Chapter 9
SPECIAL HVAC SYSTEMS

9.1 DESICCANT PRE-DRYING

9.1.1 System Description

Conventional air-conditioning systems address sensible cooling (lowering of air dry-bulb temperature) and latent cooling (dehumidification) simultaneously. Air is chilled below its dew point to remove moisture by condensation, frequently to a temperature below that required for sensible cooling. Using desiccant pre-drying, the sensible and latent air-conditioning functions are separated. A liquid or solid desiccant removes moisture from the air by absorption or adsorption, and conventional refrigeration equipment then cools air to a coil-leaving temperature (55 to 58°F [12.8 to 14.4°C]) that will meet sensible loads and is usually higher than that required for latent cooling, which results in a higher chiller COP.

The principal benefit of desiccant drying is that it can reduce refrigeration load (and associated electrical demand) by eliminating the need to chill air below its dew point solely to handle latent loads. As a result, desiccant pre-drying makes possible a reduction in air-conditioning energy costs. A careful analysis of actual system operation and economics is required to ascertain if total energy consumption is reduced along with energy costs. The pattern of energy use in desiccant cooling processes is shifted in such as way as to generally reduce energy costs, but a reduction of total energy consumption is not necessarily a given. The availability of low-grade heat for regeneration is a key to system economic viability (USDOE 1999). Handling sensible and latent cooling loads separately should enable the designer to better manipulate HVAC&R design variables and to select from a broader range of energy sources and temperatures to meet project design criteria. Other psychrometric processes, such as the use of heat pipes or hot gas reheat, might also be considered as options to reduce the energy demands of the sensible/latent air-conditioning process.

Desiccant dehumidification requires heat input to regenerate (dry) the desiccant and subsequent cooling to remove the heat released during sorption. Regeneration can occur at a relatively low temperature (as low as 120°F [49°C]) and may therefore use relatively inexpensive thermal energy sources, such as heat rejected from building process equipment, heat from cogeneration (combined heat and power: CHP), solar heat, and industrial waste heat. Cooling to remove the heat of sorption can be done at a relatively high temperature (up to 85°F [29.4°C]). Such cooling can be accomplished using non-refrigerated water from a cooling tower, from evaporatively-cooled building exhaust air, by well water, or by chilled water.

Figure 9-1 illustrates an air-and-water system with desiccant pre-drying. Outdoor air is dehumidified at the central plant (*A* to *B*) and distributed to terminals (*B* to *E*), where it is mixed with recirculated air and supplied to the occupied space (*E* to *G*). Outdoor air (*A*) is heated (not shown) and used to regenerate the desiccant. The desiccant drying process follows lines of constant enthalpy. Exhaust air (*I* to *J*) cools the reconstituted desiccant (*J* to *K*) and is then discarded.

Fig. 9-1 Terminal Recirculation Cooling with Central Desiccant Dehumidification in an Air-and-Water System.

9.1.2 Applications

Installation of desiccant pre-drying systems is most economical when the HVAC system is configured as an air-and-water system in which the quantity of dry primary air is limited to the outdoor air required for ventilation, e.g., 0.1 to 0.2 cfm/ft^2 [0.5 to 1.0 L/s per m^2]. This air should be dried to about 30 to 50 gr/lb (grains of moisture per pound of dry air), which is approximately 0.0045 to 0.0070 lb of water per lb of dry air [4.3 to 7.2 g/kg]. This low level of humidity in the supply air provides all the latent cooling required. Fan-coil units, unitary heat pumps, or induction units can then provide sensible cooling without the need for condensate drains at the terminals. In large buildings with long air distribution runs, the construction costs for the resulting smaller duct system can be substantially less than those for an all-air system. When combined with the savings from use of a smaller refrigeration unit, duct cost savings can, to a large extent, offset the cost of the desiccant equipment.

A conventional cooling system can achieve dewpoint conditions of about 50°F [10^{o}C]. In spaces requiring lower dewpoint conditions, desiccant dehumidification will be required. Desiccant pre-drying can also be cost-effective in an all-air system where: (1) 100% outdoor air is required, as in restaurants and certain laboratories, or (2) a special humidity control problem exists, as in pharmaceutical or electronic manufacturing areas. Another application of desiccants is in enthalpy wheels that are used effectively and economically in air-to-air heat exchangers. For further information on these applications see the chapter on Desiccant Dehumidification and Pressure Drying Equipment and the chapter on Air-to-Air Energy Recovery in the *ASHRAE Handbook—Systems and Equipment*.

9.1.3 Desiccants

Desiccants used in HVAC systems are of two types—solid or liquid. Solids include lithium chloride, molecular sieve, silica gel, and alumina gel. Commonly used liquids include aqueous solutions of either lithium chloride or triethylene glycol. In general, solid desiccants *adsorb* water on the surface of the desiccant, while liquid desiccants *absorb* water by chemically combining with it. In solid desiccant systems, the temperature rises in the dehumidification chamber as the sorption process releases heat (adiabatic dehumidification); the air being conditioned is precooled, aftercooled, or both. In liquid desiccant systems, cooling to remove the heat of sorption occurs as the heat is released in the dehumidification chamber (isothermal dehumidification).

Three dehumidification processes are compared schematically and psychrometrically in Figure 9-2. In the condensation dehumidification system (Figure 9-2a), a conventional refrigeration system provides simultaneous dehumidification and sensible cooling (*A* to *B* in the psychrometric diagram). The air temperature must then be raised by reheat or by mixing with warmer room air to achieve the desired supply air temperature (*B* to *C*).

In the desiccant pre-drying systems (Figures 9-2b and 9-2c), nonrefrigerated water from a cooling tower removes the heat of sorption, and the subsequent sensible cooling affects the supply air temperature only. In the solid desiccant system (Figure 9-2b), incoming air is dehumidified by passing it through a wheel that contains the solid desiccant (*A* to *B*) and is then sensibly cooled by nonrefrigerated cooling to remove the heat of sorption (*B* to *C*), followed by refrigeration cooling (*C* to *D*). The exhaust air is heated and used to regenerate the desiccant embedded in the slowly rotating wheel before being dumped. At any time, one-half (or more) of the desiccant wheel dehumidifies the incoming air, while the other half (or less) is being regenerated by the heated exhaust air.

In the liquid desiccant system (Figure 9-2c), incoming air is simultaneously precooled and dehumidified in the dehumidification chamber (*A* to *B*), followed by sensible cooling (*B* to *C*). This approach involves two chambers—a conditioner (dehumidifier) and a regenerator (concentrator). In the dehumidifier, moisture is absorbed from the incoming air by a spray of cool desiccant. Heat is removed by water from a cooling tower. The moisture-laden desiccant circulates to the regenerator, where it is heated and sprayed into a scavenging exhaust air stream, thus removing its moisture. The scavenger air is then dumped to the outside.

The theory of absorption, adsorption, and desorption of water by desiccants is explained in the chapter on Sorbents and Desiccants of the *ASHRAE Handbook—Fundamentals*. Refer to Appendix E for further information on dehumidification options, including desiccant systems.

Fig. 9-2 Dehumidification Process Options: (a) Dehumidification by Condensation; (b) Solid Desiccant Dehumidification; (c) Liquid Desiccant Dehumidification.

9.2 THERMAL STORAGE

9.2.1 General

Energy efficiency can involve using less energy, but it can also mean using energy more effectively. Thermal storage allows cooling to be produced at night for use the following day. This improves efficiency for utilities because it allows them to provide more energy with the same generating capacity. The resulting lower cost of utility operation can be passed on to customers in the form of lower rates. See Section 2.8.5 for a discussion of utility rates. Thermal storage, as such, is not an energy efficiency measure but rather a cost-saving technique. It can save the building operator money in several ways:

■ By decreasing or eliminating chiller operation during utility peak periods, demand charges are reduced.

- By displacing energy use from peak to off-peak periods, a lower energy charge may be incurred.

- Many electric utilities promote thermal storage by offering an incentive for displaced power, which can amount to several hundred dollars for each kilowatt moved from peak to off-peak demand.

It is wise to investigate whether thermal storage can be advantageous for a particular project. Among the applications most favorable to thermal storage are buildings with high cooling demands of short duration, additions to existing buildings, and projects in the service area of utilities with generation capacity problems and, consequently, high demand charges. For a more detailed discussion of thermal storage, see the chapter on Thermal Storage in the *ASHRAE Handbook—HVAC Applications*.

9.2.2 Design-Day Cooling Load

With thermal storage, air-conditioning system cooling loads require a different method of analysis. In systems without thermal storage, only the maximum instantaneous cooling load is calculated for system sizing purposes. Capacity-control devices then allow the refrigeration plant to follow the building load profile during periods of lower demand. No analysis of daily cooling loads is necessary (unless for energy consumption estimates). With thermal storage systems, however, the cooling requirement for every hour of the design day must be calculated. In Figure 9-3 for example, the base load is seen to vary from 13 tons [46 kW] at 8 AM to 20 tons [70 kW] between noon and 6 PM. After some lighting and air handlers are turned off, the load drops to 10 tons [35 kW] until the cooling systems are shut off at 8 PM. At this point, the daily load would appear to integrate to 205 ton-hours [720 kW-hours].

Fig. 9-3 Variation of Cooling Load during the Day.

It would be a mistake, however, to assume that this represents the entire cooling load. In reality, there are continuing heat gains of two types. One originates from continuously-operating heat sources such as emergency lighting and plug loads such as computers. The other derives from thermal mass (concrete floors, walls, and furniture) which absorbs radiant heat from lighting and solar radiation and releases it during unoccupied periods. These nighttime heat gains must be removed from the building prior to occupancy. They are usually handled with a precooling cycle that starts the cooling plant before occupants arrive. In this example, the aggregate of these heat gains is calculated to be 35 ton-hours [123 kW-hours]. The total daily cooling load is therefore 205 + 35 or 240 ton-hours [720 + 123 = 843 kW-hours]. Only now can the designer begin to select a thermal storage solution.

9.2.3 Design of Storage and Cooling Plant

A conventional refrigeration system requires a unit with sufficient capacity to handle the largest cooling load and a capacity control arrangement to handle all lesser loads. From

Figure 9-4, a 20-ton [70 kW] unit starting at 6 AM and following the cooling load profile would manage the load until the refrigeration system is shut off at 8 PM. Each square under the cooling load profile represents one ton-hour [3.5 kW-hour]. By counting squares, it can be seen that the cooling system must provide 240 ton-hours [843 kW-hours] during the day. A 20-ton [70 kW] capacity unit operating over 14 hours is one solution. A 10-ton [35 kW] unit operating through 24 hours, however, can also provide the needed capacity.

Fig. 9-4 Cooling Equipment Operation Options to Match the Instantaneous Load of Figure 9-3.

Trying to cover the daily cooling load with a 10-ton [35 kW] unit requires 100 ton-hours [350 kW-hours] of stored cooling between 6 AM and 6 PM to help the smaller size cooling unit deal with the load. This new distribution of cooling capacity may not be the least expensive, but it accomplishes a useful function by reducing electric demand, since a 10-ton [35 kW] unit will use less power during the time of peak demand than a 20-ton [70 kW] unit. This system is referred to as *partial storage* because it is only large enough to help an under-peak-sized cooling unit get through the day.

