ENGINEERING LETTER | 1

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SYSTEM CALCULATION

INTRODUCTION

A fan system is any combination of ductwork, hoods, filters, louvers, collectors, etc., that relies upon a fan to produce airflow. When the air moves past each of these components, resistance is created which must be considered in system calculations. It is also important to remember that fans are rated independently of a system and that fan performance will vary depending upon the accuracy of the system calculations. This Engineering Letter will explain some of the basic fundamentals of system design and calculation.

SYSTEM DESIGN

The purpose of the system will dictate the design criteria to be used. Generally they will fall into one of the following two categories:

Velocity is typically the primary consideration in dust collection, dilute pneumatic conveying, fume removal, and contaminant applications. In these applications, a capture velocity is required to redirect the flow of airborne materials into the duct system. In addition, a minimum conveying velocity is necessary to maintain the flow of the materials within the system.

Given these velocity requirements, system components can be selected to maintain the appropriate air volume and required velocity through the system.

Air Mass is the primary consideration in many drying, combustion process, and ventilating applications. These applications generally require a certain amount of air mass, usually measured in pounds of air, to support the application. Because fan manufacturers publish fan capacities in actual cubic feet per minute (ACFM), the mass of air required must be converted from standard cubic feet per minute (SCFM) to ACFM.

The velocity through a system can be determined once the ACFM is known. The relationship between velocity and airflow is defined by the equation:

$$Q = VA$$

where: Q = ACFM
V = velocity in lineal feet per minute
A = cross-sectional area in square feet

To determine the airflow requirement, the cross-sectional area is multiplied by the required velocity.

System design is really a matter of defining the required work in terms of volume or velocity and then sizing and selecting the necessary system components to accomplish that work. Of course, this must be done within the economic and space constraints of the installation.

DETERMINING SYSTEM RESISTANCE

System resistance is the sum of the resistance through each component within the system. The system depicted in Figure 1 may appear complex, but dealing with each component separately provides an orderly process for determining the overall resistance.

HOOD LOSS

To determine hood or entrance losses, resistance calculations must be made for both the acceleration loss and the entry loss. Since the air or atmosphere surrounding the hood must be accelerated from a state of rest, energy will be required to set the air in motion. This energy is equal to the velocity pressure at the entrance to the duct. Assuming the hood in this example empties into a 7" diameter duct, the required 1165 ACFM results in a velocity of 4363 FPM:

$V = Q \div A$ where: Q = 1165 CFM

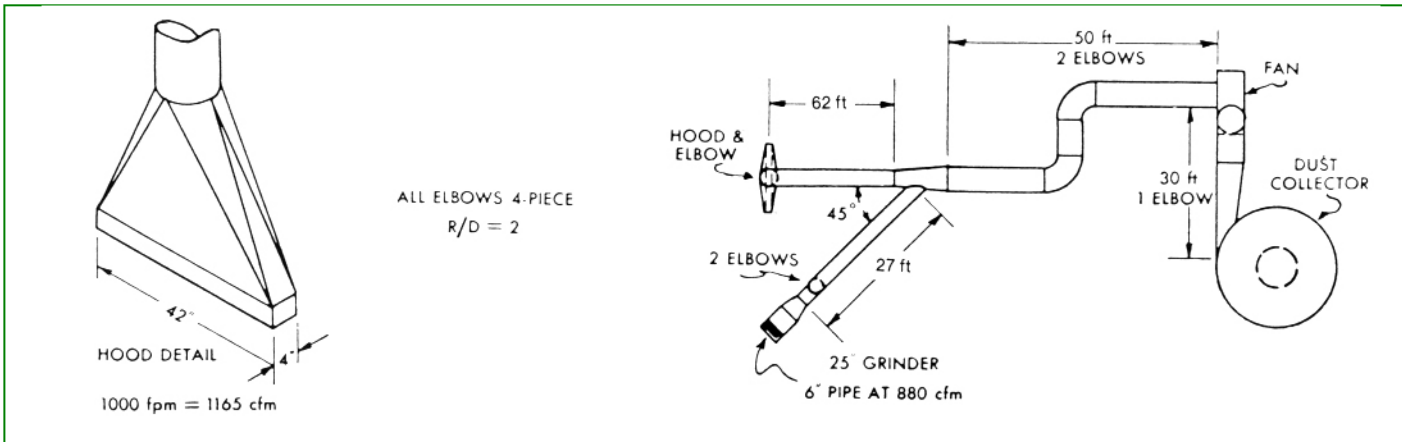


Figure 1 – Typical System

$$A = \frac{(3.5 \text{ in. radius})^2 \times 3.1416}{144 \text{ in.}^2/\text{ft.}^2} = .267 \text{ ft.}^2$$

