compared to steel contact pairs. Hybrid ceramic bearings require pressurized liquid for lubrication and cooling. Low losses enable use of refrigerants as lubricants. Application must ensure proper refrigerant lubricant properties along with low applied pressure drop through and temperature rise across the bearing to ensure refrigerant does not flash to gas in the contact zone.

Magnetic Bearings. Magnetic bearings use a combination of radial and axial bearings, position sensors, bearing controller, and bearing power source to levitate and align the shaft. An example compressor system using magnetic bearings is shown in [Figure 65](#).

Each radial bearing consists of a stationary electromagnetic armature and a series of laminations fixed to the compressor shaft. Thrust bearings typically consist of a set of electromagnetic armatures on either side of a single disc fixed to the shaft or on either end of features on the shaft. Position sensors near each bearing measure the location of the rotating shaft and transmit data to the bearing controller, which processes the information. The bearing controller sends a signal to the bearing power amplifier, which processes the information. The bearing controller sends a signal to the bearing power amplifier, which adjusts the magnetic field for each individual magnetic bearing to correct the shaft position. The shaft position is carefully monitored thousands of times per second. Magnetic bearings do not require lubrication but must be cooled, which is typically done by refrigerant gas over the armatures inside the compressor and by ambient air over the power amplifiers.

Two different systems are typically used to power the magnetic bearing system in the case of a power failure. In some, a battery back-up system is used. In others, the motor becomes an electrical generator that keeps the bearing system powered until the compressor comes to a complete stop. In the motor regeneration system, capacitors in the variable-frequency-drive system also aid in storing enough energy to maintain control of the rotor during a power loss shutdown. Back-up (touchdown) bearings support the shaft while at rest and serve as additional protection in case of a magnetic bearing system failure.

Accessories and Controls

The minimum accessories required by a centrifugal compressor with oil-lubricated bearings are an oil filter, oil cooler, and safety controls. Oil filters are usually rated for 15 to 20 μm or less. They may be built into the compressor but are more often externally mounted. Dual filters can be provided for industrial applications so that one can be serviced while the other is operating.

