CHAPTER 34

COMPRESSORS

THE COMPRESSOR is one of the four essential components of the compression refrigeration system; the others include the condenser, evaporator, and expansion device. The compressor circulates refrigerant through the system in a continuous cycle.

There are two basic types of compressors: positive displacement and dynamic. **Positive-displacement compressors** increase the pressure of refrigerant vapor by reducing the volume of the compression chamber through work applied to the compressor's mechanism: reciprocating, rotary (rolling piston, rotary vane, single-screw, and twin-screw), scroll, and trochoidal.

Dynamic compressors increase the pressure of refrigerant vapor by a continuous transfer of angular momentum from the rotating member to the vapor followed by the conversion of this momentum into a pressure rise. Centrifugal compressors function based on these principles.

This chapter describes the design features of several categories of commercially available refrigerant compressors.

POSITIVE-DISPLACEMENT COMPRESSORS

PERFORMANCE

Compressor performance is the result of design compromises involving physical limitations of the refrigerant, compressor, and motor, while attempting to provide the following:

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

Two useful measures of compressor performance are the coefficient of performance (COP) and the measure of power required per unit of refrigerating capacity [brake power input/refrigeration output (W/W)]. The COP is a dimensionless number that is the ratio of the compressor refrigerating capacity to the input power. The COP for a hermetic compressor includes the combined operating efficiencies of the motor and the compressor:

$$\text{COP (hermetic)} = \frac{\text{Capacity, W}}{\text{Input power to motor, W}}$$

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

$$\text{COP (open)} = \frac{\text{Capacity, W}}{\text{Input power to shaft, W}}$$

Power input per unit of refrigerating capacity (W/W) is a measure of performance that is used to compare different compressors at the same operating conditions. It is primarily used with open-drive industrial equipment.

$$\frac{W_{in}}{W_{out}} = \frac{\text{Power input to shaft, W}}{\text{Compressor capacity, W}}$$

Ideal Compressor

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

$$\omega = \rho V_d \quad (1)$$

where

ω = ideal mass flow of gas compressed, kg/s
ρ = density of gas entering compressor, kg/m^3
V_d = geometric displacement of compressor, m^3/s

The ideal refrigeration cycle is addressed in Chapter 1 of the *ASHRAE Handbook—Fundamentals*; the following quantities can be determined from the pressure-enthalpy diagram in Figure 8 of that chapter:

$$Q_{refrigeration\ effect} = (h_1 - h_4)$$
$$Q_{work\ of\ compression} = (h_2 - h_1)$$

Using ω, the mass flow of gas as determined by Equation (1),

$$\text{Ideal capacity} = \omega Q_{refrigeration\ effect}$$
$$\text{Ideal power input} = \omega Q_{work}$$

Actual Compressor Performance

Ideal conditions never occur, however. Actual compressor performance deviates from ideal performance because of various losses, with a resulting decrease in capacity and an increase in power input. Depending on the type of compressor, some or all of the following factors can have a major effect on compressor performance.

The preparation of this chapter is assigned to TC 8.1, Positive Displacement Compressors, and TC 8.2, Centrifugal Machines.

1. **Pressure drops within the compressor unit**
   - Through shutoff valves (suction, discharge, or both)
   - 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
2. **Heat gain to refrigerant** from
   - Hermetic motor
   - Lubricant pump
   - Friction
   - Heat of compression; heat exchange within compressor
3. **Valve inefficiencies** due to imperfect mechanical action
4. **Internal gas leakage**
5. **Oil circulation**
6. **Reexpansion.** The volume of gas remaining in the compression chamber after discharge, which reexpands into the compression chamber during the suction cycle and limits the mass of fresh gas that can be brought into the compression chamber.
7. **Deviation from isentropic compression.** When considering the ideal compressor, an isentropic compression cycle is assumed. In the actual compressor, the compression cycle deviates from isentropic compression primarily because of fluid and mechanical friction and heat transfer within the compression chamber. The actual compression process and the work of compression must be determined from measurements.
8. **Over- and undercompression.** In fixed volume ratio rotary, screw, and orbital compressors, overcompression occurs when pressure in the compression chamber reaches discharge pressure before reaching the discharge port. Undercompression occurs when the compression chamber reaches the discharge port prior to achieving discharge pressure.

Compressor Efficiency, Subcooling, and Heat Rejection

These 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 measured by the following efficiencies:

Volumetric efficiency (e_0) is the ratio of actual volume of gas entering the compressor to the geometric displacement of the compressor.

Compression efficiency (e_2) 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 the work required for isentropic compression of the gas to the work delivered to the gas within the compression volume (as obtained by measurement).

Mechanical efficiency (e_3) is the ratio of the work delivered to the gas (as obtained by measurement) to the work input to the compressor shaft.

Isentropic (reversible adiabatic) efficiency (e_4) is the ratio of the work required for isentropic compression of the gas to work input to the compressor shaft.

Actual capacity is a function of the ideal capacity and the overall volumetric efficiency e_0 of the actual compressor:

$$\text{Actual capacity} = e_0 \omega Q_{refrigeration\ effect} \tag{2}$$

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

$$P_{bp} = \frac{P_{tp} e_0}{e_2 e_3} = \frac{P_{tp} e_0}{e_4} \tag{3}$$

where

P_{bp} = actual shaft power
P_{tp} = ideal power input = ωQ_{work}
e_0 = volumetric efficiency
e_2 = compression efficiency
e_3 = mechanical efficiency
e_4 = isentropic (reversible adiabatic) efficiency

Liquid subcooling is not accomplished by the compressor. However, the effect of liquid subcooling is included in the ratings for compressors by most manufacturers.

Total heat rejection is the sum of the refrigeration effect and the heat equivalent of the power input to the compressor. The quantity of heat rejection must be known in order to size condensers.

PROTECTIVE DEVICES

Compressors are provided with one or more of the following devices for protection against abnormal conditions and to comply with various codes.

1. **High-pressure protection** as required by Underwriters Laboratories and per ARI standards and ASHRAE *Standard* 15. This may include the following:
   (a) A high-pressure cutout.
   (b) 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 the operating conditions. Care must be taken to ensure that the relief valve will not accidentally blow on a fast pull-down. Many 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.
   (c) A relief valve assembly on the oil separator of a screw compressor unit.
2. **High-temperature control** devices to protect against overheating and oil breakdown.
   (a) Motor overtemperature protective devices are addressed in the section on Integral Thermal Protection in Chapter 40.
   (b) 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).
   (c) On larger compressors, lubricant temperature is controlled by cooling with a heat exchanger or direct liquid injection, or the compressor may shut down on high lubricant temperature.
   (d) Where lubricant sump heaters are used to maintain a minimum lubricant sump temperature, a thermostat may be used to limit the maximum lubricant temperature.
3. **Low-pressure protection** may be provided for
   (a) *Suction gas.* Many compressors or systems are limited to a minimum suction pressure by a protective switch. Motor cooling, freeze-up, or pressure ratio usually determine the pressure setting.
   (b) *Compressor.* Lubricant pressure protectors are used with forced feed lubrication systems to prevent the compressor from operating with insufficient lubricant pressure.

4. **Time delay or lockouts with manual resets** prevent damage to both compressor motor and contactors from repetitive rapid-starting cycles.
5. **Low voltage and phase loss or reversal protection** is used on some systems. Phase reversal protection is used with multiphase devices to ensure the proper direction of rotation.
6. **Suction line strainer.** Most compressors are provided with a strainer at the suction inlet to remove any dirt that might exist in the 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

A gas compressor is not designed to handle liquid. The damage that may occur depends on the quantity of liquid, the frequency with which it occurs, and the type of compressor. Slugging, floodback, and flooded starts are three ways in which liquid can damage a compressor. Generally, reciprocating compressors, because they are equipped with discharge valves, are more susceptible to damage from slugging than various rotary and orbital compressors.

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 on compressor loading.

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

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

Sound Level

An acceptable sound level is a basic requirement of good design and application. Chapter 46 of the *ASHRAE Handbook—Applications* covers design criteria in more detail.

Vibration

Compressor vibration results from gas-pressure pulses and inertia forces associated with the moving parts. The problems of vibration 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 compressors, the internal compressor assembly is usually spring-mounted within the welded shell, and the entire unit is externally isolated.

Reduction of Amplitude. 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.

Chapter 46 of the *ASHRAE Handbook—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

Since 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 practice is followed is a function of 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 100 m/s^2 have been recorded; 50 m/s^2 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 fall in the range of 20 to 60 Hz with shocks to 30 m/s^2.

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

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

Testing

Testing for ratings must be in accordance with ASHRAE *Standard* 23. Compressor tests are of two types: the first determines capacity, efficiency, sound level, motor temperatures, etc.; the second predicts the reliability of the machine.

Standard rating conditions, which are usually specified by compressor manufacturers, include the maximum possible compression ratio, the maximum operating pressure differential, the maximum permissible discharge pressure, and the maximum inlet and discharge temperature.

Lubrication requirements, which are prescribed by the 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 employed.

MOTORS

Motors for positive-displacement compressors range from a few hundred watts to several megawatts. When selecting a motor for driving a compressor, the following factors should be considered:

- Power and rotational speed
- Voltage and phase
- Starting and pull-up torques
- Ambient and maximum rise temperatures
- Cost and availability
- Insulation
- Efficiency and performance
- Starting currents
- Type of protection required
- Multispeed or variable speed
- Location (high-pressure side versus low-pressure side)

Large, industrial, open compressors can be driven from 800 to 3600 rpm with three-phase, 200 to 4160 V motors. 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, wye-delta, double-delta, autotransformer, and solid-state starting methods all designed to limit inrush starting current. Care must be taken that the starting method will supply enough torque to accelerate the motor and overcome the torque required for compression.

Hermetic motors can be more highly loaded than comparable open motors because of the refrigerant lubricant cooling used.

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. The limiting factor at high loads is normally the motor temperature, while at light loads the limiting factor is the discharge gas temperature. Overdesign of the motor may increase discharge gas temperature at light loads. Consideration must be given to low-side versus high-side motor location.

The single-phase motor presents more design problems than the polyphase, because the relationship between main and auxiliary windings becomes critical together with the necessary starting equipment.

The locked-rotor rate of temperature rise 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.

The materials selected for these motors must have high dielectric strength, resist fluid and mechanical abrasion, and be compatible with an atmosphere of refrigerant and lubricant.

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

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
- 2 speed, pole switching variable speed

Central air conditioning and commercial refrigeration
- Single-phase, PSC and CSCR to 4500 W
- 3 phase, 1500 W and above, across-the-line start
- 7500 W and above; part winding; wye-delta, double-delta, and across-the-line start
- 2 speed, pole switching variable speed
- Electronically commutated dc motors (ECMs)

For further information on motors and motor protection, see Chapter 40.

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.

Single-stage compressors are primarily used for medium temperatures (–20 to 0°C) and in air-conditioning applications but can achieve temperatures below –35°C at 35°C condensing temperatures with suitable refrigerants. Chapters 2 and 3 of the *ASHRAE Handbook—Refrigeration* have information on other halocarbon and ammonia systems.

Booster compressors are typically used for low-temperature applications with R-22 or ammonia. Minus 65°C saturated suction can be achieved by using R-22, and –54°C saturated suction is possible using ammonia.

The booster raises the refrigerant pressure to a level where further compression can be achieved with a high-stage compressor, without exceeding the compression-ratio limits of the respective machines.

Since superheat is generated as a result of compression in the booster, intercooling is normally required to reduce the refrigerant stream temperature to the practical level required at the inlet to the high-stage unit. Intercooling methods include controlled liquid injection into the intermediate stream, gas bubbling through a liquid reservoir, and use of a liquid-to-gas heat exchanger where no fluid mixing occurs.

Integral two-stage compressors achieve low temperatures (–30 to –60°C), using R-22 or ammonia within the frame of a single compressor. The cylinders within 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 the high-pressure suction and low-pressure discharge stages allow an interstage gas cooling system to be connected to remove superheat between stages. This interconnection is similar to the methods used for individual high-stage and booster compressors.

Capacity reduction is typically 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.

The most widely used compressor is the halocarbon compressor, which is manufactured in three types of design: (1) open, (2) semi-hermetic or bolted hermetic, and (3) welded-shell hermetic.

Ammonia compressors are manufactured only in the open design because of the incompatibility of the refrigerant and hermetic motor materials.

Open-type compressors are those in which the shaft extends through a seal in the crankcase for an external drive.

Hermetic compressors are those in which the motor and compressor are contained within the same housing, with the motor shaft integral with the compressor crankshaft and the motor in contact with the refrigerant.

A **semihermetic compressor** (bolted, accessible, or serviceable) is a hermetic compressor of bolted construction amenable to field repair.

In **welded-shell hermetic compressors** (sealed) the motor-compressor is mounted inside a steel shell, which, in turn is sealed by welding.

Combinations of design features used are shown in Table 1. Typical performance values for halocarbon compressors are given in Table 2.

Performance Data

Figure 1 presents a typical set of capacity and power curves for a four-cylinder semihermetic compressor, 60.3 mm bore, 44.4 mm stroke, 1740 rpm, operating with Refrigerant 22. Figure 2 shows the heat rejection curves for the same compressor. Compressor curves should contain the following information:

- Compressor identification
- Degrees of subcooling and correction factors for zero or other subcool 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

Table 1 Typical Design Features of Reciprocating Compressors

Item	Halocarbon Compressor Open	Halocarbon Compressor Semi-hermetic	Halocarbon Compressor Welded Hermetic	Ammonia Compressor Open
1. Number of cylinders—one to:	16	12	6	16
2. Power range	120 W and up	0.4 to 110 kW	0.1 to 20 kW	7.5 kW 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. Hermetic compressors, electric motor		X	X	
b. Open compressors—direct drive, V belt chain, gear, by electric motor or engine	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 head	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 speeds (50 to 60 Hz)	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

Table 2 Typical Performance Values for Reciprocating Compressors

Compressor Size and Type	R-404a: Evap. Temp. = –40°C, Cond. Temp. = 41°C, Suction Gas = 18.3°C, Subcooling = 0 K	R-134a: Evap. Temp. = –17.8°C, Cond. Temp. = 43°C, Suction Gas = 18.3°C, Subcooling = 0 K	R-22: Evap. Temp. = 4.4°C, Cond. Temp. = 41°C, Suction Gas = 12.8°C, Subcooling = 0 K	R-22: Evap. Temp. = 7.2°C, Cond. Temp. = 54.4°C, Suction Gas = 18.3°C, Subcooling = 0 K
	Refrigerant Output/Power Input, W/W			
Large, over 19 kW				
Open	0.99	1.89	4.95	5.04
Hermetic	0.92	1.76	4.16	3.05
Medium, 4 to 19 kW				
Open	0.90	1.74	4.71	4.71
Hermetic	0.85	1.64	4.10	3.02
Small, under 4 kW				
Open	—	—	—	—
Hermetic	—	1.11	4.04	2.93

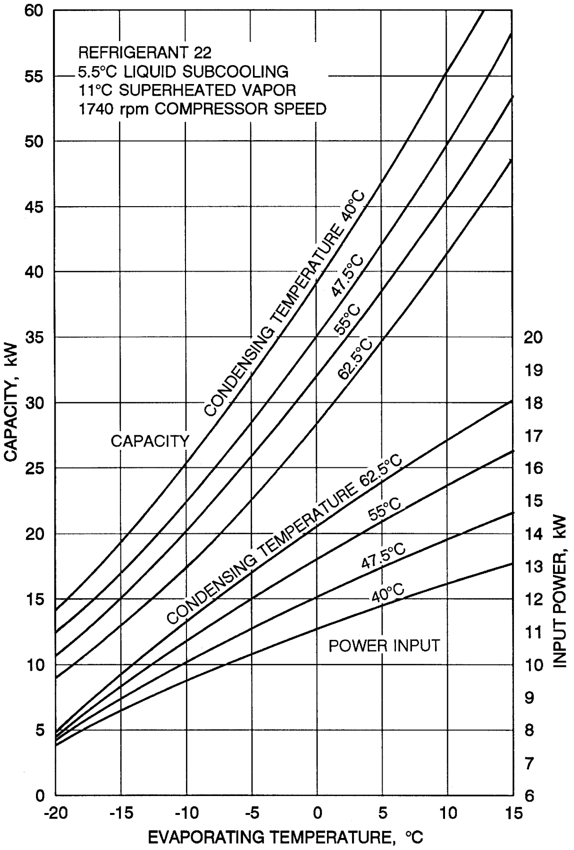


Fig. 1 Capacity and Power-Input Curves for Typical Hermetic Reciprocating Compressor

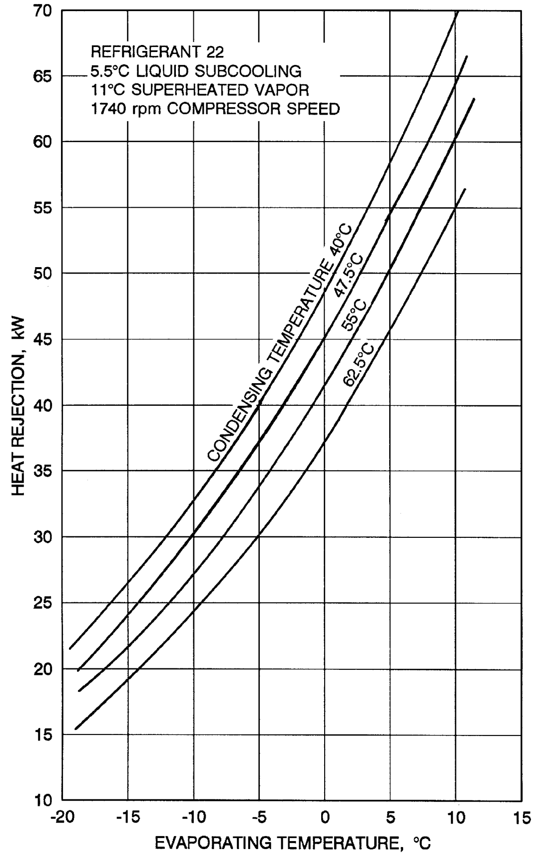


Fig. 2 Heat Rejection Curves for Typical Hermetic Reciprocating Compressor

Motor Performance

The motor efficiency is usually the result of a compromise between cost and size. Generally, the physically larger a motor is for a given rating, the more efficient it can be made. Accepted efficiencies range from approximately 88% for a 2 kW motor to 95% for a 75 kW motor. 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 and evenly spaced impulses also help reduce noise and vibration.

