design the air leaves the dehumidifier at 1 to 2 deg higher wet-bulb temperature than the spray water leaving the dehumidifier, and the difference between the dry-bulb and wet-bulb temperatures leaving the dehumidifier may be as low as 1 deg. A spray type dehumidifier having sufficient length of spray chamber and density of spray, together with proper arrangement of nozzles, may approach saturation very closely.

As explained in Chapter 3, and shown in Fig. 5, the slope of the line on the psychrometric chart connecting the room condition with the apparatus dew-point on the saturation line, determines the ratio of sensible heat absorbing capacity to the moisture absorbing capacity of the supply air. Therefore the room condition can be maintained as long as the supply air temperature lies on this line, but a greater volume of supply air must be used to satisfy the room load if the cooling coil does not contact 100 per-

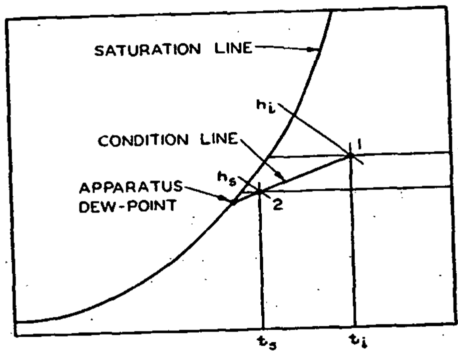

FIG. 5. APPARATUS DEW-POINT SHOWN ON A.S.H.V.E. PSYCHROMETRIC CHART

cent of the air. For a given room load, the same apparatus dew-point will be required whether the cooling appliance contacts all the air or only part of the air.

From the point of view of satisfying the given cooling load requirements, the air passing through the apparatus without being cooled below the dew-point temperature produces two effects:

1. The air quantity which must be passed through the dehumidifier must be increased. Thus, if 20 percent of the air passing is contacted, then $(20 \div 80) \times 100 =$ 25 percent more air must be used than would be necessary if all of it were contacted.

2. Passing untreated air may change the room cooling load, which in turn may change the enthalpy-humidity difference ratio (sometimes called the sensible heat factor). If return air only is passed through the dehumidifier or if room air only is by-passed, the room load will not change, but if some outside air is passed through the room sensible heat gain and room latent heat gain will be changed due to the addition of untreated outside air, which changes the enthalpy-humidity difference ratio. When a load calculation is made, it is necessary to know the percentage of air affected in the dehumidifier, and calculation must be made accordingly.

If the ventilation air is drawn through the dehumidifier before it goes into the room, only that portion of the air not saturated must be included

in the room load for the purpose of determining the apparatus dew-point and supply air quantity. It should be noted when evaluating the load added by untreated outside air that the temperature difference between room air and outside air, and the moisture content difference between room air and outside air, should be used, rather than the difference between outside air and apparatus dew-point, since the rise from the apparatus dew-point to room condition is charged against the dehumidifier as the cooling and dehumidifying load.

The procedure for determining the required air quantity is based upon the thermodynamic principles of Chapter 3 and the use of the A.S.H.V.E. psychrometric chart. Readers are advised to review these principles, paying particular attention to the illustrative examples of cooling load calculations.

Calculation of the cooling load for a conditioned space is equivalent to making, for the space, a heat balance in which all heat, moisture, and infiltration are treated as directly entering the space. As explained in the section, *Load from Outside Air, Ventilation, and Infiltration*, the outside air load normally does not become a part of the space load, because heat and moisture are removed in the air conditioner before this air gets into the conditioned space. The desired conditions are maintained by considering a certain quantity of air to be withdrawn from the space, passed through the conditioning equipment, and returned to the space with such a temperature and humidity ratio that its net effect will be to counterbalance or *remove* the given entering amounts of heat and water vapor. This quantity of *indoor air*, which is considered to be circulated in this manner, is called the *required air quantity* and its determination is normally part of every cooling-load estimate. The procedure is as follows:

1. Determine the total sensible and latent heat loads in Btu per hour for the space.

2. Compute the quantity called the enthalpy-humidity difference ratio (also referred to as heat-moisture ratio) of the room load, $\frac{h - h_s}{W_i - W_s}$. Use the equation:

$$\frac{h_i - h_s}{W_i - W_s} = \frac{\text{(Space sensible load + space latent load)}}{\text{Space latent load/1076}} \qquad (18)$$

where

h_s = enthalpy of moist air supplied to the space, Btu per pound of dry air.

h_i = enthalpy of moist air at room design conditions, Btu per pound of dry air.

W_s = humidity ratio of moist air supplied to the space, pounds of vapor per pound of dry air.

W_i = humidity ratio of moist air at room design conditions, pounds of vapor per pound of dry air.

Note that the ratio (space latent load/1076) is the equivalent of the required rate of water vapor removal in pounds per hour. If the rate of water removed is known, it may be used directly in Equation 18.

3. Draw a line through the reference point on the A.S.H.V.E. psychrometric chart and the value of $(h_i - h_s)/(W_i - W_s)$ determined above. Draw a second line through the state point of the room air (design wet-bulb and dry-bulb temperatures) parallel to this line. This is the *condition line* for the process.

4. Read the temperature where the condition line from step 3 intersects the saturation line. This is called the *apparatus dew-point*.

See Fig. 5. Note that instead of using this graphical method the left hand sid