

pressure may be conveniently maintained as high as the condensing pressure but it has been learned that a pressure about 25 percent lower gives better performance.

Valves

Suction valves are not required by rotary compressors since suction side flow is continuous. However, it is common practice to equip small machines with a check valve, at the suction inlet, to close automatically when the compressor stops.

The discharge valve is usually a simple reed valve made of high grade steel supplied for the purpose. The stock is chosen thick enough to give the desired strength and to provide good response. For machines below five horsepower, thickness is usually in the range of 0.005 to 0.012 in. In the design of these valves the material is not stressed beyond 50,000 psi. In most machines 15,000 to 35,000 psi is more typical. The lower values apply to small machines and the higher value to larger machines. Particular care is exercised in making the valve to avoid notched or rough edges particularly along the stressed areas so as to avoid stress concentration due to notch effect. For the more heavily stressed valves it is good practice to hone valve edges to obtain desired smoothness.

If the stock is too thin the valve opens quickly but closes too slowly and may fail to close entirely. On the other hand the thickness should not be appreciably greater than required to obtain prompt closing at the completion of the discharge cycle.

MECHANICAL EFFICIENCY

High mechanical efficiency is dependent upon minimizing of other sources of friction loss as well as those in the bearings. Such losses occur between the blades and the slot walls; at the blade tip on the cylinder wall and between the rotor faces and the front and rear heads.

If friction power losses are studied in relation to size and geometrical relationships, two important facts will be observed. First, as displacement is increased (for an optimized design configuration) by a given factor, friction power loss increases by a smaller factor. Second, friction power losses are related to compressor geometry such that for small machines a fairly wide range exists within which a suitable choice of dimensions may be made. However, as displacement is made larger power losses become more critically dependent upon geometry and therefore a more precise choice in dimensions must be made.

One important factor relating to high mechanical efficiency is the achievement of a successful run-in process. The run-in process materially reduces the friction losses of the blades and bearings. An improvement of about six percent in power input, of which a large part is reduced friction, is typical. This represents a much larger improvement in percent of friction. Surface finishes and surface treatments which assure a successful run-in can now be specified. The wear-in process must be non-destructive in character and when planned for correctly, is self arresting after a comparatively short period of use.

Friction losses are usually about equally divided between the losses due to viscous drag along the rotor faces and the losses due to bearing friction and blade friction. Blade friction alone is about two-thirds as great as the bearing friction. For example, a typical $\frac{3}{4}$ hp compressor was shown to have about 830 watts shaft input to the compressor (conditions 1750 rpm, 45 F suction, 130 F condensing) with total friction losses of about 110 watts made up as follows: bearing loss nearly 33 watts, blade friction about 22 watts, and viscous friction nearly 56 watts.

The amount and distribution of the friction losses will vary, depending on the design, but if care is exercised to minimize friction in each of the three important areas (blades, bearings

and viscous drag) very good mechanical efficiency can be achieved. In a typical small machine, which is well designed, these losses may amount to about 15 percent of the shaft input for a compressor producing about 900 Btu/h at 10 F suction and 135 F condensing, operating at 1750 rpm. On the other hand, a 2 hp machine (3500 rpm) at 45 F suction and 130 F condensing exhibited friction losses of just about 10 percent of the shaft input to the compressor.

LUBRICATION

The function of a good lubricating system is to circulate an ample supply of clean oil to all working surfaces, such as bearings, blades, blade slots, and seal faces. In an arrangement where the compressor shell is connected to the high-pressure or condenser side of the system the oil is automatically at a sufficiently high pressure and requires only that passageways be provided to distribute the oil within the mechanism. Larger machines may include a simple impeller to urge the oil along and thereby further insure the reliability of the system.

Low-side machines, where the housing is connected to the suction side or the evaporator circuit, require some sort of pumping means to maintain the desired oil pressure. In such systems appropriate venting is provided so that leakage gas can not intrude and prevent complete formation of the lubricating film.

Regardless of other details of the system, the passageways connect to outlets in order to permit free flow of the excess lubricant. These outlets should be carefully arranged so that minute particles are flushed through to the sump. If such particles can become trapped within the mechanism at the bearings or other working surfaces, localized areas of abrasive wear may result.

MOTOR SELECTION

The important factors to consider in selecting a hermetic motor are: insulating materials, operating temperatures, breakdown torque, starting torque, starting current, efficiency, power factor, cost and availability.

A number of good insulation systems have been developed for hermetic motor use. These systems were developed to obtain properties suited to the requirements for compatibility with the refrigerant-oil combinations which have come into use. The choice most suitable in any individual case, is the one which will meet the temperature requirements, and which achieves suitable chemical stability. The determination of suitability is made in part from the results of bench tests and laboratory measurements. Successful evaluations have been reported with the use of the mass spectrometer to determine the rate of formation of non-condensable gases on other products of chemical deterioration in the system on test. Whether a test of this kind or accelerated life tests are used, reliable information requires that the tests be made with complete systems which as nearly as possible, simulate those of actual operation.

For most systems and combinations of materials the maximum allowable operating temperature is considered to be about 250 F. Although visual tests may indicate no serious deterioration after several weeks of continuous operation at higher temperatures, say 280 F, experience has shown that high temperatures seriously shorten the life of these machines. Hence the usual practice is to limit the peak temperature to about 235 F by adequate cooling means and the use of a combination temperature-overload protector.

