heater discharge, positions preheater coil valve V-1 to maintain a constant preheater discharge temperature.

On rising outdoor temperature, between 30 F and 65 F, duct thermostat T-3 located in the outdoor-air intake, moves maximum outdoor-air damper D-2 toward the open position provided that T-1 is satisfied. At 65 F outdoor, D-2 will be fully open and return-air damper D-3 will be fully closed.

As the outdoor-air temperature rises above 65 F, duct thermostat T-3 positions V-5 in such a way as to bypass low limit thermostat T-5 so that reheater coil valve V-3 is operated directly from thermostat T-1. As the outdoor-air temperature rises from 65 F to 75 F, duct thermostat T-4 gradually closes maximum outdoor-air damper D-2 and opens return-air damper D-3.

Cooling thermostat T-2 positions cooling coil valve V-2 to admit more chilled water as the space temperature rises.

Humidistat H positions humidifier valve V-4 to maintain the desired humidity in the conditioned space.

ZONING AND ZONE CONTROL

It is apparent that while apparatus like that of Fig. 1 would be very desirable for any single room, since in that case the air could be delivered at optimum conditions, the cost of a complete individual system for each room and the space required for the equipment generally would be prohibitive. Economy is favored if the varying requirements of numerous rooms or zones can be simultaneously satisfied by air from a single central supply system.

Several control methods can be used for controlling various rooms or zones that are supplied from a single central supply system. A space thermostat controlling an automatic volume damper will provide a measure of control merely by varying the air supply to the space. Such volume damper control often is unsatisfactory due to the variable air distribution pattern obtained as well as failure to provide sufficient ventilation air when the damper is throttled to its minimum position. To obtain the best control results with volume dampers the principles of zone control outlined in Chapter 43 must be given careful consideration.

A constant-volume air supply to the space can be obtained by using reheat or booster coils to vary the supply-air temperature. Here a properly proportioned heating coil is equipped with a valve that is operated by a space thermostat. The air leaving the central fan that serves several zones should be sufficiently cool to satisfy the cooling demand of the warmest area. Other areas may require warmer air. This is supplied by the reheat coil. The use of reheat coils for summer air conditioning is recommended. For this condition the central system air is cooled and dehumidified to the desired level and then reheated as may be required by the various zones.

It is practicable in some cases to use a single central system similar to that shown in Fig. 1, in conjunction with several fan units, one for each room or zone. In such cases a separate reheater on the suction side of each fan unit can be used. The reheater would be controlled by a space thermostat.

The designer must remember that the various supply fans will compete with each other for air, against the resistance interposed by the filters, coils, etc., that are used in common under such circumstances, and consequently, unless the fans are of backward-curved blade, non-overloading type, they may alternate in carrying more than their share of the air, and thereby cause the air distribution to be chaotic and unsatisfactory.

Another method of controlling temperature in various rooms served by a central air-supply system is shown in the sectional elevation, Fig. 2. The supply fan is placed immediately after the humidifier. When cooling the air in hot weather, the humidifier is not operated. The fan will deliver the air through the heating coil and through the cooling coil to the two air-pressure chambers *A* and *B* at the right of these coils. From these chambers many separate ducts, one of which is shown, each with a double-blade mixing damper, may convey the air to the various rooms. The mixing dampers, one of which is shown, are interlocked so that as the upper one closes, the lower one opens; selecting between them, air in the required quantity from either the warmer chamber *A* or the cooler one *B*. In cold weather no refrigerant is required in the cooling coil, and in hot weather no heating medium is circulated in the heating coil. With this scheme, the control of relative humidity in warm weather is not always sufficiently precise to meet requirements, since the untreated air delivered through the upper coil may be so high in relative humidity that it cannot compensate sufficiently for the nearly saturated air leaving the lower coil. A reheater could be placed if desired, at the right of the lower coil to bring the air in the lower chamber to the desired relative humidity. The simple arrangement of Fig. 2 is admirable in winter and, except where close control of relative humidity is important, may be acceptable in summer.

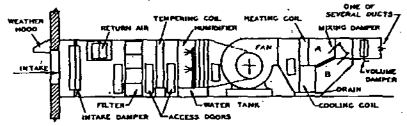

Fig. 2 .... Alternate Arrangement of Equipment for Controlling Air Conditions in Central-Air Supply System

HUMIDITY CONTROL

In winter, room relative humidities in excess of 30 percent are seldom required in a system designed for comfort conditioning only, and a low saturating efficiency may be desirable, or even necessary, especially if the same volume of air is handled as in summer. With a spray-type dehumidifier the main sprays may be shut off and only the eliminators need be flooded; which may give sufficient moisture. In other cases, such as those in which cooling coils are sprayed, the spray-water supply may be throttled. If the saturation efficiency of the sprays is too low, the spray water may be heated. The amount of heat put into the spray water by open or closed water heaters will be equal to that required to bring the dew-point temperature of the air entering the sprays up to that required before entering the preheater. It is possible, where clean steam is available, to introduce steam directly into the air stream to produce the desired dew-point temperature of supply air. However, the steam must be exceptionally clean, or objectionable odors will result.

