Table 1 .... 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	12
2. HP Range	1/6 to 150	1/2 to 150	1/6 to 10	Up to 150
3. Cylinder Arrangement				
a. Vertical, V or W, Radial	x	x		
b. Radial, Horizontal Opposed			x	
c. Horizontal, Vertical, V or W				x
4. Drive				
a. Hermetic Compressors, Induction 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, Flood	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 Thru 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
d. Refrigerant Cooled Heads				x
9. Cylinder Head				
a. Spring Loaded Safety	x	x	x	x
b. Bolted Head	x	x	x	x
10. Bearings—Sleeve, Anti-Friction	x	x	x	x
11. Capacity Control, if Provided—Manual or Automatic				
a. Suction Valve Lifting	x	x		x
b. By-Pass—Cylinder Heads to Suction	x	x		x
c. Closing Inlet	x	x		x
d. Adjustable Clearance	x	x		x
e. Variable Stroke	x	x		x
f. Variable Speed	x			x
12. Materials				
Motor Insulations and Rubber Materials must be Compatible with Refrigerant 22 and Oil Mixtures. Otherwise no Restrictions.	x	x	x	
No Copper or Brass				x
13. Oil 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. Speeds	250 to 3600	1750 to 3600	1750 to 3600	250 to 1800
15. Pistons				
a. Aluminum or Cast Iron	x	x	x	x
b. Ringless	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

b. Suction strainer.
c. Motor (hermetic compressor).
d. Suction and discharge manifold.
e. Suction and discharge valves.
f. Internal muffler.

2. Heat gain of refrigerant in compressor from the following sources:

a. Hermetic motor.
b. Friction.
c. Heat of compression—heat exchange within the compressor.

3. Mechanical action of valves—such as inertia and valve spring rate.

4. Gas leakage past the piston.

5. Oil circulation. Since oil is used in the crankcase of reciprocating compressors for lubrication of the bearing surfaces, it is virtually impossible to prevent small quantities from passing out with the discharge gas. Actually, some oil circulation is desirable because it reduces noise and lubricates parts not otherwise accessible. However, excessive oil circulation effects the efficiency of heat transfer surfaces, the compressor overall efficiency, and the refrigerant oil mixture properties. The effect on evaporator capacity is shown in Fig. 1. The amount of oil circulation is normally held below 3 percent by weight. The refrigeration system must be designed to effectively return all oil put into circulation by the compressor.

6. Clearance. All of the gas cannot be pushed out of the cylinder during the compression stroke. Some will remain in pockets in the cylinder, in the discharge ports, and in the clearance space between the piston at top dead center and the valve plate. This gas is called the clearance gas and it will expand on the suction stroke until it reaches the suction gas pressure. Only at this point can additional gas be drawn into the cylinder. For this reason, the entire compressor displacement is not available for pumping.

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 due mainly to fluid and mechanical friction within the cylinder. The actual compression cycle and work of compression must be determined from an indicator card.

It would be a difficult matter to consider all of these losses individually. They can, however, be grouped together and considered by category. Their effect upon ideal compressor performance is measured by the following efficiencies:

Compression Efficiency considers only what occurs within the cylinder. It is a measure of the deviation of the actual compression from the isentropic compression. Compression efficiency is defined as the ratio of the work required for isentropic compression of the gas to the work done within the cylinder (as obtained by an indicator diagram).

Mechanical Efficiency is defined as the ratio of the work delivered to the gas (as obtained by an indicator diagram) to the work delivered to the compressor shaft.

Volumetric Efficiency is defined as the volume of fresh vapor

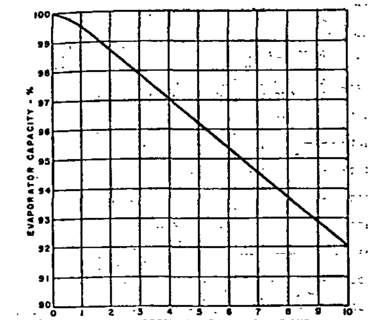

Fig. 1 .... Effect of Lubricating Oil on Evaporator Capacity

entering the cylinder per stroke divided by the piston displacement. Volumetric efficiency based on clearance volume alone can be calculated as:

$$e_1 = 1 + C\left(1 - \frac{P_2}{P_1}\right) \qquad (2)$$

where

e_1 = volumetric efficiency.
C = clearance, percent.
P_2 = absolute pressure at end of compression.
P_1 = absolute pressure at beginning of compression.

