Chapter 7
AIR-AND-WATER SYSTEMS

7.1 INTRODUCTION

Air-and-water systems condition spaces by distributing both conditioned air and water to terminal units installed in the spaces. The air and water are cooled and/or heated in a central mechanical equipment room. The air supplied is termed primary air to distinguish it from recirculated (or secondary) room air.

Air-and-water systems that have been used in buildings of various types are presented below. Not all of these systems are equally valid in the context of a given project. Not all of these systems see equal use in today's design environment. They are presented, however, to provide a sense of the possibilities and constraints inherent in the use of an air-and-water HVAC system.

Familiarity with the information on air-and-water systems presented in the In-Room Terminal Systems chapter of the *ASHRAE Handbook—HVAC Systems and Equipment* is assumed in this manual. Fundamental material from that resource is generally not repeated here.

7.2 APPLICATIONS

7.2.1 Advantages and Disadvantages

Because of the greater specific heat and the much greater density of water compared to air, the cross-sectional area of piping is much smaller than that of ductwork to provide the same cooling (or heating) capacity. Because a large part of the space heating/cooling load is handled by the water part of this type of system, the overall duct distribution requirements in an air-and-water system are considerably smaller than in an all-air system—which saves building space. If the system is designed so that the primary air supply is equal to the ventilation requirement or to balance exhaust requirements, a return air system can be eliminated. The air-handling system is smaller than that for an all-air system, yet positive ventilation is ensured. Numerous zones can be individually controlled and their cooling or heating demands satisfied independently and simultaneously. When appropriate to do so (as during unoccupied hours), space heating can be provided by operating only the water side of the system—without operating the central air system. When all primary air is taken from outdoors, cross-contamination between rooms can be reasonably controlled.

Design for intermediate season operation is critical. Changeover operation (between seasons) can be difficult and requires a knowledgeable staff. Controls are more complicated than for all-air systems, and humidity cannot be tightly controlled. Induction and fan-coil terminal units require frequent in-space maintenance.

7.2.2 Suitability to Building Types

Air-and-water systems are used primarily for perimeter building spaces, with high sensible loads and where close control of humidity is not a primary criterion. These systems work well in office buildings, hospitals, schools, apartment buildings, and other buildings where their capabilities can meet the project design intent and criteria. In most climates, these systems are designed to provide: (1) all of the required heating and cooling needs for perimeter spaces and (2) simultaneous heating and cooling in different spaces during intermediate seasons.

7.3 SYSTEM CONCEPTS

An air-and-water system includes central air-conditioning equipment, duct and water distribution systems, and room terminals. The latter can be induction units, fan-coil units, or conventional supply air outlets combined with radiant panels. Usually, the primary air system is constant volume. It provides all the ventilation air, generally covers some of the space latent cooling, and some of the space sensible cooling/heating. The water system may be a two-, three-, or four-pipe arrangement and provides the majority of the sensible cooling/heating and latent cooling (with the exception of radiant panel systems, which do no latent cooling). These three water distribution arrangements are discussed in detail in Chapter 8.

In a typical air-and-water system, primary air is cooled by chilled water produced by a chiller (although in smaller systems a direct expansion cooling coil may be used). The room terminal units are supplied by a water system connected to a chiller and a boiler. The primary air system may contain an optional reheat coil. Primary air is either all outdoor air or, if a return air system is provided, a mixture of outdoor and return air. The percentage of return air is always small. Thus, where freezing temperatures are encountered, a preheater is required at the central air handler. Central system filters should be of higher efficiency than in an all-air system to compensate for reduced dilution of space contaminants as a result of minimal air-side flow rates.

When individual zones are large, water (terminal) units are often not located within the conditioned space but some distance away, and their output air is ducted to room supply terminals through ductwork.

One system arrangement involves the use of individual terminal units in each zone to provide air conditioning. In this approach, units are installed around the perimeter of a building—one per room for small rooms; more than one when needed for larger rooms. The units are complete with casing, fans, water coil, filter, and fan motor. An opening in the back or bottom of the unit can be connected to an outdoor air intake. A grille-covered opening in the front of the unit permits room (return) air to be drawn in. The coils in the units are supplied with chilled water from a refrigeration plant in the summer and in the winter with hot water from a hot water heater or heat exchanger. Each room (zone) features individual control of temperature. Relative humidity, however, depends upon the room temperature, the length of time that the unit operates to maintain the temperature, and the chilled water temperature. To provide proper control of relative humidity, a separate heating coil is sometimes included so that dehumidification and reheat can be accomplished. This requires the use of a four-pipe water system.

In an alternative arrangement, horizontal fan-coil units are suspended in each zone. The fan-coils are furnished with chilled water in summer and hot water in winter. The ceiling of a corridor is furred down to form an insulated plenum in which conditioned (cooled and dehumidified) outdoor air is circulated. This air provides ventilation and dehumidification capability. The room terminal units draw in some air from the corridor plenum, mix it with return air from the room, further condition it, and discharge the conditioned mixture into the room.

Radiant panels for heating and cooling can also be applied to air-and-water systems. They are compatible with two-, three-, and four-pipe distribution arrangements. These panels are discussed in the chapter on Panel Heating and Cooling in the *ASHRAE Handbook—HVAC Systems and Equipment*.

The controls in air-and-water systems are usually arranged so that, under design cooling loads, the primary air is supplied at lower than room temperature and chilled water is circulated through the terminal unit water coils. In transitional months, the primary air is reheated and chilled water is still circulated through the water coils. As the outdoor temperature decreases further, the warm (reheated) primary air is switched to cold air and the chilled water is switched to hot water. This last transition, between chilled and hot water supply, is called the system changeover point. The changeover is not the same for all facades (orientations) of a building, requiring careful system design and zoning. Appropriate changeover is critical to occupant comfort in a two-pipe distribution arrangement; changeover is more forgiving in a three- or four-pipe system, where chilled and hot water can both be made available during transition.

7.4 PRIMARY AIR SYSTEMS

Figure 7-1 shows the primary air system serving a two-pipe distribution with induction terminal units. Such air systems are comparable to the single-zone systems discussed in Section 6.7. If a specific humidity ratio is to be maintained in winter, a humidifier must be installed. An air-water induction unit is shown in Figure 7-2 and a fan-coil unit in Figure 7-3.

Fig 7-1 Primary Air System.

Fig 7-2 Air-and-Water Induction Unit.

Fig 7-3 Air-and-Water Fan-Coil Unit.

The quantity of primary air for each zone is determined by the:

- Ventilation requirement

- Sensible cooling requirement at maximum cooling load—this equals the room sensible cooling load less the sensible capacity of the room (water) cooling coil
- Maximum sensible cooling requirement on changeover to the winter cycle when chilled water is no longer circulated through the coils.

