But suppose that the saturation efficiency of the apparatus is only 95 per cent, by which is meant that the air delivered by the apparatus is only 95 per cent saturated. Then the temperature at which the condition line crosses the 95 per cent saturation curve is the proper temperature at which to regulate the apparatus. This temperature is easily found to be 59.6 F dry-bulb temperature (Point *A* in Fig. 12). Of course, a somewhat larger quantity of air will have to be recirculated, namely 114,600 ÷ (31.41 − 25.51) = 19,400 lb of dry air per hour, where the enthalpy of the air at 95 per cent saturation on the condition line is 25.51 Btu per pound of dry air and h = 31.41 for the inside air. Only 18,056 lb of dry air had to be recirculated per hour with complete saturation.

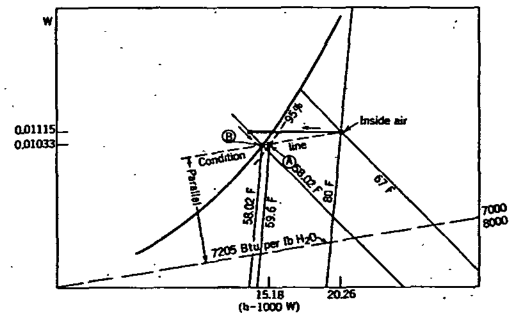

FIG. 12. DIAGRAM OF MOLLIER CHART ILLUSTRATING EXAMPLE 1

Many practicing engineers prefer an approximate method of calculating apparatus dew-point which employs what is called the *sensible heat factor*, *SHF*. The so-called *room sensible heat*, H_s, which is proportional to the quantity of supply air, Q is first estimated:

$$H_s = Q \times d \times C_p \times 60\,(t_r - t_e) \tag{1}$$

where

Q = quantity of supply air, cubic feet per minute.
d = density of air, pounds per cubic foot.
C_p = constant pressure specific heat of air.
t_r = room dry-bulb temperature, degrees Fahrenheit.
t_e = apparatus dew-point or supply air temperature, degrees Fahrenheit.

The so-called *room latent heat*, H_l, which is also proportional to the quantity of supply air, Q, is estimated:

$$H_l = Q \times d \times h_{fg} \times 60\,(W_r - W_e) \tag{2}$$

where

Q = quantity of supply air, cubic feet per minute.
d = density of air, pounds per cubic foot.
h_{fg} = an appropriate value of the latent heat of vaporization, Btu per pound.
W_r = room humidity ratio, pounds water per pound dry air.
W_e = apparatus dew-point or supply air humidity ratio, pounds water per pound dry air.

Finally the ratio

$$SHF = \frac{H_s}{H_s + H_l} \tag{3}$$

is computed. But, from Equations 1, 2 and 3

$$SHF = \frac{0.241\,(t_r - t_e)}{h_r - h_e} \tag{4}$$

where

0.241 = an appropriate value of C_p, Btu per pound.
t_r = room dry-bulb, degrees Fahrenheit.
t_e = apparatus dew-point, degrees Fahrenheit.
h_r = enthalpy at room conditions, Btu per pound dry air.
h_e = enthalpy at apparatus dew-point, Btu per pound dry air.

Thus, having estimated the value of the sensible heat factor, *SHF*, the problem of finding the apparatus dew-point reduces to solving Equation 4. Since the relation between h_e and t_e is only available in tabular or graphical form, a trial-by-error or graphical solution is required. Special charts have been devised for this purpose, but these are really unnecessary since the Mollier Chart of Chapter 1 is adaptable. Thus the quantity

$$q = \frac{h_{fg}}{1 - SHF} \tag{5}$$

where h_{fg} is an appropriate value of the latent heat of vaporization, determines, on the border scale of the Mollier Chart, the direction of the condition line which intersects the saturation curve at the apparatus dew-point.

Example 1. From a cooling load analysis of the room as determined in *Example 19*, Chapter 1, the net energy gain of 114,600 Btu per hour would be called ***room total heat***; the net moisture gain of 15.92 lb per hour, multiplied by an appropriate value of the latent heat, say 1065 Btu per pound, would be called the ***room latent heat***, namely, 16,950 Btu per hour; and 114,600 − 16,950 = 97,650 Btu per hour is the ***room sensible heat***. Determine the apparatus dew-point, if the room conditions to be maintained are 80 F dry-bulb and 67 F wet-bulb temperature.

Solution. From Equation 3 the sensible heat factor is *SHF* = 97,650 ÷ 114,600 = 0.852. Using the latent heat of vaporization at 51 F, namely 1065 Btu per pound as an appropriate value in Equation 5, q = 1065 ÷ (1 − 0.852) = 7205 Btu per pound water.

The state point of the inside air is easily located on the Mollier Chart, as shown in Fig. 12. Through this point draw a line having a slope q = 7205 Btu per pound water as determined from the border scale. This line crosses the saturation curve at 58.02 F which is the apparatus dew-point temperature (Point *B* in Fig. 12).

From the point of view of satisfying the given cooling or heating load requirements, the effect of air passing through the apparatus without being *contacted* is merely to increase the quantity to be passed through. Thus, if 20 per cent of the air admitted is uncontacted, then 25 per cent more air must be admitted than would be necessary if all of it were contacted. The failure to achieve complete saturation in commercial apparatus may be explained as failure to contact all the air admitted. Or, some of the air may be deliberately by-passed. But whatever the explanation, the only thing of importance is the end result. Of course, a low saturation efficiency or a large proportion of uncontacted air means higher capacity for the conditioning apparatus and, perhaps, an excessive number of air changes for comfort.

In winter, a degree of saturation in excess of 30 per cent is seldom required and a low saturation efficiency may be desirable or even necessary where the full summer circulation air through the conditioner is maintained. With a spray type dehumidifier the main sprays may be shut off and an eliminator flooding pump provided which may give sufficient saturation. In other cases, such as where cooling coils are sprayed, the spray water may be throttled.