device on an expansion valve or a capillary tube which equalizes pressures at shutdown, the compressor can be started without excessive effort. For multicylinder compressors, an analysis must be made of the number of cylinders that might possibly be on a compression stroke at start and the position of the rods at that moment. Since the force needed to push the piston to the top dead center is a function of how far the rod is away from the cylinder centerline, the worst possible angles these might assume will need to be graphically determined by the usual torque effort diagrams. The torques for some arrangements are shown below:

No. Cylinders	Arrangement of Cranks	Angle Between Cylinders	Approx. Torque from Equation 5
1	Single		T_s
2	Single	90°	$1.025 \times T_s$
2	180° Apart	0° or 180°	T_s
3	Single	60°	$1.225 \times T_s$
3	120° Apart	120°	T_s
4	180° {2 rods per crank	90°	$1.025 \times T_s$
6	180° {3 rods per crank	60°	$1.23 \times T_s$

Probably as important as starting torque is pull-up torque. Pull-up torque is the torque required to accelerate the compressor from rest, overcoming both inertia and gas forces to bring itself to operating speed. It has been found that the greatest pull-up torque requirement comes when starting a compressor at a pressure ratio of about 2:1. The following empirical equation has been developed experimentally for multicylinder compressors without unloaders. The equation gives a value for motor starting torque which provides adequate starting and pull-up torque to accelerate the compressor to rated speed at about 2:1 pressure ratio:

$$T_s = \frac{A_s P_s}{224}(n + 2.6) \tag{7}$$

Testing

Testing for ratings should be in accordance with the ASHRAE *Standard Methods of Testing for Rating Refrigerant Compressors* (ASHRAE Standard 23–59).

Tests of a compressor are of two types. The first is for determination of capacity, efficiency, noise, motor temperatures, etc. The second, equally necessary, is to determine the probable life of the machine. Life testing must be conducted under conditions simulating those under which the compressor must operate. A minimum set of conditions would include maximum discharge, maximum suction pressure operation, a medium condition with wet return gas, a minimum suction, high discharge pressure condition with maximum suction gas temperature. A start and stop test with enough off time to accumulate liquid in the crankcase will show weaknesses in the lubrication system and resistance to slugging. A reversing test where the motor is reversed every 3 seconds will show weaknesses in mechanical parts, motor and starter. This test may be conducted by removing the suction valves and letting a limited liquid charge drain into the compressor from a condenser mounted higher than the compressor.

Since these tests may easily take a year, it is important to remember that they can be made worthless if significant changes are made between the start of testing and the beginning of production. In addition to life testing most manufacturers conduct field tests of new compressor designs.

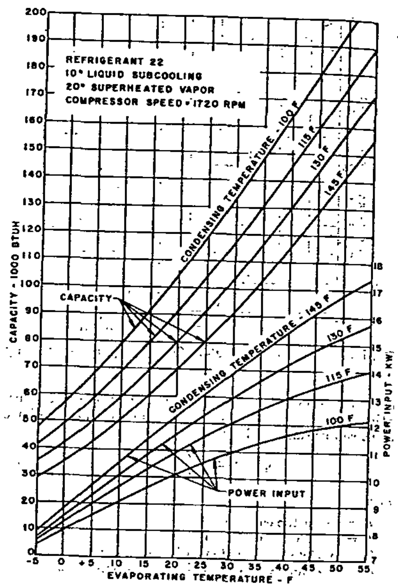

Fig. 2 . . . Typical Capacity and Power Input Curves for a Reciprocating Compressor

Standard Rating Conditions

To establish a uniform industry wide basis for rating compressors, the Air-Conditioning and Refrigeration Institute has established standard rating conditions noted in *ARI Standards* 511, 515–60, and 516–60.

Presentation of Performance Data

An example of a useful presentation of capacity data is given in Fig. 2. This is a typical set of curves for a 4 cylinder compressor, 2⅞ in. bore, 1¾ in. stroke, 1720 rpm, operating with Refrigerant 22. A set of power curves for the same compressor is also shown. Fig. 3 shows the heat rejection curves for the same compressor. Compressor curves should be labeled with the following information:

1. Degrees subcooling, or statement that data has been corrected to zero degrees subcooling.
2. Compressor speed.
3. Type refrigerant.
4. Suction gas superheat.
5. Compressor ambient.
6. External cooling requirements (if required).

FEATURES

Crankshafts

Higher speeds, the elimination of massive foundations, and more critical user requirements combine to make balancing of crankshafts necessary. Mathematical treatment of this subject can be found in texts on the subject of vibration.

