formed in the cooling operation is represented by line BC whose projection on the ordinate axis is the quantity of liquid so separated per pound of dry air. Point C is the apparatus dew-point and lies on the condition line as required.

In practice it may not be feasible to choose the apparatus dew-point as the point on the condition line to which to condition the *inside air* because to do so would require an excessive number of air changes in the given space. Or it may be that the condition line does not cross the saturation curve at all so that the apparatus dew-point as defined does not exist. Finally, it is rarely possible to obtain complete saturation in conventional air conditioning apparatus. Nevertheless the requirements of the cooling load problem can be exactly met if the conditioned air is brought to *any* point on the condition line of the problem.

Heating Load

The condition line is also useful in the analysis of heating load problems as may best be illustrated by means of an illustrative example.

Example 14. A certain space is to be maintained at 70 F and 50 per cent saturation with outside conditions at 0 F and 80 per cent saturation. The normal heat transmission through walls, partitions, floor, roof, glass and doors is estimated at 75,000 Btu per hour. Energy gained from lights and appliances is estimated at 15,000 Btu per hour.

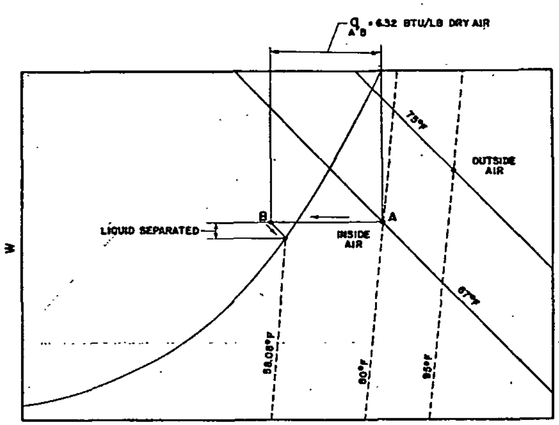

Fig. 10. Illustration of Use of Mollier Diagram in Solution of Example 13

Energy and water gains from occupants are to be disregarded in the calculations. Double doors and windows are used so that infiltration is negligible. The ventilation requirement is 30,000 cu ft per hour of *outside air*.

The requirements of the problem are to be met in the following manner; preheat the *ventilating air*; mix it adiabatically with recirculated *inside air*; saturate the mixture adiabatically with recirculated spray water; heat the resulting mixture to 105 F and return it to the conditioned space as *supply air*.

Analysis. Every pound of dry air admitted to the system (air conditioned space plus air conditioning apparatus) with the *ventilating air* displaces a pound of dry air from the system with *inside air*. Since the *ventilating air* is not admitted directly to the space, then for every pound of dry air withdrawn with *inside air* there is a pound of dry air returned with *supply air*. This has to have the net effect of adding energy of amount 60,000 Btu per hour and water of amount *zero* pounds per hour. Thus the ratio q determining the direction of the condition line is infinite, which means that the condition line is horizontal as indicated by the protractor on the Mollier Diagram.

The properties of *inside air* are: $h = 25.451$, $W = 0.007910$. Since the state point of the *supply air* must be on the condition line at 105 F, its properties are: $h = 33.986$, $W = 0.007910$. Therefore the weight of dry air withdrawn with *inside air* and returned with *supply air* is $60{,}000 \div (33.986 - 25.451) = 7029.9$ lb per hour.

The properties of *outside air* are: $h = 0.668$, $W = 0.0006298$, $v = 11.590$. Therefore the weight of dry air introduced into the system with the *ventilating air* is $30{,}000 \div 11.590 = 2588.4$ lb per hour. This *ventilating air* is to be mixed adiabatically with *inside air* containing $7029.9 - 2588.4 = 4441.5$ lb of dry air per hour; therefore, the humidity ratio of the mixture must be $(2588.4 \times 0.0006298 + 4441.5 \times 0.007910) \div 7029.9 = 0.005229$.

The condition line crosses the saturation curve at 50.86 F where the enthalpy is 20.782 and the humidity ratio is 0.007910. This is the state point to be reached by adiabatic saturation of the mixture of *ventilating air* and *inside air* with recirculated spray water. Accordingly, the state point of the mixture must lie on the 50.86 F thermodynamic wet-bulb line so that its enthalpy must have the value,

$$h = 20.782 - (0.007910 - 0.005229) \times 18.97 = 20.731$$

This requires that the enthalpy of the preheated *ventilating air* have the value,

$$h = (7029.9 \times 20.731 - 4441.5 \times 25.451) \div 2588.4 = 12.632$$

Since the humidity ratio of the preheated *ventilating air* is known to be 0.0006298, its temperature is readily found to be 49.75 F.

The quantity of heat required for preheating the *ventilating air* is $2588.4 \times (12.632 - 0.668) = 30{,}968$ Btu per hour; that to be added to the *supply air* is $7029.9 \times (33.986 -$

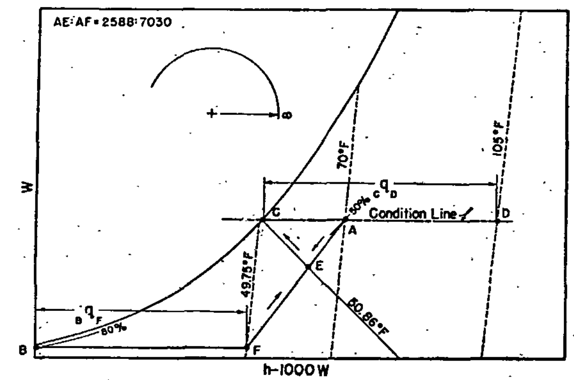

Fig. 11. Illustration of Use of Mollier Diagram in Solution of Example 14

$20.782) = 92{,}823$ Btu per hour; the energy added with the spray water is $7029.9 \times 18.97 \times (0.007910 - 0.005229) = 357$ Btu per hour; that introduced into the system with the *ventilating air* is $2588.4 \times 0.668 = 1729$ Btu per hour; that carried out of the system with the *inside air* displaced by the *ventilating air* is $2588.4 \times 25.451 = 65{,}877$ Btu per hour; therefore, the net energy added to the system is $30{,}968 + 92{,}823 + 357 + 1729 - 65{,}877 = 60{,}000$ Btu per hour as required.

On the Mollier Diagram, Fig. 11, point A is the state point of the *inside air*. The condition line is horizontal so that point D is the state point of the *supply air*. The condition line crosses the saturation curve at point C so that the state point of the mixture of preheated *ventilating air* and *inside air* before adiabatic saturation with recirculated spray water must lie somewhere on the thermodynamic wet-bulb line through C. The state point of the *ventilating air* is point B, hence that of the preheated *ventilating air* must lie somewhere on the horizontal line through B. Its exact location is determined graphically by finding the straight line AF which is cut by the thermodynamic wet-bulb line through C into two segments such that AE : AF = 2588.4 : 7029.9. The length of the line BF is the quantity of heat required for preheating the *ventilating air* per pound of dry air; the length of the line CD is the quantity of heat to be added to the *supply air*, per pound of dry air.