Table 3 .... Number of Defrosts per Year* (100 Percent Running Time)

Climate (TAC)	−10 to 5	6–8	9–11	12–14	15–17	18–20	21–23	24–26	27–29	30–32	33–35	36–39	39–41
Normally dry	475	450	400	375	350	300	250	200	175	150	100	75	50
Normally damp	800	750	700	625	575	500	425	350	300	225	175	125	100

* The figures given must be multiplied by the running-time ratio for the period considered.

Extra power consumption allowance must be made where periodic defrosting of an air-source evaporator is required, and where supplemental heating is employed.

The number of defrosting operations required per month or a complete heating season is difficult to evaluate accurately. It will depend on the detailed design and control of the heat pump, the climate, and the hours of operation. Table 3 gives the range in annual number of defrosts that may be expected on typical small unitary heat pumps.

To obtain a total yearly operating cost, the energy consumption for cooling must be added to that required for heating. Suggested procedures for calculating cooling energy consumption are given in Chapter 46 and 54.

Annual Operating Cost

A tabulation such as is shown by Table 4 may be used for estimating the operating cost for commercial and industrial installations. It is necessary to know the monthly kilowatt electric demands and the corresponding kilowatt-hour consumption for the base electric load, as well as for the heat pump or any other electric equipment, in order to apply the proper electric tariff. The base loads, shown as Columns 1, 2 and 3 of Table 4, may include lighting, elevators, office machinery, exhaust and supply fans, circulating pumps, and similar items which are normally essential in a structure and are not affected by a particular air-conditioning design. The heat-pump kilowatt demand, during the month, which is required to satisfy the net day and night heat loss or heat gain (whichever is the greater), can be obtained from manufacturers' data. These resulting kilowatt demands are multiplied by a coincident factor to obtain the coincident demands, listed in Columns 4 and 7. These coincident demands can be added to the base load demands to obtain the total electric load in Column 10. The coincident factors can best be obtained from published data on similar installations. The procedure for finding the heating and cooling kilowatt hours, Columns 5, 6 and 8, has been previously described.

For residential installations, a detailed monthly estimate generally is not justified unless a monthly demand type of rate is in effect.

In making a power consumption estimate for unitary heat pumps, a detailed breakdown into periods less than a complete season, or the breaking down into small temperature spans, is not ordinarily justified. Manufacturers of this type of equipment provide estimating procedures which are simpler to use, yet are of sufficient accuracy for the purpose intended. These methods usually consist of an empirical equation, tabular data, or plotted curves which combine the influences of the size and thermal properties of the structure, the heating design temperature, the characteristics of the heat pump, and supplemental heat usage.

HEAT-PUMP COMPONENTS

For the most part, the components used in heat pumps and the practices followed bear a direct relation to the low temperature refrigeration art. This section will outline the major components used and point out characteristics or special considerations which apply to the heat-pump field.

Compressors

Reciprocating compressors are used more than any other type on systems in the general range of ½ to 100 tons, and frequently even larger. A brief description of this type of compressor is given in Chapter 38. For the most economical application, it is general practice to select the reciprocating compressor to provide the desired cooling capacity. The cooling capacity is the evaporator capacity and represents the ability of the compressor to pump refrigerant vapor between the existing temperature limits. When heating, the capacity of a particular compressor is the sum of the evaporator capacity (i.e., heat-source coil capacity) plus the heat equivalent of the compressor work.

Figs. 13, 14 and 15 in Chapter 38 shows the capacity and power characteristics of typical reciprocating compressors. A compressor normally employed for comfort cooling use will have a clearance volume (ratio of gas volume remaining in cylinder after compression stroke to total swept-cylinder

Table 4 .... Suggested Procedure for Computing the Operating Cost of Commercial and Industrial Heat-Pump System

Period	Base Load Lighting and Miscellaneous		Cost Base Load	Heat Pump			Supplemental Resistance Heating (If Used)		Total Heating and Cooling	Total Electric Load		Total Cost
				Coincident demand	Heat	Cool						
	Kw (1)	KwH (2)	KwH (3)	Kw (4)	KwH (5)	KwH (6)	Kw (7)	KwH (8)	KwH (9)	Kw (10)	KwH (11)	(12)
June												
July				Fill in one line for each month through May								
May												

volume) of about 5 percent. The capacity drop off at low evaporator temperatures corresponding to low heat-source temperatures is also evident. The drop off in power at low evaporator and condensing temperatures is also evident.

If the compressor has low clearance volume, say 2½ percent, then it is more suitable for low-temperature operation, and will provide, for example, about 15 percent greater refrigerating capacity at an evaporator temperature of 0 F and a condenser temperature of 110 F. However, this compressor has somewhat more power demand under maximum cooling load conditions than does one of medium clearance.