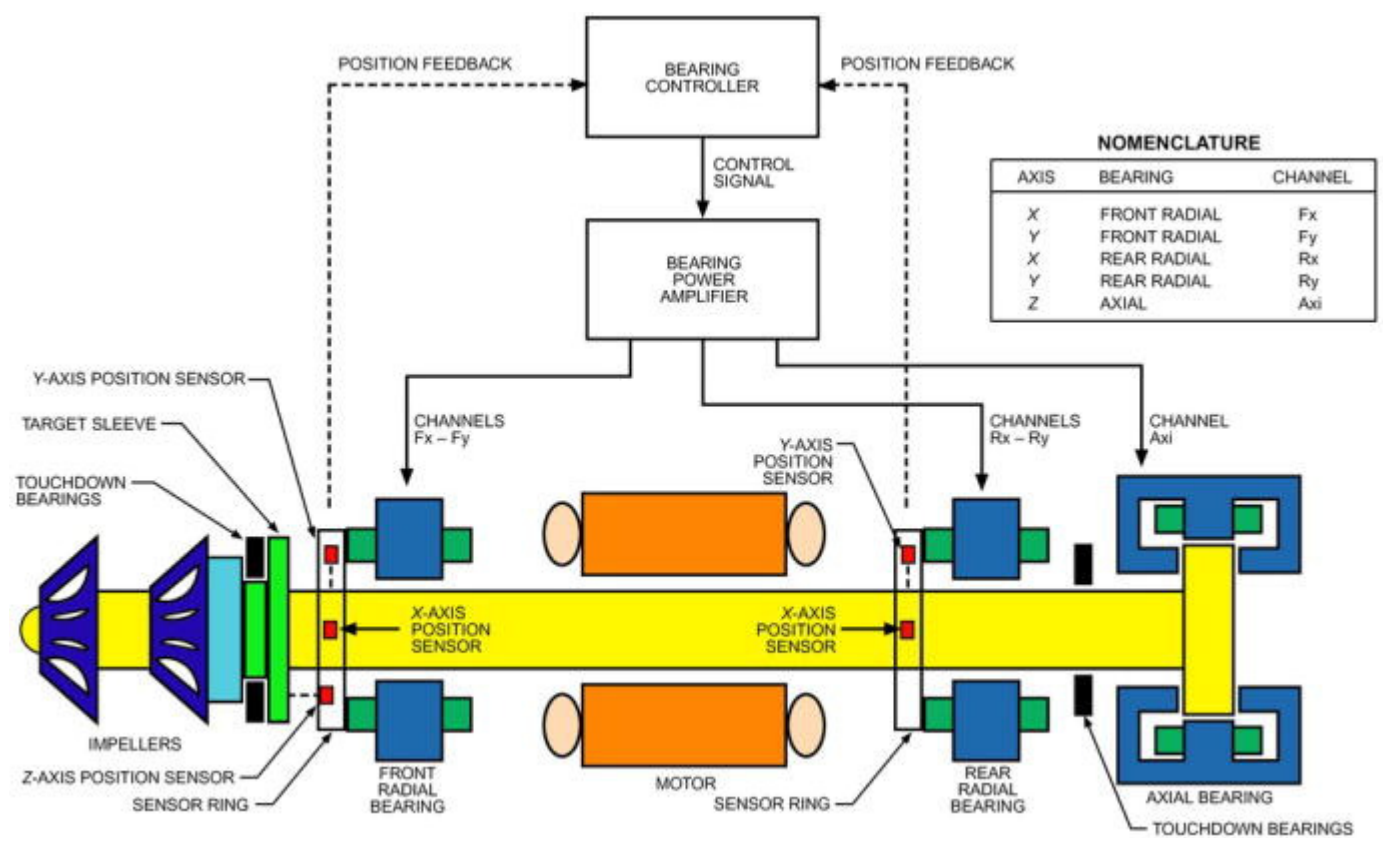


Figure 65. Magnetic Bearing Schematic on Oil-Free Centrifugal Compressor (Courtesy of Danfoss)

Single or dual oil coolers usually use condenser water, chilled water, refrigerant, or air as their cooling medium. Water- and refrigerant-cooled models may be built into the compressor, and refrigerant-cooled oil coolers may be built into a system heat exchanger. Many oil coolers are mounted externally for maximum serviceability.

Safety controls, with or without anticipatory alarms, must include a low-oil-pressure cutout, a high-oil-temperature switch, and high-discharge- and low-suction-pressure (or temperature) cutouts. A high-motor-temperature device is necessary in a hermetic compressor. Other common safety controls and alarms sense discharge temperature, bearing temperature, oil filter pressure differential, oil level, low oil temperature, shaft seal pressure, balancing piston pressure, surging, vibration, and thrust bearing wear.

Pressure gauges and thermometers are useful indicators of critical items monitored by the controls. Suction, discharge, and oil pressure gages are the most important, followed by suction, discharge, and oil thermometers. Suction and discharge instruments are often attached to components rather than to the compressor itself, but they should be provided. Interstage pressures and temperatures can also be helpful, either on the compressor or on the system. Electronic components may be used for all safety and operating controls. Electronic sensors and displays may be used for pressure and temperature monitoring.

4.3 ISENTROPIC ANALYSIS

The static pressure resulting from a compressor's work input or, conversely, the amount of work required to produce a given pressure rise, depends on compressor efficiency and the thermodynamic properties of the refrigerant. For an adiabatic process, the work input required is minimal if compression is isentropic. Therefore, actual compression is often compared to an isentropic process, and performance thus evaluated is based on an isentropic analysis.

The reversible work required by an isentropic compression between states 1 and 2, in [Figure 56](#) is known as the **adiabatic work**, as measured by the enthalpy difference between the two points:

$$W_s = J(h_2 - h_1) \tag{22}$$

where $J = 778.16 \text{ ft} \cdot \text{lb/Btu}$. Assuming negligible cooling occurs, the irreversible work done by the actual compressor is

$$W_s = J(h_2 - h_1) \tag{23}$$

Flash-cooled compressors cannot be analyzed by this procedure unless they are subdivided into uncooled segments with the cooling effects evaluated by other means. Compressors with side flows must also be subdivided. In [Figure 57](#), the two compression processes must be analyzed individually.