Since many compressors start against load, it is desirable to estimate starting torque. The following equation is for a single cylinder compressor. It neglects friction, the additional torque required to force discharge gas out of the cylinder, and the fact that the tangential force at the crankpin is not always equal to the 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 yields a conservative estimate.

$$T_s = \frac{(p_2 - p_1)As}{2N_2/N_1} \tag{4}$$

where

T_s = starting torque, N·m
p_2 = discharge pressure, Pa
p_1 = suction pressure, Pa
A = area of cylinder, m^2
s = stroke of compressor, m
N_2 = motor speed, rpm
N_1 = compressor speed, rpm

Equation (4) 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, an analysis must be made of both the number of cylinders that might be on a compression stroke and the position of the rods at start. Since 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 the following table.

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

Pull-up torque is an important characteristic of motor starting strength because it represents the lowest torque capability of the motor and occurs between 25 and 75% of the operating speed. The pull-up torque of the motor must exceed the torque requirement of the compressor or the motor will cease to accelerate and it will trip the safety overload protection device.

Features

Crankcases. The crankcase, or in a welded hermetic compressor, the cylinder block, is usually of cast iron. Aluminum is also used, particularly in small open and welded hermetic compressors. Open and semihermetic crankcases enclose the running gear, oil sump, and, in the latter case, the hermetic motor. Access openings with removable covers are provided for assembly and service purposes. Welded hermetic cylinder blocks are often just skeletons, consisting of the cylinders, the main bearings, and either a barrel into which the hermetic motor stator is inserted or a surface to which the stator can be bolted.

The cylinders can be integral with the crankcase or cylinder block, in which case a material that provides a good sealing surface and resists wear must be provided. In aluminum crankcases, cast-in liners of iron or steel are usual. In large compressors, premachined cylinder sleeves inserted in the crankcase are common. With halocarbon refrigerants, excessive cylinder wear or scoring is not much of a problem and the choice of integral cylinders or inserted sleeves is often based on manufacturing considerations.

Crankshafts. Crankshafts are made of either forged steel with hardened bearing surfaces finished to 0.20 μm or iron castings. Grade 25 to 40 (170 to 280 MPa) tensile gray iron can be used where the lower modulus of elasticity can be tolerated. Nodular iron shafts approach the stiffness, strength, and ductility of steel and should be polished in both directions of rotation to 0.40 μm maximum for best results. Crankshafts often include counterweights and should be dynamically and/or statically balanced.

While a safe maximum stress is important in shaft design, it is equally important to prevent excessive deflection that can edge-load bearings to failure. In hermetics, deflection can permit motor air gap to become eccentric, which affects starting, reduces efficiency, produces noise, and further increases bearing edge-loading.

Generally, the harder the bearing material used, the harder the shaft. With bronze bearings, a journal hardness of 350 Brinell is usual, while unhardened shafts at 200 Brinell in babbitt bearings are typical. Many combinations of materials and hardnesses have been used successfully.

Main Bearings. Both the crank and drive means may be overhung with the bearings between; however, usual practice places the cylinders between the main bearings and, in a hermetic, overhangs the motor. Main bearings are made of steel-backed babbitt, steel-backed or solid bronze, or aluminum. In an aluminum crankcase, the bearings are usually integral. 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 quite possible. For conventional shaft diameters and speeds, 4 MPa main bearing loading based on projected area is not unusual. Running clearances average 1 mm/m 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 the venting of evolved refrigerant gas and 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 and Eccentric Straps. 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 such bearing material as steel-backed babbitt or bronze, while aluminum and bronze rods can bear directly on the crankpin and piston pin. Refrigerant compressor service limits unit loading to 20 MPa 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 14 MPa has been used. Large end unit loads are usually under 7 MPa.

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 0.4 mm/m 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 2 mm/m 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 0.3 to 1.0 μm range will give good ring seating. An effective oil scraper can often be obtained with a sharp corner on the piston skirt.

The 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 2 mm/m 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 0.2 μm 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 6 m/s, determined by multiplying twice the stroke in metres by the revolutions per second, has been used successfully.

Suction and Discharge Valves. The most important components in the reciprocating compressor are the suction and discharge valves. Successful designs provide long life and low pressure loss. The life of a properly made and correctly applied valve is determined by the motion and stress it undergoes in performing its function. Excessive pressure loss across the valve results from high gas velocities, poor mechanical action, or both.

For design purposes, gas velocity is defined as being equal to the bore area multiplied by the average piston speed and divided by the valve area. Permissible gas velocity through the restricted areas of the valve is left to the discretion of the designer and depends on the level of volumetric efficiency and performance desired. In general, designs with velocities up to 60 m/s with ammonia and up to 45 m/s with R-22 have been successful.

A valve should meet the following requirements:

- Large flow areas with shortest possible path
- Straight gas flow path, minimum directional changes
- Low valve mass combined with low lift for quick action
- Symmetry of design with minimum pressure imbalance
- Minimum clearance volume
- Durability
- Low cost
- Tight sealing at ports
- Minimum valve flutter

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

1. **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 is a valve that can stand considerable abuse.

2. **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; the resulting stresses are far greater than those calculated from the curvature of the stop. Considerable care must be taken in the design to ensure reliability.
3. **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.
4. **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 larger, 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 and oil pumping; this is not a problem in package equipment but may be in remote systems where gas lines are long.

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 that quieter operation results. Since 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, the compressor may be quieter in operation.

Gear pumps are used to a large extent. Spur gears are simple but tend to promote flashing of the 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-type eccentric gear or vane pumps where a gradual opening of the suction volume takes place. The eccentric gear pump, the vane pump, or the 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 minimum clearances to pump a mixture of gas and oil. The discharge of the pump 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 fine 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 the suction gas into the compressor crankcase. A flow of gas from piston leakage opposes this oil flow, so the velocity of the leakage gas 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 permit 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-type 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 100 mm 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 such that a minimum of swelling and shrinking is encountered.

Special Devices

Capacity Control. An ideal capacity control system would have the following operating characteristics (not all of these benefits can occur simultaneously):

- Continuous adjustment to load
- Full-load efficiency unaffected by the control
- No loss in efficiency at part load
- Reduction of starting torque
- No reduction in compressor reliability
- No reduction of compressor operating range
- No increase in compressor vibration and sound level at part load

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, gas bypassing within the compressor, and gas bypassing outside the compressor.

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

- Possible increase in compressor vibration and sound level at unloaded conditions
- Minimum operating conditions as limited by discharge or motor temperatures (or both) at part-load conditions
- Good oil return at minimum operating conditions when fully unloaded
- Rapid cycling of unloaders
- Refrigerant feed device capable of controlling at minimum capacity

Crankcase Heaters. During shutdown, refrigerants tend to migrate to the coldest part of the refrigeration system. In cold weather, the compressor oil sump could be the coldest area. When the refrigerant charge is large enough to dilute the oil excessively and cause flooded starts, a crankcase heater should be used. The heater should maintain the oil at least 10 K 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 the other hazards caused by liquid refrigerant in gas compressors.

Application

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 the 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 through 4 of the *ASHRAE Handbook—Refrigeration* cover design. Chapter 6 of that volume gives details of cleanup in the event of a hermetic motor burnout.

Suction Superheat. No liquid refrigerant should be present in the suction gas entering the compressor because it causes oil dilution and gas formation in the lubrication system. If the liquid carryover is severe enough to reach the cylinders, excessive wear of valves, stops, pistons, and rings can occur; liquid slugging can break valves, pistons, and connecting rods.

Suction gas without superheat is not harmful to the compressor as long as no liquid refrigerant is entrained; some systems are even designed to operate this way, although suction superheat is intended in the design of most systems. Measuring suction superheat can be difficult, and the indication of a small amount does not necessarily mean that liquid is not present. An effective suction separator may be necessary to remove all liquid.

High suction superheat may result in dangerously high discharge temperatures and, in hermetics, high motor temperatures.

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 will have a negative effect on evaporator capacity and/or where lubricant return at reduced capacity is marginal.

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

Parallel operation of compressors in a single system has some operational advantage at part load. Careful attention must be given to apportioning returned oil to the multiple compressors so that each always has an adequate quantity. Figure 3 shows the method most widely used. Line A connects the tops of the crankcases and tends to equalize the pressure above the oil, while line B permits oil equalization at the normal level. Lines of generous size must be used. Generally, line A is a large diameter, while line B is a small diameter, which limits the possible blowing of oil from one crankcase to the other.

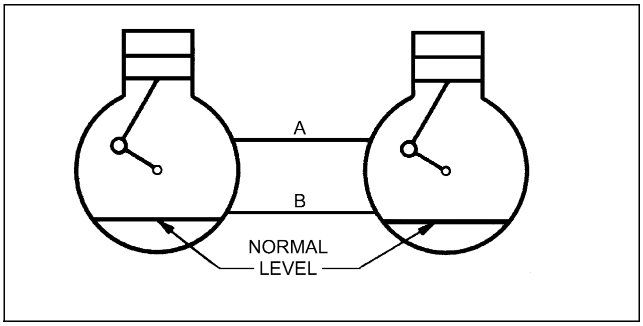


Fig. 3 Modified Oil-Equalizing System

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, the use of pumps in each oil line is necessary to maintain adequate crankcase oil level. In both cases, proper gas equalization must be provided.

ROTARY COMPRESSORS

ROLLING PISTON COMPRESSORS

Rolling piston, or fixed vane, rotary compressors are used in household refrigerators and air-conditioning units in sizes up to about 2 kW (Figure 4). 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.

Displacement for this compressor can be calculated from

$$V_d = \pi H(A^2 - B^2)/4 \qquad (5)$$

where

V_d = displacement
H = cylinder block height
A = cylinder diameter
B = roller diameter

The drive motor stator and compressor are solidly mounted in the compressor housing. This design feature is possible due to low vibration associated with the rotary compression process, as opposed to reciprocating designs which employ spring isolation between the compressor parts and the housing.

Suction gas is directly piped into the suction port of the compressor, and the compressed gas is discharged into the compressor housing shell. This high-side shell design is used because of the simplicity of its lubrication system and the absence of oiling and compressor cooling problems. Compressor performance is also improved because this arrangement minimizes heat transfer to the suction gas and reduces gas leakage area.

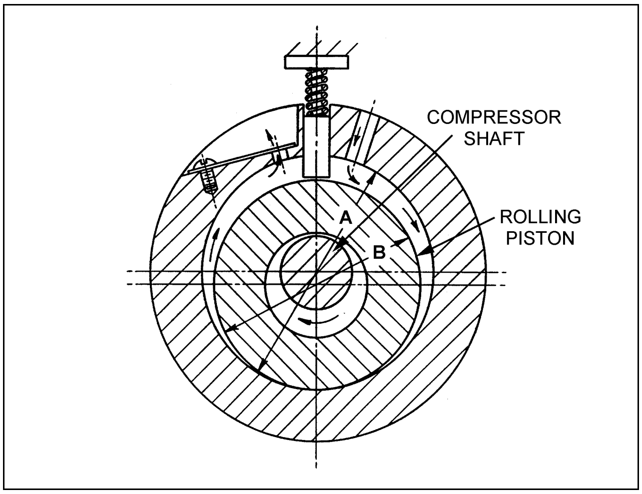


Fig. 4 Fixed Vane, Rolling Piston Rotary Compressor

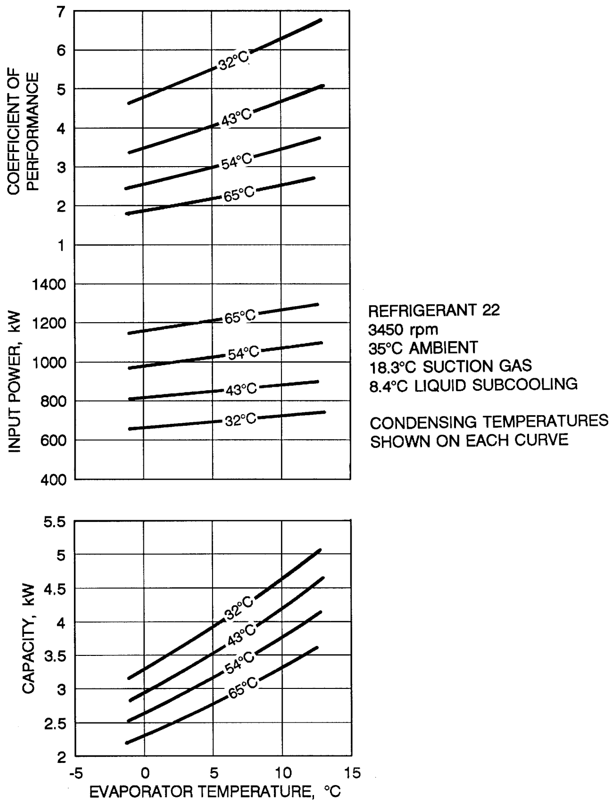


Fig. 5 Performance Curves for Typical Rolling-Piston Compressor

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 5 and Table 3 show performance of a typical rolling piston rotary compressor, commercially available for room air-conditioning and small, packaged heat pump applications.

An acceptable sound level is important in the design of any small compressor. Figure 6 illustrates a convenient method of analyzing and evaluating sound output of a rotary compressor designed for the home refrigerator. Since gas flow is continuous and no suction valve is required, rotary compressors can be relatively quiet. The sound level for a rotary compressor of the same design is directly related to its power input.

Table 3 Typical Rolling Piston Compressor Performance

Compressor speed	3450 rpm
Refrigerant	R-22
Condensing temperature	55°C
Liquid refrigerant temperature	46°C
Evaporator temperature	7°C
Suction pressure	625 kPa
Suction gas temperature	18°C
Evaporator capacity	3.5 kW
Coefficient of performance	3.22
Input power	1090 W

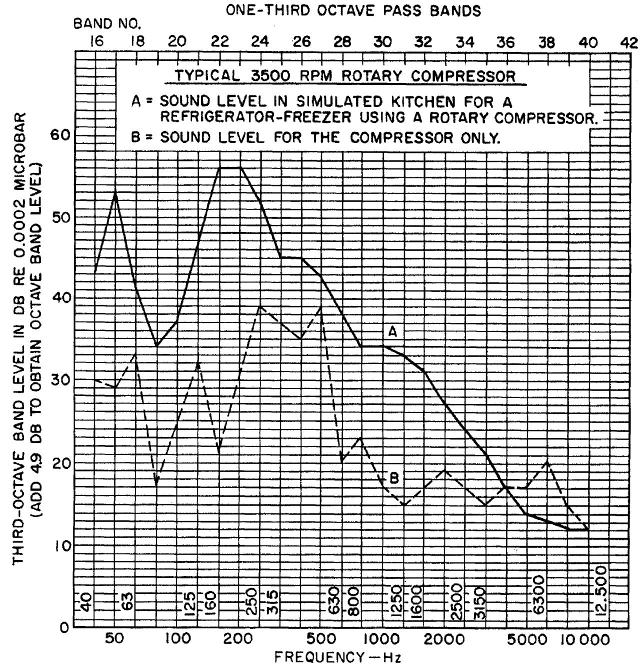


Fig. 6 Sound Level of Combination Refrigerator-Freezer with Typical Rotary Compressor

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 allow 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 shafts are generally made from steel forging and nodular cast iron. Depending on the materials chosen, a relative hardness should be maintained between the bearing and the journal. Journals are ground round to high precision and polished to a finish of 250 µm or better.

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

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. Powder metal vanes have a particular hardness typically in the 60 to 80 Rockwell C (R_c) range, and apparent hardness in the 35 to 55 R_c range.

To obtain good sealing and proper lubrication, the shape of the vane tip must conform to the surface of generation along the roller wall in accordance with hydrodynamic theory.

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.

Valves. Only discharge valves are required by rolling piston rotary compressors. They are usually simple reed valves made of high-grade steel.

Lubrication. A good lubrication system circulates an ample supply of clean oil to all working surfaces, bearings, blades, blade slots, and seal faces. High-side pressure in the housing shell ensures a sufficient pressure differential across the bearings; passageways 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 190 to 200 N·m per litre of compressor displacement per revolution. Similarly, larger compressors using R-22 for window air conditioners require about 350 to 360 N·m breakdown torque per litre of compressor displacement per revolution.

Rotary machines do not usually require complete unloading for successful starting. 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.

ROTARY VANE COMPRESSORS

Rotary vane compressors have a low mass-to-displacement ratio, which, in combination with compact size, makes them suitable for transport application. Small compressors in the 2 to 40 kW range are single-staged for a saturated suction temperature range of −40 to 7°C at saturated condensing temperature up to 60°C. By employing a second stage, low-temperature applications down to −50°C are possible. Currently, R-22, R-404a, and R-717 refrigerants are used.

Figure 7 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. While conventional valves are not required for this compressor, suction and discharge check valves are recommended to prevent reverse rotation and oil logging during shutdown.