therefore: V = 1165 CFM ÷ .267 ft.^{2} = 4363 FPM

The velocity pressure (VP) at 4363 FPM is calculated by:

$$VP = \left(\frac{\text{Velocity}}{4005}\right)^2$$

therefore: Acceleration Loss = $\left(\frac{4363}{4005}\right)^2$ = 1.19" W.G.

The same result can be obtained by interpolating from the data in Figure 2.

The entry loss of a hood is a function of its efficiency. The efficiencies of several common entry conditions are shown in Figure 3. The relative efficiencies are expressed as losses in percentage of the duct velocity pressure. Consequently, the lowest percentage is actually the most efficient.

Outlet Velocity	Velocity Pressure	Outlet Velocity	Velocity Pressure	Outlet Velocity	Velocity Pressure
800	.040	2800	.489	4600	1.32
1000	.063	3000	.560	4800	1.44
1200	.090	3200	.638	5000	1.56
1400	.122	3400	.721	5200	1.69
1600	.160	3600	.808	5400	1.82
1800	.202	3800	.900	5600	1.95
2000	.250	4000	.998	5800	2.10
2200	.302	4200	1.10	6000	2.24
2400	.360	4400	1.21	6200	2.40
2600	.422				

Figure 2 – Acceleration Loss

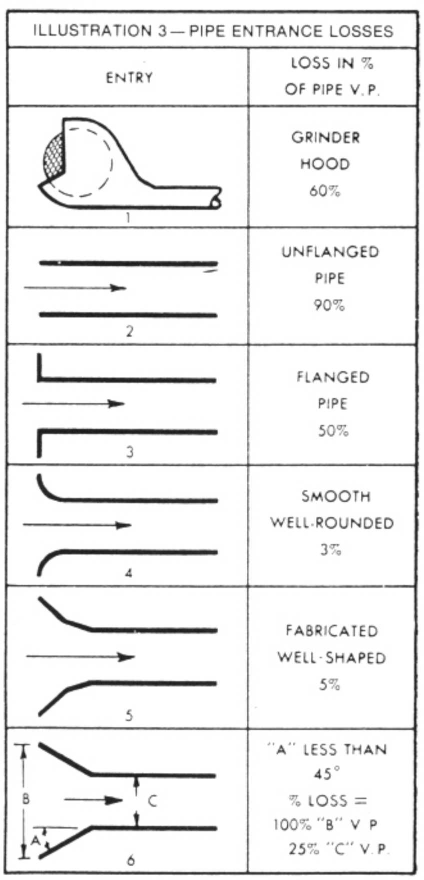


Figure 3 - Entrance Loss Percentage

The hood in this example is most similar to item 2 in Figure 3. Therefore, the entry loss from atmosphere into the hood is .90 times the entering air velocity pressure at 1000 feet per minute or:

Entry Loss = .90 x $\left(\frac{1000}{4005}\right)^2$ = .06" W.G.

This loss could have been reduced to .5 VP by simply adding a flange to the bottom edge of the hood as indicated by item 3 in Figure 3.

The total hood loss in the example is the acceleration loss added to the entry loss:

Hood loss = .06" + 1.19" = 1.25" W.G.

PRIMARY BRANCH

The duct loss from the hood to the branch junction can be determined by using the equivalent length method. This run of duct includes 62' of 7" diameter duct and one 4 piece 90° elbow of R/D = 2. According to Figure 4, the elbow has a loss equal to 12 diameters of 7" duct, or 7'. Thus, the total equivalent length of straight duct is 69'.