[Equation \(22\)](#) also assumes a negligible difference in kinetic energies of refrigerant at states 1 and 2. If this is not the case, a kinetic energy term must be added to the equation. All thermodynamic properties discussed in this section are static properties as opposed to stagnation properties; the latter includes kinetic energy.

The ratio of isentropic work to actual work is the **adiabatic efficiency**:

$$\eta_s = \frac{h_2 - h_1}{h_2 - h_1} \times \tag{24}$$

This varies from about 0.62 to about 0.83, depending on the application. Because of the thermodynamic properties of gases, a compressor's overall adiabatic efficiency does not completely indicate its individual stage performance. The same compressor produces different adiabatic results with different refrigerants and with the same refrigerant at different suction conditions.

In spite of its shortcomings, isentropic analysis has a definite advantage in that adiabatic work can be read directly from thermodynamic tables and charts similar to those presented in [Chapter 30](#) of the [2013 ASHRAE Handbook—Fundamentals](#). Where these are unavailable for the particular gas or gas mixture, they can be accurately calculated and plotted using thermodynamic relationships.

4.4 POLYTROPIC ANALYSIS

Polytropic analysis is of benefit when multistage compressors are being evaluated. The computational effort is less, because the average stage efficiency can be applied to all stages. The path equation for this reversible process is

$$\eta = v(dp/dh) \tag{25}$$

where η is the **polytropic efficiency** and v is the specific volume of the refrigerant. Reversible work done along the polytropic path is known as **polytropic work** and is given by

$$W_p = \int_{p_1}^{p_2} v dp \tag{26}$$

It follows from [Equations \(23\)](#), [\(25\)](#), and [\(26\)](#) that the polytropic efficiency is the ratio of reversible work to actual work:

$$\eta = \frac{W_p}{h_2 - h_1} \tag{27}$$

[Equation \(25\)](#) can be approximated by

$$\frac{p^m}{T} = \frac{p_1^m}{T_1} = \frac{p_2^m}{T_2} \tag{28}$$

$$p^m v^m = p_1 v_1^m = p_2 v_2^m \tag{29}$$

where

$$m = \frac{ZR \left(\frac{1}{\eta} + X \right)}{c_p} = \frac{(k-1/k)(1/\eta + X)Y}{(1+X)^2} \tag{30}$$

$$n = \frac{1}{Y - (ZR/c_p)(1/\eta + X)(1+X)} \tag{31}$$
$$= \frac{1+X}{Y[(1/k)(1/\eta + X) - (1/\eta - 1)]}$$

and

$$X = \frac{T}{v} \left(\frac{\partial v}{\partial p} \right)_p - 1 \tag{32}$$

$$Y = - \frac{p}{v} \left(\frac{\partial v}{\partial p} \right)_T \tag{33}$$

$$Z = \frac{pV}{RT} \tag{34}$$

Also, R is the gas constant and k is the ratio of specific heats; all properties are at temperature T . These equations can be used to allow integration so that [Equation \(26\)](#) can be written as follows:

$$\tag{35}$$

Further manipulation eliminates the exponent:

$$W_p = \frac{n}{(n-1)} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{(n-1)/n} - 1 \right]$$

$$W_p = \left[\frac{p_2 v_2 - p_1 v_1}{\ln(p_2 v_2 / p_1 v_1)} \right] \ln \left(\frac{p_2}{p_1} \right)$$

(36)

For greater accuracy in handling gases with properties known to deviate substantially from those of a perfect gas, a more complex procedure is required. The accuracy with which [Equations \(28\) and \(29\)](#) represent [Equation \(25\)](#) depends on the constancy of m and n along the polytropic path. Because these exponents usually vary, mean values between states 1 and 2 should be used. Compressibility functions X and Y have been generalized for gases in corresponding states by Schultz (1962) and their equivalents are listed by Edminster (1961). For usual condions of refrigeration interest (i.e., for $p < 0.9p_c$, $T < 1.5T_c$, and $0.6 < Z$), these functions can be approximated by

$$X = 0.1846(8.36)^{1/Z} - 1.539$$

(37)

$$Y = 0.074(6.65)^{1/Z} - 0.509$$

(38)

