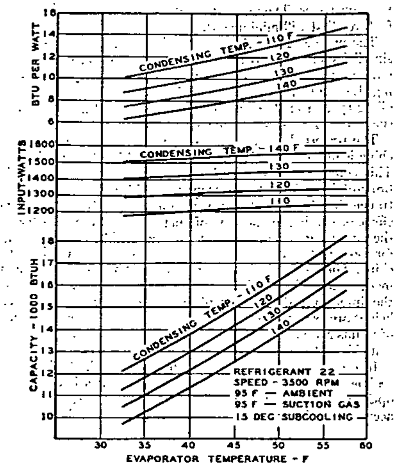

Fig. 7 .... Typical Performance Curves for a Rotary Compressor

wall meets the front head face and rear head face. The hydrodynamic sealing is dependent on clearance, surface speed, oil viscosity and surface finish of the parts. Smoother finish and closer clearance is used when low viscosity oil is used, most often in the small machines. Larger machines have greater clearance and therefore are operated with a higher viscosity oil. If the finish, clearance, alignments and oil viscosity are correctly chosen, internal leakage is reduced to a trace. There is, however, an unavoidable leakage that is due to the presence of refrigerant diluted in the oil which flows from the high pressure side to the suction side of the machine. Re-expansion losses are small since it is inherently a simple matter to make the clearance space negligibly small. It is desirable however to purposely introduce a controlled amount of re-expansion sufficient to retain in the cylinder some of the oil that is held along the leading edge of the blade. This is done by means of transfer slots in the throat of the machine. The slots perform the two important functions of supplying more oil to the cylinder wall and of providing a larger exit to the discharge ports. If these features are combined with suitable values for the radius ratio, overcompression losses are also minimized. For a given value of radius ratio, overcompression is increased as the shaft speed is increased, therefore smaller radius ratios are required in the two-pole machines than may be used in the four-pole machines. Fortunately it is not necessary to accept a severe penalty of high overcompression in the rotary machine if the compressor dimensions are selected with this requirement in mind.

Noise

The ability to achieve an acceptable noise level is an important consideration in the design of any small compressor particularly where it is intended for use on products which go into the home. Fig. 8 illustrates a convenient method for analyzing and evaluating a noise condition. This chart is typical for rotary compressors in a number of respects. Since the machine is inherently well balanced little or no difficulty arises in the first, second and third frequency bands. In the middle frequency band the noise is in large discrete frequencies associated with the discharge gas. The upper frequencies, particularly the seventh and eighth bands, are wide band frequencies mechanically induced by the moving parts. In practice various means for further reducing noise level are employed. The machine may be internally spring mounted in the hermetic shell. Such designs seek to contain the mounting problem completely in the compressor assembly. The more usual form however, is to employ an external spring mounting. In this form two additional requirements must be met. First, all possible means of transmitting vibration from the discharge gas column into the cabinet sheet metal or in some cases the condenser and tubing as well, must be scrupulously avoided. Second, when high frequency bands exhibit a high noise level the sound is absorbed either by enclosing the machine compartment or installing acoustical sound absorbing materials; a combination is usually employed.

The flow of suction gas is relatively smooth and produces so little noise that a suction side muffler is not required. A muffler on the discharge side can be used effectively particularly on larger sizes. One of the most important benefits of a discharge muffler is that it prevents the pulsating discharge gas stream from vibrating the connecting tubing and the condenser. A common cause of noise is the vibration transmitted by these members to the cabinet sheet metal panels. In general it will be observed that the noise level for a rotary compressor is also related to horsepower and to speed. Machines of successively larger horsepower rating exhibit successively higher noise levels than machines of smaller horsepower, in the same design. Increasing the speed, for example from 4-pole speed to 2-pole speed, increases the noise level but not uniformly in all the frequency bands. In this case, increased noise level is most prominent in the high frequency bands and there is very little change in most of the low and middle frequency bands.

Table 2 .... Comparison of Compressor Performance

Compressor Speed (rpm)	1735	3450
Operating conditions		
Condensing temp F	135	130
Liquid temp F	111	115
Evaporator temp F	10	45
Suct. pressure psia	29.33	91.31
Suct. gas temp F	110	95
Capacity (Cooling effect at the evaporator measured on the calorimeter)	980	21,800
Coefficient of performance	4.45	9.6
Power (Watts)	220+	2274
Power Distribution		
Work of compression	116	1580
Motor losses	67	450
Compressor losses	37.4	245

Table 3 .... Breakdown of Compressor Losses*

Compressor Size (Btuh) (Rated) Speed (rpm) Displacement (cu in. per rev)	975 (R12) 1750 0.545	12,500 (R22) 3500 1.84
Operating Conditions	10 F Evap 135 F Cond	45 F Evap 130 F Cond
Total Compressor Losses	37.4 Watts = 100%	245 Watts = 100%
1. Discharge Side Losses (%)		
A. Overcompression B. Re-expansion	12.3	51
2. Suction Side Losses (%)		
A. Pressure drop B. Heat transfer	6.42	12.25
3. Internal Leakage (%)		
A. Gas carried by oil circulation B. Leakage past clearance	6.42	3.47
4. Oil pumping (%)	2.67	2.04
5. Power to Overcome Friction (%)		
A. Bearing friction losses	19.25	10.2
B. Blade friction losses	10.16	14.7
C. Sealing film losses	42.8	16.32

* The group of machines selected for this study was made up entirely of machines which exhibited good performance.

