

manner described for the velocity-reduction method. The pressure available at each branch is divided by its equivalent length, in hundreds of feet, to obtain a design friction loss value for use with Fig. 2 or Fig. 3 in conjunction with the branch flow rate. The branch ducts are therefore sized as nearly as possible to dissipate all of the available pressure.

When using this modified method, care should be exercised that the velocities in short branches do not become excessive from a noise standpoint. This is easily guarded against during the design process, because the velocity can be read directly from the friction chart. If it is excessive, move horizontally to the left on the chart and select a diameter which yields a reasonable velocity. The damper for this run will have to dissipate the excess pressure. Since ductwork attenuates noise to some extent, the damper should be located as close to the main as possible. Sound treatment for this branch should also be considered. An alternative solution may be to revise the duct layout to increase the resistance of the run, for example, by relocating the branch take-off so that the total duct length is increased.

Example 6: (Equal-Friction Method). A duct layout is shown in Fig. 12. Outlets No. 1 and 2 deliver 750 cfm each and outlet No. 3 delivers 1000 cfm. Selecting a velocity of 1600 cfm in Section A, size the duct system and determine its static-pressure requirement.

Solution: The total cfm to be handled is 2500 cfm. From Fig. 3, with 2500 cfm and 1600 fpm velocity, read a diameter of 17 in. and a friction loss of 0.2 in. of water per 100 ft. By subtraction, the flow rate in Section B is 1750 cfm. Along the 0.2 friction line in Fig. 3, all of the ducts can be sized immediately because the flow rates are known. Results are presented in Table 7.

The rectangular equivalents were selected from Table 2 with the objective of having the same duct depth for all three branch runs.

The duct run to outlet No. 3 has the highest apparent resistance. It is decided to fabricate the elbow in Section C with a radius ratio of 1.2; hence, from Fig. 8 with $H/W = 1.9$, $L/W = 8$. Since $W = 1.25$ ft (15 in.), the additional equivalent length due to the elbow L is 10 ft. The total equivalent length of the run is therefore $(20 + 10 + 15 + 10 + 15) = 70$ ft. Therefore, at 0.2 per 100 ft the duct resistance is $0.2 \times 0.70 = 0.14$ in. of water. Adding to this the outlet pressure of 0.12 in., the static-pressure requirement of the duct system is 0.26 in. of water. The design is now complete, and dampers will be relied upon for adjusting the outlets to the design flow rates.

If refinement is deemed necessary, the modified method can be applied to Sections D and E. First, the static pressures available at the junctions with the main of the Section D and E branch ducts are obtained. For Section D it is the system pressure of 0.26 minus the friction pressure loss in Section A. The latter is $0.20 \times (20/100) = 0.04$; hence, the pressure at the entrance of Section B is 0.22 in. of water. Deducting the outlet pressure loss of 0.12, that available for the ductwork is 0.10. Assume the equivalent lengths of the branch take-off and the

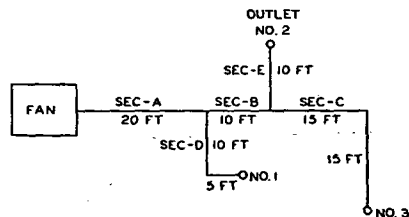


Fig. 12. Duct Layout for Example 6

Table 7. Tabulation of Results (Example 6)

Section	Flow Rate Cfm	Friction per 100 ft in. H ₂ O	Duct Dia. in.	Velocity Fpm	Rectangular Duct in.
A	2500	0.2	17.0	1600	20 × 12
B	1750	0.2	14.8	1480	15 × 12
C	1000	0.2	12.0	1200	15 × 8
D	750	0.2	10.7	1190	12 × 8
E	750	0.2	10.7	1190	12 × 8

elbow to be 10 ft each. The total equivalent length of Section D is then $(10 + 10 + 10 + 5) = 35$, and the friction loss per 100 ft required to dissipate 0.10 in. of water is $0.10 \times (100/35) = 0.29$. With this unit friction loss and a flow rate of 750 cfm, Fig. 2 yields a diameter of 10.0 in. and a velocity of 1380 fpm.