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The primary supply air is typically dehumidified to ”neutralize” outdoor air latent loads, while also contributing to space latent load control. A moisture content target for the supply air can be estimated from:

$$W_{pa} = W_r – (q_l - q_t) / (F\ Q_p) \qquad (7\text{-}1)$$

where,

W_{pa} = humidity ratio of primary air, lb water/lb dry air [g/kg]
W_r = maximum desired humidity ratio in zone, lb water/lb dry air [g/kg]
q_l = room latent load, Btu/h [W]
q_t = terminal unit latent cooling capacity, Btu/h [W]
Q_p = primary air rate, cfm [L/s]
F = 4840 [3.0]

To provide substantial primary air latent capacity, the air leaving the cooling coil must generally be 50-55°F [10.0-12.8°C] and close to saturation, which requires deep coils and a reasonably low chilled water temperature. Reheat coils are required for two-pipe systems to prevent overcooling of spaces under low load conditions.

The psychrometric behavior of an air-and-water induction system is shown in Figure 7-4 (summer peak) and Figure 7-5 (winter peak). The figures illustrate the operation of a system where, in the summer, all the latent cooling and part of the sensible cooling are performed by the primary air, while the water coil provides additional sensible cooling. During the winter, preheated primary air is shown to be humidified, while the water coil performs all heating. Other operating strategies can be employed, wherein the primary air essentially handles only ventilation loads. These patterns are also valid for systems in which primary air is supplied to a fan-coil unit through a directly connected air supply.

Fig. 7-4 Induction System Psychrometric Chart at Summer Peak.

Fig. 7-5 Induction System Psychrometric Chart at Winter Peak.

7.5 WATER-SIDE SYSTEMS

All-water systems are discussed in detail in Chapter 8; refer to that chapter for assistance in designing the water side of an air-and-water system.

Individual room temperature is controlled by varying the capacity of the terminal coil by either regulating the water flowing through the coil or the flow of air passing over it. During winter, the cooling coil may be converted to a heating coil (with a two-pipe system), or a separate heating coil may be provided (with a three- or four-pipe system). The required coil cooling capacity is the room sensible load reduced by the sensible cooling capacity of the primary air. Most air-and-water systems are designed to deliver cool primary air throughout the year. The heating capacity of the coil must therefore be designed to handle the room's heating load plus the primary air cooling effect. Select the coils conservatively to overcome any imbalances in the water system.

7.5.1 Effect of Pump Curves on System Performance

The key variables defining the performance of a pump/piping system are the pressure rise provided by the pump (to counteract pressure drop in the piping) and the water flow rate. The relationship between pump and piping determines the balance point where the system operation satisfies both. Typical performance curves for centrifugal pumps are shown in Figure 7-6 for a family of pumps operating at 1750 rpm. At low flow rates, the increase in pressure (head) is fairly constant; at high flow rates, it ultimately drops off to zero. The magnitude of the pressure rise is predominantly a function of the tip speed of the impeller, which is reflected in the higher heads available with larger diameter impellers or higher speed pumps. If the pressure provided by a pump of the largest diameter available is too low, choose a pump that operates at a higher speed, e.g., 3500 rpm.

For straight pipe, friction (head) loss is proportional to the square of the flow velocity (and thus the square of the flow rate):

$$\text{head loss} = (f)\,(L/D)\,(V^2/2g) \qquad (7\text{-}2)$$

where,

head loss is in feet [m]
f = dimensionless friction factor (see the chapter on Fluid Flow in the *ASHRAE Handbook—Fundamentals*)
L = length of pipe, ft [m]
D = internal pipe diameter, ft [m]
V = flow velocity, ft/s [m/s]
g = acceleration of gravity, 32.2 ft/s^2 [9.806 m/s^2]

For fittings, such as elbows and tees, the friction loss is proportional to the square of the flow rate, as evidenced by the expression of fitting head loss as a certain number of velocity heads ($V^2/2g$). The performance of a water distribution system is thus a combination of pump characteristics such as shown in Figure 7-6 and a squared relationship for the piping network.

Fig. 7-6 Performance of Three Pumps with Different Impeller Diameters Operating at 1750 rpm.

A further consideration is how the pump/piping system behaves over a range of demands under different methods of control. In the two two-pipe networks shown in Figure 7-7, the coils in (A) are controlled by three-way valves while in (B) they are controlled by two-way valves. The intent of three-way valves is to maintain a somewhat constant pressure drop across the coil-valve combination, while with two-way valves the pressure drop is increased in order to reduce the flow rate. The network shown in Figure 7-7(A) is "direct return," which has the drawback of a higher pressure difference across the valves and coils closest to the pump. A more balanced pressure difference occurs in a "reverse return" system, shown in Figure 7-7(B), where the return starts at the nearest coil and proceeds to the further ones before returning the total flow to the pump. This also produces a more self-balancing system relative to water flows.

Fig. 7-7 Water Flow Networks.

The relief valve at the end of the flow path in Figure 7-7(B) allows a low flow rate through the system and the pump if all the control valves close off and the pump discharge pressure rises. An operating characteristic of the two-way valve system is that, when the coils experience a low-flow-rate demand, the pressure difference across the flow control valve rises. The valves adjust to near their fully closed positions, where small changes in their stem positions can change the flow rate appreciably—bringing with this condition the danger of valve "hunting."

The combination of pump characteristics with those of a piping network under some form of flow control defines the system behavior. Figure 7-8 shows the head flow curve of one of the pumps (the one with an 11-in. [280 mm] impeller) from Figure 7-6. Pump efficiency and required pump power as functions of flow rate are also supplied. Superimposed on the pump characteristic are several piping network characteristics with their expected parabolic shapes. The lowest piping curve might represent the three-way valve network of Figure 7-7(A) or the two-way valve network of Figure 7-7(B) with the valves fully open. The intersection of the pump and the piping curves is the "balance point," indicating the head and flow rate condition where that particular combination will operate. Computer simulation permits analysis of components so that a system of 20 coils, for example, can be optimized by using 4 three-way valves at the hydraulically most remote points and 16 two-way valves without a need for relief valves.

With three-way valve control, the piping curve remains essentially constant regardless of the load demanded by the coils. The pump continues to deliver 2250 gpm [142 L/s] against a head of 100 ft [299 kPa]. With the two-way valve control, however, the piping network pressure drop increases as the flow rate is reduced—which produces a different piping system curve for each new adjustment. Two different curves with their different balance points are shown in Figure 7-8. Pump performance changes significantly over the range of operation with the changing characteristics of the piping network. As the flow rate is decreased and the operating condition moves upward and to the left along the pump characteristic, the pump efficiency increases from 75% to and through its maximum of 85%, then decreases again. The power requirement of the pump decreases as the water flow is throttled off.

