inside air. Only 18,050 lb of dry air had to be recirculated per hour with complete saturation. (Example 13, Chapter 3.)

The supply air must have the capacity to absorb the room sensible heat gain and the room moisture (latent heat) gain simultaneously and in the correct ratio. The top portion of Fig. 13 shows the relationship of sensible heat and latent heat absorbing power of air supplied to an 80 F room when room relative humidity is 50 per cent[2]. On some psychrometric charts the line connecting the room condition and the apparatus dew-point on the saturation line is equivalent to the condition line on the previously mentioned Mollier Diagram.

Most practicing engineers prefer other methods of calculating apparatus dew-point which use one of several forms to express the ratio of room sensible heat gain to room moisture (latent heat) gain. The room latent gain is the heat equivalent, in Btu, of the amount of moisture liberated within the space or introduced to the space. This heat equivalent is approximately equal to the latent heat of vaporization at the apparatus dew-point temperature. Since this is variable, an average value of 1050 Btu per pound of moisture gain may be used without serious error for normal air conditioning calculations, when the room and apparatus dew-point temperatures are above 32 F. Since the load calculation, particularly the room latent heat gain, is only approximate, it is believed that these approximate methods for calculation of apparatus dew-point are sufficiently accurate for most commercial applications. Several methods are commonly used to express the relation of room sensible heat gain to room latent heat gain and these are described in the following:

$$\text{Sensible Heat Factor, (SHF).} \quad \text{SHF} = \frac{H_s}{H_s + H_l} \tag{1}$$

$$\text{Latent Heat Factor, (LHF).} \quad \text{LHF} = \frac{H_l}{H_s + H_l} \tag{2}$$

$$\text{Multiplier Factor, (MF).} \quad \text{MF} = \frac{H_s + H_l}{H_s} \tag{3}$$

where

H_s = room sensible heat gain, Btu per hour.
H_l = room latent heat gain, Btu per hour.

These equations are equivalent to one another as shown by the following:

$$\text{LHF} = 1 - \text{SHF} \tag{4}$$

and

$$\text{MF} = \frac{1}{\text{SHF}} \tag{5}$$

The sensible heat gain in the room or space must be absorbed by the supply air in order to maintain a given room temperature. Then the following heat balance for saturated supply air may be written as:

$$H_s = (t_r - t_s)\, Q_m \times \rho \times c_p \times 60 \tag{6}$$

where

H_s = room sensible heat gain, Btu per hour.
t_r = room temperature, dry-bulb, Fahrenheit degrees.
t_s = apparatus dew-point temperature, Fahrenheit degrees.

Q_m = quantity of supply air, cubic feet per minute.
ρ = density of supply air, pounds per cubic foot.
c_p = specific heat of air, at constant pressure, Btu per (Fahrenheit degree) (pound).

Similarly the latent heat (moisture) gain in the room must be absorbed by the lower moisture content of the supply air in order to maintain a given relative humidity, and writing a heat balance for saturated supply air:

$$H_l = (W_r - W_s)\, Q_m \times \rho \times h_v \times 60 \tag{7}$$

where

H_l = room latent heat gain, Btu per hour.
W_r = moisture content of room air, pounds per pound of dry air.
W_s = moisture content of air, pounds per pound of dry air at apparatus dew-point.
h_v = heat of vaporization at apparatus dew-point temperature.

Equation 1 shows how to determine the sensible heat factor of the cooling load. The sensible heat factor of the supply air must equal the sensible heat factor of the room load in order to maintain design conditions.

If a dehumidifier that will produce a saturated condition of leaving air is used then Equations 6 and 7 may be solved for H_s and H_l and substituted in Equation 1. This for practical purposes can be shown to be equal to:

$$\text{SHF} = \frac{0.24\,(t_r - t_s)}{h_r - h_s} \tag{8}$$

where

0.24 = specific heat of air, Btu per (pound) (Fahrenheit degree).
t_r = room temperature, dry-bulb, Fahrenheit degrees.
t_s = apparatus dew-point, Fahrenheit degrees.
h_r = enthalpy of air at room conditions, room wet-bulb, Btu per pound of dry air.
h_s = enthalpy of air at apparatus dew-point, Btu per pound of dry air.

Practically, the apparatus dew-point temperature may be defined, for a cooling coil, as the average coil surface temperature; and for a spray dehumidifier of commercial design, (where the saturation efficiency approaches 95 per cent to 100 per cent) as the dew-point of the air leaving the dehumidifier.

Having estimated the room sensible heat factor the problem is reduced to the solving of Equation 8. Since the relation between h_s and t_s is only available in tabular or graphical form, a cut and try or graphical solution is required. Fig. 14 has been calculated from Equation 8 and, of course, can be used only for one room dry-bulb temperature, 80 F. The lower portion of Fig. 13 also gives the apparatus dew-point required for various sensible heat factors. Note that if air is supplied at any temperature above the room dew-point, only sensible heat is absorbed (SHF = 1.0); and that below this point moisture is absorbed in a very rapidly increasing proportion to sensible heat. A point is finally reached where the proportion of moisture absorbed to sensible heat absorbed is practically constant and a lower apparatus dew-point will not remove a greater proportion of moisture. A room condition can be maintained if the load sensible heat factor lies on the curve corresponding to the design room humidity in Fig. 14. For example, if the load SHF was 0.55 for a design room condition of 80 F dry-bulb and 50 per cent relative humidity, it would not be possible to maintain this condition unless reheat