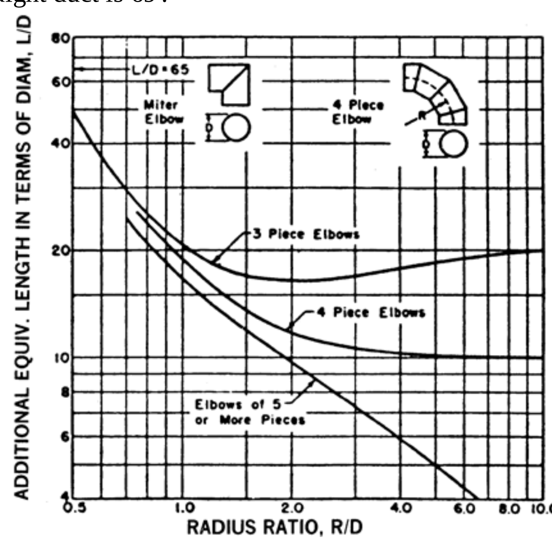


Figure 4 - Loss in 90° elbows of round cross-section

Chart I on page 4 indicates a 4.0" loss for every 100' of 7" diameter duct handling 1165 CFM. The loss for this run can be determined as:

Duct Loss = $\left(\frac{69}{100}\right)$ x 4.0 = 2.76" W.G.

Therefore, the total resistance of the hood branch to the junction is:

Branch Loss = 1.25" + 2.76" = 4.01" W.G.

SECONDARY BRANCH

A secondary branch is calculated in the same manner as the main branch. For example, a grinder hood handling 880 CFM through a 6" pipe results in a velocity of 4500 FPM, which has a 1.26" VP.

According to item 1 in Figure 3, a grinder hood has a .6 VP loss, so the total hood loss will be:

Hood Loss = 1.26" + (.60 x 1.26") = 2.02" W.G.

The duct branch from the grinder hood to the junction consists of 27' of 6" pipe and (2) 4 piece 90° elbows of R/D = 2. With an equivalent length of 39' (27' + 6' + 6') the duct loss for this run is:

Duct Loss = $\left(\frac{39}{100}\right)$ x 5.2 = 2.03" W.G.

See Chart I on page 4, which indicates a 5.2" loss for every 100' of 6" diameter duct handling 880 CFM.

The total resistance of the grinder branch to the junction is:

Branch Loss = 2.02" + 2.03" = 4.05" W.G.

Note that the resistance in both branches is nearly equal. This is because the pressures in converging branches must be equal during operation or the system will automatically equalize by adjusting the flow different than the design point. If the variation in resistance between any two converging branches exceeds 5%, further design is required to balance the loss in both branches. Where necessary, balancing can be accomplished by altering duct lengths, duct diameters, or air volumes.

MAIN DUCT

The main duct resistance calculations begin with the selection of the appropriate duct diameter. Assuming a minimum conveying velocity of 4500 FPM and an airflow requirement of 2045 ACFM (880 + 1165) in the main, a 9" diameter duct will suffice with a resulting velocity of 4630 FPM.

The junction itself represents a loss due to the mixing effect of the converging branches. The ratio of the CFM in the branch (1165 ÷ 880 = 1.3) can be used to determine the loss in percent of VP in the main. Interpolating from the data in Figure 5 results in:

Junction Loss = .19 $\left(\frac{4630}{4005}\right)^2$ = .25" W.G.

LOSS IN MAIN AT JUNCTION WITH BRANCH. (BASED ON 45° TEE & EQUAL MAIN & BRANCH VELOCITIES.)	
CFM in Upstream Main ÷ CFM in Branch	**Loss in Main in % of Main V.P.**
1	.20
2	.17
3	.15
4	.14
5	.13
6	.12
7	.11
8	.10
9	.10
10	.10
CORRECTION FACTORS FOR OTHER THAN 45° TEE.	
Tee Angle	**45° Loss X Factor**
0	0
15	0.1
30	0.5
45	1.0
60	1.7
75	2.5
90	3.4

Figure 5

Chart II on page 4 indicates a resistance of 3.3" for every 100' of 9" diameter duct handling 2045 CFM. According to Figure 4 the two elbows are equal to another 18' of duct, so the total equivalent length is 68' between the junction and the fan.

Duct Loss = $\left(\frac{39}{100}\right)$ x 3.3 = 2.24" W.G.

Note that all the losses to this point, up to the fan inlet, are expressed as negative pressure. Also that only the branch with the greatest loss is used in determining the total.

Therefore:

SP inlet = (-4.05") + (-.25") + (-2.24") = -6.54"W.G.

Assuming the same size duct from the fan to the collector, the 30' of duct and the one elbow will have a loss equivalent to the following:

Duct Loss = $\left(\frac{39}{100}\right)$ x 3.3 + 1.29" W.G.