One of the basic decisions pertaining to system cost is illustrated in Figures 7-6 and 7-8. Consider the specification of a system of large-diameter pipe with its resultant low pressure drop. While this may increase the piping cost, it may be possible to use a smaller pump, such as the one with the 11-in. [280 mm] diameter impeller shown in Figure 7-6. This pump might be slightly lower in first cost, but its big advantage lies in the lower energy costs for pumping. Both the system curve and the pump curve will lie lower on the graph in Figure 7-8, and the power requirements at the balance point will be reduced. Note how changes in one element of the system (the piping) impose changed operating conditions on another element of the system (the pump).

Fig. 7-8 Pump and Piping Network Curves.

7.5.2 Two-Pipe System

7.5.2.1 Changeover

For a system using outdoor air, there is an outdoor temperature at which mechanical cooling is no longer required. At this point, the cooling requirement can be met using outdoor air only—essentially an economizer operation. At even lower temperatures, heating rather than cooling is needed. All-air systems capable of operating with 100% outdoor air rarely require mechanical cooling at outdoor air temperatures below 50 or 55°F [10.0 or 12.8°C]. Since air-and-water systems involve considerably less air flow than all-air systems, however, they may require mechanical cooling well below an outdoor temperature of 50°F [10°C].

The transition from summer to intermediate season to winter is gradual. The changeover from cooling to heating system operation should mirror this transition. This is not easy with a two-pipe system; substantially easier with a three- or four-pipe system. With a two-pipe system the change in operation starts gradually, with an increase in the primary air temperature as the outdoor temperature decreases. This prevents zones with small cooling loads from becoming too cold. The water side provides cold water during both summer and intermediate season operation. Figure 7-9 illustrates the psychrometrics of summer-cycle operation near the changeover temperature.

Fig. 7-9 Psychrometric Chart for Off-Season Cooling with Two-Pipe System Prior to Changeover.

As the outdoor temperature drops further, the changeover temperature is reached. At this point, the water system changes from providing cold water to providing hot water for heating. This changeover temperature is empirically given (Carrier 1965) by:

$$t_{co} = t_r - [S + L + P - 1.1(t_r - t_p)]/T \qquad (7\text{-}3)$$

where,

$\cancel{L}$ = sensible heat gain from people, Btu/h [W]
$\cancel{P}$ = heat gain from lights, Btu/h [W]
S = net solar heat gain at time of changeover, Btu/h [W]
T = transmission per degree, defined as heat flow per degree difference between space temperature and outdoor temperature, which includes transmission through walls, windows, and roofs, if applicable, Btu/h °F [W/°C]
t_{co} = changeover temperature, °F [°C]
t_p = primary air temperature after changeover, normally taken as 56°F [13.3°C]
t = room temperature at changeover, normally taken as 72°F [22.2°C]

System changeover can take several hours depending upon system size, and it usually results in a temporary upset of room temperatures. Good design, therefore, includes provisions for operating the system with either hot or cold water over a range of 10 to 15°F [5.6 to 8.3°C] below the changeover point. This minimizes the frequency of changeover by making it possible to operate with warm air and cold water on cold nights when the outdoor temperature rises above the changeover temperature during the day. This limits operation below the changeover point to periods of protracted cold weather.

Optional hot or cold water operation below the changeover point is provided by increasing the primary air reheat capacity so that it can provide adequate heat during colder outdoor temperatures. Figure 7-10 shows temperature variations of primary air and water for a system with changeover.

Fig. 7-10 Typical Temperatures for System Changeover.

7.5.2.2 Non-changeover

Non-changeover systems should be considered in order to simplify system operation for buildings that experience mild winter climates. Such a system operates on the intermediate season cycle throughout the heating season, as shown in Figure 7-11. Heating may be provided during unoccupied hours either by operating the primary system with 100% return air or by switching the system and room controls to hot water for gravity heating without air circulation.

Fig. 7-11 Typical Temperatures for Non-changeover System.

7.5.2.3 A/T Ratio

A/T ratio is a factor to consider in the design of air-and-water systems.

A/T ratio = (primary airflow to space) / (transmission loss per degree) (7-4)

where the denominator is as defined under Equation (7-3). All spaces on a common primary air zone should have approximately the same A/T ratio, which is used to establish the primary air reheat schedule during the intermediate season. Spaces with A/T ratios higher than the design-base A/T will be overcooled during light cooling loads and spaces with ratios lower than the design ratio will lack sufficient heat during cool weather above the changeover point.

Calculate the minimum A/T ratio for each space by using the primary air quantity necessary to satisfy the requirements for ventilation, dehumidification, and both summer and winter cooling as defined in Section 7.4. The design-base A/T ratio is the highest ratio thus obtained, and the airflow to each space is increased as required to obtain a uniform A/T ratio across all spaces.

For each A/T ratio there is a specific relationship between the outdoor air temperature and the temperature of the primary air supplied to a terminal unit that will maintain a room at 72°F [22.2°C] or higher during conditions of minimum room cooling load. Figure 7-12 illustrates this relationship based upon an assumed minimum load of 10 times the T-value, i.e., a difference of 10°F [5.6°C] between outdoor and space temperatures.

Fig. 7-12 Primary Air versus Outdoor Air Temperatures.

Deviation from the target A/T ratio of up to 0.7 times the maximum value is permissible for massive buildings with high thermal inertia, but A/T ratios should be closely maintained for buildings with less mass, large glass areas, with curtain wall construction, or on systems with a low changeover temperature.

7.5.2.4 Zoning

A properly designed single-water-zone two-pipe system can provide good temperature control throughout the year. Its performance can be improved by zoning in several ways:

■ Primary air zoning that will permit different A/T ratios on different exposures. All spaces on the same primary air zone should have the same A/T ratio. If minimum A/T ratios for the spaces in one zone vary due to different solar exposures, air quantities can be increased for some of the spaces. It is often useful to zone primary air for northerly (north, northeast, northwest) exposures on a separate zone with reduced air quantities.

■ Primary air zoning that will permit compensation of primary air temperature. Peak cooling loads for southern exposures usually occur during winter months because of high solar loads. Therefore, primary air zoning can be used to reduce air quantities and terminal unit coil sizes for these exposures. Units can be selected for cold instead of reheated primary air, thus reducing operating costs on all solar exposures by reducing primary air reheat and water-side refrigeration needs.

■ Zoning both air and water to permit different changeover temperatures. Separate air and water zoning permits northerly exposures to operate on a winter cycle with warm water at outdoor temperatures as high as 60°F [15.6°C], whereas other exposures operate on intermediate or summer cycles at those temperatures. Primary air quantity can be lower due to a separate A/T ratio, and operating costs due to reheat and refrigeration are reduced.

7.5.2.5 Evaluation

The two-pipe air-and-water system was the first to be developed and is frequently used. Least expensive of the water distribution arrangements to install, it is less capable of handling wide variations in loads than the three- or four-pipe system arrangements. Changeover is cumbersome, and operating costs are higher than for a four-pipe system.

7.5.3 Three-Pipe Systems

Because of their wasteful energy performance (see Section 8.6), these systems are rarely installed and design information is essentially found in the *ASHRAE Handbook* archives—detailed discussions can be found in the 1973 and 1976 *ASHRAE Handbook—Systems*.

