to Case D except that a portion of the air is recirculated. All of the air circulated is passed through the washer and heater.

$$M = M_o + M_r = \frac{H}{0.24\,(t_y - t)} \text{ pounds per hour} \quad (7)$$

M_o is known from the ventilating requirements. Then $M_r = M - M_o$, weight of air recirculated. Assume 35 per cent relative humidity to be maintained in the rooms, which corresponds to a dew-point temperature of 41 F for t = 70 F, and a tempering coil to warm the entering outside air from zero to 35 F. The resulting temperature t_x of the mixture of outside and recirculated air entering the tempering coil is:

$$t_x = \frac{M_o\,(35 + 460) + M_r\,(t + 460)}{M} - 460 \quad (8)$$

A washer supplied with a water heater will raise the temperature of the air passing through from t_w to t_1 = 41 F and saturate it at this temperature.

$$H_1 = 0.24\,(t_2 - 41)M, \text{ for heater}$$

$$H_2 = 0.24\,(t_w - t_x)M, \text{ for tempering coil}$$

The heat to be supplied the washer per pound of air passing through the washer is equal to the difference between the heat content of saturated air at a temperature t_1 = 41 F (15.7 Btu per pound) and the heat content per pound of the mixture at a temperature of t_w and relative humidity to be calculated.

Assuming the outside air temperature t_o = 0, and dry; the inside air temperature t = 70 and 35 per cent relative humidity corresponding to 0.00552 lb of moisture per pound of air, there are M pounds containing $0.00552M$ lb moisture or $\frac{M}{0.00552M}$ lb moisture per pound of air entering the washer. Knowing the temperature t_w, the relative humidity is readily determined and the heat content per pound.

Case F. (Fig. 16) The temperature t_y will ordinarily be different for each room. With H and M_o fixed, $0.24\,(t_y - t)\,M_o = H$, or

$$t_y = \frac{H}{0.24\,M_o} + t \quad (9)$$

In order to provide the proper temperature t_y for each room, the so-called *hot* and *cold* or *double plenum chamber* system is employed. The weight of air drawn in from outside the building is the sum of the values for M_o as determined by the ventilating requirements for each room and is the total weight of air *passing through the tempering coil* and washer and entering the rooms. Each room is provided with an independent supply duct run to the heater plenum chamber; so that varying amounts of bypassed tempered air and hot air may be mixed to obtain the required temperature (t_y) for each room independently ($t_m = t_y - t_z$). The mixing dampers are ordinarily placed under thermostatic control.

Ducts and Outlets

The design of the duct system should be based on data contained in Chapter 32. The total friction against which the fan must operate is the sum of the resistances of all of the elements of the entire system.

Fans and Motive Power

The selection of fan and motor should be based on data contained in Chapter 34. Centrifugal fans are generally used as they are well adapted for working against the frictional resistance of the system, and reach their maximum efficiency when working against the resistance offered by the average central fan heating system.

Chapter 32

AIR DISTRIBUTION SYSTEMS

Dynamic and Friction Losses; Friction Chart; Proportioning Losses: Design of Ducts; General Rules; Procedure; Air Velocities; Proportioning for Friction; Plenum Chamber and Individual Ducts; Main Trunk Ducts with Branches; Prevention of Noise; Duct Construction: Air Distribution; Stratification and Diffusion; Downward and Upward Air Distribution; Inlets and Outlets; Grilles and Registers; Measurement of Air Flow; Pitot Tubes; Anemometers; Kata Thermometers.

THE laws governing the flow of fluids are based on the assumption that the density remains constant throughout the flow. In considering the flow of a gas such as air, these laws do not strictly hold. The velocity of flow in an air duct of uniform size through which a given weight of air is passing will vary with an increase or decrease in pressure which causes a corresponding decrease or increase in volume.

The flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated and the error occasioned by the use of the same formulae that apply to the flow of fluids may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice. The pressure difference in this field seldom exceeds more than 2 in. water gage, and it is sufficiently accurate in considering the relations between pressure head and velocity of flow for air in ducts to apply the same formulae as are used for the flow of liquids. The basic formula is:

$$V = 1096.5\sqrt{\frac{p}{W}} \quad (1)$$

where

V = velocity in feet per minute.
p = head or pressure in inches of water.
W = weight of air in pounds per cubic foot.

For standard air (70 F, and 29.92 barometer) W = 0.07495 lb per cubic foot. Substituting this value in Equation 1:

$$V = 1096.5\sqrt{\frac{p}{0.07495}} = 4005\sqrt{p} \quad (2)$$

An inspection of Equation 1 indicates that temperature affects the velocity of flow by changing the weight per cubic foot of air. Tables 5 and 6 (Chapter 39) give the velocities at various pressures and temperatures.