If a refrigeration plant needs to be shut down entirely during the daytime to avoid high demand charges, the storage must be enlarged to handle the entire cooling load, as shown in Figure 9-5. Now the system is referred to as *full storage* because it can store the entire daily cooling demand of 240 ton-hours [843 kW-hours]. If the electric utility's peak period lasts from 6 AM until 8 PM, 10 hours are left for the cooling unit to generate the full storage. In this case, the refrigeration unit would need a capacity of 240 ton-hours/10 hours, which equals 24 tons [843 kW-hours/10 hours = 85 kW]. In this case, which is typical, full storage is costlier to install than partial storage because both the refrigeration plant and the storage are larger.

Fig. 9-5 Cooling Equipment Off-Peak Operation to Satisfy Load of Figure 9-3.

Storage is particularly viable when serving cooling loads of short duration. Figure 9-6 illustrates the load profile for a church that requires 300 ton-hours [1055 kW-hours] of cooling for the entire week. Conventionally, this would require a 75-ton [265 kW] unit operating for 4 hours on Sunday morning. If storage were provided, a two-ton [7 kW] ice builder operating continuously could serve the load with 36 ton-hours [125 kW-hours] remaining for storage losses from the ice/chilled water tank (Figure 9-7). This solution not only reduces electric demand but may reduce initial cost as well. Also, consider that the three-phase service required to drive a 75-ton [265 kW] unit may not be available in some neighborhoods.

Fig. 9-6 Typical Cooling Load Profile for a Church.

Fig. 9-7 Thermal Storage Installed in a Church.

Another situation might involve a coliseum that can be maintained in the comfort zone without people or lighting using 200 tons [700 kW] of cooling but which requires 2000 tons [7035 kW] operating for 4 hours to serve 25,000 people during concerts (Figure 9-8). The conventional approach would involve the purchase of 2000 tons [7035 kW] of refrigeration to operate 4 hours per day at full load and 20 hours per day at 10% of capacity. A thermal storage approach would manage the same 12,000-ton-hour [42,200 kW-hour] daily requirement less expensively with a 500-ton [1760 kW] cooling plant operating around the clock.

Fig 9-8 Typical Cooling Load for Sports Arena.

9.2.4 Sizing and Location of Storage Tanks

Storage tanks of 50,000 gallons [190,000 L] or less are usually constructed of steel or reinforced plastic. Larger tanks are generally constructed more inexpensively of concrete. The cost of the tank is an important consideration for water storage systems because of the large volume required. Storage volume is a function of the amount of cooling to be stored. This often works out to be between 0.5 and 1.0 gal per ft^2 [20 and 40 L per m^2] of conditioned space for water storage, and to about 25% of that capacity for ice storage. Because of its bulk and weight, storage is frequently located in or under a basement or next to a building. On the other hand, "topside" storage eliminates the energy required to transfer chilled water from open storage containers into basement piping circuits pressurized by the static head of the building (see the chapter on Thermal Storage in the *ASHRAE Handbook—HVAC Applications*).

9.2.5 Chilled Water Storage Circuitry and Control

Figure 9-9 shows the elements of an appropriate control system for chilled-water storage. Once the demand limiter has been set for the chiller, it provides chilled water at a constant temperature but at some fraction of full-capacity flow. The demand limiting device forces the control valve to shift more flow to bypass. This reduces the inlet temperature to the chiller and causes it to operate using less energy. The balance of the chilled water required by the load is drawn from storage. System demand for chilled water is based upon the position of throttling valves at the coils. These, in turn, may be controlled by thermostats in the air stream leaving the coils. The total flow is regulated by a pressurestat, which controls the pump with a variable-speed motor or bypass control. In some cases, there may be an override control to maintain the temperature of water returning to storage. As shown, the controls maintain the design supply and return water temperature to ensure that the storage retains its rated capacity. The pressure-sustaining valve should be controlled from a pressurestat located at the high point of the system and set for some minimum positive pressure, such as 5 psi [240 Pa].

Fig. 9-9 Control Schematic for Typical Chilled Water Storage System.

9.2.6 Ice Storage Concepts

The idea of chilling water and storing it in tanks is straightforward. By comparison, techniques for ice storage are more varied and complex. One widely applied ice storage system consists of refrigerated pipe coils submerged in a water tank. During charging, ice builds on the coils, and during discharge, chilled water circulating through the tank melts the ice. Another type of ice storage module is made so that it can be frozen and thawed by circulating brine through coils in cylindrical water tanks. Advantages claimed for this technique include the fact that melting and refreezing always take place adjacent to the piping instead of remote from it. Other types of ice storage consist of harvester or shucker ice packages, which generate ice to be stored in an insulated ice bin. Ice is frozen to a thickness of approximately 3/16 in. [5 mm] on refrigeration plates that are up to 7 ft [2.1 m] long. A defrost cycle releases the ice, which then drops into a storage tank. As far as the refrigeration cycle is concerned, this type of equipment can be packaged in sizes up to 400 tons [1400 kW]. The tank can be of any size to suit the process. Illustrations and operating cycles for these and other ice storage types are found in the chapter on Thermal Storage in the *ASHRAE Handbook—HVAC Applications*.

9.2.7 Economic Analysis of Storage Options

Analysis of the design-day cooling load profile shown in previous sections is sufficient to determine tentative values for storage and cooling plant sizing, as illustrated in the examples given above. A more thorough analysis, however, is required to determine the economic feasibility of a thermal storage project. This is preferably done using hour-by-hour analysis over an entire year, taking into account the applicable electric utility rates. As a minimum, the analysis must include consideration of a representative day for each of the four seasons because optimum use of cool storage requires a strategy to minimize the electric demand of the chiller during peak periods. Therefore, storage operation varies from month to month as a function of the daily cooling load profile and the monthly utility billing demand, which is not necessarily identical to the actual monthly peak demand. The entire electric demand of a building (not just the cooling plant demand) must be included in economic calculations to obtain the optimum tradeoff point between chiller and storage size.

For example, Figure 9-10 illustrates a 500-ton [1760 kW] chiller managing a 12,000-ton-hour [42,200 kW-hour] cooling load on the hottest day of the year. The chiller and 7,000 ton-hours [24,620 kW-hours] of storage are both being used to full capacity. In October, the maximum cooling day requires only 9,000 ton-hours [31,650 kW-hours] (Figure 9-11). Thus, if the storage is used to its full extent, the chiller can be limited to only 200 tons [700 kW] of input capacity. It would be normal to leave the chiller at the 200-ton [700 kW] setting for other October days with lesser cooling requirements because the chiller demand has already been created, and the more cooling that is generated and directed to load, the less loss through storage. For these lower-load days, the operator would precharge the storage only as much as needed to get through the day, since partial precharging helps to limit storage losses. An exception to this procedure would involve daily use of the entire storage when off-peak rates were sufficiently lower (by, for example, 2¢/kWh or more) to overcome the penalty of transfer pumping energy.

In January, this partial storage system may be operated as full storage since the maximum cooling demand may be less than can be generated entirely off peak (Figure 9-12).

Fig. 9-10 500-ton [1760 kW] Chiller and Partial Storage Satisfying 12,000-ton-hour [42,200 kW-hour] Demand on Summer Design Day.

Fig. 9-11 500-ton [1760 kW] Chiller and Partial Storage Operating on Autumn Day.

Fig. 9-12 500-ton [1760 kW] Chiller Operating with Full Storage on Winter Day.

Some designers combine ice storage with a cold air distribution system (see Section 9.3.8) to take advantage of the lower air temperature available from ice storage. Bhansali and Hittle (1990) compared a variety of VAV systems with economizer cycles in five climates under four different utility tariffs. In spite of additional energy use (20 to 45%), the combination of ice storage and cold air resulted in lower annual utility costs than ice storage with conventional-temperature air, or regular chiller operation with conventional air or with cold air, for all cases investigated. Chillers will operate at a lower COP (higher kW/ton [kW/kW]) while making ice because of the required lower supply temperatures.

9.3 ENERGY-EFFICIENT SUBSYSTEMS

9.3.1 Introduction

In earlier times of plentiful and cheap energy, HVAC designers found it expedient to plan systems with generous air circulation rates to maintain a good sense of air movement for occupants. They also provided generous ventilation rates for the dilution of airborne contaminants. Less thought was given to achieving these effects with the least possible energy use because energy costs constituted a relatively small part of a building's operating budget. The energy disruptions of the 1970s led to a fundamental reappraisal of the role of energy use in buildings and how to design new systems to minimize energy use.

Pre-energy-crisis commercial buildings were found to be consuming anywhere from 60,000 to 300,000 Btu/ft^2 yr [682,000 to 3,408,000 kJ/m^2 yr] with an average of, perhaps, 160,000 Btu/ft^2 yr [1,818,000 kJ/m^2 yr]. Pursuit of energy-efficiency modifications subsequent to the 1970s reduced this to around 40,000 to 150,000 Btu/ft^2 yr [454,400 to 1,704,000 kJ/m^2 yr]. New buildings are typically targeted in the range of 30,000 to 85,000 Btu/ft^2 yr [341,000 to 956,000 kJ/m^2 yr] depending upon building type and design intent. The specifics of building energy use are climate-, function-, and design-specific and are fairly dynamic. Reductions in building energy use seem to have leveled off recently, and substantial improvements in lighting efficiency have been supplanted by increased plug loads. In any event, HVAC&R systems are responsible for a good percentage of building energy consumption—accounting for perhaps 40% of primary energy consumption. To understand the design philosophy of energy efficiency, it is useful to examine what was wasteful about older designs, what has been done to improve existing systems, and what is considered appropriate for new systems.

It is energy-efficient to allow the space temperature to rise above design conditions in the summer and float downward in the winter during unoccupied hours. This may be done simply by resetting the zone thermostat (or temperature sensor) automatically or manually. When there are a number of zones in a building, however, some of which are occupied while others are unoccupied, a more complex control system is required. Modular equipment, variable-speed fans and pumps, reset controls for hot and chilled water, zone control valves or mixing boxes, and unitary equipment are options to accommodate varying occupancy profiles or varying loads due to the exterior environment or building use.

9.3.2 Simple Systems

Many constant-volume systems were set up to both heat and cool a zone (Figure 9-13). Cooling could be provided totally by outdoor air when the exterior air temperature was below 55°F [12.8°C], with help from mechanical refrigeration between 55 and 70°F [12.8 and 21.1°C] and by refrigeration alone above 70°F [21.1°C]. While this system offered comfort for occupants, it wasted heating energy when the room conditions were such that 55°F [12.8°C] air proved to be too cold and the heating coil was activated to increase the supply air temperature to maintain comfort—thus, this system arrangement is no longer permitted by most energy codes. Figure 9-14 illustrates how a room thermostat can be modified to control the mixed air directly. This avoids overcooling the zone and eliminates the parasitic heating inherent in the system control depicted in Figure 9-13.

Fig. 9-13 Schematic of Conventional HVAC System.