7.5.4 Four-Pipe Systems

In four-pipe systems (see Section 8.5), the terminal unit is often provided with two independent water coils—one served by hot water, the other by chilled water. As an alternative, a common coil may be used, with this coil being served by two supply and two return pipes from the chilled and hot water loops, respectively (Figure 7-13). The primary air is cold and typically is kept at the same temperature year-round. During peak cooling and heating conditions, a four-pipe system performs in a manner similar to a two-pipe system with essentially the same operating characteristics (Figures 7-4 and 7-5). During intermediate seasons, any unit can be operated at any capacity from maximum cooling to maximum heating without regard to the operation of any other unit. Thus, distribution zoning is not required for either the air or the water system.

Fig. 7-13 Room Unit Control for a Four-Pipe System.

This system is more flexible than two- or three-pipe systems. While its installation cost is higher, its efficiency is higher, and its operating cost is lower than that of the other two distribution arrangements. Operation is simpler than for a two- or three-pipe system. No changeover is required. Standby water pumps and heat exchangers should be considered for critical applications, where loss of heating capacity would have unacceptable consequences.

7.6 REFRIGERATION LOAD

The refrigeration load equals the total building block load to be removed by the combination of air and water systems:

$$R = (4.5\ Q_p\ \Delta h) + (q_s - 1.1\ Q_p\ (t_r - t_s)) + (q_L - 4840\ Q_p\ (W_r - W_s)) \qquad (7\text{-}5a)$$

$$[R = (1.2\ Q_p\ \Delta h) + (q_s - 1.2\ Q_p\ (t_r - t_s)) + (q_L - 3.0\ Q_p\ (W_r - W_s))] \quad (7\text{-}5b)$$

where,

R = refrigeration load, Btu/h [W]
Q_p = primary airflow rate, cfm [L/s]
Δh = enthalpy difference of primary air stream across cooling coil, Btu/lb [kJ/kg]
q_s = room sensible heat for all spaces at peak load, Btu/h [W]
q_L = room latent heat for all spaces at peak load, Btu/h [W]
t_r = average room temperature for all rooms at time of peak, °F [°C]
t_s = average primary air temperature at point of delivery to rooms, °F [°C]
W_r = average room humidity ratio for all rooms at peak load, lb/lb [kg/kg]
Ws = average primary air humidity ratio at point of delivery to rooms, lb/lb [kg/kg]

The first term on the right side of Equation (7-5) is the load on the primary air cooling coil, while the remaining two terms represent the load on the terminal unit water system.

7.7 ELECTRIC HEAT OPTION

Electricity may be used as a heat source either by installing a central electric boiler and hot water terminal coils or individual electric resistance heating coils in the terminal units.

One design approach is to size the electric terminal resistance heaters for the peak winter heating load and to operate the chilled water system as a non-changeover cooling-only system, thus avoiding the problems of hot water/chilled water changeover. One disadvantage of this system is the wide swing in supply air temperature that will result if the electric coil is cycled instead of modulated.

Another approach is to use a small electric resistance terminal heater for the intermediate season heating requirements in conjunction with a two-pipe changeover-type chilled water/hot water system. The electric heater is used with outdoor temperatures no lower than 30 or 40°F [-1 or 4°C]. System or zone reheating of the primary air, as required for a two-pipe system, is greatly reduced or eliminated. Changeover to hot water occurs at lower outdoor temperatures and is thus limited to a few times per year. Simultaneous heating/cooling capacity is available, except in extremely cold weather when little cooling is required.

7.8 CLOSED LOOP WATER-SOURCE HEAT PUMPS

Many large buildings have substantial core areas that experience no transmission losses but do have internal heat gains. These internal areas require cooling year-round during occupied hours. Rather than reject the heat extracted from these core areas to the atmosphere, this heat can be used as a heat source for a heat pump system serving the perimeter of a building. A closed-loop heat pump system (Figure 7-14) can serve this purpose.

The closed water loop serves as a heat sink for the interior (cooling) heat pump units; it also serves as a heat source for water-to-water or water-to-air heat pumps used to heat the perimeter. This synergistic relationship increases the overall system coefficient of performance (COP). The temperature of the closed water loop is usually kept between 65 and 90°F [18 and 32°C], but the heat pump manufacturer should be consulted as to the range the selected unit can withstand. When the loop temperature drops below the lower limit, a boiler is used to raise it. When the higher limit is exceeded, heat is rejected from the loop via a cooling tower. No insulation is required for the water loop piping as long as the ambient temperature surrounding the piping is in the 60 to 80°F [16 to 27°C] range.

Fig. 7-14 Closed-Loop Water-to-Air Heat Pump System Schematic.

Water-source heat pumps require 2 to 3 gpm [0.13 to 0.19 L/s] in circulation per ton [3.5 kW] of refrigeration based upon the sum of the nominal tonnage of the units (usually much higher than the block load). This flow rate is generally constant during both cooling and heating modes. Since water must circulate continuously throughout the entire closed loop as long as a single unit is operating, the annual pumping costs for these systems can be high. To reduce unnecessary water circulation, control valves should be installed to shut off water flow to a heat pump when the compressor is off. When core areas need only minimal or no cooling during a significant portion of the day (e.g., apartment houses or hotels with infrequently used public areas), greater than anticipated use of the supplementary boiler to compensate for heat not being provided by the core units can reduce or eliminate the benefits of the looped heat pump system's high COP.

A closed loop heat pump system permits a landlord to wire each heat pump unit to a respective tenant's electric meter, thereby effectively charging each tenant in proportion to the tenant's use. This can be valuable when a tenant operates around-the-clock rooms with process loads or performs extensive night and weekend work. Performance comparisons of a water-source heat pump system with a fan-coil all-water system are given in Section 8.11. Frequently, the two are considered competitive system options.

For further information, refer to the chapter on Unitary Air Conditioners and Heat Pumps and the chapter on Applied Heat Pump and Heat Recovery Systems in the *ASHRAE Handbook—HVAC Systems and Equipment*.