Design of the compressor results in a fixed built-in volume ratio. Compressor volume 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. The internal compression ratio can be calculated from the following relationship:

$$p_i = V_i^k \qquad (6)$$

where

p_i = internal compression ratio
V_i = compression volume ratio
k = c_p/c_v = isentropic specific heat ratio for refrigerant being compressed

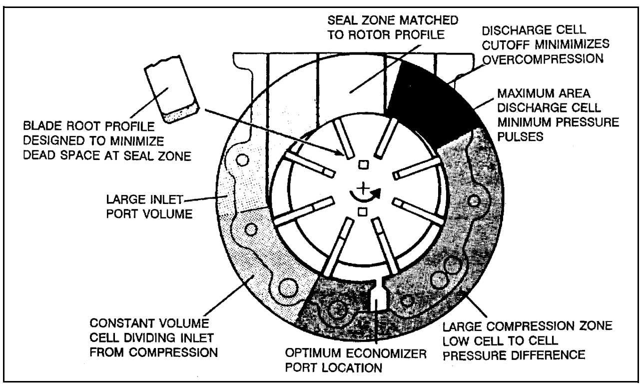


Fig. 7 Rotary Vane Compressor

The compressors currently available are of an oil-flooded, open-drive design, which requires use of an oil separator. Single-stage separators are used in close-coupled, high-temperature, direct-expansion systems, where oil return is not a problem. Two-stage separators with a coalescing-type second stage are used in low-temperature systems, in ammonia systems, and in flooded evaporators likely to trap oil.

SINGLE-SCREW COMPRESSORS

Screw compressors currently in production for refrigeration and air-conditioning applications comprise 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. The capacity range currently available is from 70 to 4600 kW.

Description

The single-screw compressor consists of a single cylindrical main rotor that works with a pair of gate rotors. Both the main rotor and gate rotors can vary widely in terms of form and mutual geometry. Figure 8 shows the design normally encountered in refrigeration.

The main rotor has helical grooves, with a cylindrical periphery and a globoid (or hourglass shape) 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. Two diametrically opposed discharge ports use a common discharge manifold located in the casing.

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 the 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. With reference to Figure 9, the process is as follows:

Suction. During rotation of the main rotor, a typical groove in open communication with the suction chamber gradually fills with

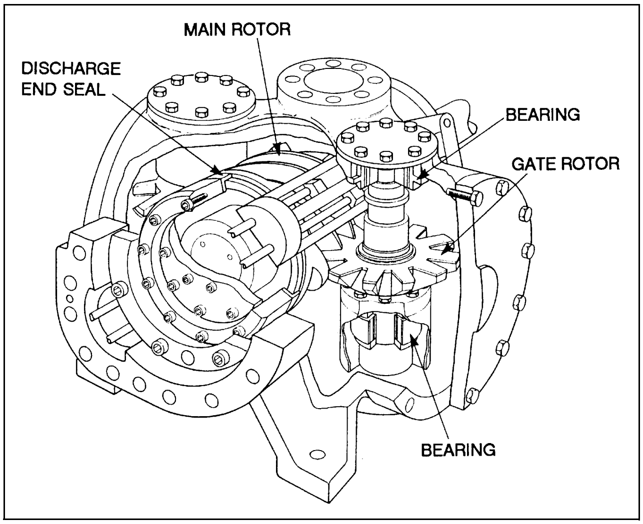
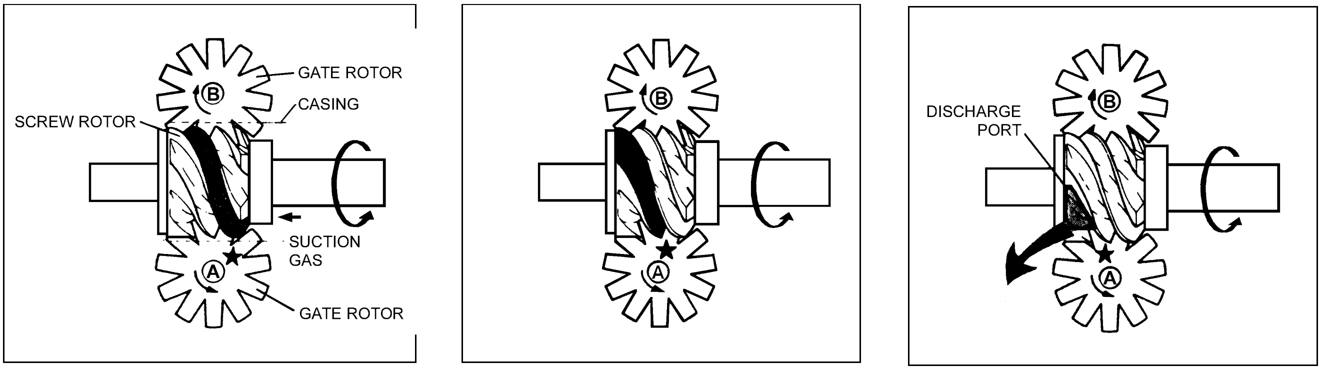


Fig. 8 Section of Single-Screw Refrigeration Compressor



Suction. During rotation of the main rotor, a typical groove in open communication with the suction chamber gradually fills with suction gas. The tooth of the gate rotor in mesh with the groove acts as an aspirating piston.

Compression. As the main rotor turns, the groove engages a tooth on the gate rotor and is covered simultaneously by the cylindrical main rotor casing. The gas is trapped in the space formed by the three sides of the groove, the casing, and the gate rotor tooth. As rotation continues, the groove volume decreases and compression occurs.

Discharge. At the geometrically fixed point where the leading edge of the groove and the edge of the discharge port coincide, compression ceases, and the gas discharges into the delivery line until the groove volume has been reduced to zero.

Fig. 9 Sequence of Compression Process in Single-Screw Compressor

suction gas. The tooth of the gate rotor in mesh with the groove acts as an aspirating piston.

Compression. As the main rotor turns, the groove engages a tooth on the gate rotor and is covered simultaneously by the cylindrical main rotor casing. The gas is trapped in the space formed by the three sides of the groove, the casing, and the gate rotor tooth. As rotation continues, the groove volume decreases and compression occurs.

Discharge. At the geometrically fixed point where the leading edge of the groove and the edge of the discharge port coincide, compression ceases, and the gas discharges into the delivery line until the groove volume has been reduced to zero.

Mechanical Features

Rotors. The screw rotor is normally made of cast iron, and the mating gate rotors are made from an engineered plastic. The inherent lubricating quality of the plastic, as well as 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 be true idlers by allowing them to dampen out transients 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 semihermetics). On the opposite side of the screw, one roller bearing is used.

Note that the compression process 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 10. 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 10).

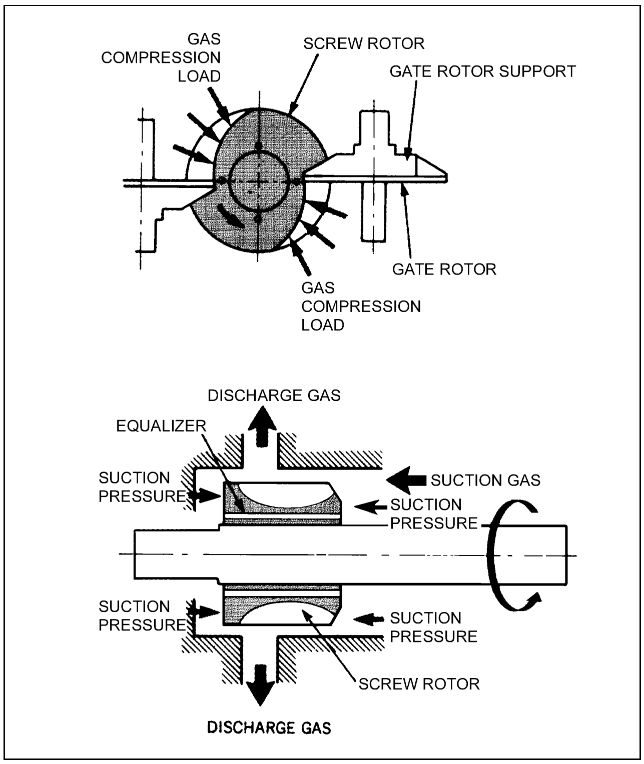


Fig. 10 Radial and Axially Balanced Main Rotor

The gate rotor bearing must overcome a small moment force due to 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. Since the single-screw compressor's physical geometry places no constraints on bearing size, lives of 200 000 h are typical.

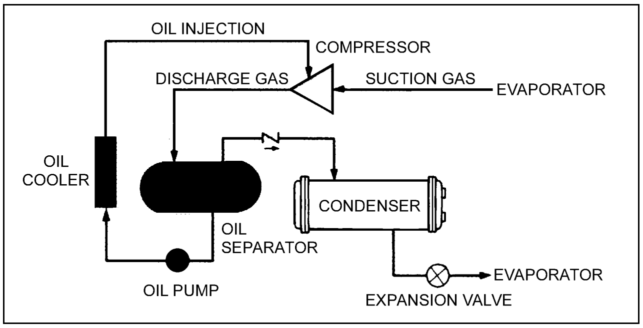


Fig. 11 Oil and Refrigerant Schematic of Oil Injection System

Cooling, Sealing, and Bearing Lubrication. A major function of injecting a fluid into the compression area is the removal of the heat of compression. Also, since the 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 allow the use of 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 reducing noise. In addition, the compressor can handle some liquid floodback because it tolerates oil.

Oil-injected single-screw compressors operate at high pressures using common high-pressure refrigerants such as R-22, R-134a, and R-717. They also operate effectively at high pressure ratios because the injected oil cools the compression process. Currently, compressors with capacities in the 15 to 1100 kW range are manufactured.

Oil injection requires an oil separator to remove the oil from the high-pressure refrigerant (see Figure 11). For those applications with exacting demands for low oil carryover, separation equipment is available to leave less than 5 mg/kg 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. A continuously running oil pump is used in some compressors to generate oil pressure 200 to 300 kPa over compressor discharge pressure. This pump requires 0.3 to 1.0% of the compressor's motor power.

Methods of oil cooling 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 0.8 kW for compressors up to 750 kW), which can result in energy savings over refrigerant injected into the compression chamber.
- **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 package unit, (3) condenser water on a package unit, (4) water from an evaporative condenser sump, (5) forced air-cooled oil cooler, and (6) high-pressure liquid recirculation (thermosiphon).

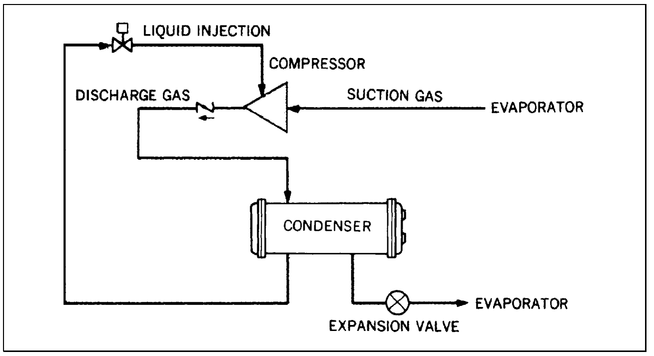


Fig. 12 Schematic of Oil-Injection-Free Circuit

The heat added to the oil during compression is the amount usually removed in the oil cooler.

Oil-Injection-Free Compressors. While the single-screw compressor operates well with oil injection, it also operates with equal or better efficiency in an oil-injection-free (OIF) mode with many common halocarbon refrigerants. This means that the 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 is replaced by liquid refrigerant. No 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%, depending on compressor design). A typical OIF circuit is depicted in Figure 12.

Oil-injection-free operation has the following advantages:

- It requires no discharge oil separator.
- Semihermetic compressors require no oil or refrigerant pumps.
- External coolers are not required.
- The compressor tolerates liquid refrigerant entering the suction and significantly reduces the size requirements of direct-expansion evaporators.

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, improves compressor-useful refrigeration and compressor efficiency (Figure 13).

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 the additional charge are then compressed together to discharge conditions. The pumping capacity of the compressor at suction conditions is not 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 H of the liquid entering the evaporator. Furthermore, the only additional mass flow the compressor must handle is the flash gas entering a closed flute, which is above suction pressure. Thus, under most conditions, the capacity improvement is accompanied by an efficiency improvement (Figure 13). Economizers become effective when the pressure ratio is 3.5 and above.

Figure 14 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 the pressure in the

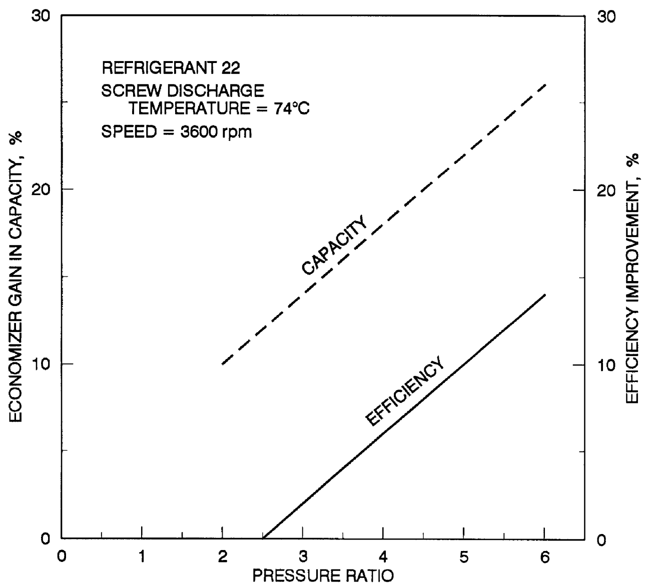


Fig. 13 Typical Improvement in Efficiency and Capacity with Economizer

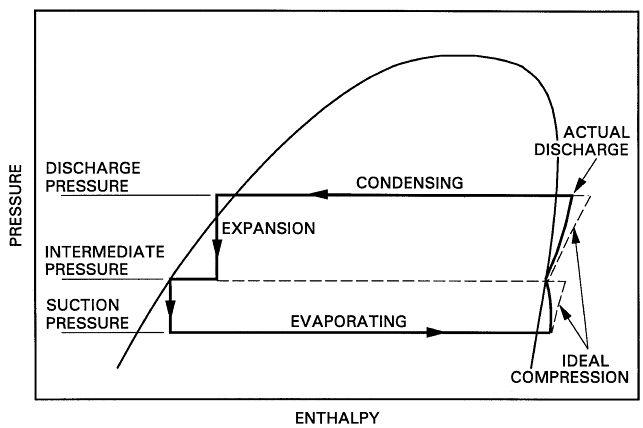


Fig. 14 Theoretical Economizer Cycle

compressor's closed flute (closed from suction). The 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 kilogram. In addition, the percentage increase in power input is lower than the percentage capacity increase.

As a screw compressor is unloaded, the 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.

Centrifugal Economizer. Some single-screw compressor designs incorporate a patented centrifugal economizer. The centrifugal economizer replaces the force of gravity in a flash-type economizer with centrifugal force to separate the flash gas generated at an intermediate pressure from the liquid refrigerant prior to liquid entering 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.

The separation is achieved by a centrifugal impeller mounted on the compressor shaft (see Figure 22); a special valve maintains a uniformly thick liquid ring around the circumference of the impeller, assuring that no gas leaves with the liquid going to the evaporator. The flash gas is then ducted to a closed groove in the compression cycle. Some designs use the flash gas with a similar liquid refrigerant to cool the motor prior to introduction into the closed compression groove.

Volume Ratio. The degree of compression within 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 ratio V_i, 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, the 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. Although such losses cause no harm to the compressor, they increase power consumption and noise and reduce efficiency.

Volume ratio selection should be made according to operating conditions. The built-in pressure ratio of a screw compressor is a function of the volume ratio:

$$p_i = V_i^k \tag{7}$$

where k is the isentropic exponent (specific heat ratio) for the refrigerant being used [see Equation (6)].

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 ease of interchangeability both during construction and after installation (although partial disassembly is required).

Single-screw compressors in refrigeration and process applications are being equipped with a simple slide valve to vary the volume ratio of the compressor while the compressor is running. The slide valve advances or delays the discharge port opening. Note that a separate slide has been designed to modulate the capacity independently of the volume ratio slide (see Figures 16, 17, and 18). Having the 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.

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 causes a reduction in groove volume, and hence in compressor throughput. As the 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 at the same time as the bypass slot is created (Figure 15) or having one valve control capacity only and a second valve independently modulate volume ratio (Figures 16, 17, and 18). A full modulating mechanism is provided in most large single-screw compressors, while two-position slide valves are used where requirements allow. The specific part-load performance will be affected by a compressor's built-in volume ratio V_i, 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 15). This mechanism is usually operated by a hydraulic or gas piston and cylinder assembly located within the compressor itself or by a positioning motor. The piston is actuated either 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 17 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 18 depicts the same system as shown in Figure 17, 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.

The single-screw compressor's two compression chambers, each having 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. Operation in this manner (asymmetrical) realizes an improvement in part-load efficiency below the 50% capacity point 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).