Fig. 9-14 HVAC System Modified to Eliminate Parasitic Heating.

9.3.3 Discriminating Temperature Controls

In Figure 9-15, a ductstat (a thermostat located inside a duct) sequences outdoor air (economizer operation) and then mechanical cooling to maintain a low, base supply air temperature to a reheat system serving more than one zone. Reheat coils respond to zone thermostats to increase this base supply air temperature, if required, to match the actual cooling requirement in each zone. The problem with this scheme is similar to that with the previous one—most of the time many of the zones will likely not require air as cold as the base supply air temperature, which was selected to meet a maximum zone design condition. A solution to reduce reheat use is to "discriminate" the supply air temperature to respond to feedback from the zone thermostats, as is seen in Figure 9-16. The supply air temperature will now be provided only as low as the temperature required for the zone with the greatest cooling load. Parasitic reheat will be a fraction of what it was with a fixed supply temperature control

This concept of discriminating controls applies equally well to most basic HVAC systems. For example, multi-zone deck temperatures need be only as high or as low as demanded by the greatest zone requirement for heating or cooling. The hot and cold supply temperatures of a double-duct system need only deviate from room temperature as required by the greatest call for heating and cooling—a situation usually marked by an extreme valve or damper position in one of the many zones. The air and water temperatures for all systems should be responsive to feedback from zone controls so that the least possible energy is required to provide comfort. In this way, flexible zone temperature control can be provided with a minimum of parasitic heating and cooling.

Fig. 9-15 Conventional Temperature Control.

Fig. 9-16 Discriminating Temperature Control to Minimize Reheat.

9.3.4 Eliminating Energy Waste in Simple Systems

An ideal HVAC&R system would use the least possible energy for air and water circulation and generate only the amount of heating and cooling necessary to offset space heat losses and gains. Reheat energy for temperature control, used extensively in the past, would be eliminated because it adds to both heating and cooling requirements. How can such an ideal system be constituted?

First, the amount of air and water circulated should be only that required, at any given instant, to convey the heating or cooling energy required to meet the loads of a space. This statement involves two cautions. First, it is wise to circulate adequate air to provide occupants with a sensation of air movement—a minimum of 2 air changes per hour is often considered appropriate. Second, fresh air as specified by ASHRAE Standard 62.1 (or 62.2) must be provided to ensure acceptable indoor air quality. Above these comfort and IAQ minimum air changes, circulation rates will vary to match fluctuating loads. Second, the heat generated must be only enough to serve the aggregate zone heating loads, and the cooling generated must be only enough to serve the aggregate zone cooling loads. If these concepts are applied to the simple system shown in Figure 9-13, the zone thermostat would reduce the volume of air being circulated as the zone temperature is satisfied. This would optimize fan energy consumption. When the minimum acceptable air change value is reached, the thermostat would have the ability to adjust the supply air conditions so that the space is not overcooled. Care needs to be applied to be sure that the adjusted supply air conditions can control both the space temperature and humidity in the summer.

Designers may recognize a flaw in the above because air that is not cooled to around 55°F [12.8°C] may carry too much moisture to be able to maintain a desired space relative humidity of 55% or less. Figure 9-17 illustrates control modifications that act to conserve fan energy while preserving minimum air circulation and a satisfactory relative humidity in a space. The return air bypass ensures that no moisture-laden outdoor air circumvents the cooling coil. Variable supply airflow allows reductions in fan energy down to a defined minimum rate of air circulation. In this scenario, mechanical cooling would be initiated when the outdoor air temperature rose to above 55°F [12.8°C]. As space temperature drops below setpoint, the thermostat would first throttle the volume of 55°F [12.8°C] air to the minimum setting and then raise the supply air temperature by opening the return air bypass. The air mixing and bypass air dampers must be provided with the minimum outdoor air volume for acceptable indoor air quality.

Fig. 9-17 Control Modifications for Energy Efficiency.

9.3.5 Eliminating Reheat in Multi-Zone Systems

Historically, the multi-zone HVAC system has wasted substantial energy. In many cases, deck temperatures were permanently set to satisfy the greatest winter heating requirement and the greatest summer cooling requirement. Automatic discrimination of deck temperatures to suit real-time zone demands can do much to reduce parasitic losses related to mixing of air streams at unnecessary temperatures, but total elimination of cooling and heating waste can only be achieved by adding a bypass deck to each zone, as shown in Figure 9-18. The zones are now able to modify the deck temperature to suit the requirement of the highest zone demand. In addition, the bypass deck eliminates the mixing of hot and cold air. The zone thermostat can call, in sequence, for full hot air, a hot air and bypass mix, full bypass, a cold air and bypass mix, and full cold air.

Fig. 9-18 Modified Multi-Zone Unit.

9.3.6 Eliminating Parasitic Heating in VAV Systems

In the past, VAV systems have saved fan energy but not without countervailing side effects. When the supply of 55°F [12.8°C] air to interior zones is reduced to match space cooling loads, the supply volume may be reduced below the minimum air change rate, leading to air quality complaints. An antidote, involving reheat of the minimum supply air volume, is also counterproductive with respect to energy efficiency. When 55°F [12.8°C] air is supplied to perimeter zones, it must be reheated to room temperature in winter before it is heated further to overcome envelope heat losses. The heat input to take air from 55°F [12.8°C] to room temperature is parasitic. One way to overcome this is to supply air that is thermostatically controlled for each solar exposure. At the least, this implies separate air supply zoning for interior, south-facing perimeter, and other perimeter zones. For compartmented air-handling systems, the air unit can be configured as shown in Figure 9-19.

Fig. 9-19 Two-Supply-Zone Air-Handling Unit.

Another popular solution is to add a fan to each air terminal, as shown in Figure 9-20. The side-pocket fan in the air terminal is sized to provide the desired minimum rate of air circulation. It is run only when the primary air supply falls below this critical value.

Fig. 9-20 Air Terminal with Side-Pocket Fan.

Reheating 55°F [12.8°C] air supply to perimeter zones can be avoided by ensuring that perimeter heat is not be activated until the cold air damper is at its minimum setting. Since the addition of fans in a plenum space means that maintenance of bearings and filters will occur in occupied spaces, the same effect can be achieved using one central fan for each floor or major zone. This central fan then recirculates air at a neutral temperature to dual-duct-type terminals connected jointly to the cold air supply and the neutral air supply. Perimeter air terminals can have optional heating coils, which are activated only after the cold air damper is at its minimum setting.

9.3.7 Economizer Cycle (Free Cooling)

Air-handling systems that have access to 100% outdoor air can provide full cooling without the assistance of mechanical refrigeration whenever the outdoor dry-bulb air temperature is lower than the required supply air temperature. Such an air-side economizer cycle (Figure 9-21), which is most effective in northern climates, is capable of saving up to 70% of mechanical refrigeration energy. In southern climates, such as seen in Florida, an air-side economizer is seldom used. This is because the number of hours (especially occupied building hours) during which the outdoor temperature falls below the controlled space temperature is insufficient to justify the investment in a relief fan, air-mixing chambers, and louvers necessary to dissipate the building pressurization caused by supplying 100% outdoor air during economizer operation.

Fig. 9-21 Air-Side Economizer Cycle.

Energy savings can be achieved with an air-side economizer cycle via the following sequences:

- The outdoor air temperature is lower than the supply air temperature required to meet the space-cooling load; compressors and chilled water pumps are turned off; and outdoor air, return air, and exhaust/relief air dampers are positioned to attain the required supply air temperature.

- The outdoor air temperature is higher than the required supply air temperature but is lower than the return air temperature; compressor and chilled water pumps are energized; and the dampers are positioned for 100% outdoor air. Outdoor air and mechanical cooling provide the desired supply air temperature.

- The outdoor air temperature exceeds the return air temperature (dry bulb economizer control), or the enthalpy of the outdoor air exceeds the enthalpy of the return air (enthalpy economizer control); the dampers are positioned to bring in the minimum outdoor air required for acceptable indoor air quality.

To be truly effective, control of air-side economizer cycles should not be based solely upon dry-bulb temperature conditions—but upon enthalpy, as illustrated in Figures 9-22 and 9-23 (Dubin and Long 1978). Enthalpy controls in the past have been difficult to keep in calibration because they must accurately sense temperature and humidity for optimum control. Because of this, many air-side economizer cycles have been scheduled to revert to a minimum setting for ventilation when the dry-bulb temperature reaches 80°F [26.7°C] (or an even lower value) , depending upon local conditions of temperature and humidity). However, recent advancements in the sensing of outdoor humidity have improved the reliability of enthalpy controls, and their usage has become more common.

Fig. 9-22 Dry-Bulb Temperature Economizer Cycle.

Fig. 9-23 Enthalpy Economizer Cycle.

Air-handling systems that lack reasonable potential for 100% outdoor air circulation may adopt a water-side-based winter free-cooling approach by interconnecting the chilled water circuit with the cooling tower. This can be done without a heat exchanger—but at the risk of pipe corrosion, more expensive water treatment, and eventual degradation of system components. The strainer shown in Figure 9-24 can effectively remove airborne dirt from the tower discharge, but oxygen, which is added to the water by the tower action, will be damaging to the chilled water piping circuit. This is generally an unacceptable risk, leading to use of a heat exchanger to separate the chilled water circuit from the cooling tower circuit. This adds first cost for the heat exchanger and reduces the effectiveness of water-side cooling because of the additional Δt imposed by the heat exchanger, but is generally a reasonable trade-off considering the potential effects of not using a heat exchanger. Water-side economizers will usually not prove as energy conserving as air-side economizers, in most climates.

Fig. 9-24 Water-Side Economizer Options

Another form of reduced-energy cooling involves purging conditioned areas with cool night air prior to occupancy the following morning. This can avoid the use of mechanical cooling energy to overcome the heat buildup from emergency/custodial lighting, 24-hour plug loads, and heat released to the conditioned spaces that was stored during the day (including solar radiation and radiant heat from lighting fixtures). This purging cycle is highly effective in dry climates with low nighttime temperatures, such as in the southwestern United States, and has been used successfully even in the Pacific Northwest (Ashley and Reynolds 1994).

9.3.8 Cold Air Supply Systems

Conventional HVAC systems are based upon a design supply air temperature of around 55°F [12.8°C] to ensure a relative humidity of 50% at a room air temperature of 75°F [23.9°F]. The advent of ice storage and the ever-increasing cost of air distribution have alerted designers to the potential of supplying less air at a lower temperature. Supply air temperatures can be obtained as close as 5°F [2.8°C] to the primary coolant temperature. In the case of water chillers, 40°F [4.5°C] water can provide 45°F [7.2°C] supply air. In the case of an ice-based storage system, the 36 to 38°F [2.2 to 3.3°C] coolant can produce air supply temperatures of 41 to 43°F [5.0 to 6.1°C]. Design information for supplying cold air in conjunction with ice storage is given in the chapter on Thermal Storage in the *ASHRAE Handbook—HVAC Applications.* Such cold supply air must be tempered with room air to prevent cold drafts. If this is done in fan-powered boxes that operate continuously, some of the energy savings are lost. Supply outlets that provide for substantial induction of room (secondary) air are an energy-efficient alternative. If the cold supply air causes system surfaces to be cooled below the ambient dew point, the surfaces must be insulated; this includes air handlers, ducts, and terminal boxes. Since leakage from cold air ducts aggravates any condensation problems, all such ducts should be reliably sealed. Condensation problems during system start-up (following shutdowns) can be reduced by lowering the supply temperature gradually.