7.9 REFERENCES

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Fig. 7-1 >>

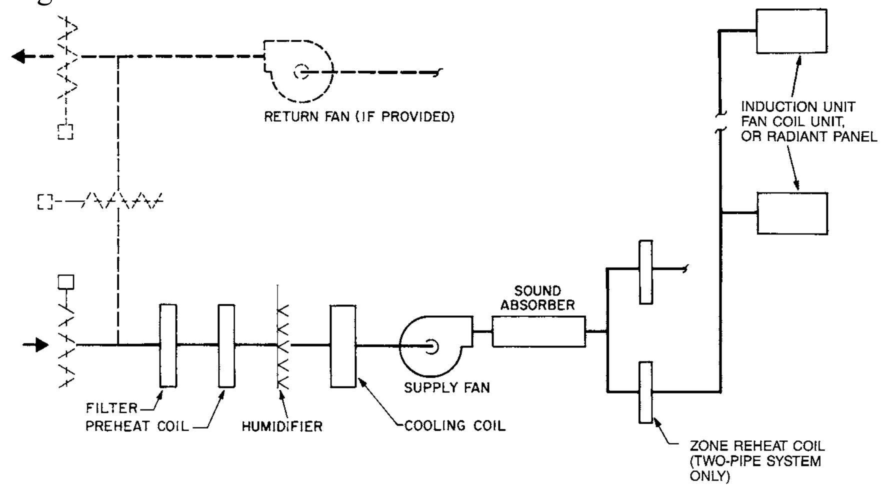

RETURN FAN (IF PROVIDED)
INDUCTION UNIT
FAN COIL UNIT,
OR RADIANT PANEL
SOUND
ABSORBER
SUPPLY FAN
FILTER
PREHEAT COIL
HUMIDIFIER
COOLING COIL
ZONE REHEAT COIL
(TWO-PIPE SYSTEM
ONLY)

Fig. 7-2 >>

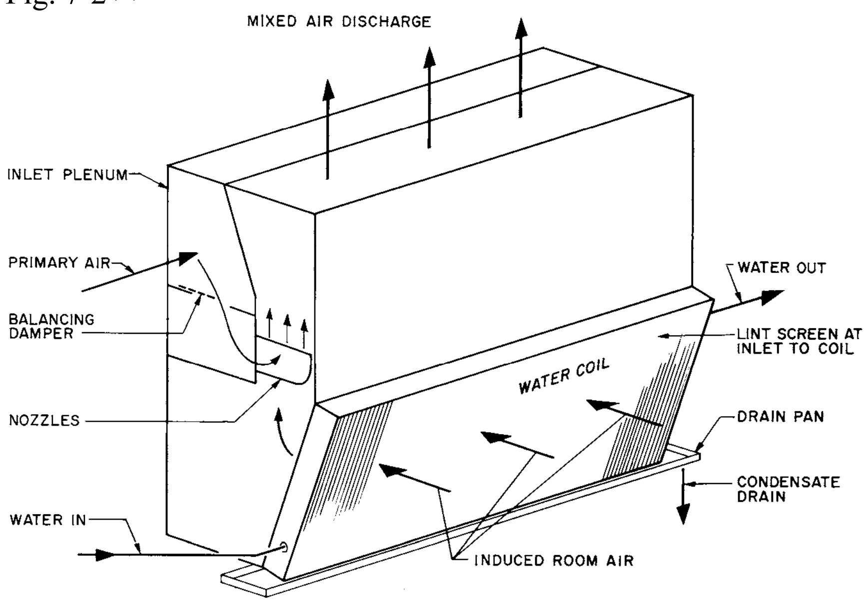
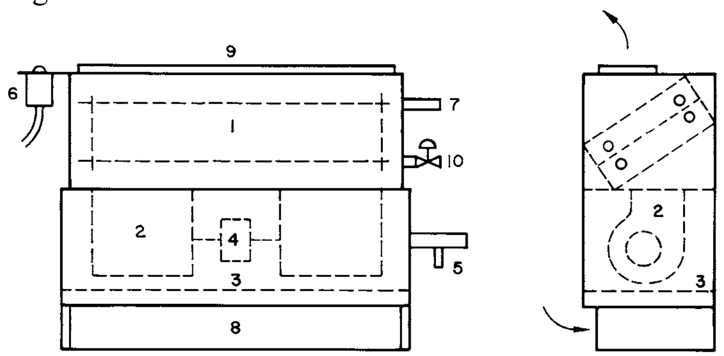

MIXED AIR DISCHARGE
INLET PLENUM
PRIMARY AIR
BALANCING
DAMPER
NOZZLES
WATER IN
WATER OUT
LINT SCREEN AT
INLET TO COIL
WATER COIL
DRAIN PAN
CONDENSATE
DRAIN
INDUCED ROOM AIR

Fig. 7-3 >>



1. FINNED TUBE COILS
2. FAN SCROLLS
3. FILTER
4. FAN MOTOR
5. AUXILIARY CONDENSATE PAN
6. FAN SPEED CONTROL SWITCH
7. COIL CONNECTIONS
8. RETURN AIR OPENING
9. DISCHARGE AIR OPENING
10. WATER CONTROL VALVES