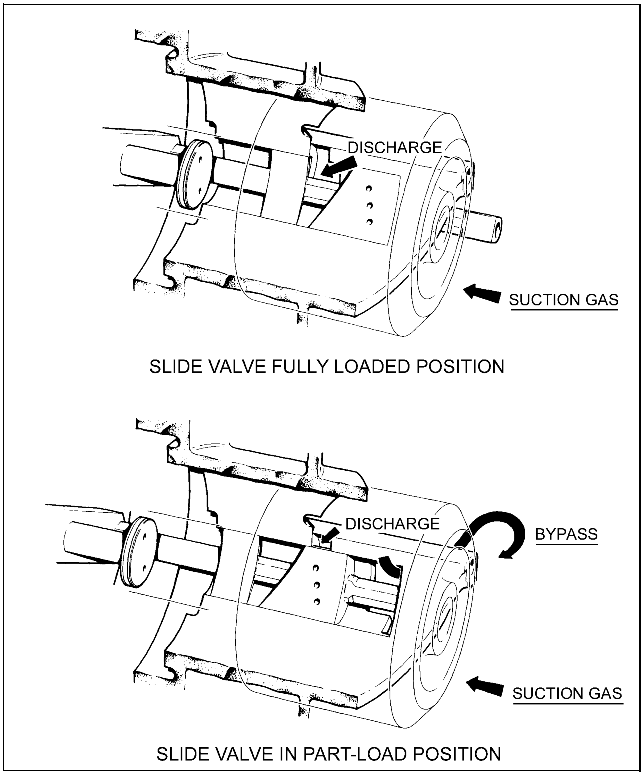


Fig. 15 Capacity Control Slide Valve Operation

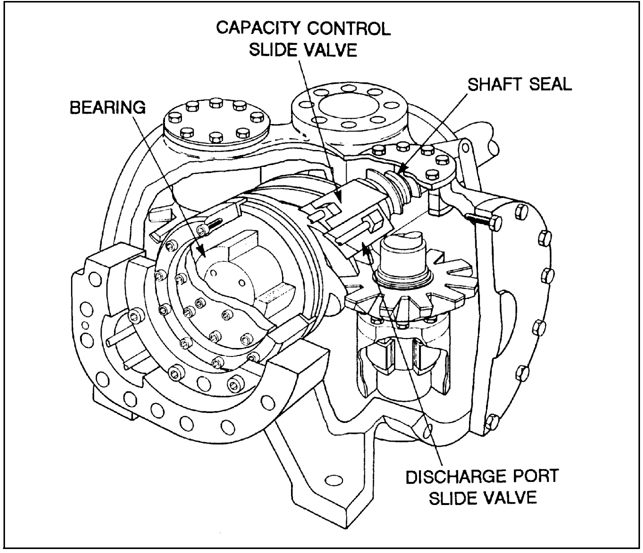


Fig. 16 Refrigeration Compressor Equipped with Variable Capacity Slide Valve and Variable Volume Ratio Slide Valve

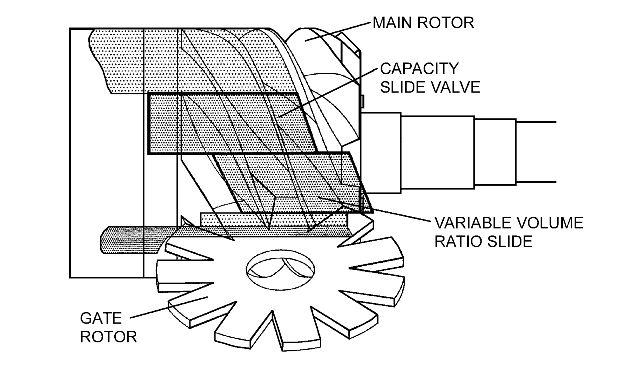


Fig. 17 Capacity Slide in Full-Load Position and Volume Ratio Slide in Intermediate Position

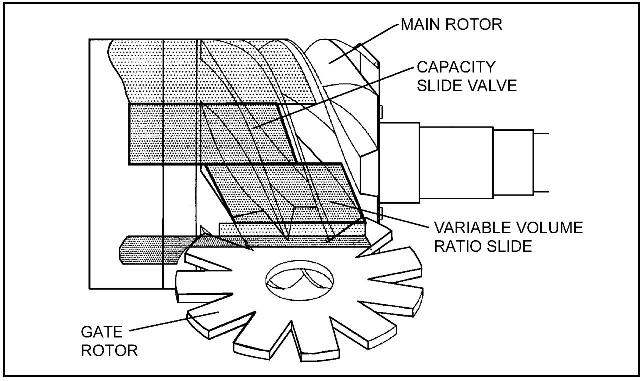


Fig. 18 Capacity Slide in Part-Load Position and Volume Ratio Slide Positioned to Maintain System Volume Ratio

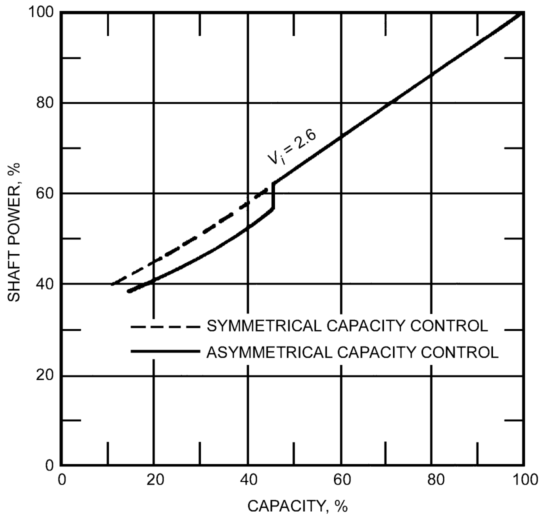


Fig. 19 Part-Load Effect of Symmetrical and Asymmetrical Capacity Control

Figure 19 demonstrates the effect of asymmetrical capacity control of a single-screw compressor.

Performance. Figures 20 and 21 show typical efficiencies of all single-screw compressor designs. High isentropic and volumetric efficiencies are the result of 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.

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

Applications. Single-screw compressors have been widely applied as refrigeration compressors, using halocarbon refrigerants, ammonia, and hydrocarbon refrigerants. A single gate rotor semihermetic version is increasingly being applied 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 140 to 1800 kW.

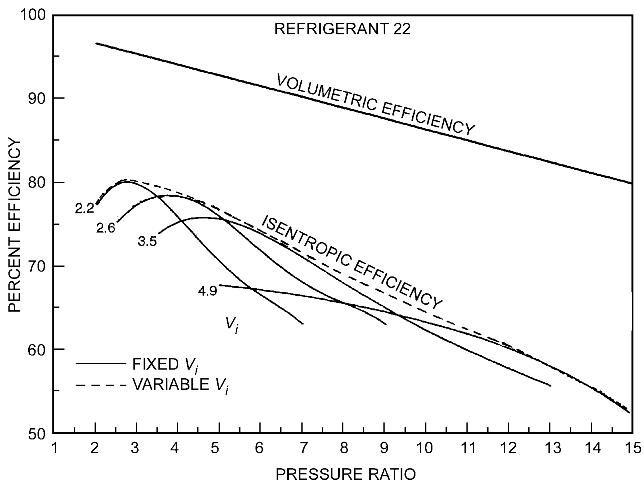


Fig. 20 Typical Compressor Performance on R-22

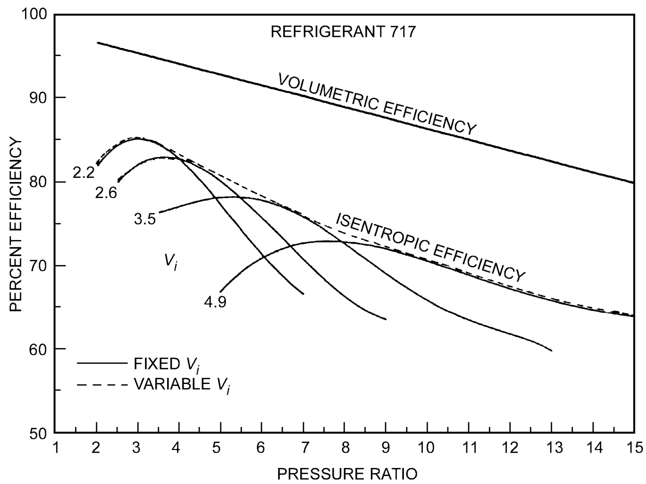


Fig. 21 Typical Compressor Performance on R-717 (Ammonia)

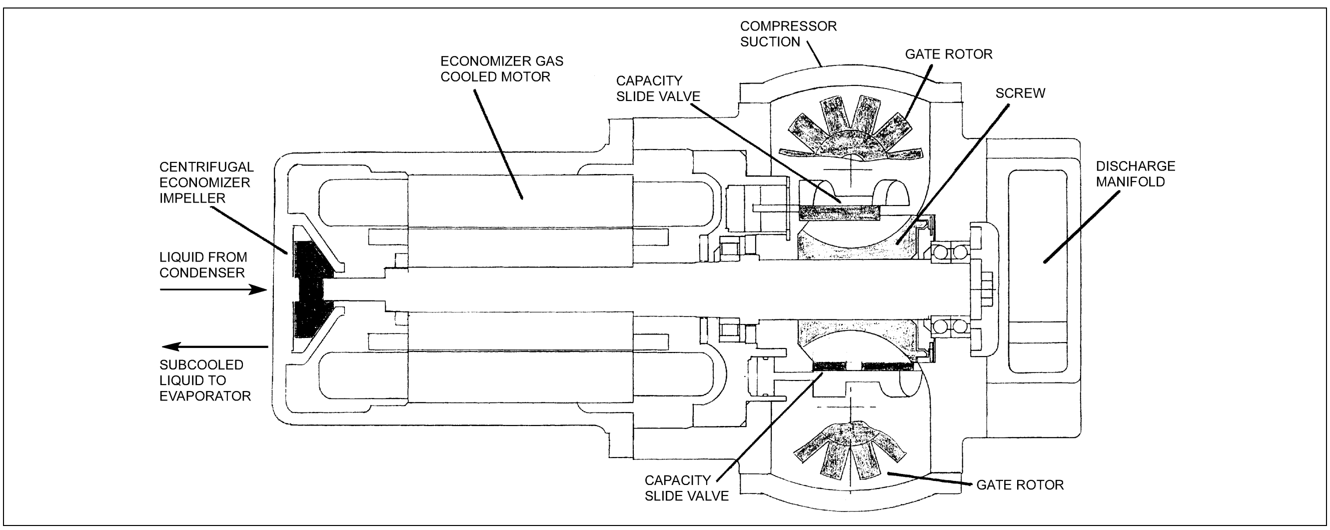


Fig. 22 Typical Semihermetic Single-Screw Compressor

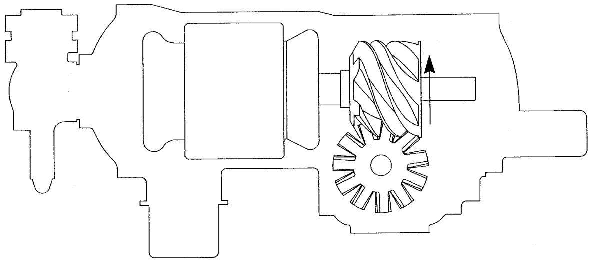


Fig. 23 Single Gate Rotor Semihermetic Single-Screw Compressor

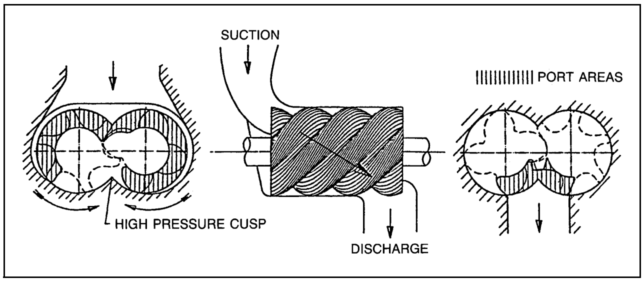


Fig. 24 Twin-Screw Compressor

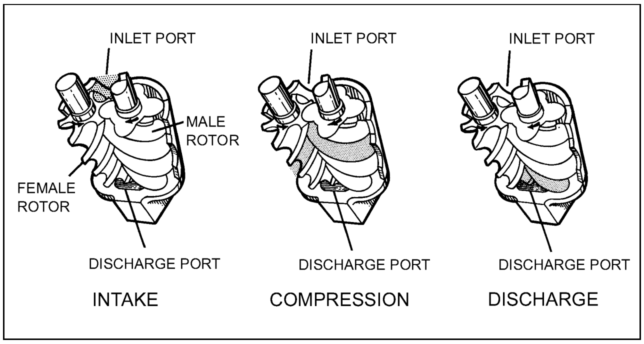


Fig. 25 Compression Process

Semihermetic Design. Figure 22 shows a semihermetic single-screw compressor. Figure 23 exhibits a semihermetic single-screw compressor using only one gate rotor. This design has found application 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 characteristics of single-screw compressors are due to small torque fluctuation and no valving required in the compression chamber. In particular, the advent of OIF technology eliminates the need for oil separators that have traditionally created noise.

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 24). The flow of gas 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.

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 often applied below 2:1 and above 20:1 compression ratios single-stage. Commercially available compressors are suitable for application on all normally used high-pressure refrigerants.

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 25).

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 in size, and gas flows continuously into the compressor. Just prior to the point at which the interlobe space leaves the inlet port, the entire length of the interlobe space is completely filled with gas.

Compression. Further rotation starts the meshing of 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 is decreased and the gas pressure consequently increased.

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. Since the intake and discharge cycles overlap effectively, a smooth continuous flow of gas results.

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 (dry compression). The symmetrical, circular rotor profile was introduced because it was easier to manufacture than the preceding profile, and it could be used without timing gears for wet or oil-flooded compression.

Current rotor profiles are normally asymmetrical and line-generated profiles, giving higher performance due to better rotor dynamics and decreased leakage area. This design allowed the possibility for female rotor drive, as well as the 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 15 to 40 m/s for wet operation (oil-flooded) and 60 to 120 m/s 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 virtually no rotor wear occurs.

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. On 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.

Male drive. The transmitted torque from male rotor to female rotor is normally 5 to 25% of input torque.

Female drive. The transmitted torque from female rotor to male rotor is normally 50 to 60% of input torque.

Rotor loads. The 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. The axial load is normally balanced with a balancing piston for larger high-pressure machines (rotor diameter above 100 mm and discharge pressure above 1100 kPa). 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 the loads. Sleeve bearings have been used historically to support radial loads in machines with male rotor diameters larger than 150 mm, while antifriction bearings were generally applied to smaller machines. However, improvements in antifriction designs and materials have led to compressors with up to 360 mm 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. 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.

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, since the bearings alone can handle the low thrust loads and still maintain good life.

High-pressure compressors have short and strong rotors (shallow grooves) and, therefore, have space for large bearings. They 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.

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) unchanged reliability. However, not all applications need ideal capacity modulation. Variable compressor displacement and variable speed are the best means for meeting these criteria. Variable compressor displacement is the most common capacity control method used. 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. They are placed within or close to the high-pressure cusp region, face one or both rotor bores, and bypass a variable portion of the trapped gas charge back to suction, depending on their position. Within this definition, there are two types of capacity slide valves.

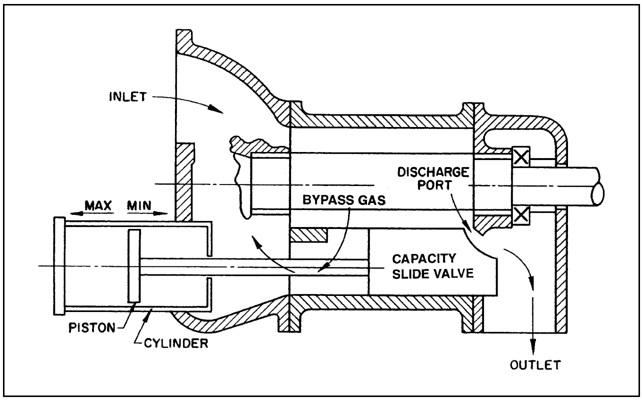


Fig. 26 Slide Valve Unloading Mechanism

1. *Capacity slide valve regulating discharge port.* This type of slide valve is located within the high-pressure cusp region. It controls 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 26 shows a schematic view of the most common arrangement.
2. *Capacity slide valve not regulating discharge port.* A slide valve outside the high-pressure cusp region controls only capacity and not the radial discharge port.

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

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 the compression space and also create leakage paths over the lobe tips. The result is somewhat lower full-load performance when 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 27).

Neither slot valves 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 the machines have the correct volume ratio for full-load operation and the compression ratio at part load does not reduce.

Volume Ratio

In all positive-displacement rotary compressors with fixed port location, the degree of compression within the rotor thread is determined by the location of the suction and discharge ports. The built-in volume ratio of screw compressors is defined as the ratio of volume of the thread at the start of the compression process to the volume of the same thread when it first begins to open to the discharge port. The suction port must be located to trap the maximum suction charge; hence, the volume ratio is determined by the location of the discharge port.

Only the suction pressure and volume ratio of the compressor determine the internal pressure achieved before opening to discharge. However, the condensing and evaporating temperatures

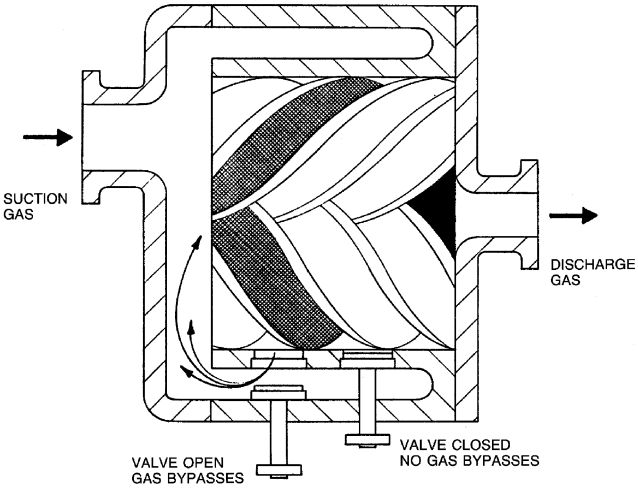


Fig. 27 Lift Valve Unloading Mechanism

determine the discharge pressure and the compression ratio in the piping that leads to the compressor. Any mismatch between the internal and system discharge pressures results in under- or overcompression loss and in 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 volume ratio for a particular application can be determined as follows.

First, determine the compression ratio of a given refrigerating system:

$$\mathrm{CR} = p_d / p_s \tag{8}$$

where

CR = compression ratio
p_s = expected suction pressure, absolute
p_d = expected discharge pressure, absolute

Then, determine the internal pressure ratio of the available compressors by approximating compression as an isentropic process as follows:

$$p_i = V_i^k \tag{9}$$

where

p_i = internal pressure ratio
V_i = compressor volume ratio
k = ratio of specific heats (isentropic exponent) for refrigerant used [see Equation (6)]

And, finally, the compressor should be selected to match as nearly as possible the internal pressure ratio of the compressor to the system compression ratio:

$$p_i = \mathrm{CR} \tag{10}$$

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 volume ratio. The difference in length basically locates the discharge port earlier or later in the compression process. Different length slide valves allow changing the volume ratio of a given compressor, although disassembly is required.

