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.

Various methods are practicable for controlling the temperature, humidity and air movement in various rooms or zones. A measure of control is attainable merely by proportioning the flow of air to each room, though usually such control by throttling dampers is difficult to maintain and should be avoided when possible.

Another scheme is to install a properly proportioned coil in the branch air-supply duct serving each room or zone to warm the air to suit the occupants. For example, the air leaving the fan that serves several zones may be cooled, before entering the fan, to the condition favorable for one zone, and the air for each other zone may be reheated by the branch duct coil to the required temperature. It is also possible to circulate a heat absorbing medium in the branch duct coils to reduce the temperature of the air passing to rooms that would be overheated if they received air at the condition leaving the central air-supply system. When heat transfer devices are placed in branch ducts for improved temperature control, mechanically-circulated water gives excellent results as a heat carrier. The water usually is warmer than the air but it is possible to use water colder than the air.

It is practicable also to use single central air-conditioning equipment similar to that shown in Fig. 1, in conjunction with several fans, one for each room or zone. In such cases there may be a separate reheater on the suction side of each relatively small supply fan.

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.

Another method of attaining temperature control in individual rooms with a year-round central air-supply system, is to install a booster fan between the main air-supply duct and the air-delivery opening to each zone or room, as shown in Fig. 3. Air can then be delivered from the central supply fan through the main duct at some desired condition, for instance, 60 F and 45 percent relative humidity. A double mixing damper near the intake opening of the booster fan, controlled by a thermostat in the room or zone that is served by the fan, is interlocked with an outlet exhaust damper in the spent-air opening, so that as more of the room air is recirculated, and as less new air from the main air-supply duct is delivered into the room, the spent-air outlet is throttled in proportion. In many large installations this principle is applied successfully for zoning different stories in multi-story office buildings, the main supply fan being on the roof, and each booster fan being used for supplying the rooms of one orientation of each story. In other cases the booster fans serve only single offices, and therefore are small enough to be concealed above ceilings alongside the main supply duct.

There may be installations in which the use of recirculated air for mixing with new refrigerated and nearly saturated air to control temperature and relative humidity is objectionable. In such cases the general recirculation arrangements of Fig. 1 may be omitted, and heat transfer coils located in the ducts may be used. In some cases where general recirculation is not acceptable, as for all the rooms in an entire building, use of the local circulation of Fig. 3 may solve the problem.

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

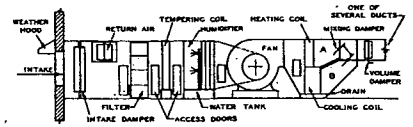


Fig. 2 . . . . Alternate Arrangement of Equipment for Controlling Air Conditions in Central-Air Supply System

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

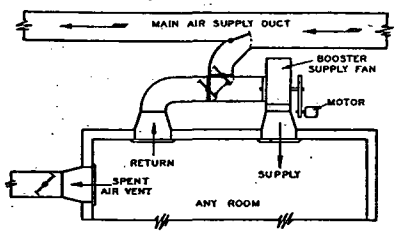


Fig. 3 . . . . Arrangement for Individual-Room Temperature Control with Central Air-Supply System

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.