Fig. 7-4 >>

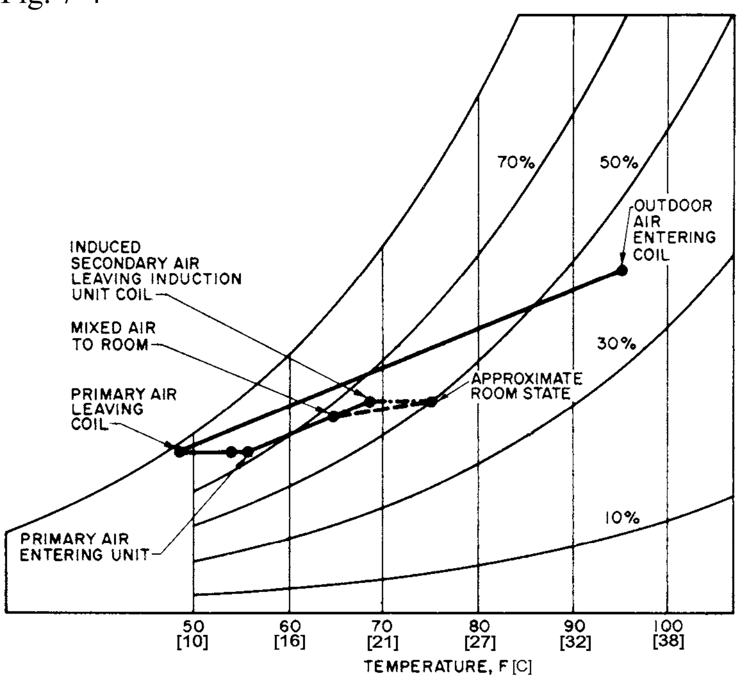
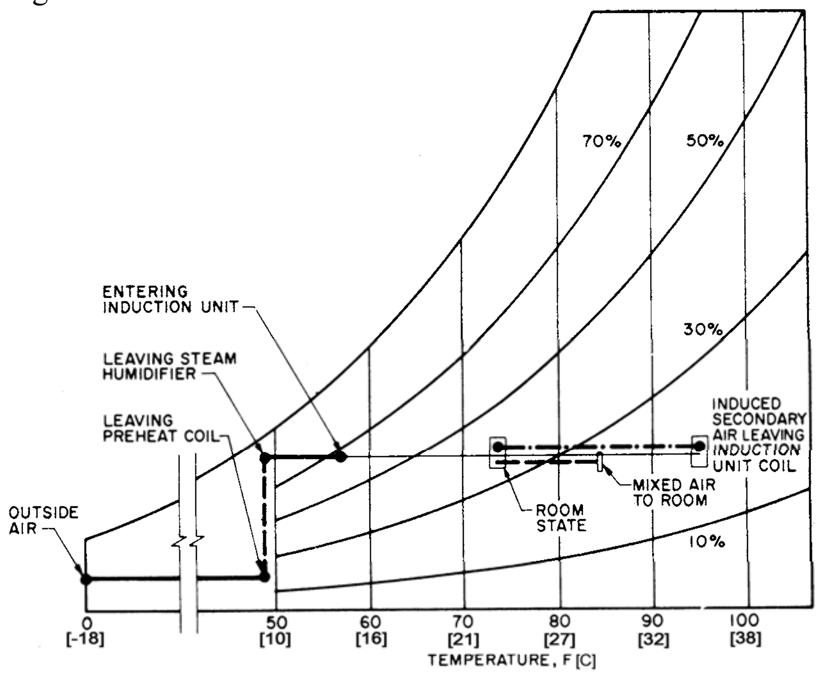
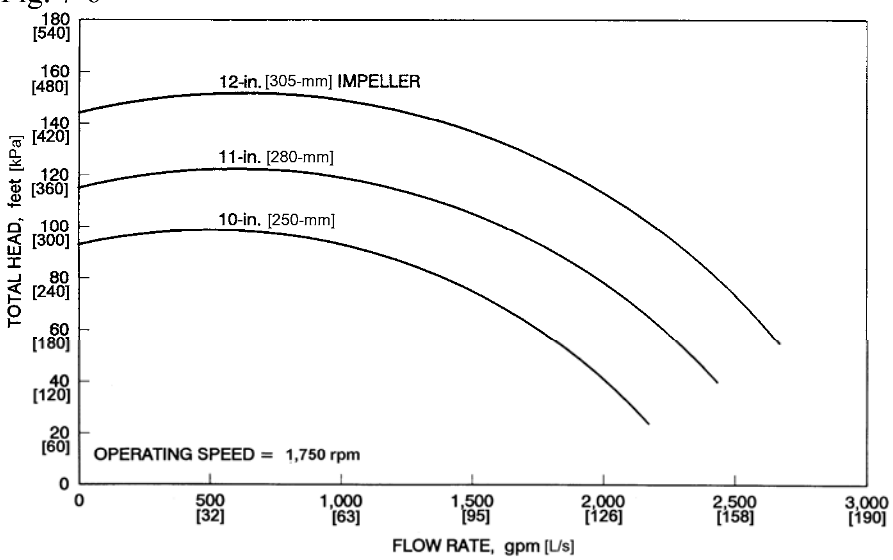


Fig. 7-5 >>



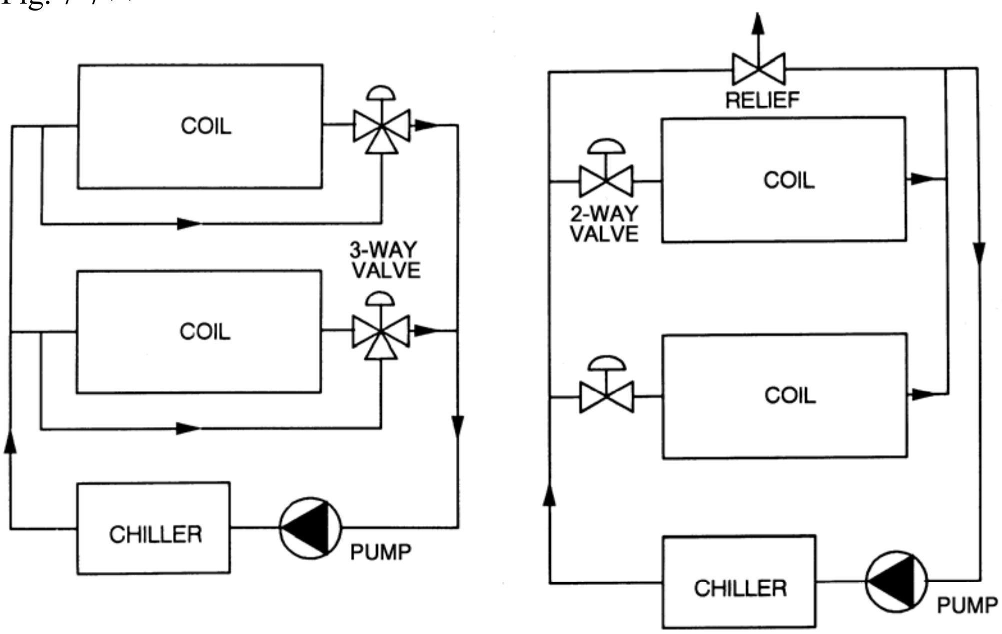
70%
50%
30%
10%
ENTERING INDUCTION UNIT
LEAVING STEAM HUMIDIFIER
LEAVING PREHEAT COIL
OUTSIDE AIR
INDUCED SECONDARY AIR LEAVING INDUCTION UNIT COIL
MIXED AIR TO ROOM
ROOM STATE
0 [-18]
50 [10]
60 [16]
70 [21]
80 [27]
90 [32]
100 [38]
TEMPERATURE, F [C]

Fig. 7-6 >>



12-in. [305-mm] IMPELLER
11-in. [280-mm]
10-in. [250-mm]
TOTAL HEAD, feet [kPa]
180 [540]
160 [480]
140 [420]
120 [360]
100 [300]
80 [240]
60 [180]
40 [120]
20 [60]
0
OPERATING SPEED = 1,750 rpm
0
500 [32]
1,000 [63]
1,500 [95]
2,000 [126]
2,500 [158]
3,000 [190]
FLOW RATE, gpm [L/s]