According to the ASHRAE comfort chart (Figure 3-1), the lower relative humidity generated with cold air means that space dry-bulb temperature setpoints can be increased up to 1°F [0.6°C] for the same occupant comfort response. To ensure full cooling and dehumidification, face velocities at cooling coils supplying cold air should be in the 350- to 450-fpm [1.78 to 2.29 m/s] range, with an absolute maximum of 550 fpm [2.79 m/s] (Dorgan 1989). Some designers have suggested slightly higher upper limits for face velocities. In spaces with high sensible heat ratios (SHR), cold air systems may produce undesirably low relative humidities. A system to remedy this situation uses cold air to satisfy the latent cooling load and supplies additional sensible cooling through 55 to 60°F [12.8 to 15.6°C] chilled water coils at the mixing boxes. Operating a chiller to supply chilled water at that temperature is very efficient, and a need for additional chilled water piping can be eliminated by using the fire sprinkler piping (insulated to prevent condensation) to supply the chilled water coils. Meckler (1988) notes that this has been permitted by NFPA Standard 13, *Standard for the Installation of Sprinkler Systems*. The currency of this provision should be checked with the local authority having jurisdiction.

The use of temperatures lower than 55°F [12.8°C] for supply air permits the use of smaller ducts and fans. The reduced ceiling space required for smaller ducts can result in significant building height savings in high-rise buildings. Perform an analysis to select an appropriate discharge air temperature, based upon a comparative economic analysis of the entire building, not just the HVAC&R system. Such an analysis must consider first-cost savings from smaller equipment, a somewhat smaller air distribution system, and a potentially lower building height in high-rise applications; increases in first cost for a chiller that operates at a lower temperature; increased operating costs from powered mixing boxes; and reduced operating costs from lower air and water mass flow rates (Dorgan 1989).

9.3.9 Water-Side Efficiency Opportunities

Savings are available from variable flow pumping of chilled water and condenser water and variable speed tower fans. Water chillers can benefit from variable speed compressor motors and variable supply water temperatures when air-handling systems operate at partial loads. Chillers can also be designed to accept a lower condensing water temperature than the conventional 85°F [29.4°F], when available, to reduce the thermodynamic head on the compressor. Check with the chiller manufacturer to determine the lowest acceptable condenser water temperature and specify operating controls accordingly. The combination of compressor motor, condenser water pump motor, and tower fan motor has historically been assumed to require one kilowatt input for each ton [3.5 kW] of output. Variable temperature and variable speed controls enable this value to be virtually halved at part load conditions for typical cooling applications.

9.3.10 Bootstrap Heating

Another form of low-cost heating can be used in winter by operating a chiller to handle cooling loads in interior spaces and using the warm condenser water thus generated to offset perimeter heat losses. This is especially the case for a facility that includes high internal loads (often process-based), which requires cooling all year. It might seem counterproductive to use any form of "electrical" heating when other, less expensive fuels are available. If properly operated, however, a water chiller can act as a heat pump with a COP of up to 6. In this situation, heat reclaim may be less than half as expensive as the next cheapest fuel. Additional piping and controls are necessary to interface heating and cooling circuitry and to add additional condenser tubing for clean heat exchange. Consider these extra costs when conducting an analysis of savings available from bootstrapping waste interior heat. Heating coils need to be much larger, because hot water supply temperatures are much lower (105°F - 115°F) than with conventional heating systems. Potential savings must be estimated and incorporated into a life-cycle cost analysis that includes estimated additional costs.

A "heat-pump chiller" should be operated only to provide the cooling capacity actually required by the building spaces. If a chiller must be false-loaded at low capacities to generate adequate heat, much of the bootstrap system operating economy will be lost. Control of a heat-pump chiller must be compatible with an economizer cycle. In other words, the mixed air temperature (or enthalpy), which determines the percentage of outdoor air, should be set to permit the cooling coil to extract the heat required by the perimeter circuits. The balance of the cooling is then performed with outdoor air, while the heat pump operation boosts the heat removed from the cooling coil to a useful heating temperature.

Buildings have occupied and unoccupied periods. During each period, there is a balance point at some outdoor temperature where heat gains (including the electrical input to the heat-pump chiller) balance heat losses. Figures 9-25 and 9-26 indicate how such balance-point temperatures can be determined. They also illustrate the range of temperatures over which heat can be salvaged and usefully transferred and where supplementary heat may be required. The bulk heat that can be bootstrapped at low cost can be analyzed and aggregated with hour-by-hour or bin-type energy analysis programs. The result can then be compared with the additional investment in hardware required to implement the concept.

Fig. 9-25 Heat Balance in a Perimeter Space.

Fig. 9-26 Heat Gain and Heat Loss for an Interior Space.

Figure 9-27 illustrates a form of control that allows a heat-pump water chiller to draw heat from interior zone cooling coils to the extent required to satisfy heating loads.

Condensing water temperatures should be established in the 90 to 105°F [32 to 41°C] range for heating. Sufficient heating element surface needs to be provided so that such relatively low-temperature water can satisfy the space heating loads. This seldom creates an investment penalty when the coils also provide for cooling, as with induction units and fan-coil units.

Fig. 9-27 Control Schematic for Heat-Pump Water Chiller Using Heat from Interior Zones to Satisfy Heating Demands in Perimeter Zones.

The chiller size required for reclaim heating is often less than one-third the size required for cooling. This may validate a two-chiller selection, with a one-third-capacity chiller operating for heating and both units operating for maximum cooling. Centrifugal chillers experience some difficulty operating at low capacity with higher condensing temperatures. Thus, chillers for heat pump duty should be selected with care and kept as fully loaded as possible.

9.3.11 Heat Reclaim/Exchange

There are normally several opportunities for the exchange of heat from an exiting fluid (air or water) to an entering fluid (also air or water) in the typical building. Many of these opportunities are cost-effective and can improve building energy efficiency. Air-to-air heat exchangers (Section 9.4.2) are an example of a commonly encountered heat-reclaim system. See the chapter on Applied Heat Pump and Heat Recovery Systems in the *ASHRAE Handbook—HVAC Systems and Equipment* for examples of such systems and their associated equipment. An increasing designer and owner interest in combined heat and power (CHP) systems brings the potential for more opportunities for beneficial heat reclaim.

9.4 ”GREEN” HVAC SYSTEMS

A recent (and likely ongoing and expanding) interest in “green” HVAC&R systems (see, for example, the ASHRAE *GreenGuide*) has focused attention on a number of HVAC&R system strategies that are seen to support the intents and expectations of green design. Ground source heat pumps, air-to-air heat exchangers, and underfloor air distribution are among these strategies. Each of these strategies is addressed in general in the *ASHRAE Handbooks* and in detail in ASHRAE special publications. The purpose of this section is simply to outline these strategies.

9.4.1 Ground Source Heat Pump

A ground source heat pump (sometimes called a ground coupled, geothermal, or water source heat pump) takes advantage of the generally more benign climate below ground as compared to that above ground. Under cooling mode operation, heat is discharged to a ground loop or well that provides a lower temperature heat sink than ambient outdoor air temperature. During winter heating operation, heat is extracted from a source that is at a higher temperature than ambient outdoor air. Numerous green buildings have utilized some variation of a ground source heat pump to improve building energy performance. First cost is higher than for a conventional air source heat pump, but operating costs are lower due to improved COP values. See the chapter on Applied Heat Pumps and Heat Recovery Systems in the *ASHRAE Handbook—HVAC Systems and Equipment* for basic information on this strategy.

9.4.2 Air-to-Air Heat Exchangers

One means of improving the energy efficiency of an HVAC system is to capture the heat or cooling effect embodied in conditioned air being exhausted from a building. Often the easiest way to utilize such a resource is to pass the heat or cooling effect to incoming outdoor air (cold in the winter; hot in the summer). Both sensible and latent exchanges are possible depending upon the type of equipment selected. See the chapter on Air-to-Air Energy Recovery in the *ASHRAE Handbook—HVAC Systems and Equipment* for basic information on appropriate equipment and its integration.

9.4.3 Underfloor Air Distribution

Supplying conditioned air to a space via an underfloor supply plenum has captured the imagination of a number of HVAC designers working on green buildings. Several benefits are claimed for this distribution approach. Carefully-considered design, construction, and commissioning are critical to the successful operation of an underfloor air distribution system. The ASHRAE *Underfloor Air Distribution Design Guide* is recommended as a starting point for this green design strategy.

9.4.4 Green Systems and CO_2

Although the design of green building HVAC systems in the United States currently tends to focus upon the energy implications of system selection and operation, internationally there is increasing environmental concern about carbon dioxide (CO_2) emissions from buildings as a key contributor to global warming. The selection of HVAC&R systems and fuels can have a major impact upon such emissions, which are conceded by most scientists to be linked to global warming. This concern is likely to eventually migrate to the United States and exert a dramatic impact on HVAC&R system design. See the ASHRAE position statement on Climate Change for background information on this issue (ASHRAE 1999).