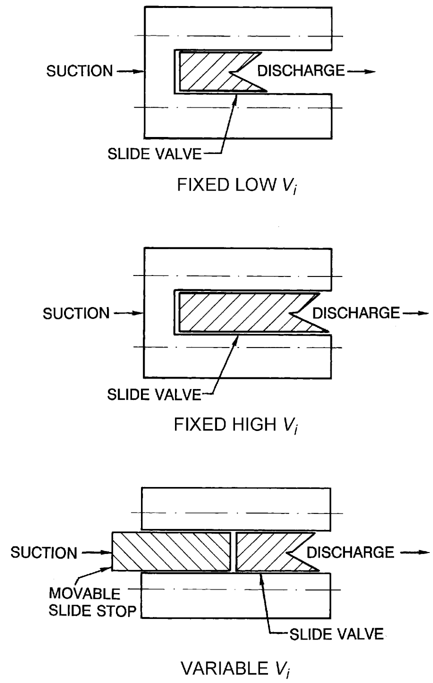


Fig. 28 View of Fixed and Variable Volume Ratio (V_i) Slide Valves from Above

Variable Volume Ratio. While operating, some twin-screw compressors adjust the volume ratio of the compressor to the most efficient ratio for whatever pressures are encountered.

In fixed volume ratio compressors, the motion of the slide valve toward the inlet end of the machine is stopped when it comes in contact with the rotor housing in that area. In the most common of the 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 28).

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 still maintaining maximum efficiency. The comparative efficiencies of fixed and variable volume ratio screw compressors are shown in Figure 29 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

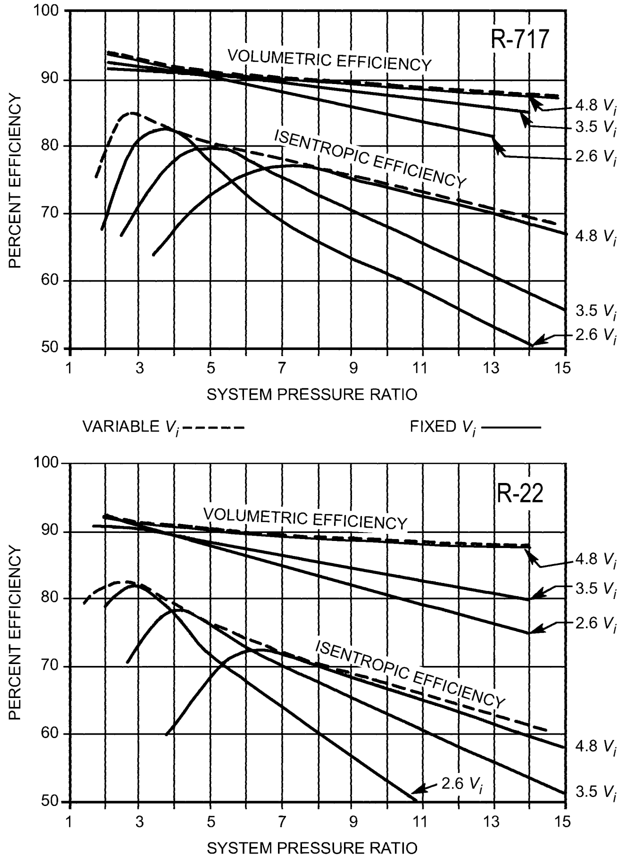


Fig. 29 Twin-Screw Compressor Efficiency Curves

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

Oil Injection

Two primary types of compressor lubrication systems are employed in twin-screw compressors—dry and oil flooded.

Dry Operation (No Rotor Contact). Since 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. Initial screw compressor designs were based on this approach, and dry screws still find application in the gas process industry.

Synchronized twin-screw compressors once required high rotor tip speed and were, therefore, 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 used in refrigeration and air conditioning. Compressor capacities range from 10 to 3000 L/s. 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 lubrication of the bearings and shaft seal prior to injection. Typically, paraffinic or naphthenic-based mineral oils are used, although synthetics are also used as required by the refrigerant or gas application. The 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 also tends to fill any leakage paths between and around the rotors. This keeps volumetric efficiency (VE) high, even at high compression ratios. Normal compressor VE exceeds 85% even at 25:1 single stage (ammonia, 190 mm rotor diameter). It also gives flat efficiency curves with decreasing speeds and quiet operation. Oil transfers much of the heat of compression from the gas to the oil, keeping the typical discharge temperature below 88°C, which allows high compression ratios without the danger of breaking down the refrigerant or the oil. The lubrication function of the oil protects bearings, seals, and the rotor contact areas.

The ability of a screw compressor to tolerate oil also permits the compressor to handle a certain amount of liquid floodback, 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. Coalescing separation equipment routinely gives less than 5 mg/kg oil in the circulated refrigerant. Some smaller compressors, primarily used on packaged units, have less efficient or no separation capability and may circulate up to 7% lubricant with the discharge gas from the compressor.

Oil injection is normally achieved by one of two methods: (l) with a continuously running oil pump capable of generating an oil pressure of 200 to 300 kPa 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 a thread during the compression process.

Since the oil absorbs a significant amount of the heat of compression in an oil-flooded operation, it must be cooled to maintain low discharge temperature. One cooling method is by direct injection of liquid refrigerant into the compression process. The amount of injected liquid refrigerant corresponds to about 0.02% of displacement volume. The amount of liquid injected is normally controlled by sensing the discharge temperature and injecting enough liquid to maintain a constant temperature. Some of the 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 of the 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 the liquid injection ports 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.

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 0.8 kW for compressors up to 750 kW).

In the third method, the oil can be cooled outside the compressor between the oil reservoir and the point of injection. Various configurations of heat exchangers are available for this purpose, and the 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 for the condenser to be reduced in size by an amount corresponding to the oil cooler capacity. Where oil cooling is carried out from 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 liquid in an external heat exchanger, (3) using chiller water on a packaged unit, (4) recirculation of 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, since the 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 thread 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. 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, a portion of the high-pressure liquid is vaporized at the side port pressure and subcools the remaining high-pressure liquid nearly to the saturation temperature at the operating side port pressure. Since this has little effect on the suction capacity of the compressor, 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 equal to about two and above (depending on volume ratio). The 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, the economizer pressure 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.

Hermetic Compressors

Hermetic screw compressors are commercially available through 700 kW of refrigeration effect using R-22. The hermetic motors can operate under discharge, suction, or intermediate pressure. Motor cooling can be with gas, oil, and/or liquid refrigerant.

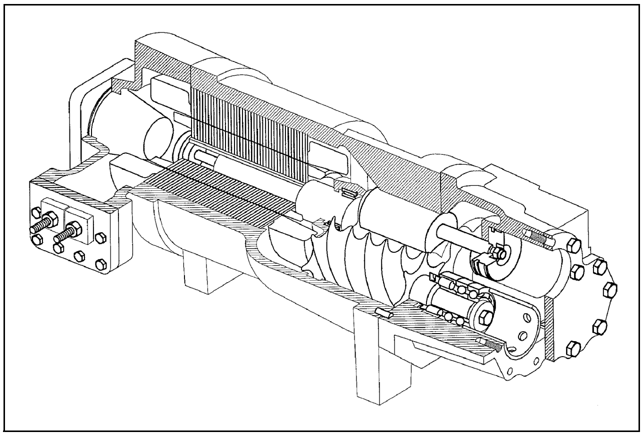


Fig. 30 Semihermetic Twin-Screw Compressor with Suction Gas-Cooled Motor

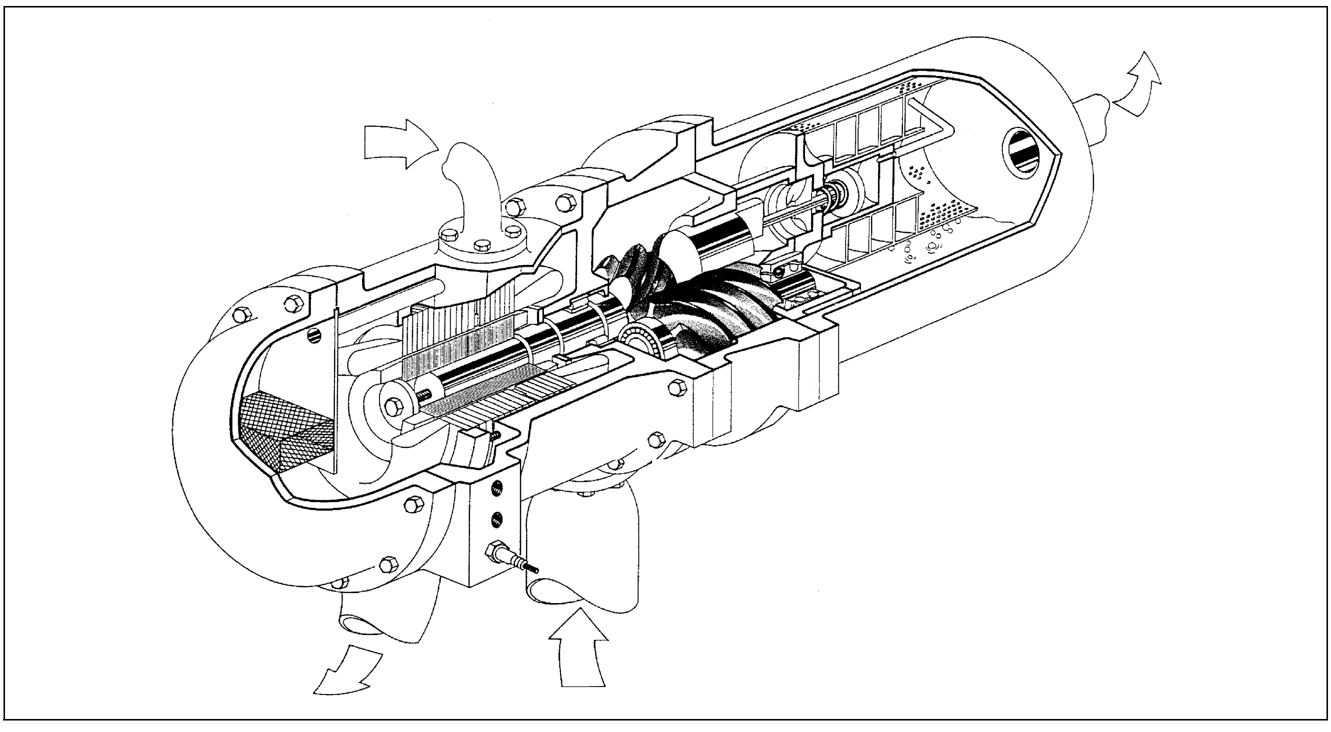


Fig. 31 Semihermetic Twin-Screw Compressor with Motor Housing Used as Economizer; Built-In Oil Separator

Oil separation for these types of compressors may be accomplished either with an integrated oil separator or with a separately mounted oil separator in the system. Figures 30, 31, and 32 show three types of hermetic twin-screw compressors.

Performance Characteristics

Figure 29 shows the full-load efficiency of a modern twin-screw compressor. Both fixed and variable volume ratio compressors without economizers are indicated. High isentropic and volumetric efficiencies are the result of internal compression, the absence of suction or discharge valves, and small clearance volume. The curves show that while 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 liquid temperature approach and pressure drop to the economizer should be specified.

ORBITAL COMPRESSORS

SCROLL COMPRESSORS

Description

Scroll compressors are orbital motion, positive-displacement machines that compress with two interfitting, spiral-shaped scroll members (Figure 33). They are currently used in residential and commercial air-conditioning, refrigeration, and heat pump applications as well as in automotive air conditioning. Capacities range from 3 to 50 kW. To function effectively, the scroll compressor requires close tolerance machining of the scroll members, which is possible due to the recent advances in manufacturing technology. This positive-displacement, rotary motion compressor includes performance features, such as high efficiency and low noise.

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 plate 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 prevented by an interconnecting coupling. An alternate approach creates relative orbital motion via two scrolls synchronously rotating about noncoincident axes. As in the former case, an interconnecting coupling maintains a relative angle between the pair of scrolls (Morishita et al. 1988).

Compression is accomplished by sealing suction gas in pockets of a given volume at the outer periphery of the scrolls and progressively reducing the size of those pockets as the scroll relative motion moves them inwards toward the discharge port. Figure 34 shows the sequence of suction, compression, and discharge phases. As the outermost pockets are sealed off (Figure 34A), the trapped gas is at suction pressure and has just entered the compression process. At stages B through F, orbiting motion moves the gas toward the center of the scroll pair, and pressure rises as pocket volumes are reduced. At stage G, the gas reaches the central discharge port and begins to exit from the scrolls. Stages A through H in Figure 34 show that two distinct compression paths operate simultaneously in a scroll set. The discharge process is nearly continuous, since new pockets reach the discharge stage very shortly after the previous discharge pockets have been evacuated.

Scroll compression embodies a fixed, built-in volume ratio that is defined by the geometry of the scrolls and by discharge port location. This feature provides the scroll compressor with different performance characteristics than those of reciprocating or conventional rotary compressors.

Both high-side and low-side shells are available. In the former, the entire compressor is at discharge pressure, except for the outer areas of the scroll set. Suction gas is introduced into the suction port of the scrolls through piping, which keeps it discrete from the rest of the compressor. Discharge gas is directed into the compressor shell, which acts as a plenum. In the low-side type, most of the shell is at suction pressure, and the discharge gas exiting from the scrolls is routed outside the shell, sometimes through a discrete or integral plenum.

Mechanical Features

Scroll Members. Gas sealing is critical to the performance advantage of scroll compressors. Sealing within the scroll set must

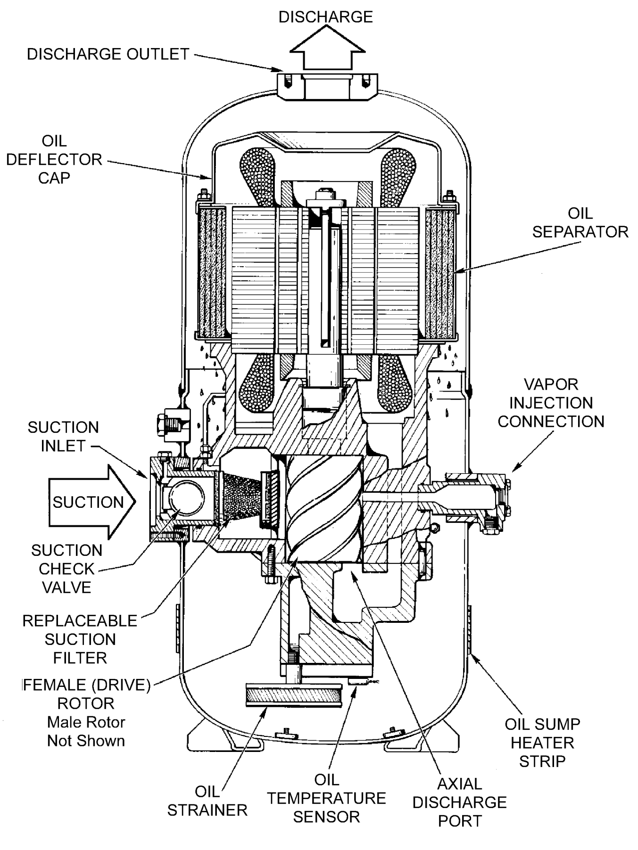


Fig. 32 Vertical, Discharge-Cooled, Semihermetic Twin-Screw Compressor

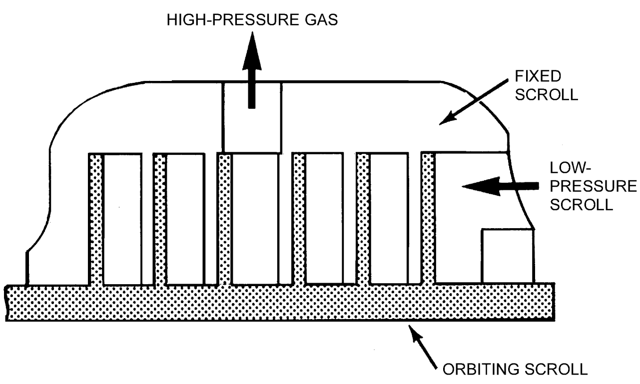


Fig. 33 Interfitted Scroll Members
(Purvis 1987)

be accomplished at the flank contact locations and between the vane tips and bases of the intermeshed scroll pair. Tip/base sealing is generally considered more critical than flank sealing. The method used to seal the scroll members tends to separate scroll compressors into compliant and noncompliant designs.

Noncompliant Designs. In designs lacking compliance, the orbiting scroll takes a fixed orbital path. In the radial direction, sealing small irregularities between the vane flanks (due to flank machining variation) can be accomplished with oil flooding. In the axial direction, the position of both scrolls remains fixed, and flexible seals fitted into machined grooves on the tips of both scrolls accomplish tip sealing. The seals are pressure loaded to enhance uniform contact (McCullough and Shaffer 1976, Sauls 1983).

Radial Compliance. This feature enhances flank sealing and allows the orbiting scroll to follow a flexible path defined by its own contact with the fixed scroll. In one type of radial compliance, a sliding "unloader" bushing is fitted onto the crankshaft eccentric pin in such a way that it directs the radial motion of the orbiting scroll. The orbiting scroll is mounted over this bushing through a drive bearing, and the scroll may now move radially in and out to accommodate variations in orbit radius caused by machining and assembly discrepancies. This feature tends to keep the flanks constantly in contact, and reduces impact on the flanks that can result from intermittent contact. Sufficient clearance in the pin/unloader assembly allows the scroll flanks to separate fully when desired.

In some designs, the mass of the orbiting scroll is selected so that centrifugal force overcomes radial gas compression forces that would otherwise keep the flanks separated.

In some other designs, the drive is designed so that the influence of centrifugal force is reduced, and the drive force overcomes the radial gas compression force (McCullough 1975). Radial compliance has the added benefit of increasing resistance to slugging and contaminants, since the orbiting scroll can "unload" to some extent as it encounters obstacles or nonuniform hydraulic pressures (Bush and Elson 1988).

