A broad division of equipment to be used for a particular installation or application may be made on the basis of the magnitude of the load. Current general practice is outlined in Table 11.

Unit or *packaged* systems, consisting of a reciprocating compressor, condenser, evaporator and fans, are generally used in the smaller sized jobs where electric power is available, as they are manufactured complete, ready to install and are the most economical (see Chapter 36).

The reciprocating compressor in the built-up central system (see Chapter 43) covers the widest range of application since it is applicable to either the direct expansion or indirect systems and can be driven by steam or gas engines, or by electric motors. The quantity of condensing cooling medium required is also less than for any other system with the exception of the centrifugal compressor, which uses the same amount.

Centrifugal compressors are used for large installations, and usually where the indirect system is required. The driving mechanism can be steam turbine or electric motor. The steam jet system is used where steam is available and cooling water can be had in large quantities.

TABLE 12. TYPICAL OPERATING CONDITIONS FOR TWO TYPES OF LOAD

Type of Enclosure	Load, Btu per Hour			Ratio Sensible to Total	Air Entering Coil		Operating Balance Point		
	Sensible	Latent	Total		F Deg	Per Cent R.H.	Evaporator Temp F Deg	Condenser Pressure Lb per Sq In.	Per Cent Sensible Heat
Restaurant	103,000	45,000	148,000	0.695	82	45	34.4	123	69.9
Office	121,000	27,000	148,000	0.820	82	45	42.2	100	82.1

It will be noted by referring to Fig. 7 that all systems using compressors have a common characteristic and that is, that the capacity varies with the evaporating temperature. Not only can the equipment be selected to produce a given result but the performance can be predicted under varying load conditions by the simple expedient of using the variable of evaporating temperature as the abscissa and the load or capacity as the ordinate in a series of curves.

Manufacturers of compressors and cooling coils furnish performance data for apparatus that can be plotted in the form of curves similar to those shown in Fig. 11. The performance of a compressor is plotted as a series of curves, each curve being drawn for a given condensing pressure. The performance of a direct expansion coil at two different air velocities is plotted on the same graph. The operating point will be, of course, where the two curves cross.

Data given in Table 12 illustrate two types of conditioned enclosures having the same total load of 148,000 Btu per hour, but with two different ratios of sensible to total heat. In the case of the office with a ratio of 82 per cent sensible to total heat, the operating point *A* in Fig. 11 is found to be 42.2 F evaporating temperature with a face velocity of 500 fpm. In the case of the restaurant, with a ratio of 69.5 per cent sensible to total heat, the air velocity is lowered to 300 fpm and the evaporating temperature is lowered to 34.4 F as shown in point *B* of Fig. 11. In order

to obtain the same capacity, a larger condensing unit is used. This illustration assumes zero pressure drop through the suction line. The pressure drop can be taken into account by shifting the compressor performance curves by the amount of pressure drop expressed in Fahrenheit degrees.

ABBREVIATIONS AND SYMBOLS IN CHAPTER

cop = coefficient of performance, ratio of refrigerating effect to the heat equivalent of the compressor work.
CVE = conventional volumetric efficiency.
d = internal diameter in inches.
H_t = cooling load in tons.
h_d = specific enthalpy of vapor at condition of discharge from compressor.
h_{lc} = specific enthalpy of liquid at discharge from compressor.
h_{ld} = specific enthalpy of liquid at discharge of expansion valve.
h_{ls} = specific enthalpy of liquid at entrance to expansion valve.
h_m = specific enthalpy of mixture.
h_v = specific enthalpy of saturated vapor.
h_{vd} = specific enthalpy of saturated vapor at discharge of valve or compressor.
h_{vs} = specific enthalpy of saturated vapor at state s entering compressor.
hp = horsepower.
p_d = pressure of saturated liquid and vapor at discharge of compressor.
p_s = pressure of saturated liquid.
psig = pressure pounds per square inch, gage.
psia = pressure pounds per square inch, absolute.
Q = quantity of heat, Btu.
Q_c = heat loss from condenser, Btu per pound refrigerant.
Q_j = heat dissipated in cooling water, Btu per hour.
s = entropy.
Δs = entropy change between suction and discharge.
T = absolute temperature, Fahrenheit degrees.
T_{avg} = average temperature, Fahrenheit degrees, absolute of gas passing through compressor.
T_c = condenser temperature, Fahrenheit degrees, absolute.
T_s = evaporator temperature, Fahrenheit degrees, absolute.
t_{sd} = degrees superheat at discharge condition of vapor leaving compressor.
t_d = discharge temperature, Fahrenheit degrees.
V_c = clearance, percentage of volume, swept by piston, which is contained in spaces at end of cylinder when piston is at end of stroke (clearance includes valve spaces, etc.)
v_s = specific volume of gas at suction, cubic feet per pound.
v_d = specific volume of gas at discharge, cubic feet per pound.
W_r = refrigerant rate, pounds per minute.
x = proportion of liquid in mixture of vapor and liquid.

REFERENCES

[1]—Application and Economy of Steam Jet Refrigeration to Air Conditioning, by A. R. Mumford and A. A. Markson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 33).

[2]—The Application of Storage Refrigeration to Air Conditioning, by C. F. Boester (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 675).

[3]—Use of Cold Accumulators in the Air Conditioning Field, by R. W. Evans and C. J. Otterholm (A.S.H.V.E. TRANSACTIONS, Vol. 48. 1942. p. 123).

[4]—Cooling Homes, A Field for Refrigeration, by A. R. Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926. The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T. G. N. Haldane (*Electric Review*,