9.5 REFERENCES

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Fig. 9-1 >>

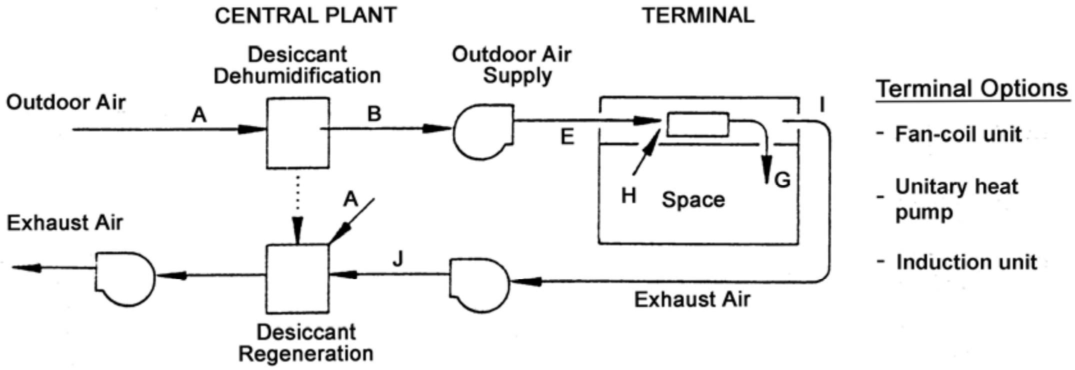
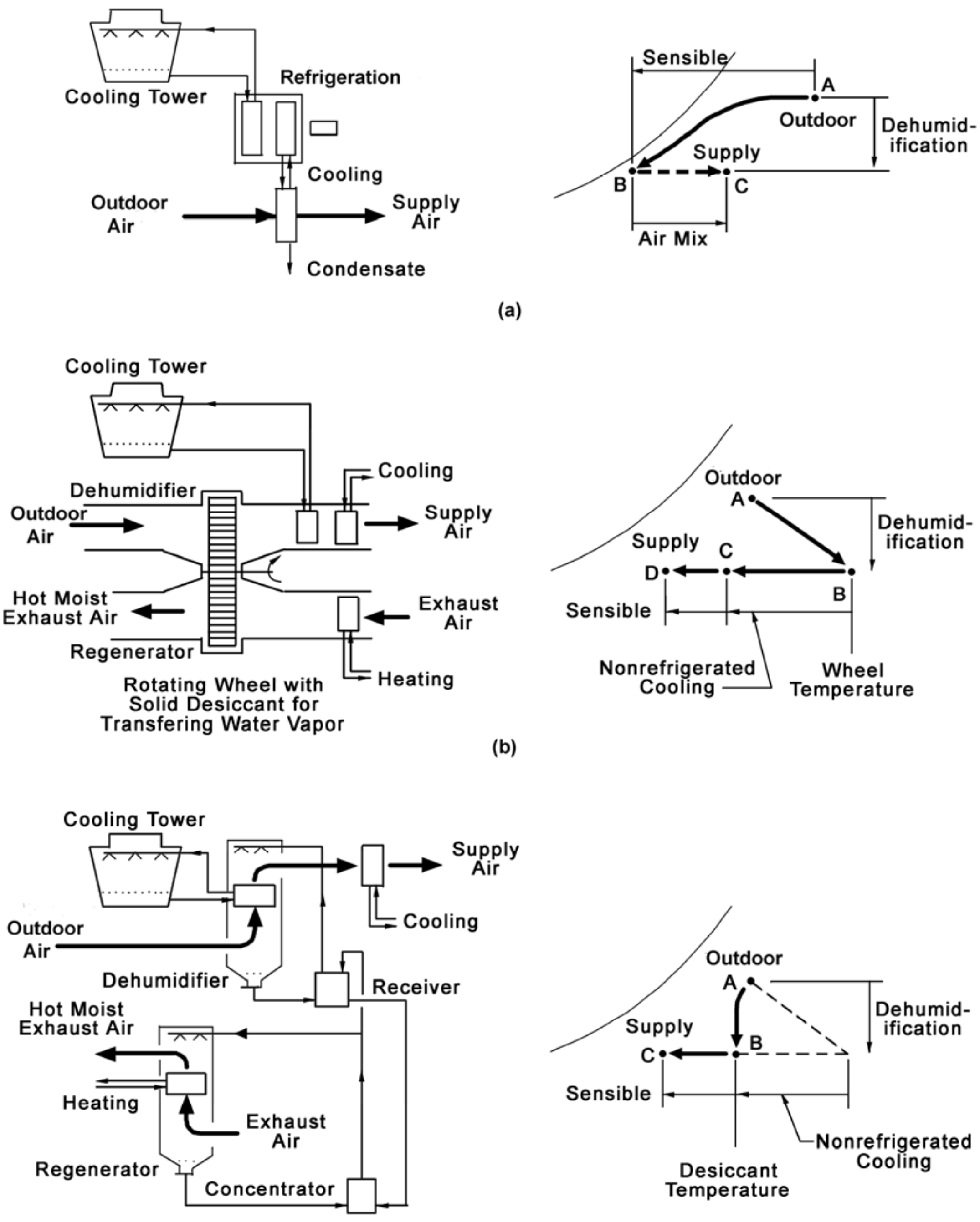

CENTRAL PLANT
TERMINAL
Desiccant
Dehumidification
Outdoor Air
Supply
Outdoor Air
A
B
E
I
H
G
Space
A
Exhaust Air
J
Exhaust Air
Desiccant
Regeneration
Terminal Options
- Fan-coil unit
- Unitary heat
pump
- Induction unit

Fig. 9-2 >>



Cooling Tower
Refrigeration
Cooling
Outdoor Air
Supply Air
Condensate
Sensible
A
Outdoor
Dehumid-ification
Supply
B
C
Air Mix
(a)
Cooling Tower
Cooling
Dehumidifier
Outdoor Air
Supply Air
Hot Moist Exhaust Air
Exhaust Air
Regenerator
Heating
Rotating Wheel with Solid Desiccant for Transfering Water Vapor
Outdoor
A
Dehumid-ification
Supply
C
D
B
Sensible
Nonrefrigerated Cooling
Wheel Temperature
(b)
Cooling Tower
Supply Air
Cooling
Outdoor Air
Dehumidifier
Receiver
Hot Moist Exhaust Air
Heating
Exhaust Air
Regenerator
Concentrator
Outdoor
A
Dehumid-ification
Supply
B
C
Sensible
Nonrefrigerated Cooling
Desiccant Temperature
(c)

Fig. 9-3 >>

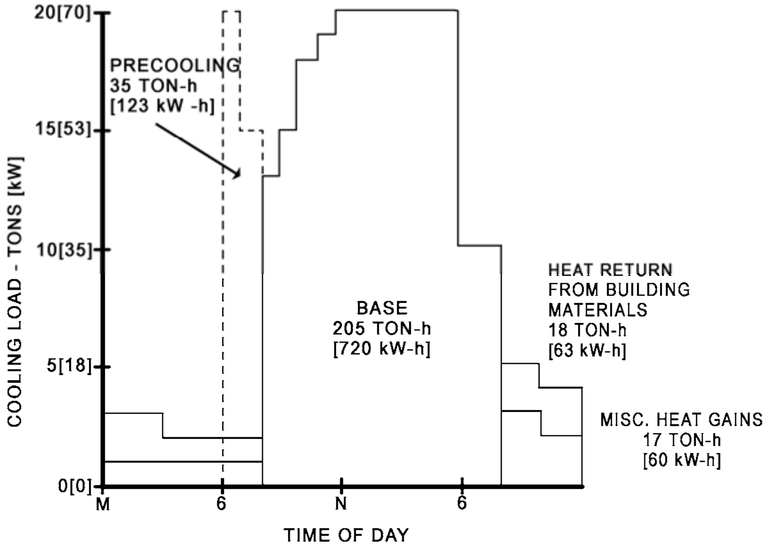

PRECOOLING
35 TON-h
[123 kW -h]
BASE
205 TON-h
[720 kW-h]
HEAT RETURN
FROM BUILDING
MATERIALS
18 TON-h
[63 kW-h]
MISC. HEAT GAINS
17 TON-h
[60 kW-h]
COOLING LOAD - TONS [kW]
20[70]
15[53]
10[35]
5[18]
0[0]
M
6
N
6
TIME OF DAY

Fig. 9-4 >>

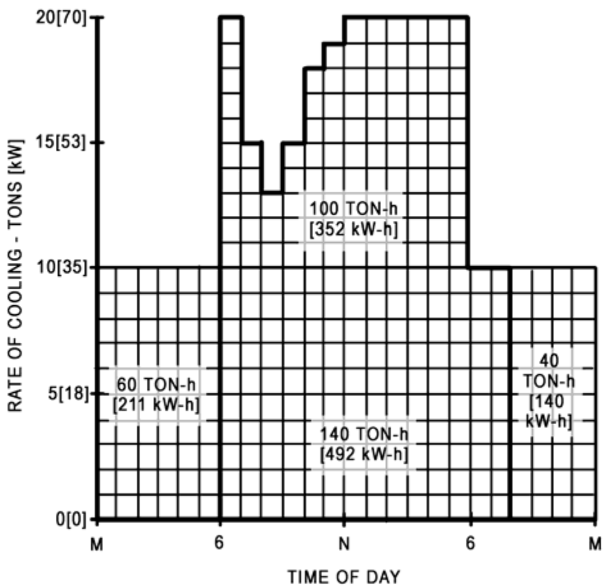

100 TON-h
[352 kW-h]
60 TON-h
[211 kW-h]
140 TON-h
[492 kW-h]
40
TON-h
[140
kW-h]
RATE OF COOLING - TONS [kW]
20[70]
15[53]
10[35]
5[18]
0[0]
M
6
N
6
M
TIME OF DAY

Fig. 9-5 >>

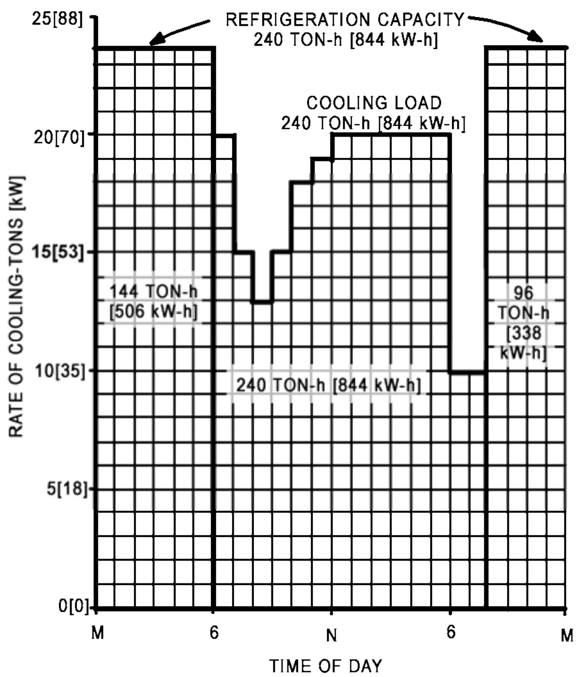

REFRIGERATION CAPACITY
240 TON-h [844 kW-h]
COOLING LOAD
240 TON-h [844 kW-h]
144 TON-h
[506 kW-h]
240 TON-h [844 kW-h]
96
TON-h
[338
kW-h]
25[88]
20[70]
15[53]
10[35]
5[18]
0[0]
RATE OF COOLING-TONS [kW]
M
6
N
6
M
TIME OF DAY

