The velocity head corresponding to a velocity of 1340 fpm is $\left(\frac{1340}{4005}\right)^2 = 0.112$ in.

Taking 50 diameters as one head loss, then $\frac{81.2}{50} \times 0.112 = 0.182$ in. static loss in duct.

Where the connection pieces are made with long easy slopes and the general workmanship is good, a regain in static pressure may be deducted from the foregoing pressure loss. This can be taken as approximately two-thirds the difference in velocity pressures at the fan outlet and the last run of pipe. The velocity in the riser is 667 fpm with a corresponding velocity pressure of 0.033 in. The fan outlet velocity is 1400 fpm with a corresponding velocity pressure of 0.122 in. The regain equals ⅔ (0.122 − 0.033) = 0.059 in.

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 manufacturer's tables) ..... 0.100 in.
(3) Air washer loss (from manufacturer's tables) ..... 0.250 in.
(4) Reheating coil loss (from manufacturer's 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 manufacturer's 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 16⅓ 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.

TABLE 3. PIPE SIZES FOR EXAMPLE 4[a]

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

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

[1]See Chapters 17 and 41.

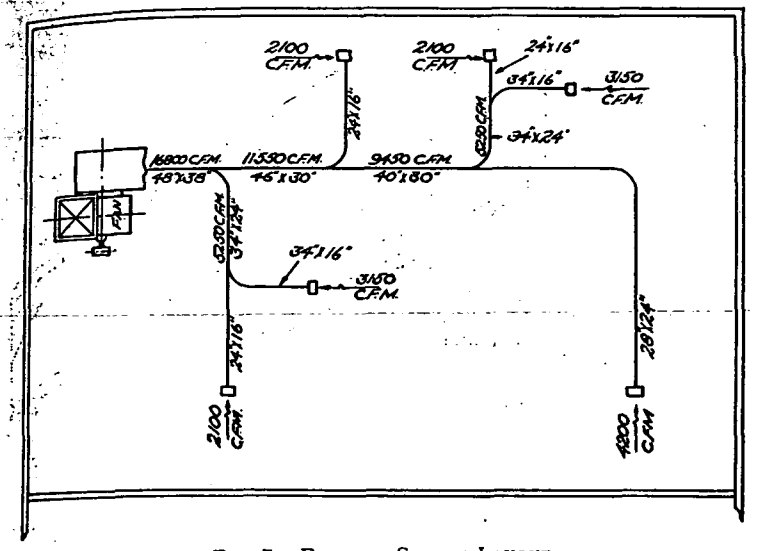

FIG. 7. EXHAUST SYSTEM LAYOUT

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).

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 head depending upon the nature of the change. For average systems 0.20 velocity head is a close approximation.)

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