Fig. 7-7 >>



A Direct return with three-way valves B Reverse return with two-way valves

Fig. 7-8 >>

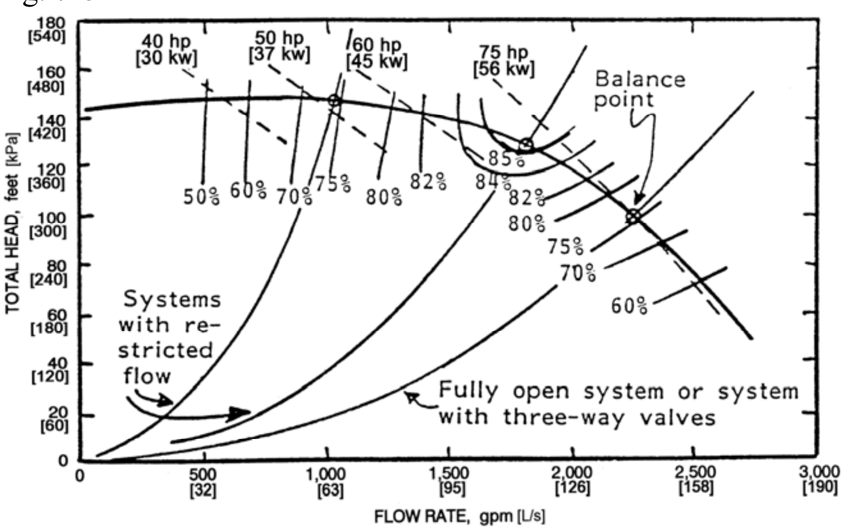


Fig. 7-9 >>

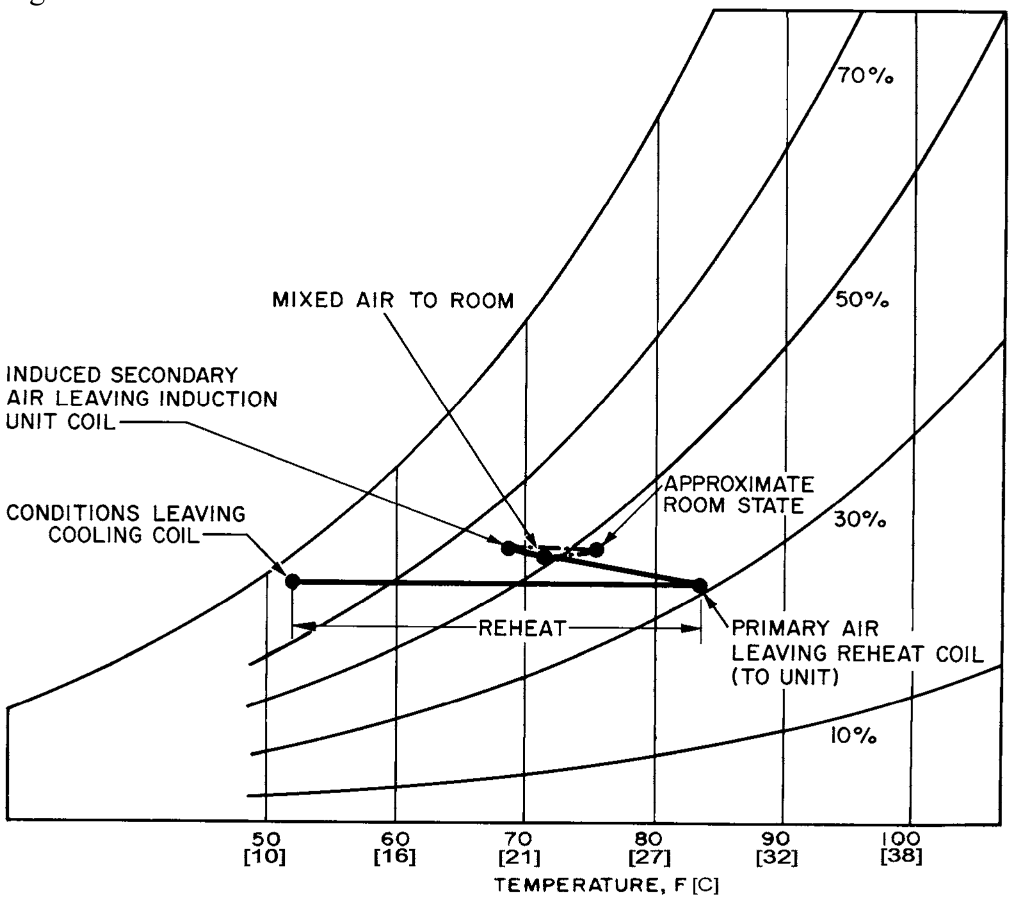

70%
50%
30%
10%
MIXED AIR TO ROOM
INDUCED SECONDARY AIR LEAVING INDUCTION UNIT COIL
APPROXIMATE ROOM STATE
CONDITIONS LEAVING COOLING COIL
REHEAT
PRIMARY AIR LEAVING REHEAT COIL (TO UNIT)
50 [10]
60 [16]
70 [21]
80 [27]
90 [32]
100 [38]
TEMPERATURE, F [C]