Fig. 9-6 >>

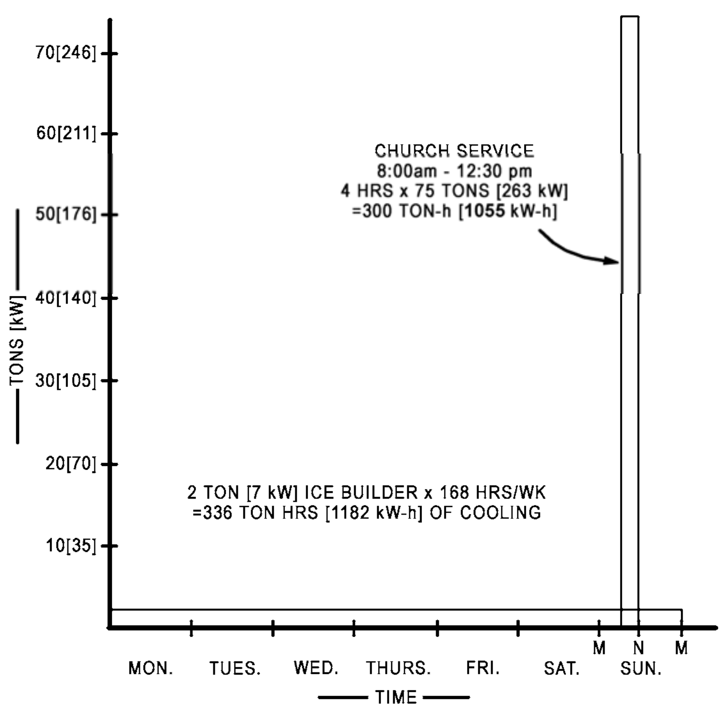
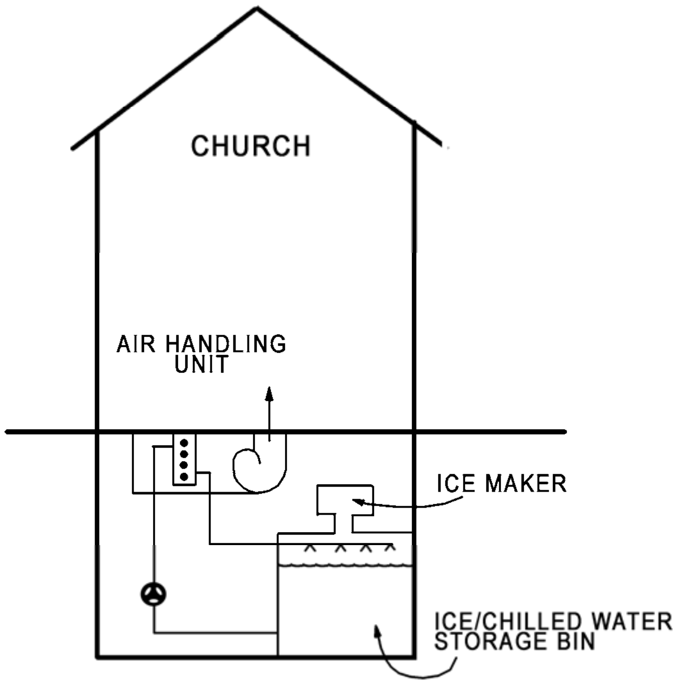
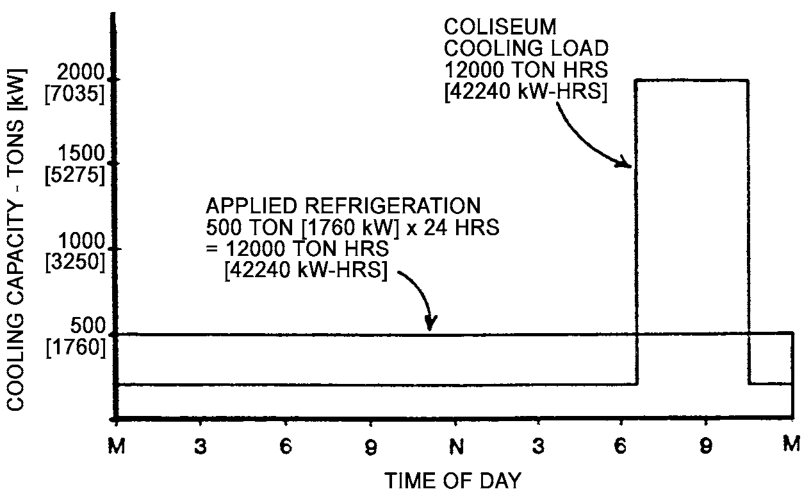

CHURCH SERVICE
8:00am - 12:30 pm
4 HRS x 75 TONS [263 kW]
=300 TON-h [1055 kW-h]
2 TON [7 kW] ICE BUILDER x 168 HRS/WK
=336 TON HRS [1182 kW-h] OF COOLING
70[246]
60[211]
50[176]
40[140]
30[105]
20[70]
10[35]
TONS [kW]
M
N
M
MON.
TUES.
WED.
THURS.
FRI.
SAT.
SUN.
TIME

Fig. 9-7 >>



CHURCH
AIR HANDLING UNIT
ICE MAKER
ICE/CHILLED WATER STORAGE BIN

Fig. 9-8 >>



COLISEUM COOLING LOAD 12000 TON HRS [42240 kW-HRS]
APPLIED REFRIGERATION 500 TON [1760 kW] x 24 HRS = 12000 TON HRS [42240 kW-HRS]
COOLING CAPACITY - TONS [kW]
2000 [7035]
1500 [5275]
1000 [3250]
500 [1760]
M 3 6 9 N 3 6 9 M
TIME OF DAY

