To reduce auxiliary fuel costs, it may be advantageous to segregate exhaust streams requiring combustion at their source rather than diluting them with less noxious exhaust gas. This will permit use of a smaller oxidation unit.

The major expense associated with combustion systems is the auxiliary fuel needed to heat incoming exhaust gas and assure complete combustion. Because most of the local ventilation systems covered in this book exhaust mostly room air with very low levels of contaminants, combustion is often not cost-effective. Combustion devices find more application with process vents or similar sources where the contaminant concentration is relatively high.

Summary

Air cleaner selection depends on the physical state of the contaminants, the characteristics of the contaminant and exhaust gas, and the required air cleaner efficiency. Air cleaners often provide the largest single source of resistance in the ventilation system and may also present the greatest maintenance and operating problems. Careful initial characterization of the contaminants and exhaust gas is vital to a successful system.

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Chapter 8

Ventilation System Design

This chapter describes the design procedure and contains several step-by-step design examples of varying complexity. It builds on the background information on local exhaust systems and hoods in Chapters 5 and 6, applying these principles to the design of ventilation systems. Even for readers who do not design systems, at least "walking through" the *Single Hood* example will be an excellent way to reinforce the local exhaust principles covered in Chapter 5.

There are several example calculations and four sample designs in this chapter. In order to make the calculations easier to follow, in most cases the design information from various tables and figures is shown in the examples exactly as it appears in the tables and figures. Often this results in more "significant digits" being shown than are justified by the precision of the calculations. For example, the cross-sectional area of a 5-inch diameter duct may be shown as 0.1364 ft^2 in the calculation because this is the value in the table of standard duct areas. In practice this value could be rounded to 0.14 ft^2 without any loss of precision. The complete value is used so the reader can locate the value in the table and identify where it appears in the sample calculations. The calculations themselves are rounded to three or four significant digits as appropriate, including in the intermediate steps shown in the examples. This will cause some slight differences in the answers if the reader uses a calculator for the multi-step calculations and only rounds the final answer.

Design Overview

The first step in system design is choosing the hood types and location, type of air cleaner (if any), and other parameters for the system. Table 8-1 lists the typical components of a local exhaust system with references to the chapters containing more detailed information and with comments on other relevant design considerations.

After the design parameters have been chosen, the design calculations determine the diameter for each duct segment, actual total airflow through the system, fan size (ft^3/min and Fan Static Pressure rating) and other information needed to build the system so it functions properly and is economical to install and operate. Table 8-2 shows the information needed to begin design and also the "output" of the design process.

In order to size the ducts and fan, design data (such as hood entry loss coefficients and duct friction loss factors) are needed. It is also helpful to have some

TABLE 8-1. Overview of Local Exhaust System Components

Component	See Chapter	Comments
Hoods	5, 6	It is important to select proper type of hood (enclosure, capturing hood or receiving hood) for the application, and use reliable hood design information from the *Industrial Ventilation Manual* or other source.
Ducts	5	
Air cleaner	7	Often the largest source of resistance in system. It is important to select the proper air cleaner since later upgrades after system is installed may require replacement of fan with larger unit.
Fan	5, 9	The ducts leading into and out of fan are important for proper fan operation (See Chapter 9).
Exhaust stack	5, 9	Every system needs at least a short straight stack (other other duct section) on the fan outlet to optimize fan performance.

TABLE 8-2. Design Information: Input and Output

"Input" for System Design*	"Output" of System Design
• Location of each hood, air cleaner, fan, etc., plus route of ducting. • Type, desired flowrate and hood entry loss for each hood. • Minimum duct velocity. • Duct characteristics: – Materials of construction – Friction loss – Elbow radius and construction – Branch entry angle – Specifications for duct enlargement or contractions • Type of air cleaner, and pressure loss.	• Actual airflow for each hood. • Diameter and air velocity for each duct segment. • Fan specifications: – Airflow (Q), ft^3/min. – Fan Static pressure, in. of H_2O • Hood static pressure (useful for balancing system after installation).

*Not all components and features apply to simple systems.

design aids such as a calculation worksheet and chart of the standard duct diameters that are generally available. These items are discussed in the next two sections.

Design Information

For the systems illustrated in this chapter, the design information described in this section is needed. Systems with more complex features, such as duct enlargements or contractions, are beyond the scope of this book. For design information on these features, refer to the *Industrial Ventilation Manual*.[1]

Hood Information

Wherever possible, use hood design plates from the ACGIH *Manual* or a similar source. The *Manual* contains over 140 separate drawings showing different hood designs. The drawings recommend key design parameters for each hood such as airflow, hood entry loss coefficient, and minimum duct and slot velocities. They may also contain other important design or operating recommendations.

Table 8-3 lists figures in other chapters that illustrate hood design data and also describes hood design features that influence hood entry loss.

Duct Design Data

Proper duct design is important for good system performance. The magnitude of the pressure losses in the system depends on the square of air velocity through the ducts and other components. Duct velocity, in turn, depends on the diameter of the ducts. Smaller diameter ducts are less expensive to fabricate and install than larger diameter ducts; however, the resulting higher duct velocities in smaller diameter ducts increase pressure losses, thus requiring a larger fan with higher power consumption.

