SURFACE COOLING UNIT METHOD

The fourth type of system employing a surface cooling unit of the extended-surface type will cool and dehumidify. A refrigerant is circulated or expanded within the unit and air is blown over it. This system has the advantages of lower initial cost and low operating cost, and it does not require as much attention as a spray-type system.

For comfort cooling, water is usually used as the refrigerant. If a refrigerant with a temperature lower than 32 F is used, care must be exercised in the design to prevent frosting. Low temperature refrigerants often reduce the moisture content of the air lower than is usually required. Water at 45 to 50 F is quite practical, using a 10 to 15 deg rise in water temperature depending on the quantity of water used and the velocity of the water required through the tubes.

Although surface cooling units have advantages over the conventional spray-type systems, they are usually adaptable for both cooling and heating, and they can be used to control humidity in summer. The effective cooling accomplished by the unit is dependent upon many variable factors. The air velocity through the unit, air temperature, moisture content, water temperature, and velocity of the water through the tubes must all be considered in designing the unit. If any of these factors vary without a corresponding variation of the other factors, the effective cooling obtained by the unit will drop off.

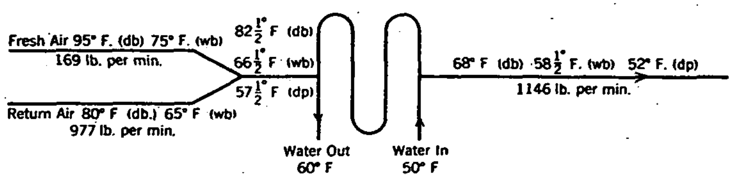

FIG. 5. DIAGRAM OF SURFACE COOLING METHOD

Example 4. Surface Cooling Method. (Fig. 5). Assuming the same required conditions and load as in Example 1, a surface cooling unit of proper size and capacity may be determined by the following calculations:

Outside air	169 lb per minute at 95 F (db), 75 F (wb)
Recirculated air	977 lb per minute at 80 F (db), 65 F (wb)
Air leaving unit	1146 lb per minute at 68 F (db), 59.56 F (wb), 54.17 F (dp)
Water, in	50 F
Water, out	60 F

Water flowing counter to air flow.

It will be noticed from Fig. 5 that the dew-point temperature of the entering mixture is lower than the leaving water temperature but higher than the entering water temperature. Condensation of the moisture in the entering air will therefore occur some place in the unit where the temperature of the pipe or surface is lower than 57½ F. Condensation will continue to the leaving end of the unit and will result in a dew-point temperature of the leaving air between 50 F and 54.17 F, or approximately 52 F. With this dew-point temperature and the dry-bulb temperature of the air leaving of 68 F, which has been set by pre-determined calculations, the new wet-bulb temperature will be 58½ F*.

*It can readily be shown that if the entering water temperature were 54.17 F and the other conditions were such that the leaving dew-point temperature were 54.17 F as desired, the refrigeration load would be the same as for the by-pass and local recirculation systems. These conditions, however, have not been approached in practice to date.

The total heat to be removed by the water will be

$$\begin{aligned}\text{Total heat at } 66\tfrac{1}{2}\text{ F (wb)} &= 30.77\\ \text{Total heat at } 58\tfrac{1}{2}\text{ F (wb)} &= 25.20\\ \hline &\quad 5.57 \text{ Btu per pound of air}\end{aligned}$$

$$1146 \times 5.57 = 6383 \text{ Btu per minute}$$

$$\text{Refrigeration} = \frac{6380}{200} = 31.90 \text{ tons}$$

$$\text{Quantity of water required} = \frac{6380 \text{ Btu per minute}}{10 \text{ deg} \times 8.33} = 76.6 \text{ gpm}$$

As the cooling efficiency is dependent partly on the velocity of water through the tubes, this must be determined from a manufacturer's catalog. For instance, one manufacturer gives the formula

$$V = \frac{(\text{gallons per minute}) \times 1.235}{36} \tag{3}$$

where

V = velocity in feet per second.
36 = number of tubes per unit.
1.235 = a constant for the particular sized pipe used.

Using Formula 3, the velocity would be,

$$V = \frac{76.6 \times 1.235}{36} = 2.63 \text{ fps}$$

To determine the amount of cooling surface required it is necessary to ascertain the mean temperature difference between the air and water, and the coefficient of transmission. The formula for the mean temperature difference is:

$$T_d = \frac{(T_1 - T_2) - (T_3 - T_4)}{\text{Log}_e\left(\dfrac{T_1 - T_2}{T_3 - T_4}\right)} \tag{4}$$

where

T_d = mean temperature difference.
T_1 = entering dry-bulb temperature of air.
T_2 = leaving water temperature.
T_3 = leaving dry-bulb temperature of air.
T_4 = entering water temperature.

$$T_d = \frac{(82\tfrac{1}{2} - 60) - (68 - 50)}{\text{Log}_e\left(\dfrac{82\tfrac{1}{2} - 60}{68 - 50}\right)} = \frac{22\tfrac{1}{2} - 18}{\text{Log}_e\left(\dfrac{22\tfrac{1}{2}}{18}\right)} = 20.2 \text{ F}$$

The coefficient of transmission is usually taken from the manufacturers data, knowing the water velocity through the unit and the air velocity over the tubes. The velocity of the water through the tubes has been determined, and assuming a velocity of 600 fpm for the air the coefficient of transmission is (from manufacturer's curves):

K = 9.7 Btu per square foot surface per mean temperature difference.

The cooling surface required is usually based upon the sensible heat load. The latent heat due to condensation is taken out at the same time the sensible heat is extracted, and no extra surface is required unless the latent heat exceeds approximately 40 per cent of the total heat. This is due to the higher coefficient of transmission factor because of the wetted surface. This factor holds fairly consistent up to a point where the latent heat