Fig. 9-9 >>

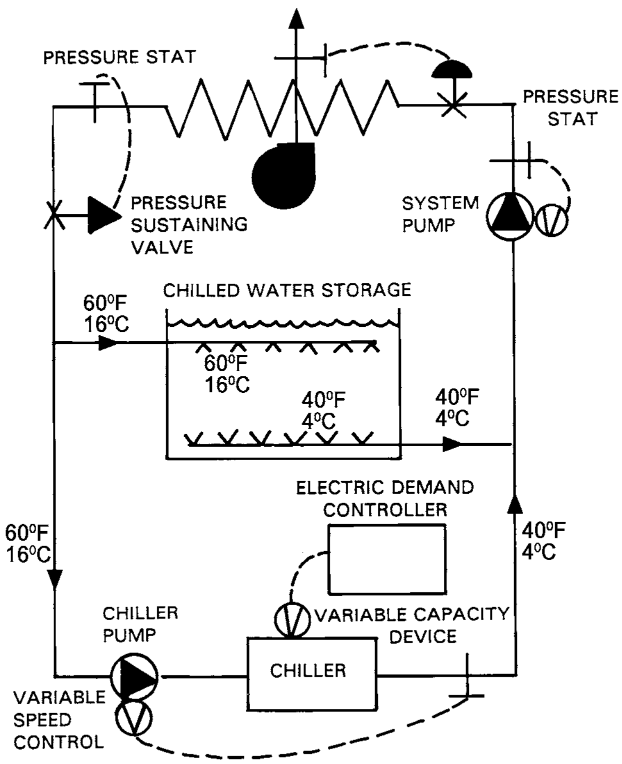


Fig. 9-10 >>

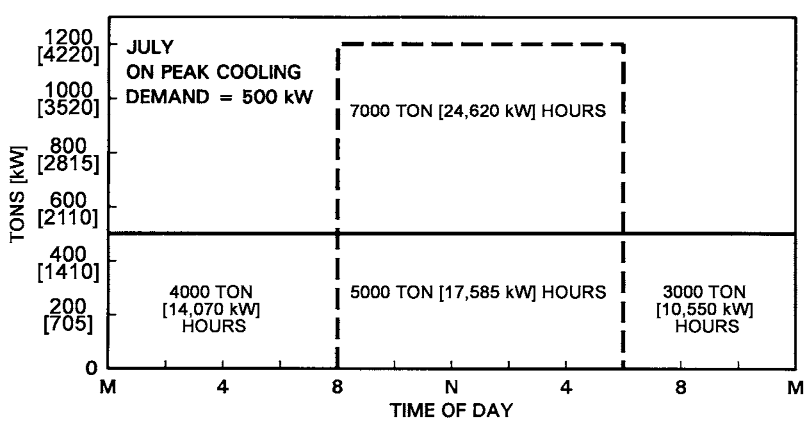


Fig. 9-11 >>

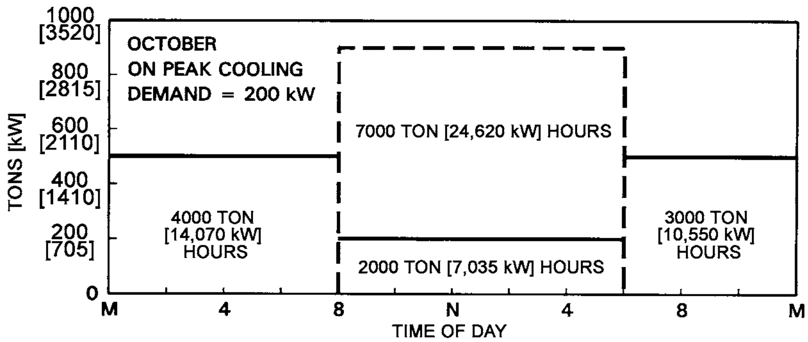


Fig. 9-12 >>

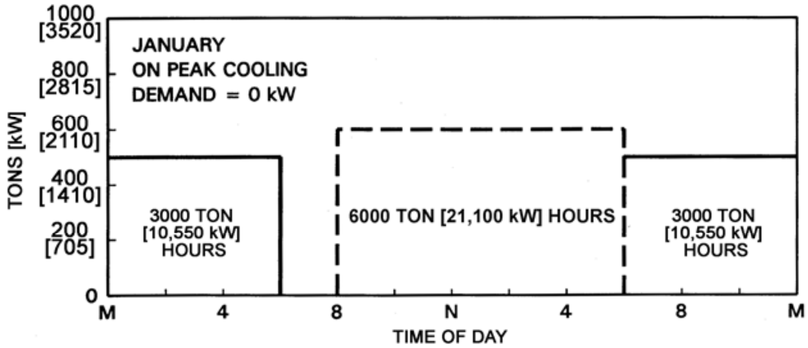


Fig. 9-13 >>

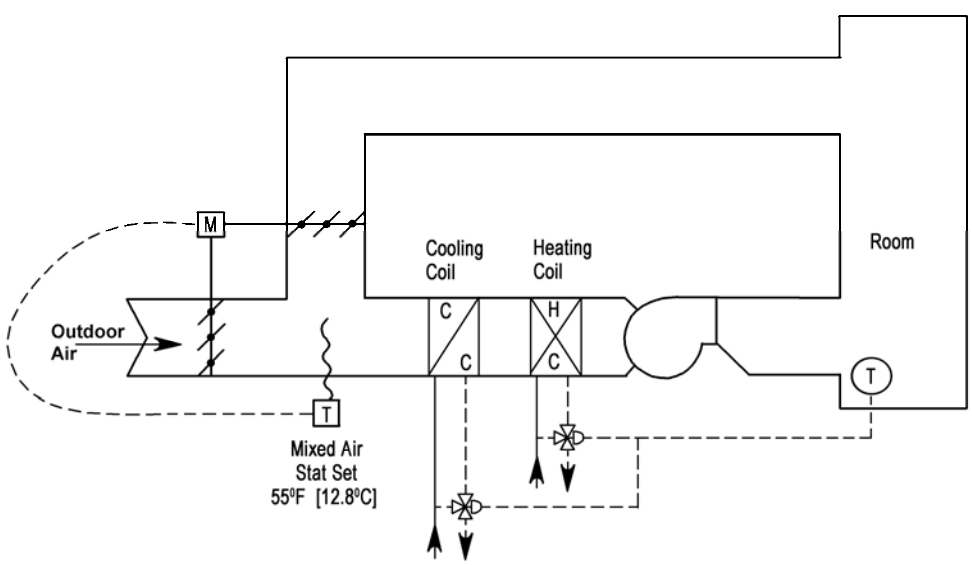


Fig. 9-14 >>

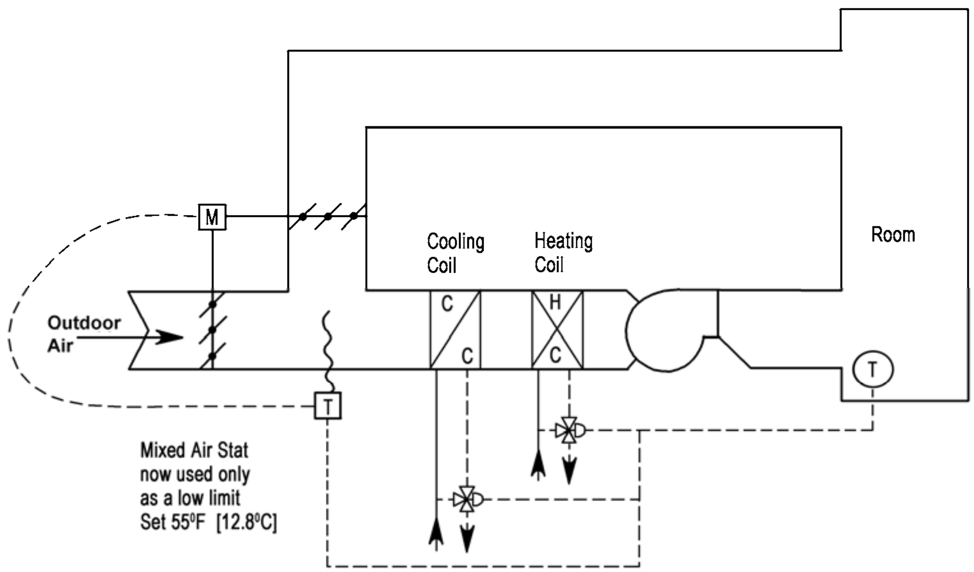


Fig. 9-15 >>

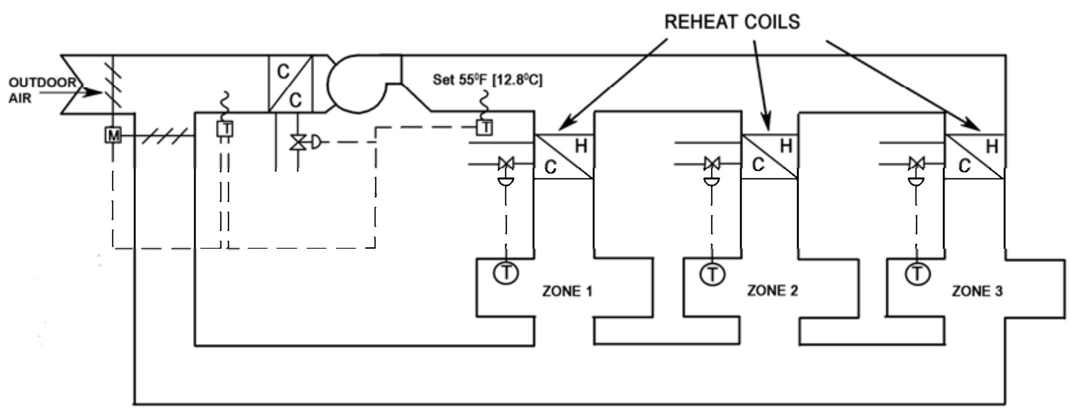


Fig. 9-16 >>

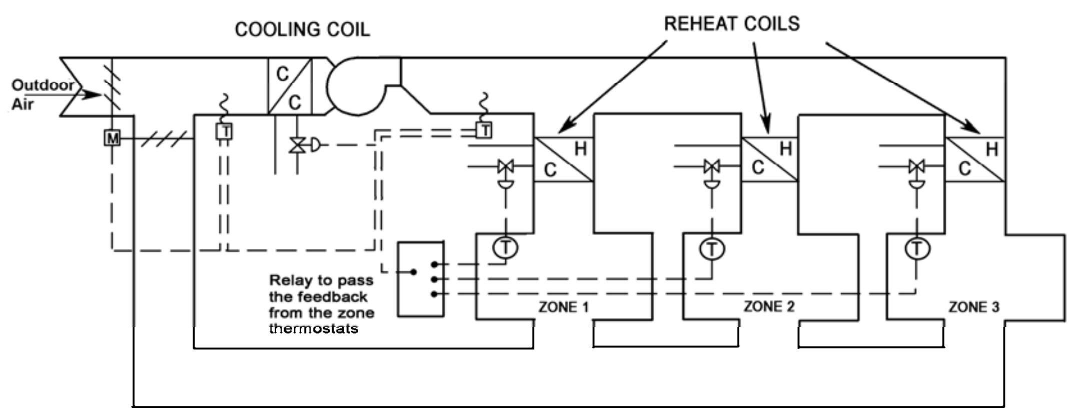


Fig. 9-17 >>

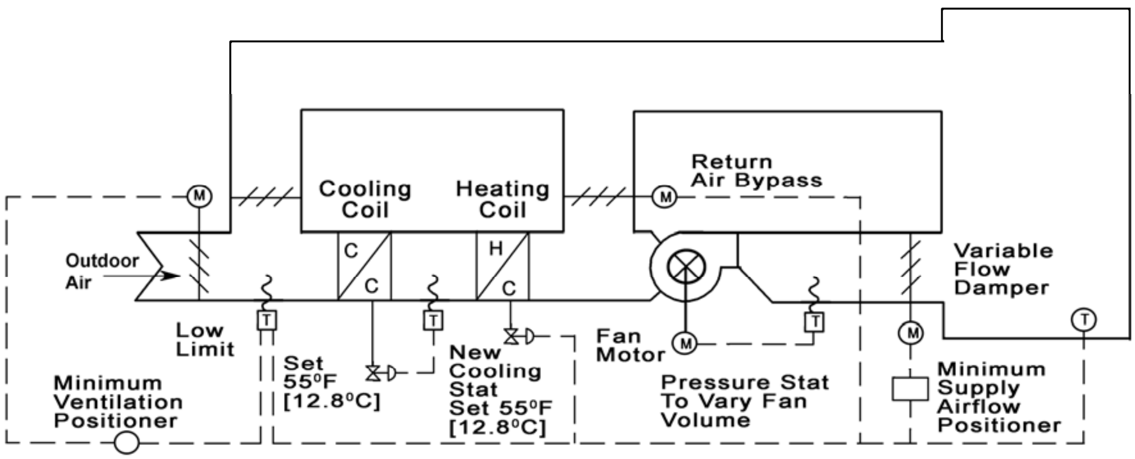

Cooling
Coil
Heating
Coil
Return
Air Bypass
Outdoor
Air
Variable
Flow
Damper
Low
Limit
Set
55°F
[12.8°C]
New
Cooling
Stat
Set 55°F
[12.8°C]
Fan
Motor
Pressure Stat
To Vary Fan
Volume
Minimum
Supply
Airflow
Positioner
Minimum
Ventilation
Positioner

Fig. 9-18 >>

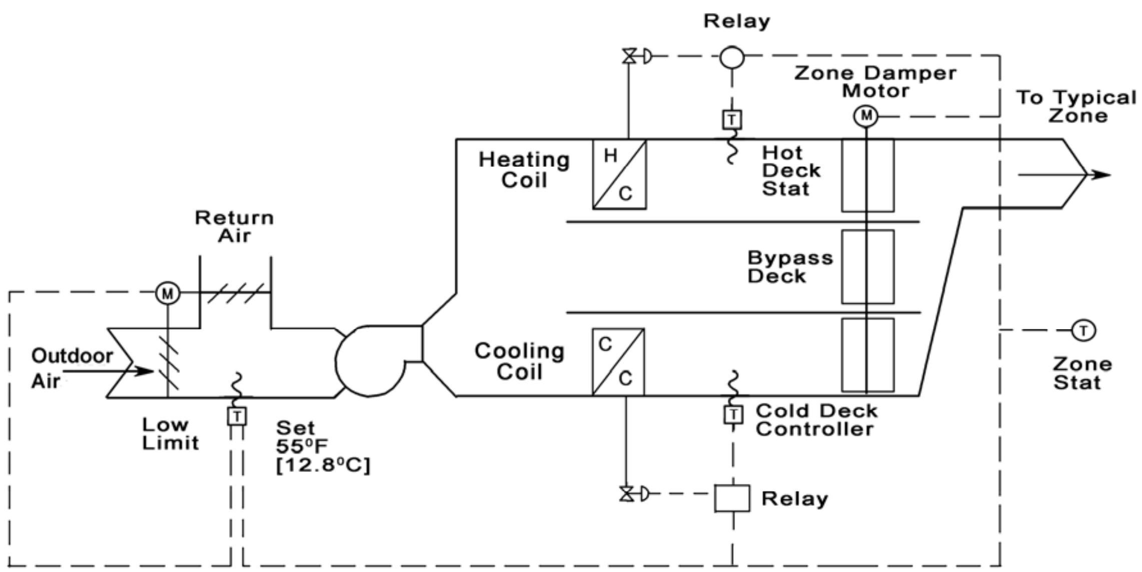

Relay
Zone Damper
Motor
To Typical
Zone
Heating
Coil
Hot
Deck
Stat
Return
Air
Bypass
Deck
Outdoor
Air
Cooling
Coil
Zone
Stat
Low
Limit
Set
55°F
[12.8°C]
Cold Deck
Controller
Relay