Experience indicates that the overall volumetric efficiency will run 15 to 25 percent lower, depending on the compression ratio, than the theoretical volumetric efficiency based on clearance alone.

Brake Horsepower is a function of the power input to the ideal compressor and the compression, mechanical, and volumetric efficiencies of the compressor.

Converting the ideal power input per pound of gas to horsepower:

$$thp = \frac{Q_w}{2544.65} \qquad (3)$$

$$bhp = \frac{(thp) \times V_d \times e_1}{V_b \times e_2 \times e_3} \qquad (4)$$

where

bhp = brake horsepower.
thp = theoretical adiabatic horsepower per pound of gas per hour.
V_d = displacement of compressor, cubic feet per hour.
V_b = specific volume of gas, cubic feet per pound.
e_1 = volumetric efficiency.
e_2 = compression efficiency.
e_3 = mechanical efficiency.

Liquid Subcooling is not accomplished by the compressor, and the increase in capacity resulting from liquid subcooling should not be credited to the compressor.

Suction Gas Superheat

The amount of suction gas superheat has several effects on compressor performance. Less than 6–10 F deg superheat is generally undesirable because it results in continuous liquid bleed back to the compressor with resulting oil dilution and foaming and possibly liquid slugging. Unless all liquid is evaporated in or before it enters the cylinder, it will evaporate during re-expansion and thereby reduce the capacity. It also makes expansion valve control difficult. Where liquid carry-over is necessary in a flooded evaporator to insure oil return to the compressor, heat exchangers should be used to insure that dry gas enters the compressor. Beyond 10 F deg an increase in superheat has very little effect on the compressor capacity and efficiency. Increase in superheat will increase the discharge temperature and, in hermetic compressors, the motor temperature. Less refrigerant is dissolved in the oil, and compression becomes noisier. Tests on several compressors showed an increase in capacity of 0.3 to 1.0 percent per 10 F deg increase in superheat in the range from 10–50 F deg superheat. Power input was reduced approximately 0.5 percent per 10 F deg superheat. The superheat was measured at the suction shut-off valve before the gas passed over the motor.

Heat Rejection

The heat rejected by the condenser is the sum of the refrigeration effect and the heat equivalent of the power input to the compressor. In a hermetic compressor, this heat rejection includes the inefficiency of the motor. Compressor heat rejection factors are necessary for sizing condensers.

Motor Efficiency, Performance, and Starting Torque

The motor efficiency usually represents a compromise between cost and size. A motor can generally be made more efficient the larger it is for a given rating. Accepted efficiencies range from approximately 80 percent for a 3-hp motor to 92 percent for a 100-hp 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 will reduce the impulse effect.

Small and evenly spaced impulses are also beneficial from a noise and vibration standpoint.

Since many compressors start against load, it is desirable to estimate starting torque requirements. The following equation is for a single cylinder compressor and neglects the 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, little or no leakage past the piston rings, and will yield a conservative estimate of starting torque.

$$T_s = \frac{(P_2 - P_1)As}{24d_1/d_2} \qquad (5)$$

$$T_s = \frac{(P_2 - P_1)As}{24N_2/N_1} \qquad (6)$$

where

T_s = starting torque, pound feet.
P_2 = discharge pressure, pounds per square inch, absolute.
P_1 = suction pressure, pounds per square inch, absolute.
A = area of cylinder, square inches.
s = stroke of compressor, inches.
N_2 = motor speed, revolutions per minute.
N_1 = compressor speed, revolutions per minute.

Equation 5 shows that if pressures in the system are balanced or almost equal, ($P_2 = P_1$), the torque requirements are considerably reduced. Thus, by using a pressure balancing