Compressibility factor Z has been generalized by Edminster (1961) and Hougen et al. (1959), among others. Generalized corrections for specific heat at constant pressure c_p can also be found in these works. [Equations \(28\) and \(22\)](#) make possible the integration of [Equation \(26\)](#):

$$W_p = f \left(\frac{n}{n-1} \right) p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{(n-1)/n} - 1 \right]$$

(39)

In [Equation \(39\)](#), polytropic work factor f corrects for whatever error may result from the approximate nature of [Equations \(28\) and \(29\)](#). Because the value of f is between 1.00 and 1.02 in most refrigeration applications, it is generally neglected. Once the polytropic work has been found, efficiency follows from [Equation \(22\)](#). Polytropic efficiencies range from about 0.70 to about 0.84, a typical value being 0.76. The highest efficiencies are obtained with the largest compressors and densest refrigerants because of a Reynolds number effect discussed by Davis et al. (1951). A small number of stages is also advantageous because of parasitic loss associated with each stage. Overall, polytropic work and efficiency are more consistent from one application to another because they represent an average stage aerodynamic performance. Instead of using [Equations \(25\) to \(32\)](#), it is easier and often more desirable to determine the adiabatic work by isentropic analysis and then convert to polytropic work by

$$\frac{W_p}{W_s} \approx \eta \left[\frac{(p_2/p_1)^{(k-1)/k\eta} - 1}{(p_2/p_1)^{(k-1)/k} - 1} \right]$$

(40)

[Equation \(40\)](#) is strictly correct only for an ideal gas, but because it is a ratio involving comparable errors in both numerator and denominator, it is of more general utility. [Equation \(40\)](#) is plotted in [Figure 66](#) for $\eta = 0.76$. To obtain maximum accuracy, the ratio of specific heats k must be a mean value for states 1, 2, and 2s. If c_p is known, k can be determined by

$$k = \frac{1}{1 - (ZR/c_p)(1 + X)^2/Y}$$

(41)

Gas compression power is

$$P = \dot{m} W$$

(42)

where $\dot{m}$ is mass flow. To obtain total shaft power, add the mechanical friction loss. Friction loss varies from less than 1% of gas power to more than 10%. A typical estimate is 3% for compressor friction losses.

