Table 7 . . . . Approximate Suction-line Capacity Factors for Equal Pressure Drop of CCl_2F_2 (Refrigerant 12)

Saturated Suction Temperature, F.	50	40	30	20	10	0	−10	−20
Factor	1.09	1.00	0.92	0.86	0.80	0.74	0.66	0.56

difluoromethane, and 0.64 psi per foot for monofluorotrichloromethane. Where there is a possibility of vaporization of some of the liquid before reaching the expansion valves, means for subcooling should be provided.

Since a reduction of suction pressure at the compressor results in an appreciable reduction in capacity and more power input per ton of refrigeration, great care should be given to the proper sizing of suction lines between the evaporator and the compressor. Although comparatively high velocities, 500 to 5000 fpm, may be used, the optimum value will depend upon the refrigerant and the operating-pressure range. Since return of the oil to the compressor must be considered in the case of the fluorinated hydrocarbons and methyl chloride, for these refrigerants the minimum velocity should be 500 fpm for horizontal runs and 1000 fpm for vertical runs. For the former, the usual design velocities range between 1000 and 2000 fpm. Too high velocities create noise problems and excessive pressure drops. The total pressure drop in the suction line should be between one and two psi, if the velocity can be kept within the specified limits.

Compressor discharge or hot gas lines may be designed with velocities from 1000 to 5000 fpm, except for dense gases such as carbon dioxide, where noise considerations will reduce the upper limit. A pressure drop of 2 to 4 psi is recommended for the discharge lines. Extensive tables are available in the literature for the determination of pressure drops through refrigerant lines with various refrigerants. The capacities listed in Tables 5, 6, and 7 are published in *ARI Equipment Standards* 520 (1946), of the Air-Conditioning and Refrigeration Institute, and are used by permission. Table 5 shows the tonnage capacity normally allowed for CCl_2F_2 liquid lines per foot equivalent length of pipe, and Table 6, the maximum tonnage for suction and discharge CCl_2F_2 lines. Table 7 presents suction-line capacity factors for equal pressure drop.

ACCESSORIES

Dehydrators, oil separators, strainers, vibration eliminators, sight glasses, and various types of valves are accessories frequently needed for the proper installation and operation of refrigeration systems. Refrigerant-line dehydrators or dryers usually consist of copper containers fitted with tubing connections at either end, and contain a desiccant such as silica gel, activated alumina, or calcium chloride. The liquid refrigerant is circulated through the dryer during operation of the system, and the moisture content of the refrigerant charge is thus kept to a minimum.

Oil separators are installed between the compressor and condenser to prevent excessive oil removal from the compressor crankcase and its passage into the condenser and evaporator. The oil is separated from the gaseous refrigerant by gravity during its passage through a chamber of sufficient size to reduce the velocity. A float-operated valve maintains a maximum oil level in the separator, and additional oil is forced by pressure difference through a line back to the crankcase.

Screen strainers are frequently installed in the liquid line piping before solenoid valves and expansion valves, as well as before regulating valves in water lines leading to water cooled condensers. Sight glasses that permit visual inspection of the condition of the refrigerant are sometimes installed on factory-assembled commercial unit systems. It is particularly advisable to place such a fitting before the expansion valve, if the evaporator is located above the condenser.

Flexible vibration eliminators, usually consisting of a bellows design covered with woven copper wire, are sometimes installed in copper lines where units such as compressors are installed on flexible mountings, or where vibration is otherwise a problem. Packed or packless shut-off valves are necessary where it may be required to isolate portions of a system.

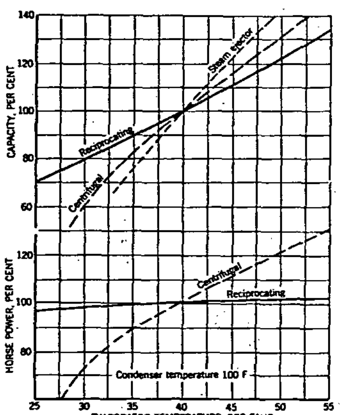

Fig. 16 . . . . Performance Characteristics of Compression Refrigeration Machines at Constant Speed

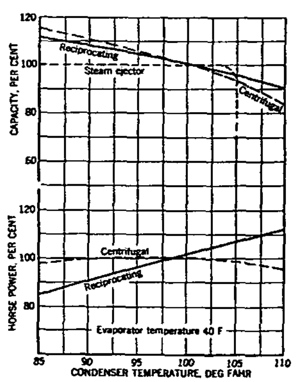

Fig. 17 . . . . Performance Characteristics of Compression Refrigeration Machines at Constant Speed

Table 8 . . . . Basis of Equipment Selection

Capacity Tons	Majority Used	Some Used	Few Used
0 to 10	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built-up central systems.
10 to 25	Unit central systems using duct distribution.	Unit systems in conditioned space.	Built-up central systems using reciprocating compressors, adsorption, and absorption systems.
25 to 100	Built-up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Central systems using adsorption systems or centrifugal refrigeration.
100 to 200	Built-up central systems using reciprocating compressors.	Built-up central systems using absorption and centrifugal compressors.	Built-up central systems using steam jet.
200 and Over	Built-up central systems using centrifugal compressors.	Built-up central systems using steam jet or absorption systems.	Built-up central systems using reciprocating compressors.

EQUIPMENT CHARACTERISTICS AND SELECTION

The various types of compression systems have quite different characteristics of capacity and power with varying evaporator and condenser temperatures, as may be noted from curves in Figs. 16 and 17.

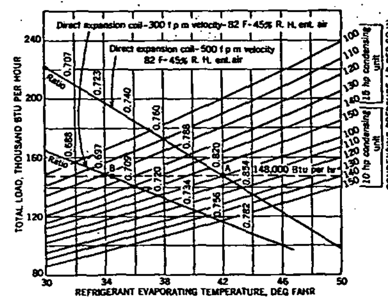

Fig. 18 . . . . Compressor and Coil Performance

From Fig. 16 it may be observed that power requirements for the centrifugal compressor increase much more rapidly than for the reciprocating compressor, with increase in evaporator temperature. Similarly, the capacities of the steam ejector and centrifugal compressors increase more rapidly than those of the reciprocating compressor with increase in evaporator temperature. Thus, both the steam-jet and centrifugal machines tend to be more self-regulating than the reciprocating. It is also evident from Fig. 16 that the steam-jet equipment is best suited for operation at high evaporator temperatures.

The effect of condenser temperature upon the power and capacity of the different types of compressors is shown in Fig. 17. It may be noted that the power required by the reciprocating compressor increases rapidly with increase in condenser temperature, while the power curve for the centrifugal compressor is relatively flat. It is also evident that the capacity of the steam-jet compressor is independent of condenser temperature until a certain point is reached,

Table 9 . . . . Typical Operating Conditions for Two Types of Load

Type of Enclosure	Load Btu per Hr.	Ratio Sensible to Total Heat	Air Entering Coil: Temp. F	Air Entering Coil: Rel. Hum. %	Operating Balance Point: Evaporator Temp. F.	Operating Balance Point: Condenser Pressure Psi	Operating Balance Point: Sensible Heat %
Restaurant							
Sensible	103,000						
Latent	45,000						
Total	148,000	0.695	82	45	34.4	123	69.9
Office							
Sensible	121,000						
Latent	27,000						
Total	148,000	0.820	82	45	42.2	100	82.1