Solution. From a table of air properties, (Chapter 1, Table 6), it is found that the sensible heat content of air at 80 F is 19.19 Btu per pound, and that of air at 68 F is 16.31 Btu per pound. Hence the sensible heat load which can be absorbed by 1 lb of air is 19.19 − 16.31 = 2.88 Btu. Then the total air quantity required is 100,000/2.88 = 34,700 lb per hour. Expressed in terms of standard air at 0.075 lb per cubic foot, this is: 34,700/(60 x 0.075) = 7,720 cfm. Since a latent heat load of 33,000 Btu per hour must be absorbed by 34,700 lb of air per hour, the latent heat to be absorbed per pound of air is: 33,000/34,700 = 0.952 Btu. The latent heat per pound of *vapor* is approximately 1060 Btu (from steam tables), and since 1 lb = 7,000 grains, the moisture to be added to each pound of air is: (0.0952 x 7000)/1060 = 6.3 grains. Hence the dew-point temperature of the conditioned air supply will correspond to 85.5 − 6.3 = 79.2 grains per pound, and from the psychrometric chart or tables this is found to be 60.7 F dew-point. At a dry-bulb temperature of 68 F this corresponds to a wet-bulb temperature of approximately 63.3 F, or a relative humidity of 78 per cent. Summarizing, the required air supply is: 7,720 cfm at 68 F dry-bulb and 63.3 F wet-bulb.

In the previous example and solution, no consideration was given to the type of air conditioner to be used. *The solution is dependent on room conditions only, and the method is generally applicable regardless of the heat load ratio or the size of the system.* Of course if the air quantity or the temperatures obtained in the solution are not practicable for the actual installation, the original selection of a dry-bulb temperature of the inlet air may be changed as required, providing the design of the distribution system is modified accordingly. A higher dry-bulb temperature of the air supply, i.e., a smaller temperature differential, will call for a larger quantity of air and a larger wet-bulb depression at the supply inlets.

Heat Removed by Apparatus

The total cooling and dehumidifying load will depend on the total room load, the duct losses, the outside air or ventilation load, and the amount of reheat, if any. The necessity for reheat will in turn be determined by the type of system, and by the dry-bulb temperature required at the room supply inlets.

Example 2. To maintain a room at 80 F dry-bulb and 55 per cent relative humidity requires a conditioned air supply of 7,720 cfm at 68 F dry-bulb and 63.3 F wet-bulb temperature (see Example 1.). The air is to be conditioned in a central plant unit of the type shown in Fig. 2. The conditioned air is to consist of 30 per cent outside air (2320 cfm) and 70 per cent recirculated air (5400 cfm). The outside air enters the conditioner at 95 F dry-bulb and 75 F wet-bulb temperature, and the return air is assumed to enter at 80 F dry-bulb and 68.5 F wet-bulb, (neglecting radiation and duct losses). Find the total refrigeration load, and the air conditions entering and leaving the dehumidifier, for both spray and surface type units.

Solution. The simplest method of solution is on the basis of wet-bulb temperatures and total heats, calculating the return air and outside air loads separately. Before this can be done, the air conditions at the exit of the dehumidifier must be determined. It is known that the dew-point of the conditioned air must be 60.7 F. The dry-bulb temperature at the exit of the dehumidifier will then depend on whether sprays or coils are used, and on the design of each. Assume in this case that the spray dehumidifier saturates the air, and that the air discharged from the surface dehumidifier has a 3 F wet-bulb depression. (These are common assumptions, but other values may be obtained, depending on the designs). Air conditions at the discharge of the dehumidifier are then: For the spray dehumidifier, dry-bulb = wet-bulb = dew-point = 60.7 F. For the surface type dehumidifier, dry-bulb = 65.5 F, wet-bulb = 62.5 F, dew-point = 60.7 F. The total refrigeration load for each type of dehumidifier may be calculated as follows:

Spray Type Dehumidifier. 2320 cfm of outside air cooled from 75 F wet-bulb (38.46 Btu per pound, total heat content), to 60.7 F wet-bulb (26.86 Btu per pound, total heat content.) Refrigeration load: 2320 × 0.075 × 60 × (38.46 − 26.86) = 121,000 Btu per hour. 5400 cfm of return air, cooled from 68.5 F wet-bulb (32.71 Btu per pound), to 60.7 F wet-bulb (26.86 Btu per pound). Refrigeration load: 5400 × 0.075 × 60 × (32.71 − 26.86) = 142,000 Btu per hour. The total refrigeration load for the dehumidifier is therefore 263,000 Btu per hour or 21.9 commercial tons of refrigeration.

Surface Type Dehumidifier. 2320 cfm of outside air cooled from 75 F wet-bulb (38.46 Btu per pound), to 62.5 F wet-bulb (28.12 Btu per pound). Refrigeration load: 2320 × 0.075 × 60 × (38.46 − 28.12) = 108,000 Btu per hour. 5400 cfm of return air, cooled from 68.5 F wet-bulb (32.71 Btu per pound) to 62.5 F wet-bulb (28.12 Btu per pound). Refrigeration load: 5400 × 0.075 × 60 × (32.71 − 28.12) = 111,500 Btu per hour. The total refrigeration load for the dehumidifier is therefore 219,500 Btu per hour or 18.3 commercial tons of refrigeration.

A summary of the results with the two types of dehumidifiers is given in Table 2.

TABLE 2. COMPARISON OF SPRAY AND SURFACE TYPE DEHUMIDIFIERS FOR EXAMPLE 2

AIR CONDITIONS OR LOAD	SPRAY TYPE DEHUMIDIFIER	SURFACE TYPE DEHUMIDIFIER
Exit Air Conditions:		
Dry-bulb temperature, deg F	60.7	65.5
Wet-bulb temperature, deg F	60.7	62.5
Dew-point temperature, deg F	60.7	60.7
Total refrigeration load, tons	21.9	18.3
Reheat necessary to raise dry-bulb temperature of exit air to 68 F, Btu per hour	60,900	20,800

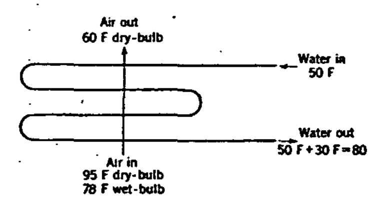

FIG. 5. COUNTER-FLOW SURFACE COOLING DIAGRAM

The design and selection of both spray and surface dehumidifiers is usually made largely on the basis of manufacturers' data, although there are certain general precautions to be observed. Air velocities in spray dehumidifiers are usually limited to 500 or 600 fpm, and the highest temperature of the spray water should be 2 or 3 F below the required exit dew-point. Surface units using cold water should be designed to obtain as near true counterflow as possible as shown in Fig. 5. Face velocities are usually from 400 to 600 fpm, and water velocities should be high enough to obtain good heat transfer, but low enough to avoid excessive pumping costs (preferred range is usually 1 to 3 fps). A close approximation of surface coil area can be made on the basis of the sensible heat transfer, if the latent heat load is not more than 25 or 30 per cent of the total. In this calculation the dry coil heat transfer coefficients are used, and the computation is the same as that used in selecting an air heating coil. If the coil loads, the entering air conditions, and the refrigerant and air velocities are specified by the designer, then the refrigerant temperature and the depth of coil to be used are dictated by the coil design, and cannot be arbitrarily selected. Surface coil performance is greatly affected by the refrigerant temperature, and if the refrigerant temperature