The net static pressure loss in the duct only is then:

0.182 in. − 0.059 in. .... 0.123 in.

Other friction losses are as follows:

(1) Fresh air intake 1000-fpm velocity (1½ heads × 0.0625) .... 0.094 in.
(2) Tempering coil loss (from manufacturers tables) .... 0.100 in.
(3) Air washer loss (from manufacturers tables) .... 0.250 in.
(4) Reheating coil loss (from manufacturers tables) .... 0.100 in.
(5) Allowance for regulating dampers and diffusers .... 0.100 in.

Static pressure loss of system .... 0.767 in.

The fan should be selected from the manufacturers ratings which, according to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[1], will deliver 22,935 cfm at a static pressure of 0.767 in. and which has an outlet area of 16½ sq ft.

The method of design used in Example 3 is the *equal friction method* described under the heading Procedure for Duct Design. This involves the arbitrary reduction of velocity from the fan outlet to the point of discharge to the room, and the friction is calculated by adding the pressure losses of each section of duct. This method requires dampering in the risers.

Example 4. Fig. 7 shows an exhaust system layout for exhausting from buildings of the same type as in Example 3. Assume the air requirements based on the number of air changes per hour to be 16,800 cfm. Using a velocity of 1400 fpm in the main duct at the fan inlet, which is an average velocity for this type of system, the area of the main is 12 sq ft, which corresponds to a 47-in. pipe. Referring to Example 3, and using the charts, Figs. 4 and 5, the pipe sizes are as indicated in Table 3.

All risers will require dampering as in Example 3. The calculation of the friction is as follows:

The longest run from the intake grille to fan inlet is 100 ft.

(1) Duct friction 100 ft of 47-in. pipe $\left(\frac{100 \times 12}{47}\right)$ .... 25.6 dia.

Two 28½-in., 90-deg elbows in riser $\left(\frac{2 \times 28.5 \times 30}{47}\right)$ .... 36.4 dia.

(Two bad elbows in riser each equivalent to 30 diameters of duct).

TABLE 3. PIPE SIZES FOR EXAMPLE 4

Volume of Air (cfm)	Per Cent of Total Volume	Diameter of Pipe (Inches)	Equivalent Size of Rectangular Duct (Inches)
16,800	100.0	47	38 x 48
11,550	68.8	41	30 x 46
9,450	56.2	38	30 x 40
5,250	31.3	31	24 x 34
4,200	25.0	28.5	24 x 28
3,150	18.8	25.3	16 x 34
2,100	12.5	21.6	16 x 24

[Velocity through intake grilles (not shown) to be approximately 400 fpm].

[1]See Chapters 17 and 42.

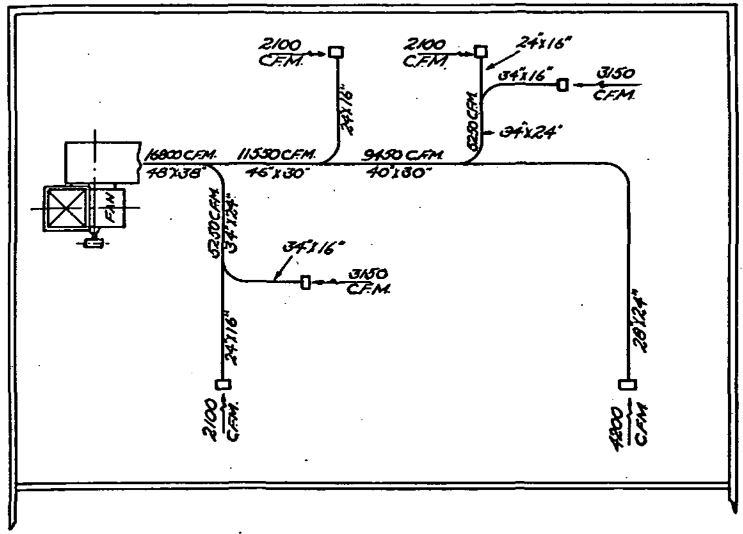

FIG. 7. EXHAUST SYSTEM LAYOUT

One 28½-in., 90-deg elbow in horizontal run $\left(\frac{28.5 \times 12}{47}\right)$ .... 6.0 dia.

Total diameter of 47-in. pipe .... 68.0 dia.

Velocity head corresponding to 1400 fpm is $\left(\frac{1400}{4005}\right)^2 = 122$ in.

Taking 50 diameters as one head loss, then $\frac{68 \times 0.122}{50}$ .... 0.166 in.

(2) Intake loss from grille (1½ heads at a 400 fpm velocity 1½ × 0.01) .... 0.015 in.
(3) Static pressure required to produce one velocity head at 1400 fpm .... 0.122 in.
(4) Loss occasioned by step-up of velocity (0.20 × 0.122) .... 0.024 in.

(This loss varies from 0.05 to 0.40 velocity heads depending upon the nature of the change. For average systems 0.20 velocity heads is a close approximation).

Static pressure loss on inlet side .... 0.327 in.

To this must be added the resistance on the discharge side of the fan. A fan outlet velocity of approximately 1500 to 1600 fpm may be used. Assuming the fan outlet to be equivalent in area to a 45-in. pipe, the velocity is 1525 fpm.

Loss on discharge (15 ft from fan outlet to discharge):

$$\frac{15 \times 12}{45} = 4 \text{ diameters of 45-in. pipe.}$$

The velocity head corresponding to a velocity of 1525 fpm is 0.145 and the discharge-side loss is $\frac{0.145 \times 4}{50} = 0.012$ in. The total static pressure loss of the system is then:

$$0.012 + 0.327 = 0.339 \text{ in.}$$

The fan will be selected to handle 16,800 cfm at a static pressure of 0.339 in. and to have an outlet velocity of 1525 fpm. Outlet area 11 sq ft.