sufficient saturation. In other cases, such as where cooling coils are sprayed, the spray water may be throttled.

When it does become necessary to increase the saturation efficiency of the sprays, 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 reheater. It is possible where clean steam is available, to introduce the 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. This precaution should be observed also where open water heaters or ejector water heaters are used.

Present day practice, for spray type dehumidifiers of good design, assumes that the air leaves the dehumidifier at 1 to 2 F higher than the temperature of the spray water leaving the dehumidifier. A spray dehumidifier having sufficient length of spray chamber and density of spray together with a proper arrangement of nozzles may closely approach complete saturation.

Air Quantity and Effective Temperature Difference

The difference between the room air temperature and the supply air temperature at the outlet to the room is known as the *effective temperature difference.* In the theoretical case of a dehumidifier having a 100 per cent saturating efficiency and where this air is delivered directly to the room without temperature increases due to heat gain, then the *effective temperature difference* 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 determined by the solution of Equations 2 and 3, with 4 or by the charts.

It has been indicated that where a percentage of air is passed through the dehumidifier without being treated that the relationship was modified in direct proportion, and that if room air passed through untreated no effect on the heat balance resulted. 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 *effective temperature difference.* Thus if a solution of Equations 2 and 3 in conjunction with Equation 4 indicates that 10,000 cfm at 30 F below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 F below the room, then mixed with 10,000 cfm of room air resulting in a supply air quantity of 20,000 cfm and an *effective temperature difference* of 15 F instead of 30 F. Supply air outlets and grilles that have a high induction ratio (that is, a large amount of room air is mixed with the air leaving the outlet within a short distance of the outlet through the induction effect of the air stream) are available as well as induction units. A proper selection of outlets or units 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. Lower *effective temperature differences* may be required for this reason. While the use of a high *effective temperature difference* results in a saving in initial cost of fans and ducts and in the operating cost of fans, this difference should be carefully considered. If the sensible heat load of a space is subjected to substantial variations, lower *effective temperature differences* should be considered, since systems employing a *low effective temperature difference* will be less exacting in control requirements.

Assume that a space has a sensible heat load so that 10,000 cfm of air supplied at a 30 F *effective temperature difference* is required to maintain a room temperature of 80 F. If the load is suddenly reduced 50 per cent with the air supplied at the same temperature, the resultant room temperature would become 65 F. On the other hand, if 20,000 cfm were supplied at an *effective temperature difference* of 15 F and the load suddenly reduced 50 per cent the resultant room temperature would be 72.5 F. This, while a rather extreme example, indicates the less exacting demand on the controls brought about by the use of the lower *effective temperature difference.* Of even greater importance is the case where two or more spaces are controlled from an average condition such as by a thermostat located in the return air stream of all the spaces. From the previous example, it can be seen that a large variation in the load in one of the spaces will not reflect itself in such a large change in room temperature. Thus in the long run the larger *effective temperature difference* may not be the most economical.

The analysis in the foregoing applies largely to summer air conditioning. The same analysis will apply to some extent in winter. The mathematical relationship is revised due to a heating requirement rather than cooling. Present practice indicates a high temperature difference in winter in comparison to that for summer. It is due to the fact that the heat losses in winter in Btu per hour far exceed the summer heat gains in Btu per hour, and particularly to the fact that in winter, reheating the air to produce desired room conditions is not reflected as a load on the system as it is in the summer, but merely accomplishes the necessary work.

In line with the latter, reduction of air quantity by slowing down the fans for the winter season and increasing the temperature difference often is feasible, creating a saving in fan horsepower at no expense to the final heat balance, providing the air distribution is not seriously affected.

Extremes should be avoided in all cases. For summer air conditioning low supply air temperatures will result in larger heat gains to the air passing through the ducts, poor control, etc. Too high a supply air temperature may result in excessive initial and operating costs. Suggested limits for the *effective temperature difference* are from 12 to 25 F, 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 mixture is insured, while too low supply air temperature may