

heat (or wet-bulb line) to its intersection with the 70 per cent relative humidity curve and read 94 F dry-bulb, which will be the temperature of the air leaving the dryer.

Moisture per cubic foot at 94 F and 70 per cent relative humidity = 11.8 grains  
 Moisture per cubic foot at 95 F and 78 F wet-bulb = 8.0 grains

Moisture added per cubic foot of air handled = 3.8 grains

$$\frac{1700 \times 7000}{24 \times 60 \times 3.8} = 2170 \text{ cfm.}$$

No allowance is made for heat lost in the transmission to and from the dryer or for the heat required to raise the product from its entering temperature to that maintained in the dryer. This would necessitate a trial and error solution common to all drying problems.

6 • It is proposed to install a central fan type air conditioning system comprised of fan, air washer, filters, and heating coils to provide ventilation and to maintain proper humidity in a small library during periods of winter operation. The heat loss has been estimated at 450,000 Btu per hour in maintaining a condition of 72 F dry-bulb and 45 per cent relative humidity. Assuming that the air washer completely saturates the air, what must be the leaving dry- and wet-bulb temperatures to provide the required condition?

49.85 F is the dew-point temperature corresponding to the stated required condition.

7 • Assuming a maximum permissible air delivery temperature of 100 F in Question 6, what air volume will be required?

$$\frac{450,000 \times 55.2}{(100 - 72) \times 60} = 14,800 \text{ cfm.}$$

8 • If in Questions 6 and 7 it is assumed that winter humidity control will consist simply of a dew-point thermostat at the exit of the air washer, controlling the dew-point temperature by operating automatic dampers, and thereby proportioning the respective volumes of outside and recirculated air admitted:

- What volume of air should be recirculated?
- What volume of air will be exfiltrated from the buildings?
- What reheating capacity will be required?

a. Btu per pound at 72 F and 45 per cent relative humidity = 25.38  
 Btu per pound at 0 F (assumed saturated) = 0.85  
 Btu per pound at 49.85 F saturated = 20.11

$$\text{Recirculated air} = \frac{(20.11 - 0.85)}{(25.38 - 0.85)} \times 14,800 = 11,600 \text{ cfm.}$$

b. The same volume as is introduced as fresh outside air, namely,  
 $14,800 - 11,600 = 3200 \text{ cfm.}$

c. The reheaters must be of such capacity as to reheat the volume of air handled from 49.85 (the dew-point) to 100 F.

$$\frac{14,800 \times (100 - 49.85) \times 60}{55.2} = 808,000 \text{ Btu per hour.}$$

## Chapter 4

# NATURAL VENTILATION

*Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation*

VENTILATION by natural forces, supplemented in certain cases with mechanical forces, finds extensive application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for the displacement of air in buildings are the wind and the difference in temperature of the air inside and outside the building. The arrangement and control of ventilating openings should be such that the two forces act coöperatively and not in opposition.

## Wind Forces

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings and trees.

Table 1, Chapter 8, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 7, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings:

$$Q = EAV \quad (1)$$

where

Q = air flow in cubic feet per minute.  
 A = free area of inlet (or outlet) openings in square feet.  
 V = wind velocity in feet per minute,  
 = miles per hour  $\times 88$ .  
 E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle.)