Fig. 7-10 >>

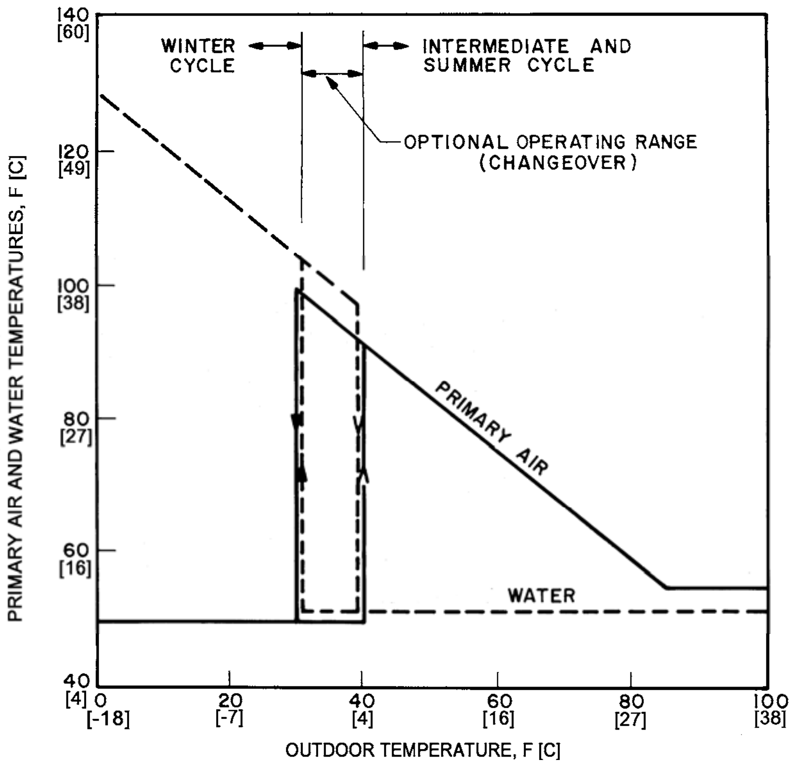


Fig. 7-11 >>

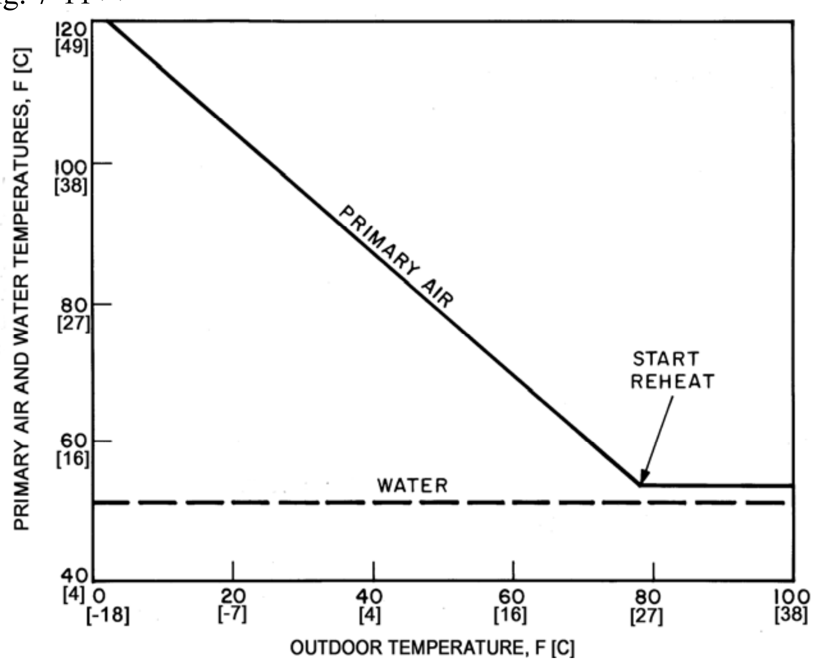


Fig. 7-12 >>

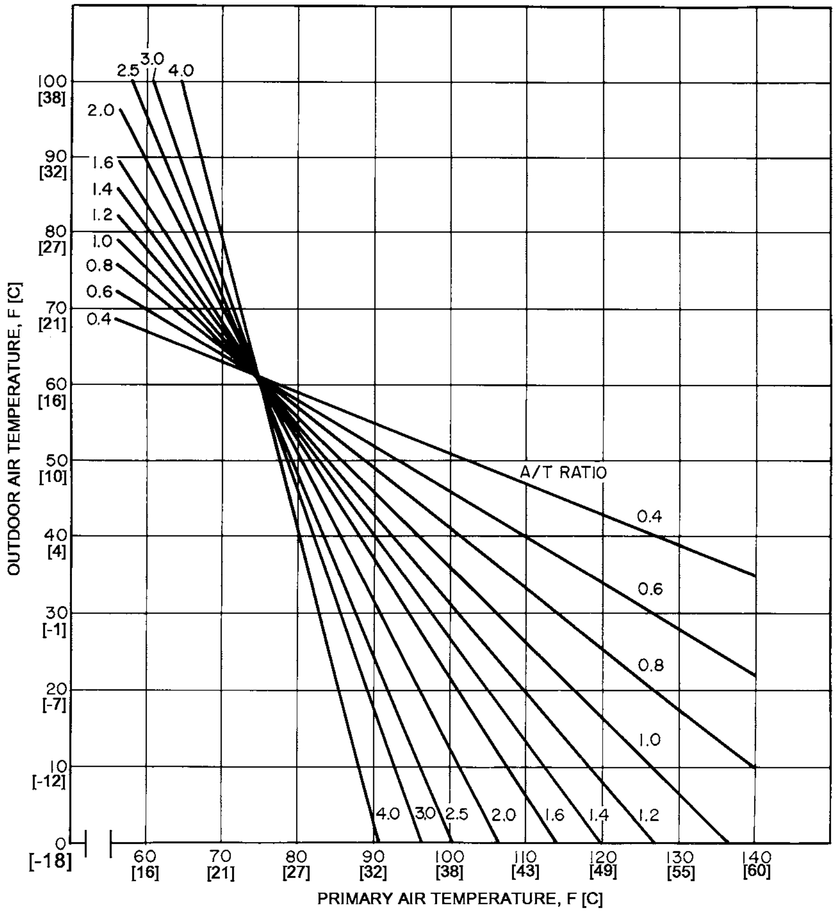
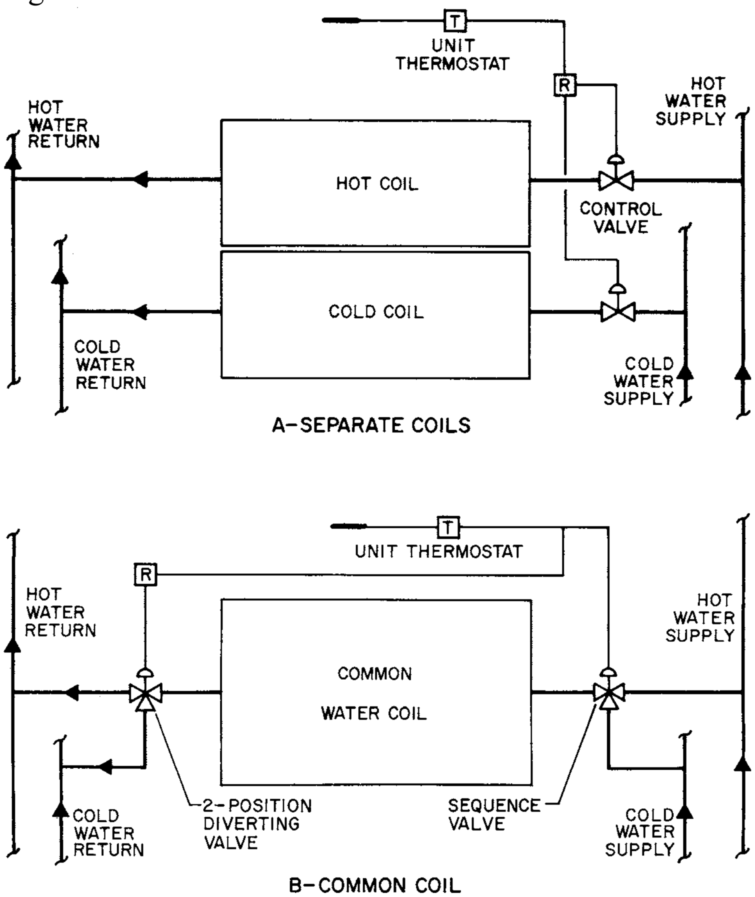

OUTDOOR AIR TEMPERATURE, F [C]
PRIMARY AIR TEMPERATURE, F [C]
A/T RATIO
100 [38]
90 [32]
80 [27]
70 [21]
60 [16]
50 [10]
40 [4]
30 [-1]
20 [-7]
10 [-12]
0 [-18]
60 [16]
70 [21]
80 [27]
90 [32]
100 [38]
110 [43]
120 [49]
130 [55]
140 [60]
0.4
0.6
0.8
1.0
1.2
1.4
1.6
2.0
2.5
3.0
4.0

Fig. 7-13 >>


UNIT
THERMOSTAT
HOT
WATER
RETURN
HOT
WATER
SUPPLY
HOT COIL
CONTROL
VALVE
COLD COIL
COLD
WATER
RETURN
COLD
WATER
SUPPLY
A-SEPARATE COILS
UNIT THERMOSTAT
HOT
WATER
RETURN
HOT
WATER
SUPPLY
COMMON
WATER COIL
2-POSITION
DIVERTING
VALVE
SEQUENCE
VALVE
COLD
WATER
RETURN
COLD
WATER
SUPPLY
B-COMMON COIL