Axial Compliance. With this feature, an adjustable axial pressure maintains sealing contact between the scroll tips and bases while running. This pressure is released when the unit is shut down, allowing the compressor to start unloaded and to approach full operational speed before a significant load is encountered. This scheme obviates the use of tip seals, eliminating them as a potential source of wear and leakage. With the scroll tips bearing directly on the opposite base plates and with suitable lubrication, sealing tends to improve over time. Axial compliance can either be implemented on the orbiting scroll or the fixed scroll (Tojo et al. 1982, Caillat et al. 1988). The use of axial compliance requires auxiliary sealing of the discharge side with respect to the suction side of the compressor.

Antirotation Coupling. To ensure relative orbital motion, the orbiting scroll must not rotate in response to gas loading. This rotation is most commonly accomplished by an Oldham coupling mechanism, which physically connects the scrolls and permits all planar motion, except relative rotation, between them.

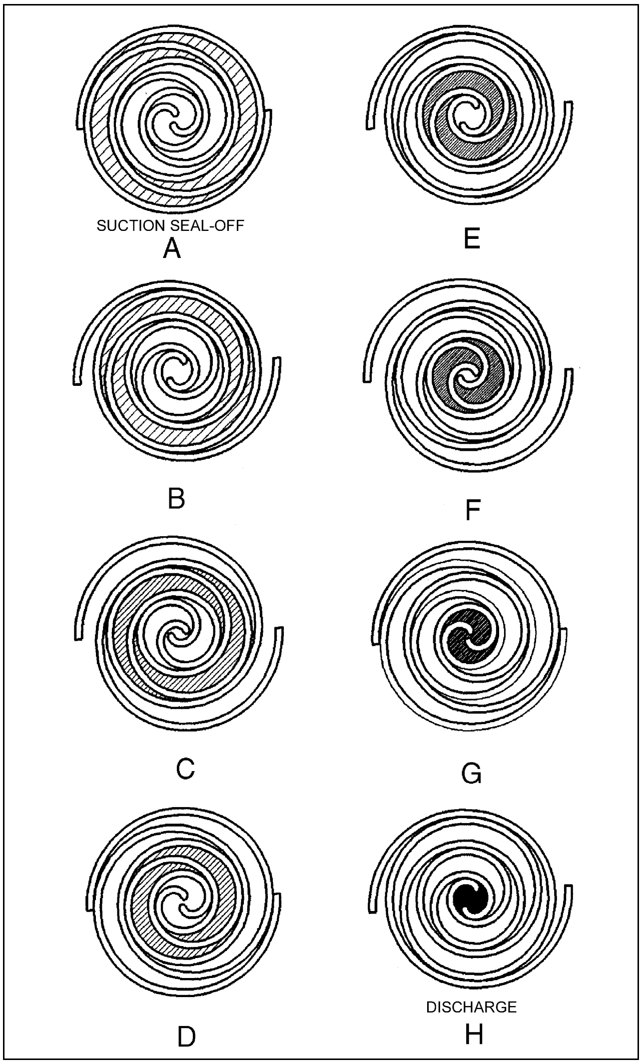


Fig. 34 Scroll Compression Process
(Purvis 1987)

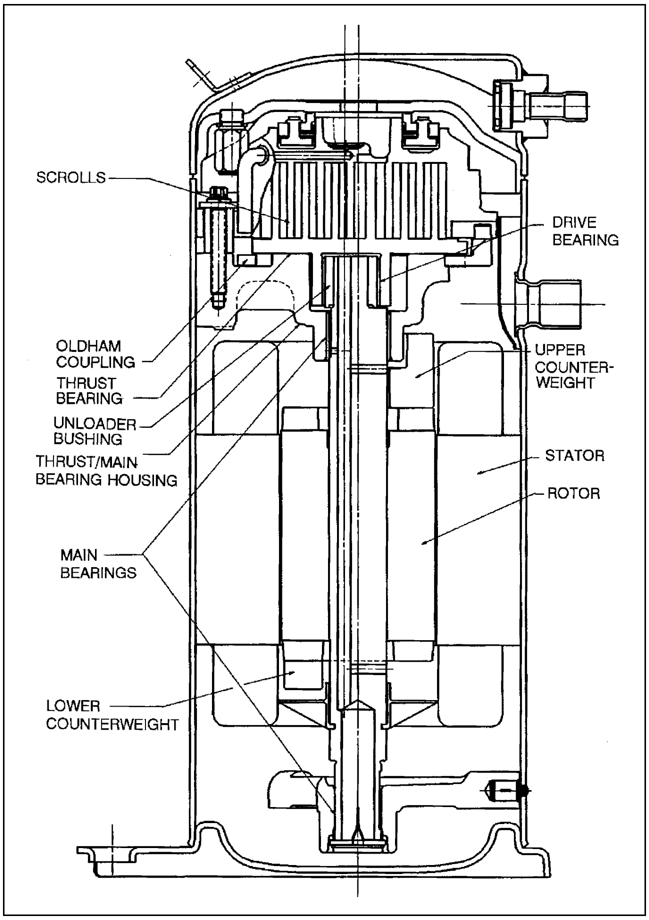


Fig. 35 Bearings and Other Components of Scroll Compressor
(Elson et al. 1990)

Bearing System. The bearing system consists of a drive bearing mounted in the orbiting scroll and generally one of two main bearings. The main bearings are either of the cantilevered type (main bearings on same side of the motor as the scrolls) or consists of a main bearing on either side of the motor (Figure 35). All bearing load vectors rotate through a full 360° due to the nature of the drive load.

The orbiting scroll is supported axially by a thrust bearing on a housing which is part of the internal frame or is mounted directly to the compressor shell.

Capacity Control

Two different capacity control mechanisms are currently being used by the scroll compressor industry.

Variable-Speed Scroll Compressor. Conventional air conditioning uses a constant speed motor to drive the compressor. The variable-speed scroll compressor uses an inverter drive to convert a fixed frequency alternating current into one with adjustable voltage and frequency, which allows the variation of the rotating speed of the compressor motor. The compressor uses either an induction or a permanent magnet motor. Typical operating frequency varies between 15 and 150 Hz. The capacity provided by the machine is nearly directly proportional to its running frequency. Thus, virtually infinite capacity steps are possible for the system with a variable-speed compressor. The variable-speed scroll compressor is now widely used in Japan.

Variable-Displacement Scroll Compressor. This capacity control mechanism incorporates porting holes in the fixed scroll member. The control mechanism disconnects or connects compression chambers to the suction side by respectively closing or opening the porting holes. When all porting holes are closed, the compressor runs at full capacity; opening of all porting holes to the suction side yields the smallest capacity. Thus, by opening or closing a different number of porting holes, variable cooling or heating capability is provided to the system. The number of different capacities and the extent of the capacity reduction available is governed by the locations of the ports in reference to full capacity suction seal-off.

Performance

Scroll technology offers an advantage in performance for a number of reasons. Large suction and discharge ports reduce pressure losses incurred in the suction and discharge processes. Also, physical separation of these processes reduces heat transfer to the suction gas. The absence of valves and reexpansion volumes and the continuous flow process results in high volumetric efficiency over a wide range of operating conditions. Figure 36 illustrates this effect. The built-in volume ratio can be designed for lowest over- or undercompression at typical demand conditions (2.5 to 3.5 pressure ratio for air conditioning). Isentropic efficiency in the range of 70% is possible at such pressure ratios, and it remains quite close to the efficiency of other compressor types at high pressure ratio (Figure 36). Scroll compressors offer a flatter capacity versus outdoor ambient curve than reciprocating products, which means that they can more closely approach indoor requirements at high demand conditions. As a result, the heat pump mode requires less supplemental heating; the cooling mode is more comfortable, because cycling is less as demand decreases (Figure 37).

Scroll compressors available for the North American market are typically specified as producing ARI operating efficiencies (COPs) in the range of 3.10 to 3.34.

Noise and Vibration

The scroll compressor inherently possesses a potential for low sound and vibration. It includes a minimal number of moving parts

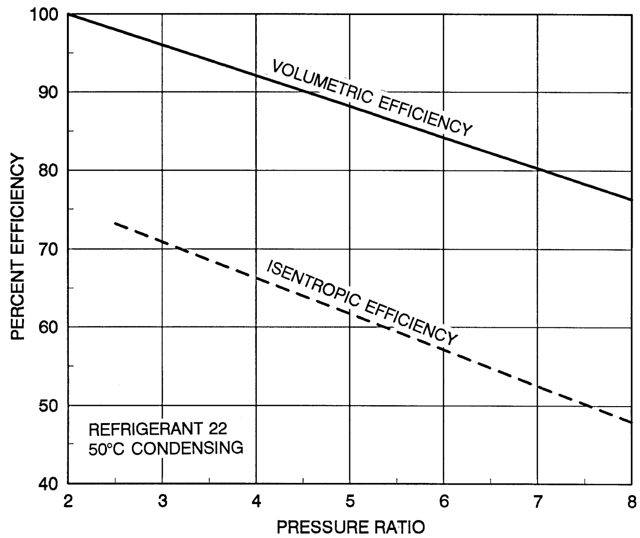


Fig. 36 Volumetric and Isentropic Efficiency Versus Pressure Ratio for Scroll Compressors
(Elson et al. 1990)

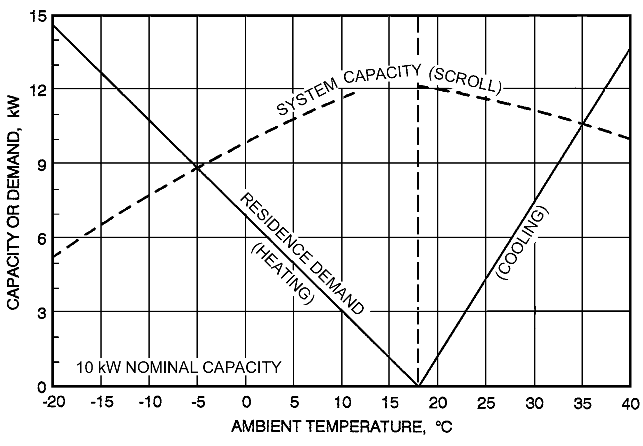


Fig. 37 Scroll Capacity Versus Residence Demand
(Purvis 1987)

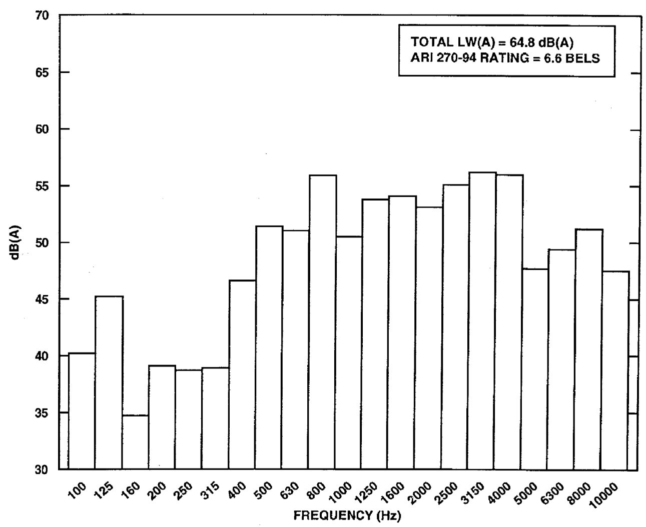
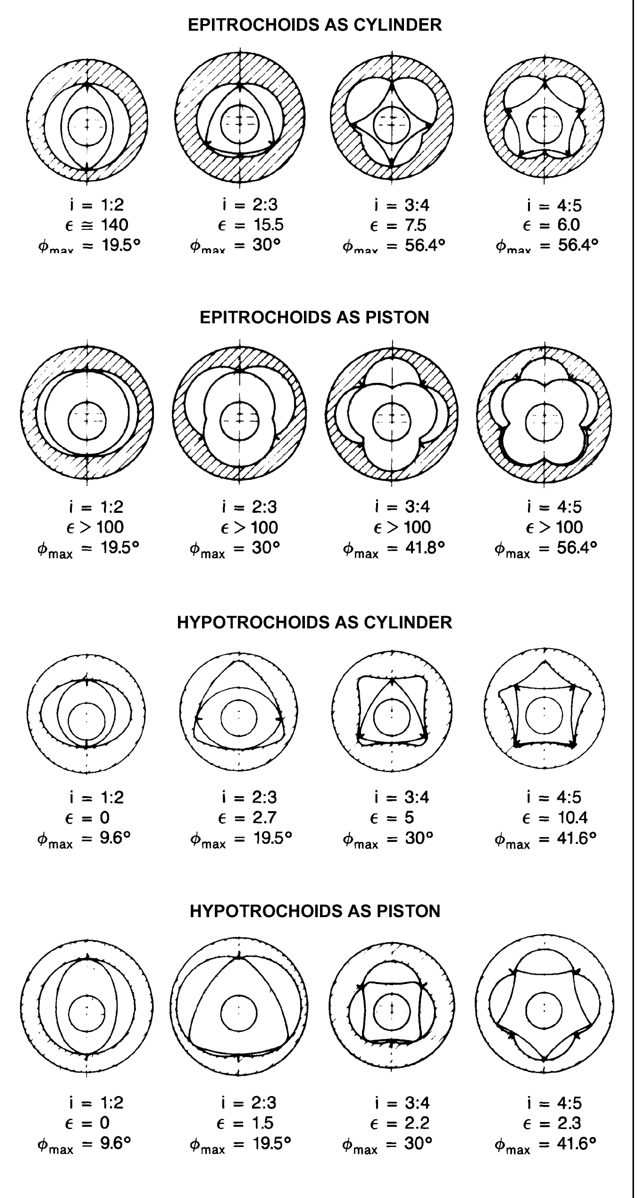


Fig. 38 Typical Scroll Sound Spectrum

compared to other compressor technologies. Since scroll compression requires no valves, impact noise and vibration are completely eliminated. The presence of a continuous suction-compression-discharge process and low gas pressure pulsation help to keep vibration low. A virtually perfect dynamic balancing of the orbiting scroll inertia with counterweights eliminates possible vibration due to the rotating parts.

Also, smooth surface finish and accurate machining of the vane profiles and base plates of both scroll members (requirement for small leakage) aids in minimal impact of the vanes.

A typical sound spectrum of the scroll compressor is shown in Figure 38.



i = Diameter ratio of generating circles
ε = Theoretical compression ratio
ϕ_{max} = Maximum inclination angle of sealing elements against trochoid

Fig. 39 Possible Versions of Epitrochoidal and Hypotrochoidal Machines

Operation and Maintenance

Most scroll compressors used today are of the hermetic type, which require virtually no maintenance. However, the compressor manufacturer's operation and application manual should be followed.

TROCHOIDAL COMPRESSORS

The trochoidal compressor is a small, rotary, positive-displacement compressor which 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 39).

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 a slipping motion along the inner cylinder surface; for trochoidal piston design, the piston shows a gear-like motion. As seen in Figure 40, a built-in theoretical 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 pressure 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.

In the past, trochoidal machines were designed much like those of today. However, like other positive-displacement rotary concepts that could not tolerate oil injection, the early trochoidal equipment failed because of sealing problems. The invention of a closed sealing border by Wankel changed this (Figure 40). Today, the Wankel trochoidal compressor with a three-sided epitrochoidal piston (motor) and two-envelope cylinder (casing) is built in capacities of up to 7 kW.

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 and expensive manufacturing tolerances; neither do they need oil for sealing,

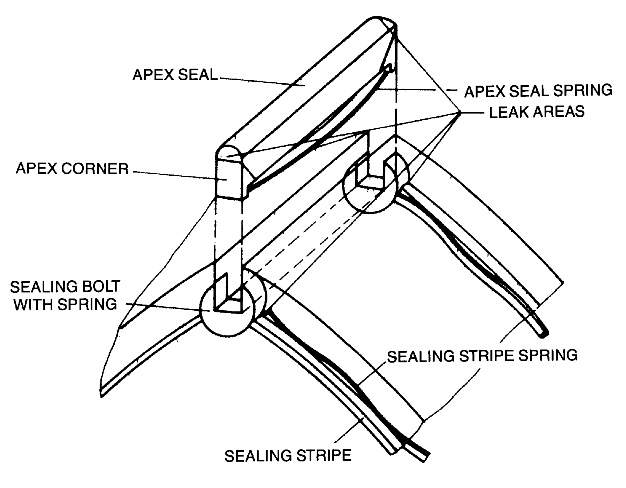


Fig. 40 Wankel Sealing System for Trochoidal Compressors

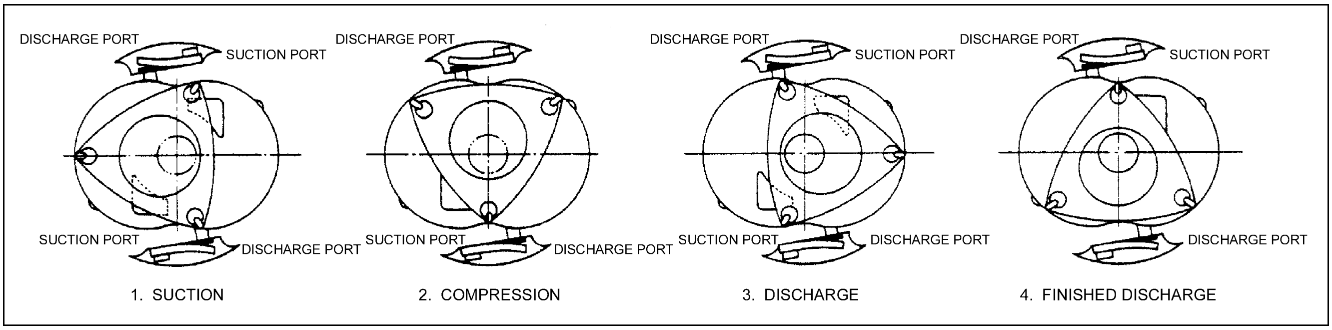


Fig. 41 Sequence of Operation of Wankel Rotary Compressor

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 version of the trochoidal compressor can also be built. Figure 41 shows the operation of the Wankel rotary compressor (2:3 epitrochoid) with discharge reed valves.

The 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 the resulting higher mechanical efficiency improves the 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.