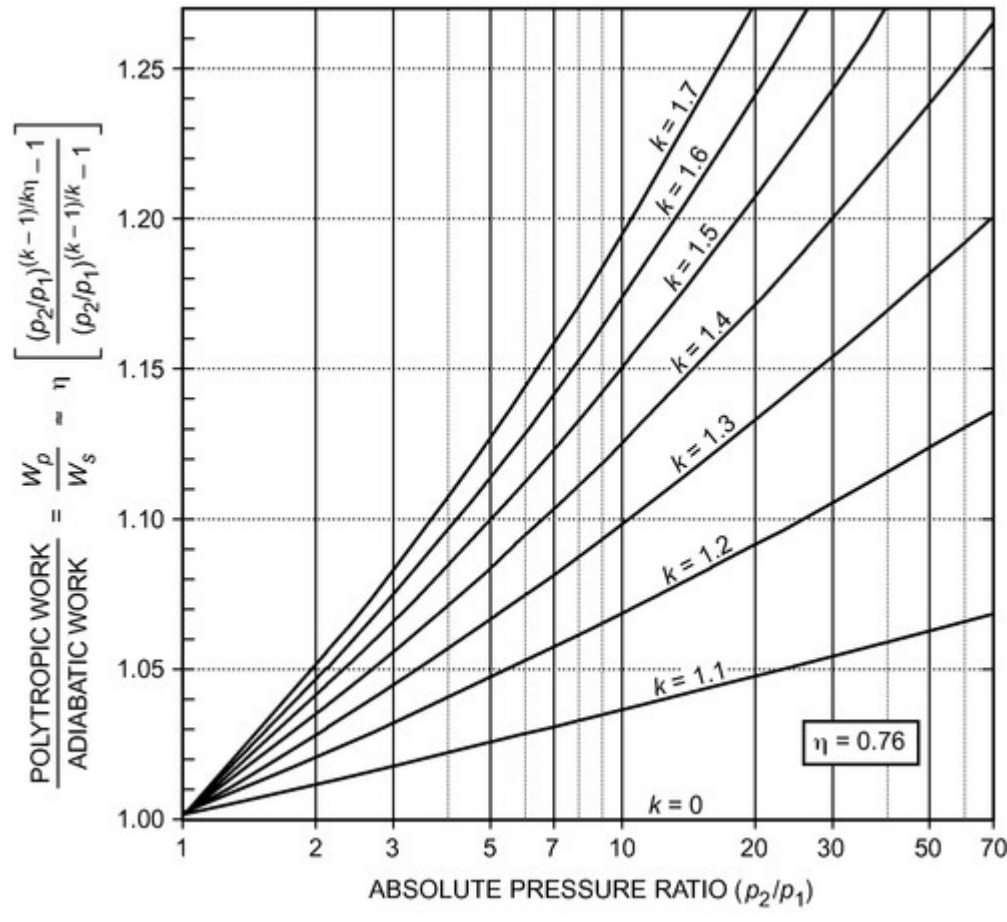


Figure 66. Ratio of Polytropic to Adiabatic Work

Nondimensional Coefficients

Some nondimensional performance parameters used to describe centrifugal compressor performance are flow coefficient, polytropic work coefficient, Mach number, and specific speed.

Flow Coefficient. Desirable impeller diameters and rotational speeds are determined from blade tip velocity by a dimensionless flow coefficient Q/ND^3 in which Q is the volumetric flow rate. Practical values for this coefficient range from 0.02 to 0.35, with good performance between 0.11 and 0.21 and optimum results between 0.15 and 0.18. Impeller diameter D and rotational speed N follow from

$$Q/ND^3 = \pi Q_t / u_t D_i^2 = \pi^3 Q_p N^2 / u_i^3$$

(43)

where u_t is tip speed.

The maximum flow coefficient in multistage compressors is found in the first stage and the minimum in the last stage (unless large side loads are involved). For high-pressure ratios, special measures may be necessary to increase the last stage (Q/ND^3) to a practical level, as stated previously. Side loads are beneficial in this respect, but interstage flash cooling is not.

Polytropic Work Coefficient. Polytropic work and polytropic head can be used interchangeably. Because this chapter is concerned with polytropic work, that term will be used exclusively. Polytropic head is related to it by $H_p = W_p/\rho g$. A greater restriction on stage work capability is often imposed by the impeller Mach number M . For adequate performance, M must be limited to about 1.8 for stages with impellers overhung from the ends of shafts and to about 1.5 for impellers with shafts passing through their inlets because flow passage geometries are moved out. For good performance, these values must be even lower. Such considerations limit maximum stage work to about $1.5a^2$, where a is the acoustic velocity at the stage inlet.

Besides the power requirement, polytropic work also determines impeller blade tip speed and number of stages. For an individual stage, stage work is related to speed by

$$W_{pi} = \mu_i u_i^2 / g_c$$

(44)

where μ_i is the stage work coefficient.

The overall polytropic work is the sum of the stage works:

$$W_p = \sum W_{pi}$$

(45)

and the overall work coefficient is

$$\mu = g_c W_p / \sum u_i^2$$

(46)

Values for μ (and μ_i) range from about 0.42 to about 0.74, with 0.55 representative for estimating purposes. Compressors designed for modest work coefficients have backward-curved impeller blades. These impellers tend to have greater part-load ranges and higher efficiencies than do radial-bladed designs.

Maximum tip speeds are limited by strength considerations to about 1400 fps. For cost and reliability, 980 fps is a more common limitation. On this basis, the maximum polytropic work capability of a typical stage is about 15,000 ft·lb/lb.

A greater restriction on stage work capability is often imposed by the impeller Mach number M . For adequate performance, M must be limited to about 1.8 for stages with impellers overhung from the ends of shafts and to about 1.5 for impellers with shafts passing through their inlets because flow passage geometries are moved out. For good performance, these values must be even lower. Such considerations limit maximum stage work to about $1.5a^2$, where a is the acoustic velocity at the stage inlet.

A compressor user with a particular installation in mind may prefer more explicit curves, such as pressure ratio and power versus volumetric flow at constant rotational speed. Plots of this sort may require fixed suction conditions to be entirely accurate, especially if discharge pressure and power are plotted against mass flow or refrigeration effect.