It should be noted that the quantity of outdoor air to be introduced is affected by infiltration and leakage. Infiltration will reduce the quantity to be introduced by the system, while leakage may have to be offset by an increase in the quantity of outdoor air.

COOLING LOAD

The method of determining the cooling load for a conditioned space or spaces is outlined in Chapter 13. As pointed out therein, many of the items of heat gain are variable and do not reach their maximum values simultaneously. Proper consideration of these peaks and the avoidance of pyramiding these peaks in the cooling load calculations are stressed. Maximum solar heat gain on an east exposure is seldom coincident with the maximum outdoor wet-bulb temperature. A discussion of the factors affecting the actual instantaneous cooling load will be found in Chapter 13.

HEATING LOAD

Methods of calculating the heating load are shown in Chapter 12. Many of the factors outlined in Chapter 13, Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirements instead of the cooling requirements. However, these factors affect the heating load from the standpoint of control of inside conditions, overall performance, and economy of operation more than from a capacity of equipment standpoint. It is not only necessary to heat a building or space to its design conditions when there is but the merest fraction of normal occupancy, and when there are practically no lights, internal heat, or solar radiation, but it is also necessary to provide capacity to heat the building quickly when sudden cold follows relatively warm weather, as may occur after a week-end or holiday shut-down. However, in normal operation during week-ends and holidays, buildings are usually kept at a holding temperature to prevent the freezing of services. In many cases, less fuel is required to operate the heating plant at a near-normal rate and maintain the building or space at a temperature of 50 to 65 F at such times, than to shut the system down and then bring the temperature back to normal through forced operation of the heat-generating equipment with a consequent loss in efficiency.

AIR QUANTITY AND TEMPERATURE DIFFERENTIAL

The difference between the room-air temperature and the supply-air temperature at the outlet to the room is known as the temperature differential. In the theoretical case of a dehumidifier having 100 percent saturating efficiency, and where this air is delivered directly to the room without temperature increases due to heat gain, then the temperature differential is the difference between room temperature and apparatus dew-point temperature. If duct heat gains are considered a part of the room load, this still holds true. The apparatus dew point, as outlined previously, is fixed by the latent and sensible loads of the space, but in many cases, it is desirable to deliver more air to the spaces than is indicated by the difference between the room temperature and the apparatus dew point.

It has been indicated that where a percentage of air is passed through the dehumidifier without being treated, the relationship is modified in direct proportion, and that if room air is passed through untreated, no effect on the heat balance results. Similarly, if room air is passed around the dehumidifier and mixed with the treated air, the heat balance is not adversely affected. Therefore, if the quantity of air passed through the dehumidifier is determined by the usual methods, room air can be passed around the dehumidifier and mixed with the dehumidified air, increasing the supply-air quantity and temperature and decreasing the temperature differential. Thus if the difference between the room temperature and the apparatus dew point indicates that 10,000 cfm at 30 deg below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 deg below the room temperature, then mixed with 10,000 cfm of room air, resulting in a supply-air quantity of 20,000 cfm and a temperature differential of 15 deg instead of 30 deg. Air-supply outlets and grilles having a high induction ratio are available, and through their induction effect cause a large amount of room air to be mixed with the supply air within a short distance of the grille. A proper selection of outlets may make it possible to introduce air at low temperatures and high velocities without causing objectionable drafts or cold spots, but care must be used to see that too little air motion is not a result. Small temperature differentials may be required for this reason. While the use of a large temperature differential results in a saving in initial cost of fans and ducts, and in the operating cost of fans, this differential should be carefully considered. If the sensible heat load of a space is subjected to substantial variations, small temperature differentials should be considered, since systems employing small temperature differentials require less precision in controls.

Reduction of air quantity by slowing down the fans for the winter season, and increasing the temperature differential, often is feasible. A saving in fan power can thus be effected, provided the air distribution remains adequate.

Extremes should be avoided in all cases. For summer air conditioning, low supply-air temperatures result in larger heat gains to the air passing through the ducts, as well as in poor control. Too high a supply-air temperature may result in excessive initial and operating costs. Suggested limits for the temperature differential are from 12 to 25 deg, the actual selection being based on the requirements of the particular case. For winter air conditioning, too high supply-air temperatures result in excessive heat losses from the ducts and stratification within the room unless thorough mixing is assured, while too low supply-air temperatures may cause drafts, high operating costs, etc. Suggested limits are from 15 to 35 deg. There can be no set rule, and each case should be judged according to its particular requirements of the installation.

Reference may be made to Chapter 20 for further discussion of the most satisfactory design difference between the entering air temperature and volume in relation to the desired room condition.

UNITARY-CENTRAL SYSTEMS

Many different types of central air-conditioning systems with room units of various designs have been developed for multi-room buildings, such as office buildings, hotels, and hospitals. The primary object in using these systems is to save space by reduction of duct sizes or by entirely eliminating ducts. In new buildings, small ducts may reduce the overall building height; in existing buildings, the use of small ducts is frequently imperative and may even play a decisive role in the acceptance of air conditioning for these buildings. Also, multi-room buildings frequently require a high degree of zoning or individual room control, which the systems must supply.

These buildings usually have a large perimeter relative to the floor area. Air-conditioning units are usually installed beneath the windows. Where the spaces to be conditioned extend a considerable distance from the outer wall into the