Wrist pin bushings in cast iron or forged rods are usually made of bronze with loadings up to 3000 psi. The finish here as well as the aluminum or bronze rods should be 16 microinch. Clearances of 0.0002 to 0.0004 have been used successfully. Pressure lubrication to these areas isn't necessary but splash holes and open end oil grooves must be placed in the unloaded zone and some grooving is often carried into the loaded zone.

Clearances on the crankpin end of the rod should be 0.00075 to 0.001 per inch of diameter.

When the eccentric strap is used some means of retaining it on the crankpin must be made. This can be accomplished by the use of washer type springs between the strap at the wrist pin end and the piston pin bosses, to obtain a close fit between the strap and piston pin bosses or shields attached to the eccentric of the shaft. Bearing load on the eccentric diameter will be very low due to the large size of the eye, but the rubbing speed must be held below 3000 fpm.

In small hermetic compressors in sizes below ¼ hp, the scotch yoke type piston-connecting rod arrangement is used advantageously. It is an economical assembly and readily formed by hydrogen brazing steel components.

Piston, Piston Ring and Wrist Pin

Pistons are usually made of cast iron or aluminum. Cast iron is used for tight fits to keep leakage to a minimum without using rings. Aluminum is used where weight is of importance and one or more rings are always provided.

Clearances between cylinder and piston vary from 0.0003 in. per inch diameter for small cast iron pistons to 0.003 in. per inch (or more) for large aluminum pistons. The minimum clearance is determined by the theoretical expansion of the materials, but the actual clearance used must be determined by tests because temperature differences between various points of the piston may cause distortions which greatly exceeds the expansion at uniform temperatures. Tapered or stepped pistons may be used to compensate for heat distortion without sacrificing the sealing effect of the piston skirt.

Piston rings are used extensively, but care should be taken to determine if the second or third ring is improving or reducing the performance. It will often be found that the addition of a second or third piston ring will increase, the motor watts without increasing the capacity.

Piston rings in a compressor will normally scrape oil from the cylinder walls in the wrong direction, that is, into the cylinder space. If this is a problem, an oil scraping ring may be used. It must be provided with reliefs in the form of drain holes in the piston or suitable recesses for oil return, otherwise the ring is next to useless. A cheap and often sufficiently effective oil scraping effect can be obtained by leaving the bottom edge of the piston as sharp as possible instead of breaking the corner as is usual for other machined parts.

The finishes of the piston and cylinder wall should be better than 32 microinch.

The shape of the piston is determined to a great extent by the space required for the pin and the connecting rod. A one-to-one ratio between length and diameter, with the pin midway between top and bottom will always provide an ample shoe for the connection rod. For economical design, however, the pin may be placed as close to the piston top as possible and the skirt should be made no longer than necessary. The minimum length of the piston is determined to a large degree by the cylinder-piston clearance. Larger clearances will cause more pronounced edge loadings for the same piston length. A length to a diameter ratio of 0.80 is considered satisfactory with a 0.001 to 0.002 in. clearance per inch diameter for a connecting rod to crank arm ratio of 4.5 to 5.

There is no well defined limit to the piston speed that can be used in a compressor. Speeds up to 1200 fpm as determined by Equation 8, have been applied. The limiting factor is gas velocity through the valves. It is, therefore, the ratio between piston area and valve area in a design that limits the piston speed.

Piston speed:

$$\text{fpm} = 2S(\text{rpm})/12 \qquad (8)$$

where

S = stroke, inches.

The wristpin usually is made of steel, case hardened to 50–60 Rockwell C and ground to an 8 microinch finish. It can be fitted tightly in the piston bosses and held in place with spring clips or can be made floating but restrained from striking the cylinder walls by similar clips. Pin diameter is determined by the size of connecting-rod bearing needed.

Lubrication

Lubrication systems range from the simplest splash system to the most elaborate forced feed systems with filters, vents and equalizers. The type of lubrication required depends largely on bearing loads and application.

For low to medium bearing loads and for 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, combined with the splashing effect of the dippers in the oil and the freer bearings, there is a tendency for the compressor to operate somewhat noisily. Further, the splash at high speed encourages frothing and oil pumping which, however, is no problem in package type equipment but might prove so in remote systems where gas lines are long.

A flooded system is so called to cover those arrangements which include discs, screws, grooves, oil-ring gears, or other devices which lift the oil to the shaft or bearing level. These devices flood the bearing and are not much superior to splash systems except that the oil is not agitated as violently, and quieter operation results. Since little or no pressure can be developed by this method, it is not correct to consider it as forced feed.

Forced feed lubrication develops pressure by means of a pump, either gear, vane, or plunger, which forces oil under pressure into the bearing. Smaller bearing clearances can be used because there is adequate pressure to feed oil in sufficient quantity for proper bearing cooling. As a result, the compressor is quieter in operation.

Gear pumps are used to a large extent. Spur gears are simple but have a tendency to promote flashing of the refrigerant dissolved in the oil. This is due to 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. It will, therefore, be seen that the eccentric gear type pump, the vane pump or the piston pump will give better performance than simple gear pumps when pump is not submerged in the oil.

Oil pumps must be made with minimum clearances to permit pumping a mixture of gas and oil. There should be provision on the discharge side of the pump to bleed a small quantity of oil into the crankcase. This vents the pump, prevents excessive pressure and assures more prompt priming. A strainer should be inserted in the suction line to keep foreign substances from the pump and bearings.

The strainer will not catch very fine particles. If large quantities of such particles are present and if bearing loadings are high, it may be necessary to use an oil filter in the discharge side of the pump.