CENTRIFUGAL COMPRESSORS

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

Centrifugal compressors are well suited for air-conditioning and refrigeration applications because of their ability to produce a high pressure ratio. The suction flow enters the rotating element, or impeller, in the axial direction and is discharged radially at a higher velocity. The change in diameter through the impeller increases the velocity of the gas flow. This dynamic pressure is then converted to static pressure, through a diffusion process, which generally begins within the impeller and ends in a radial diffuser and scroll outboard of the impeller.

Centrifugal compressors are used in a variety of refrigeration and air-conditioning installations. Suction flow ranges between 0.03 and 15 m^3/s, 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 15 m^3/s compressor being about 1.8 to 2.1 m in diameter.

Suction temperature is usually between 10 and −100°C, with a suction pressure between 14 and 700 kPa and discharge pressure up to 2 MPa. Pressure ratios range between 2 and 30. Almost any refrigerant can be used.

Refrigeration Cycle

Typical applications might involve a single-, two-, or three-stage halocarbon compressor or a seven-stage ammonia compressor.

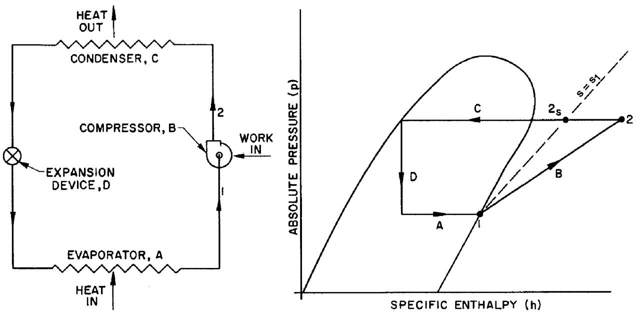


Fig. 42 Simple Vapor Compression Cycle

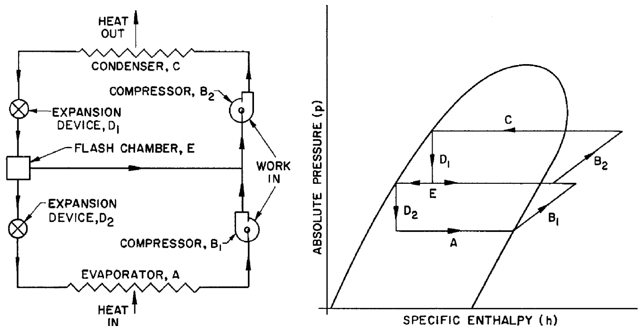


Fig. 43 Compression Cycle with Flash Cooling

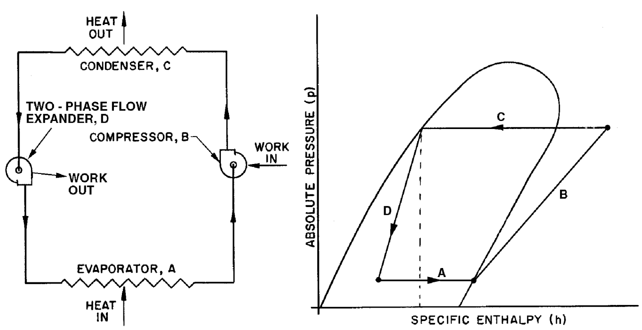


Fig. 44 Compression Cycle with Power Recovery Expander

Figure 42 illustrates a simple vapor compression cycle in which a centrifugal compressor operates between states 1 and 2.

Figure 43 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 44 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 (Brasz 1995). Power recovery during the expansion process reduces the enthalpy of the two-phase flow mixture, thus increasing the refrigeration effect of this cycle.

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 1 of the *ASHRAE Handbook—Fundamentals*.

Angular Momentum

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

$$W_i = u_i c_u \qquad (11)$$

where

W_i = impeller work input per unit mass of refrigerant, J/kg
u_i = impeller blade tip speed, m/s
c_u = tangential component of refrigerant velocity leaving impeller blades, m/s

These velocities are shown in Figure 45, 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 (11) assumes that the refrigerant enters the impeller without any tangential velocity component or swirl. This is generally the case at design flow conditions. If the incoming refrigerant was 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.

Some of the work done by the impeller increases the refrigerant pressure, while the remainder only increases its kinetic energy. The ratio of pressure-producing work to total work is known as the impeller reaction. Since this varies from about 0.4 to about 0.7, an appreciable amount of kinetic energy leaves the impeller with magnitude $c^2/2$.

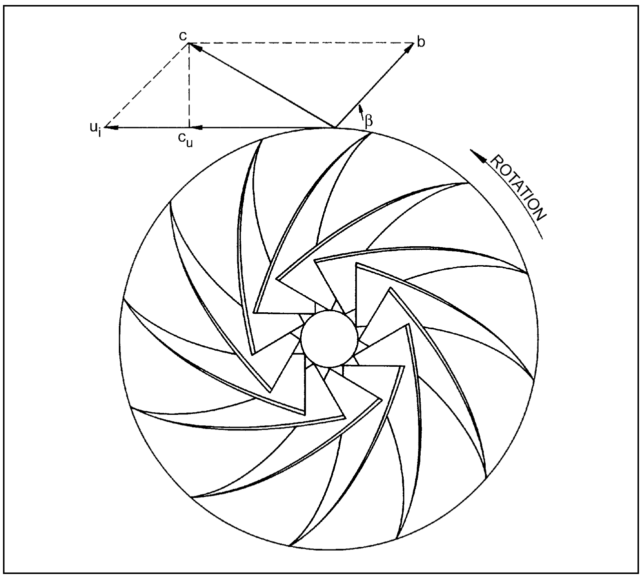


Fig. 45 Impeller Exit Velocity Diagram

To convert this kinetic energy into additional pressure, a diffuser is located after the impeller. Radial vaneless diffusers are most common, but vaned diffusers, scroll diffusers, and conical diffusers are also used.

In a multistage compressor, the flow leaving the first diffuser is guided to the inlet of the second impeller and so on through the machine, as can be seen in Figure 46. The total compression work input per unit mass of refrigerant is the sum of the individual stage inputs:

$$W = \Sigma W_i \qquad (12)$$

provided that the mass flow rate is constant throughout the compressor.

Description

A centrifugal compressor can be single-stage, having only one impeller, or it can be multistage, having two or more impellers mounted in the same casing as shown in Figure 46. For process refrigeration applications, a compressor can have as many as ten stages.

The suction gas generally passes through a set of adjustable inlet guide vanes or an external suction damper before entering the impeller. The vanes (or suction damper) are used for capacity control as will be described later.

The 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 they can be adjustable. Adjustable diffuser vanes can be used for capacity modulation either in lieu of or in conjunction with the inlet guide vanes.

For multistage compressors, the 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 in a volute or collector chamber. From there, the high-pressure gas passes through the compressor discharge connection.

When multistage compressors are used, side loads can be introduced between stages so that one compressor performs several functions at several temperatures. Multiple casings can be connected in tandem to a single driver. These can be operated in series, in parallel, or even with different refrigerants.

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 the efficiency of the compressor and the thermodynamic properties of the refrigerant. For an adiabatic process, the work input required is minimal if the compression is isentropic. Therefore, actual compression is often compared to an isentropic process, and the performance thus evaluated is based on an isentropic analysis.

The reversible work required by an isentropic compression between states 1 and 2_s in Figure 42 is known as the **adiabatic work**, as measured by the enthalpy difference between the two points:

$$W_s = h_{2s} - h_1 \qquad (13)$$

Assuming negligible cooling occurs, the irreversible work done by the actual compressor is

$$W_s = h_2 - h_1 \qquad (14)$$

Flash-cooled compressors cannot be analyzed by this procedure unless they are subdivided into uncooled segments with the cooling

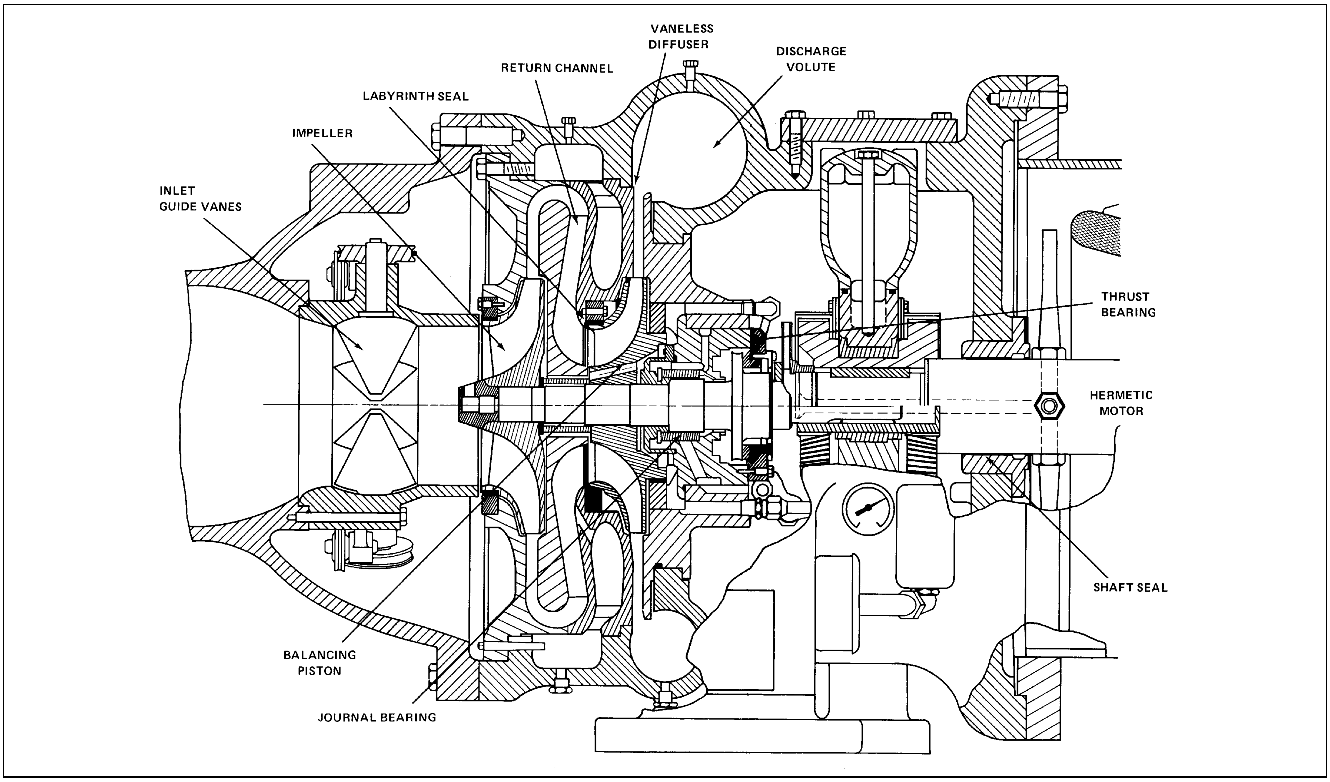


Fig. 46 Centrifugal Refrigeration Unit Cross Section

effects evaluated by other means. Compressors with side flows must also be subdivided. In Figure 43, the two compression processes must be analyzed individually.

Equation (14) also assumes a negligible difference in the kinetic energies of the refrigerant at states 1 and 2. If this is not the case, a kinetic energy term must be added to the equation. All the thermodynamic properties throughout the section on Centrifugal Compressors 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**:

$$\eta_s = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{15}$$

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 also 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 20 of the *ASHRAE Handbook—Fundamentals*. Where these are unavailable for the particular gas or gas mixture, they can be accurately calculated and plotted using thermodynamic relationships and a computer.

POLYTROPIC ANALYSIS

Polytropic work and efficiency are more consistent from one application to another, because a reversible polytropic process duplicates the actual compression between states 1 and 2 in Figure 42. Therefore, values calculated by the polytropic analysis have greater versatility than those of the isentropic analysis.

The path equation for this reversible process is

$$\eta = v(dp/dh) \tag{16}$$

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

$$W_p = \int_{p_1}^{p_2} v\,dp \tag{17}$$

It follows from Equations (14), (16), and (17) that the polytropic efficiency is the ratio of reversible work to actual work:

$$\eta = \frac{W_p}{h_2 - h_1} \tag{18}$$

Equation (16) can be approximated by

$$\frac{p^m}{T} = \frac{p_1^m}{T_1} = \frac{p_2^m}{T_2} \tag{19}$$

$$pv^n = p_1 v_1^n = p_2 v_2^n \tag{20}$$

where

$$m = \frac{ZR}{c_p}\left(\frac{1}{\eta} + X\right) = \frac{(k - 1/k)(1/\eta + X)Y}{(1 + X)^2} \tag{21}$$

$$n = \frac{1}{Y - (ZR/c_p)(1/\eta + X)(1 + X)}$$

$$= \frac{1 + X}{Y[(1/k)(1/\eta + X) - (1/\eta - 1)]} \quad (22)$$

and

$$X = \frac{T}{v}\left(\frac{\partial v}{\partial p}\right)_p - 1 \quad (23)$$

$$Y = -\frac{p}{v}\left(\frac{\partial v}{\partial p}\right)_T \quad (24)$$

$$Z = \frac{pv}{RT} \quad (25)$$

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 permit integration so that Equation (17) can be written as follows:

$$W_p = \frac{n}{(n-1)}p_1 v_1\left[\left(\frac{p_2}{p_1}\right)^{(n-1)/n} - 1\right] \quad (26)$$

Further manipulation eliminates the exponent:

$$W_p = \left[\frac{p_2 v_2 - p_1 v_1}{\ln(p_2 v_2/p_1 v_1)}\right]\ln\left(\frac{p_2}{p_1}\right) \quad (27)$$

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 (19) and (20) represent Equation (16) 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 conditions of refrigeration interest (i.e., for $p < 0.9p_c$, $T < 1.5T_c$, and $0.6 < Z$), these functions can be approximated by

$$X = 0.1846(8.36)^{1/Z} - 1.539 \quad (28)$$

$$Y = 0.074(6.65)^{1/Z} + 0.509 \quad (29)$$

The compressibility factor Z has been generalized by Edminster (1961) and Hougen et al. (1959), among others. Generalized corrections for the specific heat at constant pressure c_p can also be found in these works.

Equations (19) and (20) make possible the integration of Equation (17):

$$W_p = f\left(\frac{n}{n-1}\right)p_1 v_1\left[\left(\frac{p_2}{p_1}\right)^{(n-1)/n} - 1\right] \quad (30)$$

In Equation (30), the polytropic work factor f corrects for whatever error may result from the approximate nature of Equations (19) and (20). Since 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, the efficiency follows from Equation (18). 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 the 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 (16) through (30), it is easier and often more desirable to determine the adiabatic work by an isentropic analysis and then to convert to polytropic work by

$$\frac{W_p}{W_s} \approx \eta\left[\frac{(p_2/p_1)^{(k-1)/k\eta} - 1}{(p_2/p_1)^{(k-1)/k} - 1}\right] \quad (31)$$

Equation (31) 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 (31) is plotted in Figure 47 for $\eta = 0.76$. To obtain maximum accuracy, the ratio of specific heats k must be a mean value for states 1, 2, and 2_s. If c_p is known, k can be determined by

$$k = \frac{1}{1 - (ZR/c_p)(1 + X)^2/Y} \quad (32)$$

The gas compression power is

$$P = wW \quad (33)$$

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

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 the blade tip velocity by a dimensionless flow coefficient Q/ND^3 in which Q is the volumetric flow rate.

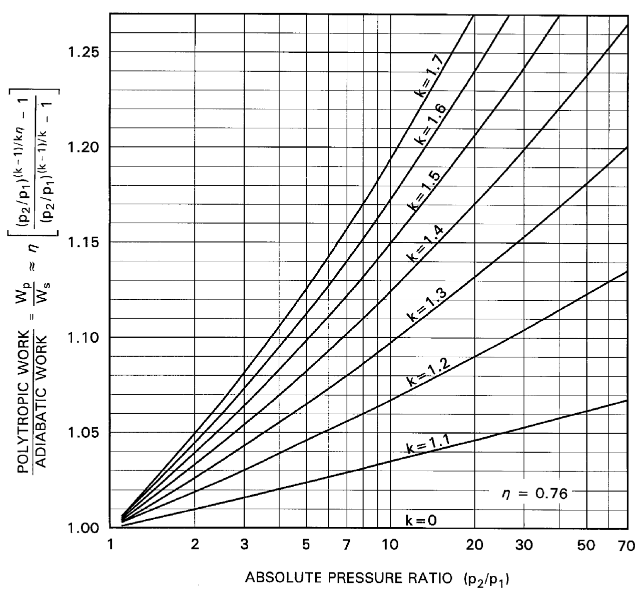


Fig. 47 Ratio of Polytropic to Adiabatic Work

Practical values for this coefficient range from 0.02 to 0.35, with good performance falling between 0.11 and 0.21. Optimum results occur between 0.15 and 0.18. Impeller diameter D_i and rotational speed N follow from

$$Q/ND^3 = \pi Q_i/u_i D_i^2 = \pi^3 Q_i N^2/u_i^3 \tag{34}$$

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

$$W_{pi} = \mu_i u_i^2 \tag{35}$$

where μ_i is the stage work coefficient.

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

$$W_p = \sum W_{pi} \tag{36}$$

and the overall work coefficient is

$$\mu = W_p/\Sigma u_i^2 \tag{37}$$

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 430 m/s. For cost and reliability, 300 m/s is a more common limitation. On this basis, the maximum polytropic work capability of a typical stage is about 50 kJ/kg.

A greater restriction on stage work capability is often imposed by the impeller Mach number M_i. For adequate performance, M_i 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 1.5 a_i^2, where a_i is the acoustic velocity at the stage inlet.