Testing

When a centrifugal compressor is tested, overall μ and η versus Q/ND^3 at constant M are plotted. They are useful because test results with one gas are sometimes converted to field performance with another. When side flows and cooling are involved, the overall work coefficient is found from [Equations \(45\) and \(46\)](#) by evaluating mixing and cooling effects between stages separately. The **overall efficiency** in such cases is

$$\eta = \frac{\sum w_i W_{pi}}{\sum w_i W_i}$$

(47)

Testing with a fluid other than the design refrigerant is a common practice known as **equivalent performance testing**. Its need arises from the impracticability of providing test facilities for the complete range of refrigerants and input power for which centrifugal compressors are designed. Equivalent testing is possible because a given compressor produces the same μ and η at the same (Q/ND^3) and M with any fluids whose volume ratios (v_2/v_1) and Reynolds numbers are the same. Thermodynamic performance of a compressor can be evaluated according to either the stagnation or static properties of the refrigerant, and it is important to distinguish between these concepts. The **stagnation efficiency**, for example, may be higher than the **static efficiency**. The safest procedure is to use static properties and evaluate kinetic energy changes separately.

4.5 OPERATION AND MAINTENANCE

Refer to the compressor manufacturer's operating and maintenance instructions for recommended procedures. A planned maintenance program, as described in [Chapter 39 of the 2015 ASHRAE Handbook—HVAC Applications](#), should be established. As part of this program, operating documentation should be kept, tabulating pertinent unit temperatures, pressures, flows, fluid levels, electrical data, and refrigerant added. ASHRAE *Guideline* 4 has further information on documentation. These can be compared periodically with values recorded for the new unit. Gradual changes in data can signify the need for routine maintenance; abrupt changes indicate system or component difficulty. A successful maintenance program requires the operating engineer to recognize and identify the reason for these data trends. In addition, by knowing the component parts and their operational interaction, the designer can use these symptoms to prescribe proper maintenance procedures.

The following items deserve attention in establishing a planned compressor maintenance program:

- A tight system is important. Leaks on compressors operating at subatmospheric pressures allow noncondensables and moisture to enter the system, adversely affecting operation and component life. Leakage in higher-pressure systems allows oil and refrigerant loss. ASHRAE *Guideline* 3 can be used as a guide to ensure system tightness. Vacuum leaks can be detected by a change in operational pressures not supported by corresponding refrigerant temperature data or the frequency of purge unit operation. Pressure leaks are characterized by symptoms related to refrigerant charge loss such as low suction pressures and high suction superheat. Such leaks should be located and fixed to prevent component deterioration.
- Compliance with the manufacturer's recommended oil filter inspection and replacement schedule allows visual indication of the compressor lubrication system condition. Repetitive clogging of filters can mean system contamination. Periodic oil sample analysis can monitor acid, moisture, and particulate levels to assist in problem detection.
- Operating and safety controls should be checked periodically and calibrated to ensure reliability.
- Electrical resistance of hermetic motor windings between phases and to ground should be checked (megged) regularly, following the manufacturer's outlined procedure. This helps detect any internal electrical insulation deterioration or the formation of electrical leakage paths before a failure occurs.
- Water-cooled oil coolers should be systematically cleaned on the water side (depending on water conditions), and operation of any automatic water control valves should be checked.
- For some compressors, periodic maintenance (e.g., manual lubrication of couplings and other external components, shaft seal replacement) is required. Prime movers and their associated auxiliaries all require routine maintenance. Such items should be made part of the planned compressor maintenance schedule.
- Periodic vibration analysis can locate and identify trouble (e.g., unbalance, misalignment, bent shaft, worn or defective bearings, bad gears, mechanical looseness, electrical unbalance). Without disassembling the machine, such trouble can be found early, before machinery failure or damage can occur. Dynamic balancing can restore rotating equipment to its original efficient, quiet operating mode. Such testing can help avoid costly emergency repairs, pinpoint irregularities before major problems arise, and increase the useful life of components.
- The necessary steps for preparing the unit for prolonged shutdown (i.e., winter) and specified instruction for starting after this standby period, should both be part of the program. With compressors that have internal lubrication systems, provisions should be made to have their oil heaters energized continuously throughout this period or to have their oil charges replaced before putting them back into operation.