Fig. 9-19 >>

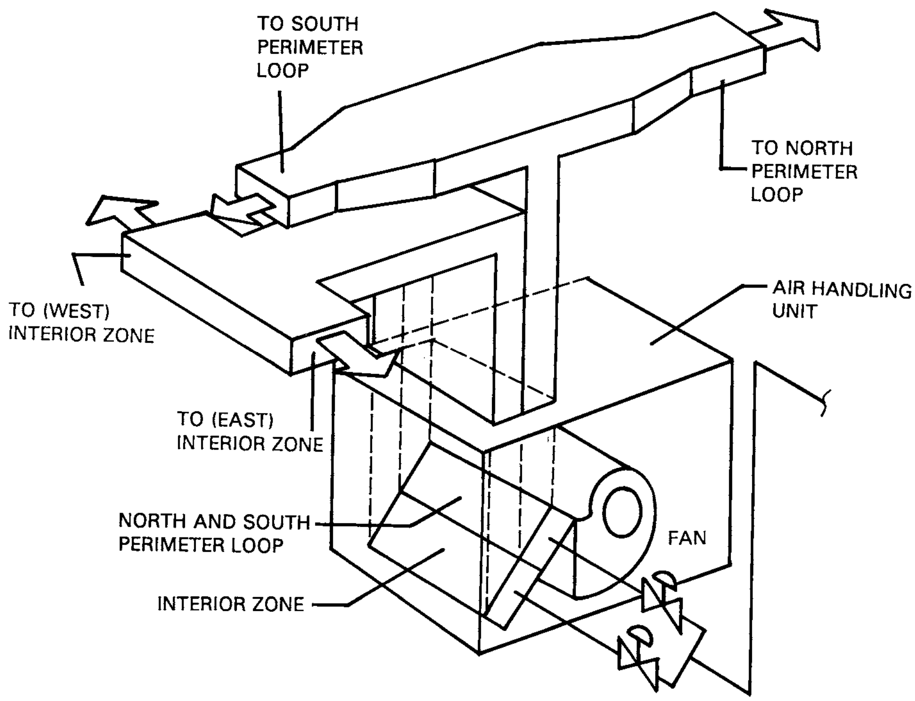


Fig. 9-20 >>

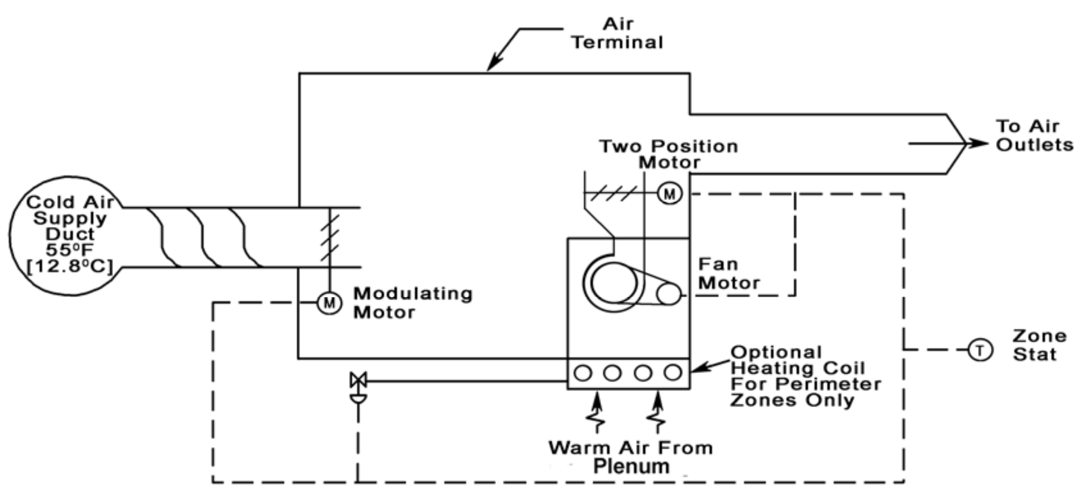


Fig. 9-21 >>

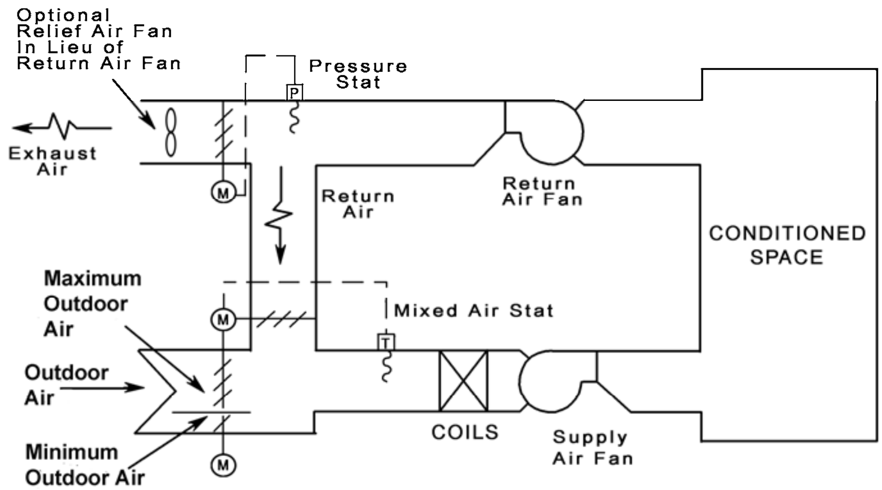


Fig. 9-22 >>

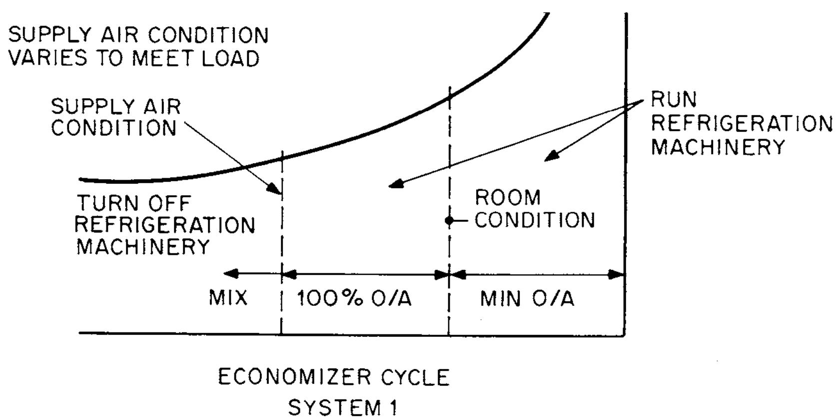


Fig. 9-23 >>

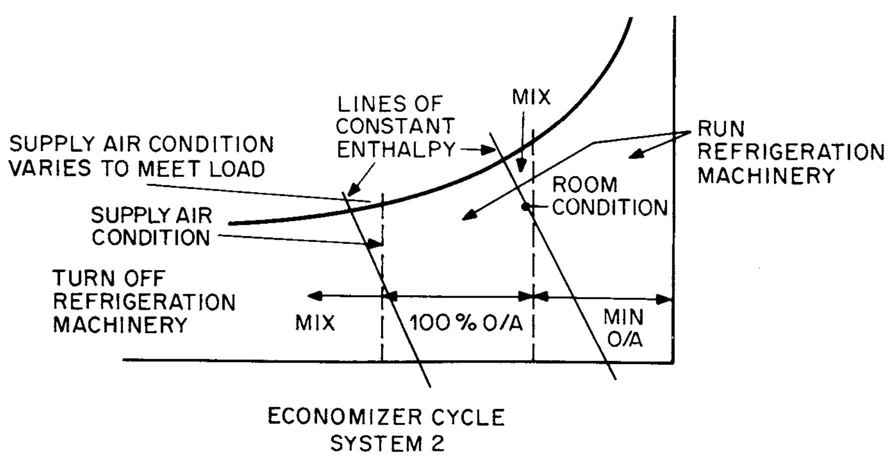


Fig. 9-24 >>

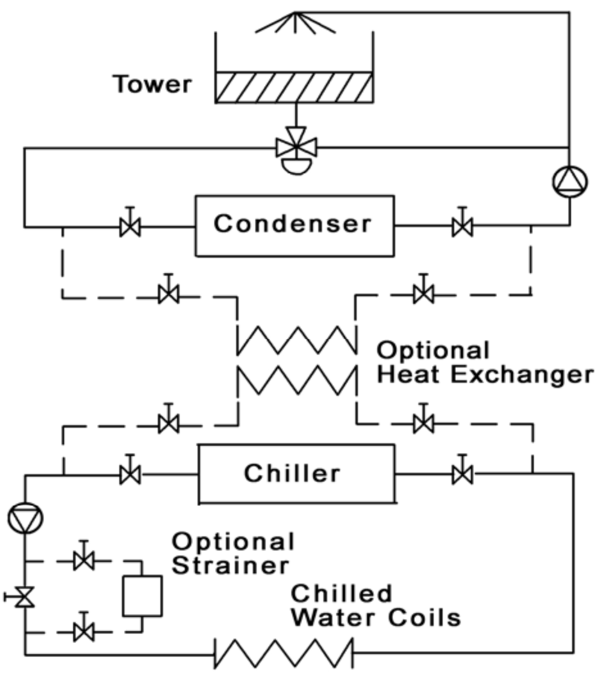


Fig. 9-25 >>

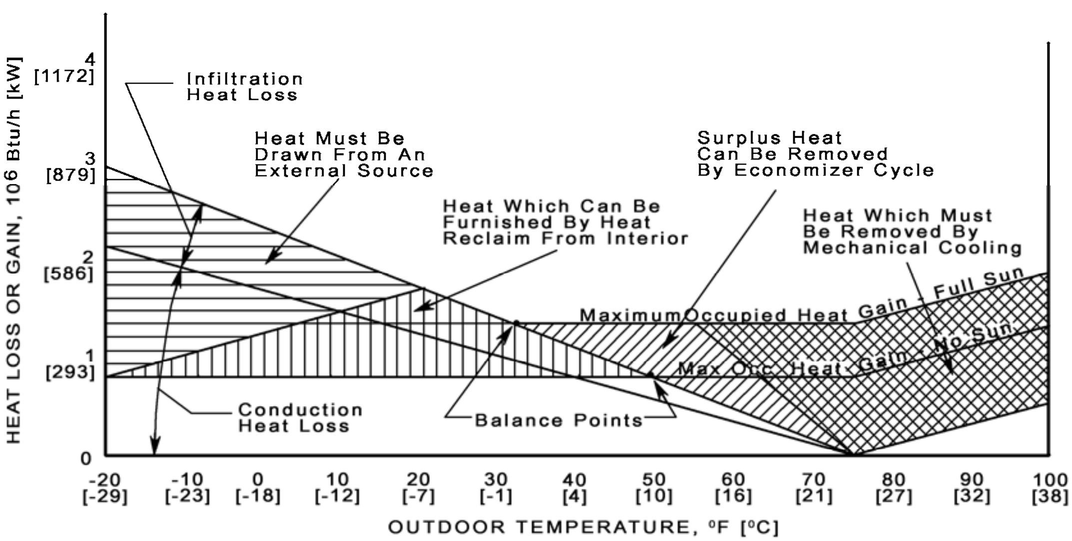


Fig. 9-26 >>

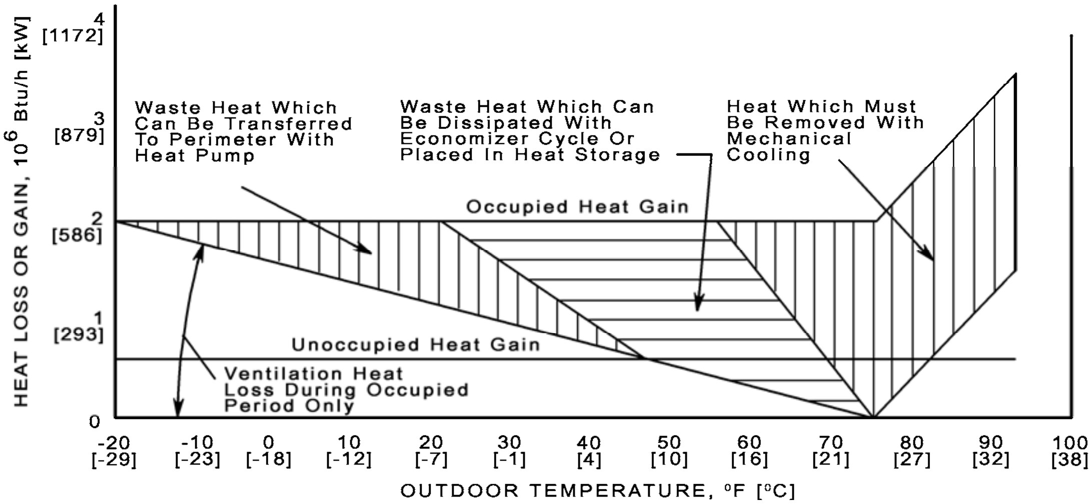


Fig. 9-27 >>