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

$$N_s = N\sqrt{Q_i}/W_{pi}^{0.75} = (1/\pi^3\mu_i^{0.75})\sqrt{Q_i/ND_i^3} \tag{38}$$

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:

$$M = c/a \tag{39}$$

Table 4 Acoustic Velocity of Saturated Vapor, m/s

	Evaporator Temperature, °C					
Refrigerant	–110	–80	–50	–20	10	40
11			124	131	136	139
12	115	124	131	135	136	133
13	125	131	133	128	117	
13B1	104	111	116	116	113	104
22	140	150	158	163	163	159
23	157	166	170	167	157	
113			104	110	115	118
114		101	108	113	116	116
123		108	116	122	127	130
124	106	115	122	127	129	127
125	114	122	127	129	124	113
134a	124	134	142	147	147	143
142b		135	144	150	153	153
152a	158	170	180	187	189	186
500	128	138	145	150	151	147

Source: Gallagher et al. (1993).

where

$$a = v\sqrt{-(\partial p/\partial v)_s} = \sqrt{n_s p v} \tag{40}$$

Values of acoustic velocity for a number of saturated vapors at various temperatures are presented in Table 4.

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_i at the stage inlet:

$$M_i = u_i/a_i \tag{41}$$

Performance

From an applications standpoint, more useful parameters than μ and (Q/ND^3) are Ω and Θ (Sheets 1952):

$$\Omega = W_p/a_i^2 = \mu(\Sigma u_i^2/a_i^2) \tag{42}$$

$$\Theta = Q_1/a_1 D_1^2 = (M_1/\pi)(Q_1/ND_1{}^3) \tag{43}$$

They are as general as the customary test coefficients and produce performance maps like the one in Figure 48, with speed expressed in terms of first-stage impeller Mach number M_1.

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.

A typical compressor performance map is shown in Figure 48 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.

To reduce first cost, refrigeration compressors are selected for pressure and capacity beyond their peak efficiency, as shown in Figure 48. The opposite selection would require a larger impeller and additional stages. Refrigerant acoustic velocity and the ability

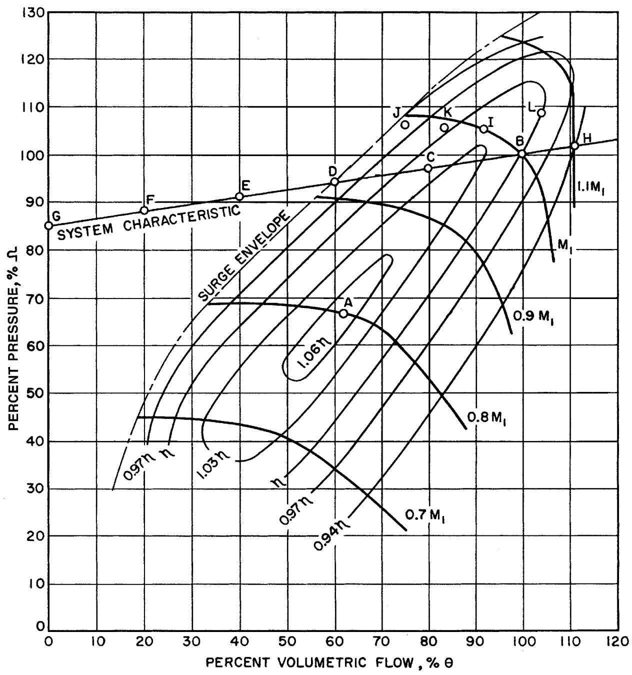


Fig. 48 Typical Compressor Performance Curves

to operate at a high enough Mach number are also of concern. If the compressor shown in Figure 48 were of a multistage design, M_1 would be about 1.2; for a single-stage compressor, it would be about 1.5.

Another acoustical effect is seen on the right of the performance map, where increasing speed does not produce a corresponding increase in capacity. The maximum flow at M_1 and $1.1M_1$ approach a limit determined by the relative velocity of the refrigerant entering the first impeller. As this velocity approaches a sonic value, the flow becomes choked and further increases become impossible. Another commonly used term for this phenomenon is stonewalling; it represents the maximum capacity of an impeller.

Testing

When a centrifugal compressor is tested, overall μ and η versus Q_1/ND_1^3 at constant M_1 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 (36) and (37) by evaluating the mixing and cooling effects between stages separately. The **overall efficiency** in such cases is

$$\eta = \frac{\sum w_i W_{pi}}{\sum w_i W_i} \tag{44}$$

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_i with any fluids whose volume ratios (v_1/v_2) and Reynolds numbers are the same.

The thermodynamic performance of a compressor can be evaluated according to either the stagnation or the 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.

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 the 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.

The 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 involves the formation of 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. The driver load is steady during rotating stall, which is harmless to the compressor, but it may, however, vibrate components excessively.

System Balance

If a refrigeration system characteristic is superimposed on a compressor performance map, it shows the speed and efficiency at which the compressor operates in that particular application. A typical brine cooling system curve is plotted in Figure 48, 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 a hot gas bypass. The volume flow at the compressor suction must be at least that for Point D in Figure 48; 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 is realized as the load decreases. The compressor is being artificially loaded to stay out of the surge envelope. The increased volume due to hot gas recirculation performs no useful refrigeration.

Capacity Control

When the driver speed is constant, a common method of altering capacity is to swirl the refrigerant entering one or more impellers. Adjustable inlet guide vanes, or **prerotation vanes** (Figure 46), produce the swirl. Setting these vanes to swirl the flow in the direction of rotation produces a new compressor performance curve without any change in speed. Controlled positioning of the vanes can be accomplished by pneumatic, electrical, or hydraulic means.

Typical curves for five different vane positions are shown in Figure 49 for the compressor in Figure 48 at the constant speed M_1. With the prerotation vanes wide open, the performance curve is identical to the M_1 curve in Figure 48. 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 48, to provide a comparison of these two modes of operation. In Figure 49, 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, except at low Mach numbers. Hot gas bypass is still necessary at Points F and G

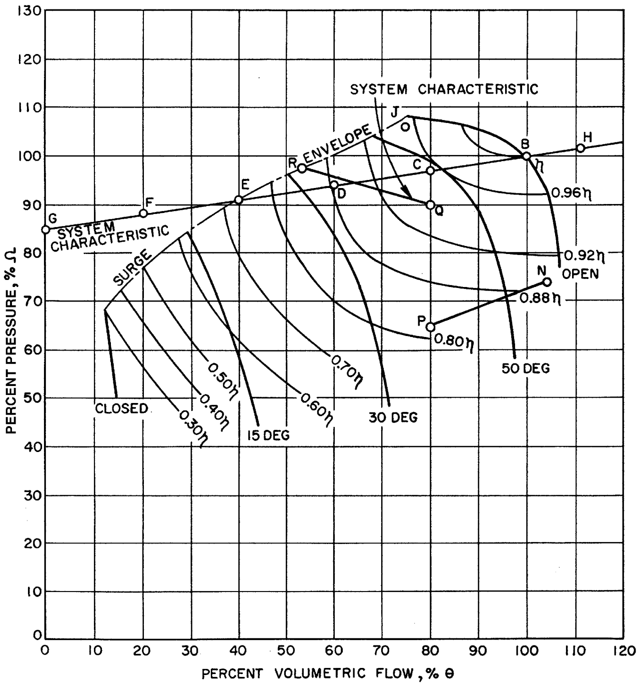


Fig. 49 Typical Compressor Performance with Various Prerotation Vane Settings

Table 5 Typical Part-Load Gas Compression Power Input for Speed and Vane Controls

System Volumetric Flow, %	Power Input, %	
	Speed Control	Vane Control
111	120	—
100	100	100
80	76	81
60	59	64
40	55	50
20	51	46
0	47	43

with prerotation vane control, but to a lesser extent than with variable speed.

The gas compression powers for both control methods are listed in Table 5. For the compressor and system assumed in this example, Table 5 shows that speed control requires less gas compression power down to about 55% of rated capacity. Prerotation vane control requires less power below 55%. For a complete analysis, friction loss and driver efficiency must also be considered.

Variable-speed control requires decreasing polytropic work (pressure) with decreasing flow (capacity) to perform more efficiently. Decreasing polytropic work is generally accomplished by reducing condensing pressure. Impeller tip speed must remain constant if the lift requirements do not change.

Refrigeration capacity is 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, a 50% reduction in compressor speed will result in a 50% reduction in refrigeration capacity; however, the available pressure for the compressor will be 25% of its value at 100% speed. If this is not consistent with actual operating conditions, hot gas bypass will be required to false load the compressor and prevent it from entering into a surge condition.

In Figure 49, for example, the compressor could operate at 15% of its rated capacity with 28% of its rated power if the polytropic work requirement could be reduced by 29%.

Since fixed-speed motors are the most common drivers of centrifugal compressors, prerotation vane control is more prevalent than speed variation. This is generally the best control mode for applications where pressure requirements do not vary significantly at part load. Less common control methods are (1) suction throttling, (2) adjustable diffuser vanes, (3) movable diffuser walls, (4) impeller throttling sleeve, and (5) combinations of these with prerotation vanes and variable speed. Each method has advantages and disadvantages in terms of performance, complexity, and cost.

SYMBOLS

a = acoustic velocity at a particular point
a_i = acoustic velocity at impeller inlet
c = flow velocity
C_p = specific heat at constant pressure
D = impeller diameter
f = polytropic work factor
g_c = gravitation constant
h = enthalpy at a specific state point
k = ratio of specific heats, Equation (32)
m = exponent, Equation (21)
M = flow Mach number, Equation (39)
M_i = flow Mach number impeller, Equation (41)
n = exponent, Equation (22)
N = rotational speed
N_s = specific speed
P = gas compression power
p = pressure at a specific state point
p^m = pressure raised to the power of m
Q = volumetric flow rate
Q/ND^3 = dimensionless flow coefficient
Q_i = volumetric flow rate in impeller
R = gas constant
T = absolute temperature at a specific state point
u_i = impeller tip speed
v = specific volume
V^n = volume raised to the power of n
W = total work input
w = mass flow
W_i = impeller work input
W_p = polytropic work input
W_{pi} = polytropic work by impeller
W_s = adiabatic work input
X = compressibility function, Equation (23)
Y = compressibility function, Equation (24)
Z = compressibility function, Equation (25)
Ω = head parameter, Equation (42)
η_s = adiabatic efficiency
η = polytropic efficiency
μ = overall work coefficient
Θ = flow parameter, Equation (43)

APPLICATION

Critical Speed

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.

The operating speeds of hermetic compressors are fixed, and each manufacturer has full responsibility for making sure the critical speeds are not too close to the operating speeds. For open-drive compressors, however, operating speed depends on the required flow of the application. Thus, the designer must make sure 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, the drive turbine or motor, and the coupling(s). In geared systems, the gearbox design is also involved. Manufacturers of centrifugal compressors use computer programs to calculate the torsional natural frequencies of the entire system, including the driver, the coupling(s), and the 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.

Vibration

Excessive vibration of a centrifugal compressor is an indication of malfunction, which may lead to failure. Periodic checks of the vibration spectrum at suitable locations or continuous monitoring of vibration at such locations are, 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 the 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 of the frequency components of the vibration spectrum above the baseline will 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 due to bearing instability, and increases at twice the running frequency are usually the result of deterioration of alignment, particularly coupling alignment.

As a general guide to establishing satisfactory vibration, a constant velocity criterion is sometimes used. In many cases, a velocity amplitude of 5 mm/s constitutes a reasonable criterion for vibration measured on the bearing housing.

Although measurement of the 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 the displacement of the shaft relative to the bearing housing, either instead of, or in addition to, monitoring the vibration of the bearing housing (Mitchell 1977). Such sensors are also useful for monitoring the 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 isolated properly from the building. Conducting vibration tests of the installed machine under operating conditions gives a base comparison for future reference.

Noise

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

There are two aspects of importance in noise considerations for applications of centrifugal compressors. In the equipment room, OSHA regulations specify employer responsibilities with regard to exposure to high sound levels. Increasing liability concerns in this regard 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 the transmission of compressor noise to such areas should be considered in the building layout and weighed against cost factors for alternative locations of the equipment in the building.

If the equipment room is to be located close to noise-sensitive building areas, it is usually cost-effective to have the noise and vibration isolation designed by an experienced acoustical consultant, since 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. Data on the noise output of a specific machine should be obtained from the manufacturer; the request should specify that the measurements are to 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 25 to 900 kW. Sometimes the driver is coupled directly to the compressor; often, however, there is a gear set between them, usually because of low driver speed. Flexible couplings are required to accommodate the angular, axial, and lateral misalignments that may arise within a drive train. Additional information on prime movers may be found in Chapter 7 and Chapter 40.

Centrifugal refrigeration compressors are used in many special applications. An outstanding example is their use in factory-packaged water chilling systems of 280 to 7400 kW capacity. These units use single-, two-, and three-stage compressors driven by open and hermetic motors. These designs have internal gear and direct drives, both of which are quieter, less costly, and more compact than external gearboxes. Internal gears are used when compressors operate at rotative speeds higher than two-pole motor synchronous speed. Chapter 43 of the *ASHRAE Handbook—Refrigeration* discusses centrifugal water chilling systems in greater detail.

A hermetic compressor absorbs the heat of the motor because the motor is cooled by the refrigerant. An open motor is cooled by air in the equipment room while the heat rejected by a hermetic motor must be considered in the design of the refrigeration system. However, heat must still be removed from the equipment room, generally by mechanical ventilation. Because they operate at a lower temperature, hermetic motors are generally smaller than open motors for a given power rating. But, if a motor burns out, a hermetic system will require thorough cleaning, while 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 to both speed squared and to the 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

The problems associated with paralleling turbine-driven centrifugal compressors at reduced load are illustrated by Points I and J in Figure 48. 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, while 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 best solution of all is to install a flow meter in the discharge line of each compressor and to use a master-slave control in which the original controller signals only one turbine, the master, while a second controller causes the slave unit to match the master's discharge flow.

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

The paralleling of 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 will be operating 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 an 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 Special Applications

Other specialized applications of centrifugal compressors are found in petroleum refineries, marine refrigeration, and in the chemical industry, as discussed in Chapters 30 and 36 of the *ASHRAE Handbook—Refrigeration*. Marine requirements are also detailed in ASHRAE *Standard* 26.

MECHANICAL DESIGN

Impellers

Impellers without covers, such as the one shown in Figure 45, are known as open or unshrouded designs. Those with covered blades (Figure 46), are known as shrouded impellers. Open models must operate in close proximity 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 from 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-mass ratio than steel, up to about 150°C, which permits higher rotating speeds with lighter rotors. Steel impellers retain their strength at higher temperatures and are more resistant to erosion. Lead-coated and stainless steels can be selected 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, having been used for temperatures as low as −100°C and pressures as high as 2 MPa. Nodular iron and cast or fabricated steel are also used for low temperatures, high pressures, high shock, and hazardous applications. Multistage casings are usually split horizontally, although unsplit barrel designs can also be used.

Lubrication

Like motors and gears, the bearings and lubrication systems of centrifugal compressors can be internal or external, depending on whether or not they operate in refrigerant atmospheres. For reasons of simplicity, size, and cost, most air-conditioning and refrigeration compressors have internal bearings, as shown in Figure 46. 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 backup service.

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 55 and 65°C are required, depending on pressure. Once a compressor has started, its oil should be cooled to increase oil viscosity and maximize refrigerant retention during the pull-down period.

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 on the solubility of refrigerants in oil can be found in Chapters 1, 2, and 7 of the *ASHRAE Handbook—Refrigeration*.

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 a minimal loss of refrigerant must be provided in external bearing systems.

Bearings

Centrifugal compressor bearings are generally of a hydrodynamic design, with sleeve bearings (one-piece or split) being the most common for radial loads; tilting pad, tapered land, and pocket bearings are customary for thrust. The usual materials are aluminum, babbit-lined, and bronze.

Thrust bearings tend to be the most important in turbomachines, and centrifugal compressors are no exception. Thrust comes from the pressure behind an impeller exceeding the pressure at its inlet. In multistage designs, each impeller adds to the total, unless some are mounted backward to achieve the opposite effect. In the absence of this opposing balance, it is customary to provide a balancing piston behind the last impeller, with pressure on the piston thrusting opposite to the stages. To avoid axial rotor vibration, some net thrust must be retained in either balancing arrangement.

Accessories

The minimum accessories required by a centrifugal compressor are an oil filter, an oil cooler, and three 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.

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 gages and thermometers are useful indicators of the 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.

OPERATION AND MAINTENANCE

Reference should be made to the compressor manufacturer's operating and maintenance instructions for recommended procedures. A planned maintenance program, as described in Chapter 37 of the *ASHRAE Handbook—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 be used to signify the need for routine maintenance; abrupt changes should indicate system or component difficulty. A successful maintenance program requires the operating engineer to be able to recognize and identify the reason for these data trends. In addition, by having a knowledge of the component parts and their operational interaction, the designer will be able to use these symptoms to prescribe the proper maintenance procedures.

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

1. 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. The existence of 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.
2. Compliance with the manufacturer's recommended oil filter inspection and replacement schedule allows visual indication of the condition of the compressor lubrication system. Repetitive clogging of filters can mean system contamination. Periodic oil sample analysis can monitor acid, moisture, and particulate levels to assist in problem detection.
3. Operating and safety controls should be checked periodically and calibrated to ensure reliability.
4. The electrical resistance of hermetic motor windings between phases and to ground should be checked (megged) regularly, following the manufacturer's outlined procedure. This will help detect any internal electrical insulation deterioration or the formation of electrical leakage paths before a failure occurs.
5. Water-cooled oil coolers should be systematically cleaned on the water side (depending on water conditions), and the operation of any automatic water control valves should be checked.
6. For some compressors, periodic maintenance, such as manual lubrication of couplings and other external components, and 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.
7. Vibration analysis, when performed periodically, can locate and identify trouble (e.g., unbalance, misalignment, bent shaft, worn or defective bearings, bad gears, mechanical looseness, and electrical unbalance). Without disassembly of the machine, such trouble can be found in its early stages before machinery failure or damage can occur. Dynamic balancing can restore rotating equipment to its original, efficient and quiet operating mode. Such testing can help avoid costly emergency repairs, pinpoint irregularities before becoming major problems, and increase the useful life of components.

8. 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 prior to putting them back into operation.

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