Breakdown torque requirements are dependent on the displacement of the compressor, the choice of refrigerant, and the operating conditions. A small compressor for refrigeration

duty (home freezers or household refrigerators) with R12 refrigerant employs about 36 to 38 oz-ft of breakdown torque per cubic inch of compressor displacement (displacement in cubic inches per revolution). Larger compressors such as required for window air conditioners (with Refrigerant 22) employ about 67 to 69 oz-ft of breakdown torque per cubic inch of displacement. Of particular interest is the fact that due to the high volumetric efficiency exhibited by the rotary machine the motor torque required for a given condensing range is almost constant over a wide range of suction pressure conditions. For example, at 10 F evaporator temperature the Btu output of a given machine is about half of the Btu output of the same machine at 45 F evaporator temperature. The wattage however for a constant head pressure over the same range is nearly constant because the coefficient of performance varies from about 4.75 (approx. 135 F condensing) at the 10 F condition to about 9.6 at the 45 F condition.

For most applications, rotary machines do not require complete unloading for successful starting and therefore it is rarely necessary to specify exceptional or high starting torque. The starting torque available in standard split-phase motors is ample for the small sizes. Start-capacitor motors are used on $\frac{1}{4}$, $\frac{1}{2}$ and some $\frac{3}{4}$ hp sizes to limit the starting current as required by regulations of the Underwriter's Laboratories. Permanent split-capacitor motors for the air conditioner sizes provides sufficient starting torque and improve the power factor to the required range. Motor manufacturers' test data and recommendations should be consulted in selecting a suitable motor for any specific application. These motors range in efficiency from nearly 70 percent for small split-phase motors to about 85 percent for the larger PSC motors.

LARGE ROTARY COMPRESSORS

The preceding information is representative of small rotary compressors as used in household refrigerators and small air-conditioning units up to 2 tons capacity.

However, there presently exists another important field of application for larger rotary compressors. This is the low-temperature field in which the rotary serves the useful purpose of a high cfm low-stage or booster compressor. These booster compressors are applied at saturated suction conditions ranging from -125 F to -5 F with Refrigerants 12, 22, and 717 (Ammonia). Available units range in horsepower from 10 to 600 hp and in displacement from 60 cfm to 3600 cfm in a single unit. The basic design of the compressor is similar to that shown in Fig. 6 with the exception that many equally spaced blades are used in place of the two shown and no discharge valve is used. Gas is trapped between successive blades and is compressed by the volume reduction resulting as the compressor rotates. The discharge port is located so as to communicate at the desired point during compression. This is called the design point of the compressor, and although the compressor will operate at compression ratios either above or below this design point, either undercompression (back-flow) or overcompression losses occur resulting in increased

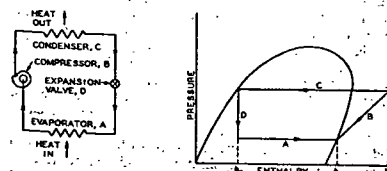


Fig. 9 Simple Compression Refrigeration Cycle

horsepower requirements. In current practice these compressors are limited to operation at compression ratios not exceeding 7:1 and the actual point is determined by the specific job in question. They are also limited to operation at relatively low differential pressures because of stresses resulting in the blades and bearings, and rotor deflection.

PART III: CENTRIFUGAL COMPRESSORS

This chapter covers the more important overall design features and performance capabilities of centrifugal refrigeration compressors. Specific design information relating to the various components of centrifugal compressors, such as the impellers and diffusers, are not discussed. Information can be found in the Bibliography at the end of the chapter.

Centrifugal compressors are essentially high speed, high capacity machines. Single-stage or multi-stage arrangements are available to meet specific design requirements in air-conditioning and refrigeration applications.

Fig. 9 illustrates the basic compression cycle which is frequently used with single- or two-stage centrifugal compressors. Although the basic cycle has an efficiency somewhat less than that of more complex cycles, it is often used on packaged centrifugal chillers. Improvement in the basic cycle efficiency can be effected through the use of liquid sub-coolers.

Multi-stage centrifugal compressors are readily adaptable to cycles involving intercooling, including interstage liquid flash cooling (sometimes referred to as economizers). Fig. 10 illustrates this cycle, where part of the condensed refrigerant is flashed to gas at the interstage pressure. This results in a power saving because the enthalpy of the liquid is reduced and the resulting flash gas does not pass through the first stage of compression. In a three-stage compressor, two stages of intercooling can be utilized. For the same overall temperature lift, two stages of intercooling will result in better cycle efficiency than one stage, and each added stage with intercooling will result in better cycle efficiency and further improvement, but with diminishing returns.

The improvement in cycle efficiency through the use of economizers is greater for those refrigerants having low basic cycle efficiencies. This is because the amount of flash gas required to cool the liquid is greater for these refrigerants than for refrigerants having higher basic cycle efficiencies. Thus, a larger proportion of refrigerant vapor bypasses the first stage of compression, and, correspondingly, a greater reduction in horsepower is effected. The resulting cycle efficiency is still lower than for the refrigerant having high basic efficiency with the same number of economizing stages.

REFRIGERANTS

Since a general discussion of refrigerants will be found in Chapter 19, this discussion is limited to the more important factors which are unique to centrifugal refrigeration.

As far as the compressor is concerned, the following factors

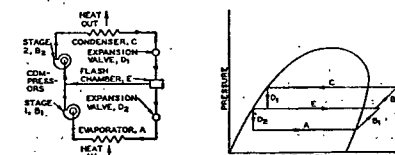


Fig. 10 Compression Refrigeration Cycle With Economizer