with the Inside Air displaced by the Ventilating Air is 2099 × 31.41 = 65,900 Btu per hour; the weight of water leaving is 2099 × 0.01115 = 23.41 lb per hour.

Each occupant may be regarded as a *normal person standing at rest* and therefore evaporating 0.198 lb of water per hour at about 79 F (Table 3, Chapter 2). Therefore the energy added to the store by such evaporation is 50 × 0.198 × 1096.2 (enthalpy of saturated vapor at 79 F, Table 8) = 11,000 Btu per hour, the weight of water added being 50 × 0.198 = 9.90 lb per hour. In addition each person loses 225 Btu of heat per hour by conduction, convection and radiation, making a total for 50 persons of 11,300 Btu per hour.

An energy balance shows a net gain of 16,000 + 48,000 + 13,900 + 80,300 − 65,900 + 11,000 + 11,300 = 114,600 Btu per hour. A water balance shows a net gain of 29.43 − 23.41 + 9.90 = 15.92 lb per hour. The slope of the condition line is determined by the ratio q = 114,600 ÷ 15.92 = 7205 Btu per pound of water. The temperature at which the condition line crosses the saturation curve is 58.02 F which is, therefore, the apparatus dew-point. This temperature is found by solving Equation 25,

$$\frac{31.41 - h_s}{0.01115 - W_s} = 7205$$

The fact that a trial-by-error solution is required is not a serious complication.

In order to calculate the cooling load it will be assumed that the air conditioning process consists of cooling and separating. The thermodynamic properties entering the calculations are:

	Inside Air	*After Cooling*	*After Separating*
t	80.0	58.02	58.02
W	0.01115	0.01115	0.01027
h	31.41	25.086	25.063

It follows that the refrigeration required is 31.41 − 25.086 = 6.324 Btu per pound dry air. But the weight of dry air involved is 114,600 ÷ (31.41 − 25.063) = 18,056 pounds per hour; hence the total refrigeration required, namely, the cooling load, is 18,056 × 6.324 = 114,185 Btu per hour, or 114,185 ÷ 12,000 (Btu extracted per hour per ton of refrigeration) = 9.49 tons.

The weight of water removed is 18,056 × (0.01115 − 0.01027) = 15.92 lb per hour as required. This water is removed as liquid at 58.02 F and therefore removes energy of amount 15.92 × 26.1 (specific enthalpy of liquid water at 58.02 F, Table 6) = 415 Btu per hour. This plus the refrigeration accounts for the total removal of 114,600 Btu per hour as required.

In practice the point at which the condition line crosses the saturation curve may dictate an excessive number of air changes. If so, it may be necessary to cool to a lower temperature. But, if the requirements of the problem are to be exactly met both as regards removal of energy and removal of water, the mixture returned to the conditioned space must then contain a certain amount of liquid. In other words, its state point must lie on the condition line.

It may be that the condition line does not cross the saturation curve at all, in which case the apparatus dew-point as defined previously does not exist. In this case the actual dew-point of the apparatus can be set at any temperature provided the air is then reheated to a point on the condition line before being returned to the conditioned space.

In actual practice it is rarely possible to obtain complete saturation at the dew-point temperature at which the apparatus is set. This may be due to insufficient contact; or a portion of the air may be deliberately by-passed. But either is equivalent to reheating and, if the final condition still lies on the condition line, the requirements of the problem can be exactly met.

Heating Load

The idea of the condition line is also useful in calculating heating load problems. Its use is best illustrated by means of an illustrative example.

Example 20. The clothing store of Example 19 is to be maintained at 70 F dry-bulb, 50 per cent saturation in winter, with outside design conditions being 0 F dry-bulb, 80 per cent saturation. In order to avoid window condensation with the given inside and outside conditions, double doors and windows are provided. Show windows are sealed. The ventilation requirements of 10 cfm per person for 50 persons, or 30,000 cfh, will build up a slight pressure. For these three reasons, infiltration is reduced to a negligible amount. The normal heat transmission through walls, partition, floor, roof, glass, and doors is estimated at 73,750 Btu per hour. Considerable energy is gained from lights and occupants, but only after the store is raised to the proper conditions; hence this item should be disregarded in figuring the maximum heating load. Analyze the problem as shown in Fig. 9.

Solution. The thermodynamic properties of Outside Air are: v = 11.59 cu ft per pound of dry air, h = 0.67 Btu per pound of dry air, and W = 0.00063 lb water per pound of dry air. Accordingly, the Ventilating Air introduces dry air of amount 30,000 ÷ 11.59 = 2590 lb per hour, energy of amount 2590 × 0.67 = 1730 Btu per hour, and water of amount 2590 × 0.00063 = 1.63 lb per hour. Since infiltration is negligible, none of the Ventilating Air will be admitted directly to the store, but will enter with the Supply Air after having been processed in the air conditioning apparatus. Nevertheless, it displaces an equal weight of dry air from the store.

The thermodynamic properties of Inside Air are: v = 13.51 cu ft per pound of dry air, h = 25.38 Btu per pound of dry air, and W = 0.00787 lb water per pound of dry air. Accordingly, the Ventilating Air displaces energy of amount 2590 × 25.38 = 65,730 Btu per hour, and water of amount 2590 × 0.00787 = 20.38 lb per hour.

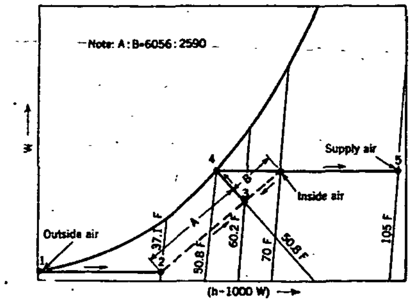

Fig. 9. Diagram Illustrating Example 20

Using these data, it appears that the total energy to be added to the store is 73,750 + 65,730 = 139,480 Btu per hour while the total water to be added is 20.38 lb per hour. But it would be a mistake to determine the condition line by the ratio of these two quantities, since the dry air returned with the Supply Air exceeds that recirculated from the store by the amount introduced with the Ventilating Air. It is correct, however, to lump the Recirculated Air and the Displaced Air together, since both leave the store under the conditions of Inside Air. Then the Supply Air must return more energy to the store than both of these remove by an amount equivalent to the normal heat transmission or 73,750 Btu per hour, and more water by amount *zero*. The ratio of these two quantities determines the condition line. Since the ratio is infinite, the condition line is horizontal on the Mollier Chart; in other words, the humidity ratio of the Supply Air must be the same as that of Inside Air.

Good practice is to limit the temperature of the Supply Air to 105 F. The condition line crosses the 105 F isotherm at 15.6 per cent saturation. The thermodynamic properties of the Supply Air are: h = 33.91 Btu per pound of dry air, and W = 0.00787 lb water per pound of dry air. Accordingly, the weight of dry air to be recirculated is 73,750 ÷ (33.91 − 25.38) minus 2590 = 6056 lb per hour.

The dew-point temperature of the Supply Air is the same as that of Inside Air, namely, 50.8 F. A suitable conditioning process is to (see Fig. 9): (1-2) preheat the Ventilating Air to temperature t; (2-3) mix it adiabatically with Recirculated Air; (3-4) saturate the resulting mixture adiabatically with recirculated spray; (4-5) reheat to 105 F on the condition line. The temperature t must be chosen so that, after preheating, the wet-bulb of the Ventilating Air is 50.8 F.

The determination of the preheating temperature t using the data of Table 6, though straightforward, is somewhat tedious. Graphical solution using the Mollier Chart is easier. The answer 37.1 F is obtained by drawing the line A + B so that the length of