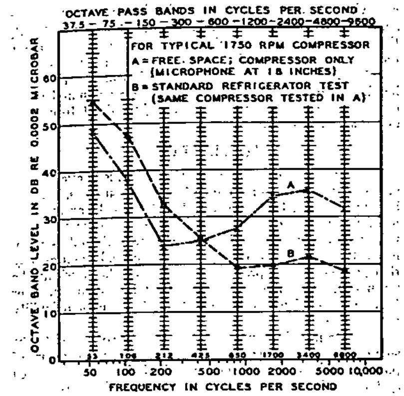

Fig. 8 .... Typical Sound Level Analysis of Rotary Compressor

FEATURES

Shafts

The requirements for rigidity in the shaft to meet deflection allowances that are compatible with minimum permissible film thickness in the bearing, results in a shaft which surpasses the strength requirements based on the other load and torque considerations. An exception to this criterion may occur when an outboard bearing is used, particularly in small machines when a small-diameter shaft seems adequate so far as the compressor itself is concerned. Where a small-diameter shaft is to be considered, particular attention must be given to avoid flexing at the armature end. The flexing of a small diameter shaft permits a gyrating motion of the armature to develop and permits the attendent forces together with the eccentric pull in the air gap to build up excessively unless enough rigidity is provided by the shaft.

The selection of the shaft materials and shaft dimensions is based on the stiffness required and a deflection analysis as well as the hardness and finish requirements of the journals. The journals are ground as accurately as possible and honed or polished to a finish of 5 microinch RMS or better. The hardness should be 56 R_c or harder.

In small machines the cantilever loaded shaft has been used very successfully. In larger sizes and for higher head pressure, an outboard bearing should secure support on both sides of the load.

Journals and Bearings

Stated briefly the bearing must support the rotating member under all conditions over the intended range of use. The bearing must be completely reliable, immune to possible damage during *run in*, trouble free during a very long service life, require no attention or maintenance of any kind, and exhibit a minimum of wear. Fortunately, a well designed sleeve bearing will meet these requirements.

The chief disadvantage of the sleeve bearing is that there is that there is no fast simple way of designing one. A procedure that relies on past experience and merits of the bearing in another situation, judged to be sufficiently similar, is not likely to provide the best design for the application.

Experience shows that a cast-iron bearing with a hardened steel journal possesses superior qualities. The reason for this is not completely understood but the experience is verified by actual test. One possible explanation may be that this combination of materials maintains a tough film of metallic compounds on the bearing and journal surfaces and thereby retard oil degradation and prevent metallic contact.

The grey cast-iron bearing must be a high quality fully pearlitic nonporous casting. It should be ASTM Type A graphite and contain no ferrite, with a desired hardness range 187 to 220 BHN.

Blades

Blades are designed for maximum reliability. Particular attention is given to the choice of materials and lubrication. The slots are hardened, ground and polished to the best finish obtainable; above 56 R_c and about 3 micro-inch RMS. A cast iron carefully selected for resistance to wear seems to be one of the best blade materials. It lubricates well and has a modulus of elasticity that permits adequate load distribution at the slot edge. One material that performs well is a grey cast iron, ASTM Type A graphite, fine grain full pearlitic, ferrite free, and in the hardness range 245-320 BHN. Peak loads in excess of 200 lb per inch of cylinder height have been used satisfactorily.

From the designers point of view a decision is first made as to the number of blades to be used and the blade dimension required for the blade loading at maximum conditions. The slots are made long enough to support the blade adequately at maximum blade extension. This usually requires that the slot be made as long as possible.

The usual number of blades is two although three or more blades may be used where the analysis of the compression cycle shows that discharge from the compression chamber can begin before over-compression occurs.

Blade thickness is made sufficent to limit the deflection under load to 0.00005 in. or less per inch of blade length. The shape of the blade tip must be made to conform to surface of generation along the cylinder wall through 360 deg of rotation of the shaft since neither too large a radius nor too small a radius will lubricate properly or seal well.

Springs or other forms of mechanical linkage to control blade movement are not used in the sliding vane machine. Better performance and greater reliability have been achieved through hydrodynamic control of blade dynamics. The general requirement is that the blade tip maintain continuous contact with the cylinder wall. The under blade hydrodynamic pressure is the most important of the several forces acting on the blade against the force of compression tending to raise the tip from the cylinder wall. The underblade pressure is cyclic due to the motion of the blade in the slot and the presence of oil behind the blade. The average underblade