The pressure drop across the dust collector, like coils or filters, must be obtained from the manufacturer of the device. Assuming a 2.0" loss for this example, the resistance at the fan outlet is:

SP outlet = 1.29" + 2.0" = 3.29" W.G.

FAN SELECTION

At this point enough information is known about the system to begin fan selection. Because fans are rated independent of a system, their ratings include one VP to account for acceleration. Since the system resistance calculations also consider acceleration, fan static pressure can be accurately determined as follows:

Fan SP = SP outlet **-** SP inlet **-** VP inlet

In this example with 4630 FPM at the fan inlet, and a 1.33" VP

Fan SP = 3.29" **-** (-6.54") **-** 1.33" = 8.5" W.G.

For this example, a fan should be selected for 2045 ACFM at 8.5" SP and have an outlet velocity of at least 4500 FPM to prevent material settling. This presumes a standard airstream density of .075 lbs./ft.3 If the density were other than standard, the system-resistance calculations would have been the same but the resulting fan SP would have been corrected. Refer to Engineering Letter 4 for density correction procedures.

This example also assumes that the fan inlet and outlet connections are aerodynamically designed. Fans are sensitive to abrupt changes in airflow directly adjacent to the fan inlet or outlet. The effects of abrupt changes and other “system effect” problems are discussed in Engineering Letter 5.

CONCLUSION

It is the responsibility of the system designer to ensure that there are adequate air flows and velocities in the system and that the selection of duct components and fan equipment has been optimized. While computer programs do the bulk of system calculations today, this Engineering Letter should help to provide a common set of methods and terminology to assist in that effort.