Section E is sized in a similar manner. The pressure available is 0.26 minus the friction loss in Sections A and B; hence, 0.20. With the outlet pressure loss of 0.12 deducted, the available ductwork pressure loss is 0.08 in. Assuming that the branch take-off loss is equivalent to 10 ft of duct, the total equivalent length is 20 ft. The required friction loss is $0.08 \times (100/20) = 0.40$. With this unit friction loss and a flow rate of 750 cfm, Fig. 2 yields a diameter of 9.4 in. and a velocity of 1530 fpm. An equivalent rectangular size is 9 × 8 in.

Comparing these results with those in Table 7, it is evident that the modified method has reduced the size of Section D somewhat and that of Section E considerably. The reduced sizes accomplish more economically what would otherwise have to be done with dampers.

Static-Regain Method

Consider a straight run of duct with several branch take-offs attached. The flow rate of air along the run is progressively reduced by the amount diverted into each successive take-off. If, for example, the size of the run were the same throughout its length, the velocity would become progressively less in accordance with Equation 3. When velocities are reduced, a conversion of velocity pressure into static pressure occurs (as well as a loss in total pressure). The principle of the static-regain method is to size a duct run so that the increase in static pressure (regain) at each take-off junction just offsets the pressure loss of the succeeding section of the run.

The method provides a convenient means of designing a long run of duct having several take-offs so that essentially the same static pressure exists at the entrance to each branch. If, instead of branch ducts, supply outlets are connected directly to the run, then essentially the same static pressure will exist behind each outlet. As a consequence, outlet selection and system balancing is simplified. The method is particularly suited to large installations having several long runs of duct, with each run having many take-offs or supply outlets attached. For this type of application, little or no dampening is ordinarily required to balance the system.

The initial velocity in the main duct is selected from noise and pressure loss considerations, and the branch ducts are sized by the modified equal-friction method.

If the distance between branch take-offs is either very small or very great, it may not be feasible or economically desirable to design for the same static pressure at each junction. In such cases, the method can be used to size the main for either a progressively lower static pressure (net static-pressure loss) or a progressively higher pressure (net static-pressure gain).

If no friction or dynamic losses occurred at the junction, there would be no loss in total pressure, and the change in

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velocity pressure would be completely converted into a regain (rise) in static pressure, which for standard air would be:

$$P_r = \left(\frac{V_1}{4005} \right)^2 - \left(\frac{V_2}{4005} \right)^2 \quad (12)$$

where

P_r = theoretical static-pressure regain, inches of water.

V_1 = velocity in main upstream of branch, feet per minute.

V_2 = velocity in main downstream of branch, feet per minute.

Under the best practice, 0.7 to 0.8 of the change in velocity pressure is actually recovered, but for practical design an average recovery of 0.5 is assumed. Hence, the actual regain P_r is:

$$P_r = 0.5 \left[\left(\frac{V_1}{4005} \right)^2 - \left(\frac{V_2}{4005} \right)^2 \right] \quad (13)$$

Design charts based on Equation 13 and rectangular ducts having aspect ratios of 3 to 1 or less are presented in Figs. 14 and 15.

The duct length of any section should include the equivalent length of any elbows or transitions within the section. The charts apply to constructions where regain takes place unaccompanied by radical change in direction; namely, to straight-through sections of divided-flow fittings.

Example 7: (Static-Regain Method). The duct shown in Fig. 13 handles 8000 cfm. Determine the duct sizes in Section A, B, C, D, E, F, and G, maintaining an operating pressure of 0.12 in. water in the duct behind each outlet. Find the total pressure loss of the system.

Solution: The following nine steps indicate the solution:

1. Assume the velocity in Section A to be 1500 fpm. This results in an initial duct size of 48 × 16 in. The 16-in. duct depth will be maintained throughout the system.

2. The circular equivalent of a 48 × 16-in. duct is 29.2 in. (from Table 2), and with 8000 cfm flowing in this duct, the friction loss from Fig. 3 is 0.13 in. per 100 ft.