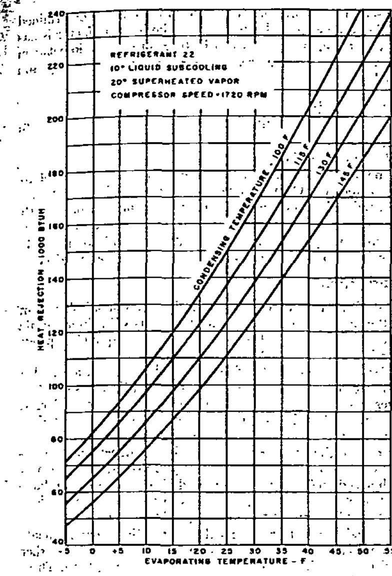

Fig. 3 . . . Typical Heat Rejection Curves for a Reciprocating Compressor

While the stress in the shaft is the important factor in shaft life, the deflection of the shaft determines bearing life and plays a significant role in noise, vibration, and efficiency. A shaft with too much deflection will cause excessive edge loading resulting in increased clearance, a high degree of noise and vibration, an eccentric and insufficient air gap in hermetic compressors, and eventual bearing failure.

Shafts have in general been made of forged steel with hardened bearing surfaces to reduce wear, while the core properties giving good ductility and strength have been retained. The finished crankpins and journals are commonly 8 microinch and polished.

Current practices have utilized cast shafts of either the alloy, or nodular irons, or higher tensile gray irons. These are generally of lower cost than the forgings, have lower notch sensitivity and superior damping qualities. The gray cast irons in the range of 40,000 psi tensile can be employed where the lower modulus of elasticity material can be tolerated, as these are most economical on a first cost basis and machineability. The alloy or nodular irons are slightly more expensive than the gray irons and the controls necessary for good machineability are more expensive and critical. These materials have excellent wear resistance but must be highly polished to avoid scoring the bearings when rotating in either direction. To obtain minimum shaft wear and maximum bearing life, a minimum Brinell hardness of 220 should be maintained on all crankpins and main journals. A hardness in this range will give a good pearlitic structure and insure removal of all splinters during the grinding and polishing operations. The finish requirements for both crankpins and main journals is 16 microinch.

Main Bearings

It is possible to have one wide outer bearing and to overhang both motor and crank; some designs are so arranged. The usual practice, however, is to place the cylinders between main bearings and overhang the motor. This arrangement is especially convenient if a forced-feed pump is applied on the end of the shaft. The bearing on one end serves as a convenience seal and mounting for the pump.

Main bearings are usually made of steel backed babbitt, aluminum or leaded bronze or cast aluminum which is an integral part of the crankcase or bearing head. It can generally be said that the harder bearing materials require harder shafts, better finishes and more cleanliness in order to operate at higher loadings. Bearing manufacturers publish bearing loadings for limited bearing life (usually 500 hr) and for perfect lubrication. These loadings should never be approached in refrigeration systems because lubrication can be far from perfect where oil and refrigerants mix readily. Since heat-pump and refrigeration service can easily total more than 4000 hr per year, bearings must be chosen to give at least 20,000 hr life.

Clearances depend on the speed and materials involved. For a steel backed babbitt bearing with a cast-iron shaft, the clearance should be 0.00075 to 0.001 in. per inch of diameter for 1800 rpm operation.

Oil grooves are usually required in the main bearings and grooves should always be placed in the unloaded area of the bearing. Grooves in shafts are used but should be avoided because they have potential oil scraping characteristics.

While it is necessary to feed oil to a bearing, it is not the only requirement. Oil must also have a way of escaping from the bearing in the quantities necessary to keep the bearing cooled. This is of particular importance where the bearing also locates the shaft or takes a thrust load.

Since shell-type bearings do not have enough side area to take thrust loads, thrust surfaces with or without thrust washers must be provided. Some type of oil groove should be provided in the thrust face or washer to allow oil to escape from the bearing. This will also help the lubrication of the thrust surface. The grooves should be placed in the stationary part to avoid the accidental construction of a centrifugal oil pump which may remove oil too fast from the bearing. Depending on the thrust load, the shaft face may bear against the bearing housing face itself, a hardened steel washer, a steel backed babbitt washer or a bronze washer.

Connecting Rods

Connecting rods are either the eccentric strap (not split) or blade and cap (split) type. The blade and cap type can be either made of forged steel, cast iron, bronze or aluminum while the eccentric strap is normally made of aluminum or bronze. Where the connecting rod is made of forged steel or cast iron, a steel backed insert employing babbitt or aluminum is fitted in the crankpin bore requiring no further machining. The crankpin end is normally narrow, while the diameter is large. This keeps the distance between main bearings to a minimum. The width of wristpin end should preferably be the same as that at the crankpin to facilitate machining. If the rod is of bronze or aluminum, it is satisfactory for use as a bearing without the need for inserts or wristpin bushings. In this case, however, the shaft should be hard. Rods should be drilled opposite the point of maximum bearing pressure to admit oil for splash lubricated machines. Splash machines may have dippers as part of the rod.