It is obvious that more total heat capacity can be obtained at low outdoor temperatures by deliberately oversizing the compressor. The compressor is oversized in order to obtain more heating capacity so it is necessary to provide some type of capacity reduction to bring the cooling capacity into balance. This can be done by means of two-speed motor drives, cylinder cutouts, or other methods as described in Chapter 38. The disadvantage of this arrangement results from the fact that the greater number of operating hours that occur at the higher suction temperatures must then be served with the compressor in the unloaded condition which generally causes lower efficiency. Therefore, the annual operating cost will tend to rise. It is also true that the additional first cost of the oversized compressor must be economically justified by the gain in heating capacity. One method proposed for increasing the heating output at low temperatures involves the use of *staged compression* in which one compressor may pump from −20 F suction temperature to 40 F condensing temperature and a second compressor compress the vapor from 40 F to 120 F. In such an arrangement it is possible to interconnect any two compressors so that they are in parallel, both pumping from say 45 F to 120 F at the normal cooling rating point, while at some predetermined outdoor temperature on heating they are reconnected so that they pump in staged relationship.

Fig. 8 shows the performance of such a pair of compressors for compressors of both medium and low clearance volume. It is apparent that at low suction temperatures the reconnection into a staged relationship does provide some added capacity. Also, it should be understood that the motor selection involved must be based upon the maximum loading conditions for summer operation even though the low stage compressor has a greatly reduced power requirement under the heating condition.

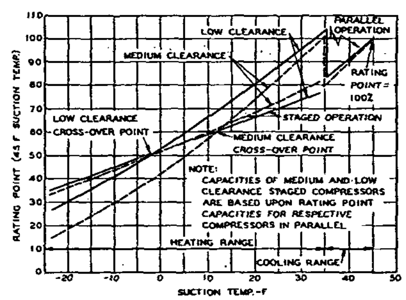

Fig. 8 .... Comparison of Parallel and Staged Operation

The coefficient of performance will be approximately the same whether they are coupled in parallel or compound-staged at any set of operating conditions depending somewhat on the motor characteristics when lightly loaded.

A *rotary* compressor has characteristics similar to a reciprocating compressor except that it is by nature possessed of low clearance and high volumetric efficiency. From this standpoint it is well suited to heat-pump service, providing about 30 percent greater capacity at 0 F−110 F lift than a medium-clearance reciprocating compressor. However, this characteristic tends also to increase the power demand at the maximum cooling load conditions. At this time, reciprocating compressors are most widely used for heat pumps, with rotary compressors restricted to the first stage of a staged-compression system and not used during the cooling cycle.

Heat Transfer Components

Refrigerant-to-air and refrigerant-to-water heat exchangers, as previously described in the section Heat Sources and Sinks, are similar to heat exchangers used in current air-conditioning practice. A refrigerant subcooler coil may be employed either in conjunction with an indoor air coil, or, on systems with a ventilation air supply, to preheat ventilation air. A substantial gain in capacity and coefficient of performance can result.

Refrigeration Components

Refrigerant piping, receivers, expansion devices and refrigeration accessories in heat pumps are usually the same as those used in other types of refrigeration and air-conditioning systems (see Chapter 38).

A *reversing valve* is used to change the system from the cooling to the heat operation. This change-over requires the use of a valve, or valves, in the refrigerant circuit,* except where the change is accomplished in fluid circuits external to the refrigerant circuit (see Table 1). Reversing valves are usually pilot-operated by means of solenoid valves which admit head and suction pressures to move the operating elements.

Expansion devices for controlling the refrigerant flow are normally thermostatic expansion valves as described in Chapter 38. Special requirements may sometimes be presented. If the circuiting is so arranged that the refrigerant line upon which the control bulb is placed can become the compressor discharge line, the resulting pressure developed in the power element of the valve may be excessive, requiring the use of a special control charge or pressure-limiting element. When a thermostatic expansion valve is applied to an outdoor air coil, a special *cross-charge* is desirable to limit the superheat at low temperatures, and thereby obtain a better utilization of the coil.

When an expansion valve is attached to a coil that is operated as a condenser, a bypass with a check valve is normally provided as indicated in Table 1.

On small factory-built systems, capillary tubes are normally used as expansion devices. While a single capillary tube on both heating and cooling is sometimes employed, better efficiency and performance can be obtained by using a more restrictive capillary tube for heating than for cooling. This may be accomplished by using two capillary tubes in either a series or parallel arrangement with a check valve to bypass one for cooling or to block one for heating, respectively.

On an air-source heat pump that must operate over a wide range of evaporating temperatures, a capillary tube will tend to pass refrigerant at an excessive rate at low back pressures, causing liquid floodback to the compressor. In some cases suction line accumulators or charge-control devices are employed to minimize this effect.