3. To size the branch ducts, determine the shortest equivalent length of duct up to the first outlet. Section B = 25 ft; Section F = 10 + equivalent length of elbow. Assume the width of the duct in Section F to be 16 in.; therefore $H/W = 15/16 = 0.94$. Also the radius of the elbow should not be less than the width ($R/W = 1.00$), resulting in an equivalent length L , from Fig. 8, of $10 \times (15/12) = 12.5$ ft. Equivalent length of Section F = $10 + 12.5 = 22.5$ ft. Therefore, Section F being shorter than Section B, size Section F first, using the same friction rate as in Section A, 0.13 in. per 100 ft. In Fig. 2, with 2000 cfm and a friction rate of 0.13 in., find an equivalent diameter of 17 in. for Section F. The rectangular equivalent of a 17-in. duct is 15 × 16 in. (from Table 2), resulting in a velocity of 1200 fpm.

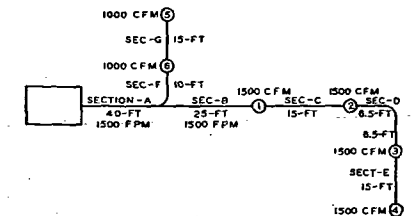


Fig. 13. Duct Layout for Example 7

Table 8. Tabulation of Results (Example 7)

Section	Air Volume Cfm	Equivalent Length ft	Velocity Fpm	Rectangular Duct in.	Dia. in.	Friction Per 100 ft in. H ₂ O	Net Pressure Loss in. H ₂ O
A	8000	40	1500	48 × 16	29.2	0.13	0.05
B	6000	25	1500	36 × 16	—	—	0.03
C	4500	15	1300	31 × 16	—	—	0
D	3000	26*	1040	26 × 16	—	—	0
E	1500	15	860	16 × 16	—	—	0
F	2000	22.5	1200	15 × 16	17	0.13	.03
G	1000	15	900	10 × 16	—	—	0

* Includes 13 equivalent ft for elbow (see Step 7 in Example 7).

Note: The shorter equivalent length line is sized first to prevent velocities in other branches from exceeding recommended values.

4. Loss in Section F = $0.13 \times 0.225 = 0.03$ in. water.
5. Since the operating pressure of Outlet 1 is to be the same as that of Outlet 2, the loss in Section B must equal the loss in Section F. Using the Static-Regain Chart, Fig. 15, size Section B for a net loss of 0.03 in. water.

The procedure for using these charts is indicated by arrow-heads and dashed lines on Fig. 15. Proceed as follows:

a. Locate the velocity of the preceding (upstream) duct section along the velocity scale on the left margin (1500 fpm).
b. Proceed horizontally to the air volume abscissa (6000 cfm).

c. Proceed parallel to the curved lines to intersect the diagonal base line.
d. Go vertically to the net loss desired (0.03 net loss).

e. Go horizontally to the air velocity base line.

f. Proceed parallel to the curved lines to intersect ordinate for 26-ft duct length.
g. Move horizontally to the air velocity scale and read the velocity in the downstream section of duct (1500 fpm).

6. This procedure is repeated for Section C except that the no gain or loss line is used in previous Step 5 d. Outlet No. 2 will then have the same static pressure behind it as Outlet No. 1, thus fulfilling the problem condition of the same operating pressure for all outlets.

7. At this point, the equivalent length of the elbow in Section D should be estimated. Its width will be somewhat less than that of Section C. Assume $W = 26$ in.; hence $H/W = 16/26 = 0.6$. With a typical radius ratio of 1.25, from Fig. 8, $L/W = 6$; therefore $L = 6 \times (26/12) = 13$ ft. Adding this to the actual length of Section D results in a total equivalent length of 26 ft.

8. Using the no gain or loss line in Fig. 14, size Sections D, E, and G. Results are listed in Table 8.

9. The total pressure loss of the system is the sum of the losses in Sections A and B (or F) and the outlet operating pressure:

Loss in Section A = $0.13 \times 0.40 = 0.05$

Loss in Section B (or F) = 0.03

Outlet Operating Pressure = 0.12

Total Pressure Loss = 0.20 in.

DESIGN OF HIGH-VELOCITY DUCTS

The transmission of air at high velocities, though common in industrial exhaust systems for many years, has gained wide acceptance in comfort air-conditioning and ventilation systems in the past few years only. This acceptance is due partly to the use of improved fans and of special sound-attenuation and control equipment, and partly to improved design and installation methods based upon a better understanding of the general problems connected with the design