4.6 SYMBOLS

a = acoustic velocity at a particular point
 A = area of cylinder [Equation \(11\)](#), in²
 a_s = acoustic velocity at impeller inlet
 c = flow velocity
 c_p = specific heat at constant pressure
CR = compression ratio
 c_u = tangential component of refrigerant velocity leaving impeller blades, fps
 D = tangential component of refrigerant velocity leaving impeller blades, fps
 f = polytropic work factor
 g_c = gravitation constant
 h_c = enthalpy at a specific state point
 H = cylinder block height
 J = mechanical equivalent of heat, 778.1 ft·lb/Btu
 k = ratio of specific heats, [Equation \(41\)](#); isentropic coefficient for refrigerant used, from refrigerant tables [e.g., Lemmon et al. (2002)]
 m = exponent in [Equation \(28\)](#), defined in [Equation \(30\)](#)
 $\dot{m}$ = ideal mass flow of compressed gas, lb/h
 M = flow Mach number, [Equation \(17\)](#)
 M_1 = first-stage impeller Mach number
 M_i = flow Mach number impeller, [Equation \(19\)](#)
 n = exponent, [Equation \(29\)](#)
 N = rotational speed
 N_c = compressor speed, rpm
 N_1 = motor speed, rpm
 N_s = specific speed
 p = pressure at a specific state point
 P = gas compression power
 p_1 = pressure on other side of piston, psi
 p_2 = discharge pressure, psi
 p_d = expected discharge pressure (absolute)
 P_e = power input to motor
 P_{sh} = power input to shaft
 $p^{(p)}$ = pressure raised to power of m
 P_p = ideal power input, Btu/h
 P_v = power required for isentropic compression
PR = pressure ratio = p_d/p_s
 p_s = expected suction pressure (absolute)
 Q = volumetric flow rate; actual capacity, Btu/h
 Q/ND^3 = dimensionless flow coefficient
 Q_i = volumetric flow rate in impeller
 Q_p = ideal capacity, Btu/h
 s = stroke of compressor, in.
 R = gas constant
 T = absolute temperature at specific state point
 T_s = starting torque, lb·in.
 u_i = impeller tip speed
 v = specific volume
 V_d = geometric displacement of compressor, ft³/h
 V^p = volume raised to power of n
 $\dot{w}$ = mass flow
 W = total work input
 w_p = work delivered to gas within the compression volume
 w_e = work input to motor, Btu/lb
 W_i = impeller work input per unit mass of refrigerant, ft·lb/lb_m
 w_{pi} = work input to compressor shaft, Btu/lb
 w_{ps} = specific work of isentropic compression, Btu/lb
 W_p = polytropic work input
 W_{pi} = polytropic work by impeller
 W_s = adiabatic work input
 X = compressibility function, [Equation \(32\)](#)
 Y = compressibility function, [Equation \(33\)](#)
 Z = compressibility function, [Equation \(34\)](#)
 η = polytropic efficiency
 η_{comp} = total compressor efficiency
 η_e = motor efficiency
 η_i = isentropic efficiency
 η_m = mechanical efficiency
 η_{pi} = compression isentropic efficiency
 η_s = adiabatic efficiency
 η_v = volumetric efficiency
 μ = overall work coefficient
 μ_i = stage work coefficient
 ρ_s = density of gas entering compressor (at suction port), lb/ft³
 Θ = flow parameter, [Equation \(24\)](#)
 μ = overall work coefficient
 Ω = head parameter, [Equation \(20\)](#)

REFERENCES

AHRI. 2004. Performance rating of positive displacement condensing units. ANSI/AHRI Standard 520-2004. Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA.

AHRI. 2004. Performance rating of positive displacement refrigerant compressors and compressor units. *ANSI/AHRI Standard 540-2004*. Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA.

AHRI. 2008. Method of measuring machinery sound within an equipment space. *ANSI/AHRI Standard 575-2008*. Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA.

ASHRAE. 2008. Preparation of operating and maintenance documentation for building systems. *Guideline 4-2008*.

ASHRAE. 2010. Safety standard for refrigeration systems. *ANSI/ASHRAE Standard 15-2010*.

