toward the duct openings to the room. To find the total static pressure against which the fan must operate, the static pressure loss of each section must be calculated separately and the total loss found by adding the individual losses of the various sections of the run having the highest resistance. Usually this is the longest run but in some cases a shorter run may have more elbows, transformations, booster heaters, etc., which will cause it to have a higher resistance pressure. This method requires judgment and experience in choosing the proper velocities to approach equal friction for all lengths of run but many engineers believe that the velocity method is handier to use than other methods and will give satisfactory results for most practical applications. The air velocities given earlier in this chapter are helpful in choosing proper velocities. Adjustable dampers or splitters are used to regulate air quantities delivered.

Equal Friction Method

The equal friction method of design is sometimes preferred because it does not require nearly so much judgment and experience in selecting the proper velocities in the various sections of a system. The usual procedure in this method of design is to select the main duct velocity to be consistent with good practice from a standpoint of noise for a particular type of building. This velocity should be less than the fan outlet velocity. All main ducts and branch ducts are sized for equal friction by the use of Fig. 2 and Table 1 or Fig. 4.

In cases where the fan or factory assembled air conditioning unit has a limited external resistance, it is necessary to divide the available resistance by the total equivalent length of the longest or most complicated run of duct to determine the resistance per 100 ft and then to size all ducts at this resistance value, which will automatically determine the duct velocities and give the desired total duct resistance. A further refinement which is sometimes used in large systems is to size each branch duct so that it has a resistance equal to the resistance of the main system at the point of juncture. Even when this refinement is added, regulating dampers are recommended in each branch.

After the duct system is designed the frictional resistance is calculated and tabulated together with the resistance of all component parts. The fan is then selected for the required volume of air, static pressure and outlet velocity.

Example 2. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices. The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-min air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The free area of the outdoor air inlet is based on a velocity of 1000 fpm or 22,935 ÷ 1000 = 22.94 sq ft. The main duct velocity selected from Table 3 is 1250 fpm which gives a main duct area of 22,935 ÷ 1250 = 18.354 sq ft (60 × 44 in.). From Table 1 a 60 × 44 in. duct is approximately equivalent to 56 in. diameter.

Referring to Fig. 2, a volume of 22,935 cfm through a 56 in. diameter duct gives a resistance of 0.028 in. per 100 ft. The amount of air to be handled by each section of pipe is shown in Fig. 6, and by locating each of these values on the 0.028 in. friction line, the round pipe sizes are obtained and then, referring to Table 1, the equivalent rectangular sizes are selected as shown in Table 4.

TABLE 4. PIPE SIZES FOR EXAMPLE 2[a]

VOLUME OF AIR (CFM)	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	56	60 x 44
12,510	45	58 x 30
10,425	42	50 x 30
8,340	39	42 x 30
6,255	35	42 x 24
4,170	29.5	30 x 24
2,085	23	30 x 15

[a]Velocity through grilles (not shown) to be approximately 300 fpm.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser, or in each branch duct, and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 x 30 in. and 50 x 30 in. branches.

Resistance Losses for the System

(1) Outdoor air intake, 1000 fpm velocity (1.5 heads × 0.0625) ..... 0.094 in.
(2) Filters (from manufacturer's tables) ..... 0.250 in.
(3) Tempering coil loss (from manufacturer's tables) ..... 0.074 in.
(4) Air washer loss (from manufacturer's tables) ..... 0.250 in.
(5) Reheating coil loss (from manufacturer's tables) ..... 0.083 in.
(6) Duct resistance:

The longest run is ..... = 150 ft

Two, 58 x 30 in. elbows (150% ratio) $\frac{2 \times 13 \times 58}{12}$ ..... = 126 ft

Two, 30 x 15 in. elbows (150% ratio) $\frac{2 \times 13 \times 30}{12}$ ..... = 65 ft

Three, 15 x 30 in. elbows (75% ratio) $\frac{3 \times 35 \times 15}{12}$ ..... = 131 ft

Total equivalent run ..... 472 ft

472 ft at 0.028 in. per 100 ft ..... 0.132 in.

(7) Allowance for damper adjustment, 25% of 0.132 ..... 0.033 in.
(8) Supply grille resistance (from manufacturer's tables) ..... 0.036 in.

Total static pressure loss of system ..... 0.952 in.

The fan is selected from the manufacturer's ratings to deliver 22,935 cfm at a static pressure of 0.952 in. as outlined in Chapter 30.

Example 3. If the rooms and offices of the hotel building of Example 2 are to be served from a manufactured unit with a capacity of 22,935 cfm against an external resistance of 0.35 in., the known resistances are calculated as:

(1) Outdoor air inlet ..... 0.094 in.
(2) Allowance for damper adjustment ..... 0.033 in.
(3) Supply grille resistance (from manufacturer's tables) ..... 0.036 in.

Total known resistance ..... 0.163 in.