In designing a compressor, some provision must be made to return oil from the suction gas. The returning oil must flow into the compressor crankcase. Opposing this oil flow there is a flow of gas from piston leakage, 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 entering the suction. This check valve must be provided with 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.

Hermetic Motors

Hermetic motors can be loaded considerably greater than open motors because of the suction gas cooling that is employed. Loads of 200–250 percent above nominal have been applied to these motors.

The cost of a hermetic motor is normally determined by D^2L where D is the outside diameter or frame size, and L is the stack length. It can be seen that on a fixed design D is a constant and L is the only variable which will effect a cost reduction. The D^2L relationship can also be used as a parameter in an attempt to redesign for the next lower frame size.

To effectively use a hermetic motor, it is important to design the maximum load as close to breakdown torque at 85 percent nominal voltage as possible. This is normally 80 percent of breakdown torque on single-phase motors and 85 percent on polyphase. Doing this will afford better operation at the low load and possibly higher voltage conditions. The limiting factor at high loads is normally the motor temperature, while at light loads the limiting factor is the discharge temperature. Overdesign of the motor will result in higher minimum load conditions because of the larger I^2R losses which will reflect an increased actual suction gas and thus a higher discharge gas temperature.

It is therefore important to keep losses to a minimum if the compressor is to be capable of operating at high compression ratios. The single-phase motor presents more problems in design in this area than does the polyphase as the relationship between main and auxiliary windings becomes more important and critical along with necessary starting equipment.

The locked rotor rate of rise must be kept low enough to insure against excessive motor temperatures with the motor protection available. The maximum temperature under these conditions should be held to 300 F. With better protection, a higher rate of rise can be tolerated, and thus a less expensive motor may be used.

The materials employed in these motors must display qualities of high dielectric strength and resistance to fluid and mechanical abrasion, as well as being compatible with an atmosphere of Refrigerant 22 or Refrigerant 12, and oil.

Hermetic motor selection:

1. Types of motors in use (by application)
   Refrigerator compressors
   Low to medium torque—Split phase or P.S.C.
   High torque—C.S.C.R. or P.S.C.
   Room air-conditioner compressors
      P.S.C. or C.S.C.R.
   Central air conditioning and commercial refrigeration
      C.S.C.R., 1-phase, to 6 hp
      3-phase, 2 hp to 100 hp
2. Factors in selection
   a. Insulating system
   b. Efficiency and performance factor
   c. Starting and breakdown torques
   d. Temperatures and starting currents
   e. Cost and availability

Seals

Stationary and rotary type seals have been employed extensively on reciprocating compressors. The older stationary type usually employed a metallic bellows and hardened shaft for a wearing surface. Its use has dropped off considerably with one of the main reasons being a high first cost.

The rotary seal which is more prominently used is considerably less expensive and trouble free. A synthetic seal tightly fitted to the shaft prevents leakage at this point and also seals against the back face of a carbon nose. The front face of this carbon nose seals against a stationary cover plate. This type of design has been used on shafts up to 4-in. diameter. The rotary seal should be designed so that the carbon nose is never subjected to the full thrust of the shaft. The design should be capable of absorbing movement through the end play of the shaft. The spring should be so designed that a minimum cocking force is applied. Material should be such that a minimum of swelling and shrinking is encountered.

Suction and Discharge Valves

The limiting factors of piston speed are the valve area that can be accommodated and the gas velocities permissible without excessive throttling losses. The actual gas velocities vary for different points of the stroke. For design purposes, gas velocity = bore area × average piston speed /valve area. The permissible gas velocity through the restricted area of the valve will depend upon the loss in volumetric efficiency and the excess power requirements the designer is willing to take. In general, with ammonia, velocities up to 12,000 fpm and with Refrigerant 12, Refrigerant 22, velocities up to 6,000 fpm result in no material loss in volumetric efficiency or increase in horsepower. The velocity in the restricted valve area is not the only factor; the shape of the valve and adjacent ports, and the number of directional changes of the gas flow affects the efficiency of the conversion of velocity head to pressure head.

The ideal valve would meet the following specifications:

1. Largest possible restricted area, of the shortest possible length.
2. Straight gas flow, no directional changes.
3. Light weight combined with low lift for quick action.
4. *Minimum unbalance.*
5. No harmful clearance.
6. Rugged.
7. Inexpensive.
8. Tight seating.

Most valves used today are in one of the following groups:

a. A free floating reed valve with a backing to limit the movement. It seats against a flat surface with elongated ports. It is extremely simple and stresses can be calculated readily. It is limited to relatively small port areas and multiples are often used. As more reeds are used the back-stop machining or arrangement becomes quite expensive. Totally backed with a curved back-stop, it is probably the only valve that will withstand continuous slugging.

b. A reed, clamped at one end and with full back-stop support or a stop at the tip to limit the movement. This cantilever reed is more complicated than the free floating reed because it does not move in a simple way. Even where a curved back-stop is used the reed does not roll up on the curved surface as might be expected, but has a waving, snaking motion when opening. This motion will cause stresses far greater than those calculated from the curvature of the back-stop. The waving movement must be timed so that the reed tip is not whipped into the valve plate or tip breakage will result.

c. Free floating ring valve. A truly free floating ring is seldom used because it would have only gas forces to return it to the seat after it had opened. Since this gives unnecessary losses some type of spring return is provided in the form of coil springs or a wavy, flexing ring in back of the valve. Ring valves are particularly adaptable to designs using cylinder sleeves. Without sleeves coring and machining become difficult and expensive.