ASHRAE. 2005. Methods of testing for rating positive displacement refrigerant compressors and condensing units. *ANSI/ASHRAE Standard 23-2005*.

ASHRAE. 2010. Mechanical refrigeration and air-conditioning installations aboard ship. *ANSI/ASHRAE Standard 26-2010*.

ASHRAE. 2002. Reducing the release of halogenated refrigerants from refrigeration and air-conditioning equipment and systems. *ANSI/ASHRAE Standard 147-2002*.

Blazer, W.E., Jr. 1972. Chiller noise: Its impact on building design. *ASHRAE Transactions* 78(1):268.

Brasz, J.J. 1995. Improving the refrigeration cycle with turbo-expanders. *Proceedings of the 19th International Congress of Refrigeration*.

Bush, J., and J. Elson. 1988. Scroll compressor design criteria for residential air conditioning and heat pump applications. *Proceedings of the 1988 International Compressor Engineering Conference*, vol. 1, pp. 83-97. Office of Publications, Purdue University, West Lafayette, IN.

Caillat, J., R. Weatherston, and J. Bush. 1988. Scroll-type machine with axially compliant mounting. U.S. *Patent 4,767,292*.

CEN. 2005. Refrigerant compressors—Rating conditions, tolerances and presentation of manufacturer's performance data. *EN Standard 12900*. 2005. European Committee for Standardization, Brussels.

Davis, H., H. Kottbus, and A.M.G. Moody. 1951. The influence of Reynolds number on the performance of turbomachinery. *ASME Transactions* (July):499.

Edminster, W.C. 1961. *Applied hydrocarbon thermodynamics*, 22 and 52. Gulf Publishing, Houston.

Elson, J., G. Hundy, and K. Monnier. 1990. Scroll compressor design and application characteristics for air conditioning, heat pump, and refrigeration applications. *Proceedings of the Institute of Refrigeration* (London) 2:1-2.10 (November).

Elson, J.P., N. Kaemmer, S. Wang, and N. Perevozchikov. 2008. Scroll technology: An overview of past, present and future developments. *Paper C1204. Proceedings of the 2008 International Compressor Engineering Conference at Purdue*. Office of Publications, Purdue University, West Lafayette, IN.

Hoover, R.M. 1960. Noise levels due to a centrifugal compressor installed in an office building penthouses. *Noise Control* (6):136.

Hougen, O.A., K.M. Watson, and R.A. Ragatz. 1959. *Chemical process principles, Part II—Thermodynamics*. John Wiley & Sons, New York.

Lemmon, E.W., M.O. McLinden, and M.L. Huber. 2002. *NIST reference fluid thermodynamic and transport properties—REFPROP*, version 7.0. Standard Reference Data Program. National Institute of Standards and Technology, Gaithersburg, MD.

McCullough, J. 1975. Positive fluid displacement apparatus. U.S. *Patent 3,924,977*.

McCullough, J., and R. Shaffer. 1976. Axial compliance means with radial sealing for scroll type apparatus. U.S. *Patent 3,994,636*.

Mitchell, J.S. 1977. Monitoring machinery health. *Power* 121(1):46; (II):87; (III):38.

Morishita, E., Y. Kitora, T. Suganami, S. Yamamoto, and M. Nishida. 1988. Rotating scroll vacuum pump. *Proceedings of the 1988 International Compressor Engineering Conference* (July). Office of Publications, Purdue University, West Lafayette, IN.

Purvis, E. 1987. Scroll compressor technology. Heat Pump Conference, New Orleans.

Saids, J. 1983. Involute and laminated tip seal of labyrinth type for use in a scroll machine. U.S. *Patent 4,411,605*.

Schaffer, M.E. 2005. A practical guide to noise and vibration control for HVAC systems, 2nd ed. ASHRAE.

Schultz, I.M. 1962. The polytropic analysis of centrifugal compressors. *ASME Transactions* (January):69 and (April):222.

Sheets, H.E. 1952. Nondimensional compressor performance for a range of Mach numbers and molecular weights. *ASME Transactions* (January):93.

Tojo, K., T. Hosoda, M. Ikegawa, and M. Shibayashi. 1982. Scroll compressor provided with means for pressing an orbiting scroll member against a stationary scroll member and self-cooling means. U.S. *Patent 4,365,941*.

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