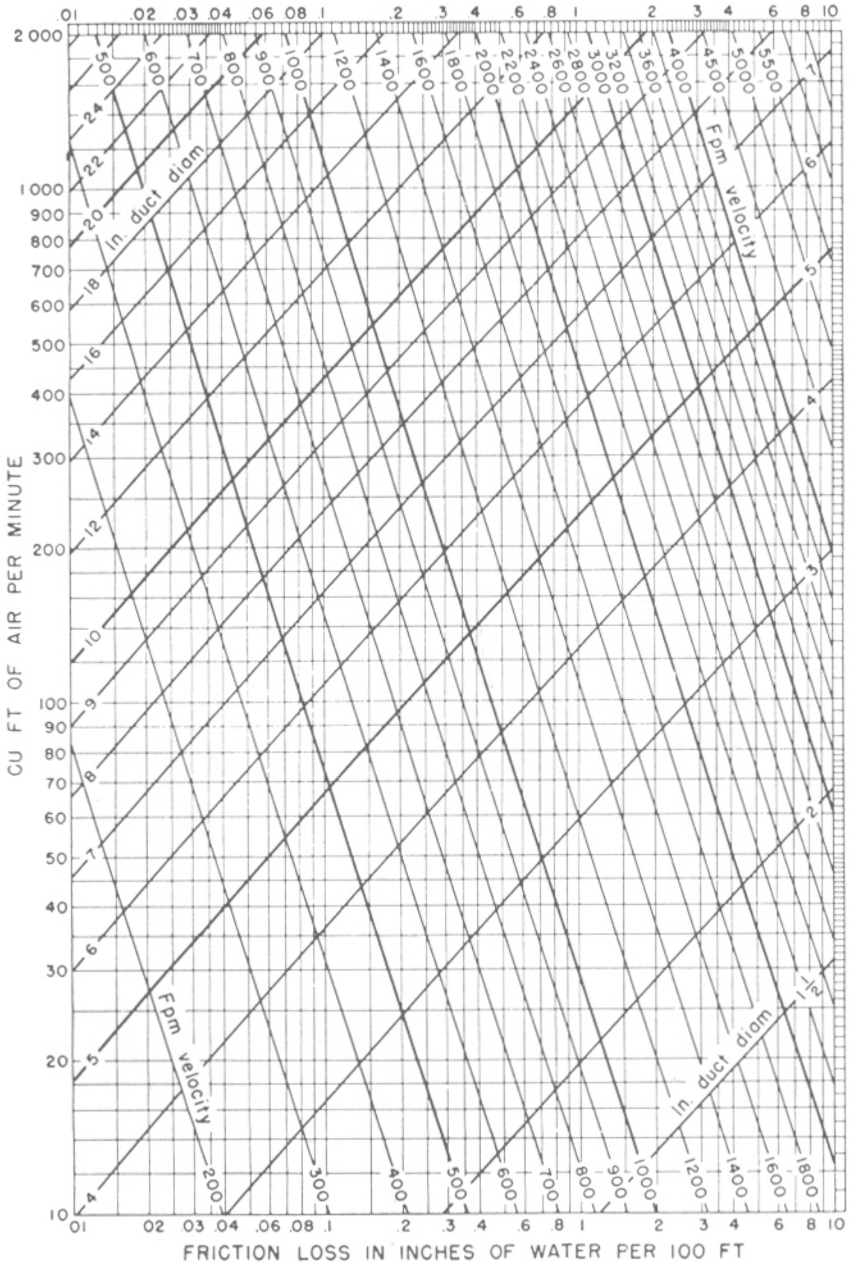
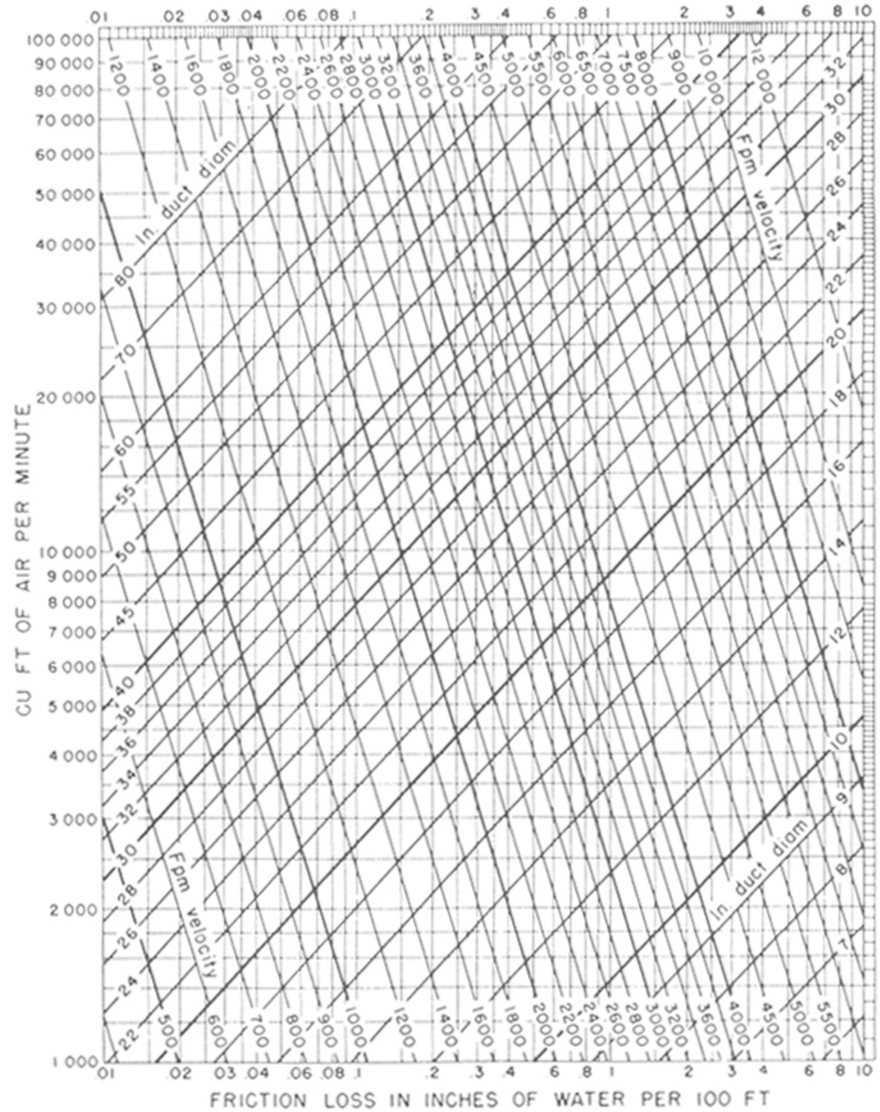


Chart I



FRICTION OF AIR IN STRAIGHT DUCTS
Heating Ventilation Air Conditioning Guide 1950
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Based on standard air of 0.075 lb. per cu. ft. density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.
No safety factor included. Caution: Do not extrapolate below chart.

Chart II