In addition, one method of designing multiple-hood systems relies on choosing duct diameters that cause the correct amount of pressure loss to distribute airflow through each hood. This is called a *balanced* design. Another design method, called *blast gate* design, uses adjustable blast gates (i.e., sliding dampers) in the ducts to generate the additional resistance or pressure loss

TABLE 8-3. Hood Entry Loss for Typical Hood Types

Hood Type	Hood Entry Loss (h_e)	Typical Examples
Low Entry Loss Hood • Open area of hood is large enough to allow smooth airflow pattern into the hood. • Transition from the hood into the duct is tapered to minimize turbulence.	0.25 VP_{duct}	Figures 1-3, 5-1, 6-2 and 6-6 (right)
Moderate Entry Loss Hood • Hood is similar to "Low Entry Loss Hood" above except that transition into the duct is abrupt (not tapered). • Hood with baffles plates to distribute airflow or other features that interfere with smooth airflow patterns through the hood. • Hood with restricted open area that results in higher velocity and turbulence in hood.	0.50 VP_{duct}	Figures 3-5, 5-2, 6-3, 10-5 and 10-6
High Entry Loss Hood • Slot hood where air distribution across the face of the hood is achieved by use of a slot under suction. After entering the slot, the air passes through a plenum chamber into the duct, causing additional entry loss as air enters the duct.	1.78 VP_{slot} + 0.25 VP_{duct}	Figures 6-1, 6-4, 6-6 (left) and 6-34

needed to achieve proper balance. These two design methods are described later in this chapter.

Because velocity distribution is more uniform in round ducts than in rectangular ducts, round ducts are used when possible, especially in dust conveying systems.

Round ducts are also stronger under suction than rectangular ducts. The ACGIH *Industrial Ventilation Manual* contains techniques for calculating equivalent diameters of non-round ducts so that design tables based on round ducts can be used to perform design calculations for the system.

Minimum Duct Velocity

Systems carrying particulates generally need to maintain a certain minimum transport velocity to avoid material settling in the ducts. For common dusts, this velocity is often 3000–4000 ft/min. For more dense materials, larger particles or sticky materials, the value is higher. Table 8-4 lists typical minimum transport velocities for materials; this information can be used to supplement the general recommendations in the hood design figures in the ACGIH *Industrial Ventilation Manual*.

Although systems handling vapors and gases have no minimum duct velocity criteria, as a rule-of-thumb duct velocities of 2000–3000 ft/min usually result in a good balance between initial duct construction cost and fan operating cost.

TABLE 8-4. Typical Minimum Duct Transport Velocities

Operation	Typical Duct Transport Velocity, ft/min
Barrel filling or dumping	3500–4000
Belt conveyors	3500
Bins and hoppers	3500
Metallizing booth	3500
Melting pot and furnace	2000
Oven hood	2000
Buffing and polishing	
Dry dust	3000–3500
Sticky dust	3500–4000
Grinding dust	5000
Sandblast dust	4000
Sawdust	
Dry	3000
Wet	4000
Shavings	
Dry	3000
Wet	4000
Metal turnings	5000
Lead dust	5000
Welding dust	1000–3000
Soldering fumes	2000
Paint spray	2000
Grain dust	3000
Cotton dust	3000
Cotton lint	2000

Friction Loss

To calculate the friction loss in the ducts in a system, it is necessary to know the friction loss per foot of duct (expressed as "VP_d lost per foot"). This loss is a function of the material of construction (interior roughness), how the ducts are constructed (e.g., whether formed from sheet metal with a longitudinal seam vs. extruded with no seam), and duct diameter. The duct diameter is a factor because for narrow ducts the ratio of the inside perimeter to duct area is greater than for larger diameter ducts, which results in proportionately more friction loss as air "rubs" against the duct walls.

Friction loss is calculated from Equation 5.10; experimental constants for this equation are given in Chapter 5 for these common duct materials: galvanized sheet metal, aluminum, stainless steel, polyvinyl chloride (PVC), and wire-wrapped fabric flexible duct. Stainless steel is used where protection against corrosion is needed. Wire-wrapped flexible fabric ducts are often not recommended for local exhaust systems carrying particulates because of their tendency to accumulate settled material and sag unless rigidly supported and because of the difficulty in cleaning out settled material.

For use in the design examples, Table 8-5 lists the friction loss factors for some common sizes of galvanized sheet metal duct as a function of duct velocity. The ACGIH *Industrial Ventilation Manual* has more extensive data on sheet metal ducts, and similar information for other duct materials. When using Table 8-5, the friction loss for velocities that fall between the given values can be determined by interpolation or an estimate.

Example: What is the friction loss in an 8-in. duct with 3500 ft/min duct velocity that is 35 ft long?

Answer: From Table 8-5, the friction loss factor must be estimated from the given information:

Dia., in.	3000 ft/min vel.	4000 ft/min vel.
8.0	0.0311 VP_d/ft	0.0304 VP_d/ft

The loss factor for 3500 ft/min is halfway between the two given values, or 0.0308 VP_d/ft.

The friction loss for a duct 35 ft long is:

$$35\text{ ft} \times \frac{0.0308\ VP_d}{\text{ft}} = 1.08\ VP_d$$

To convert this to a loss expressed in "inches of water," use equation 5.3:

TABLE 8-5. Friction Loss Factors for Galvanized Sheet Metal Ducts*

	Friction Loss, Number of VP_d per foot				
Diameter, inches	**2000 ft/min**	**3000 ft/min**	**4000 ft/min**	**5000 ft/min**	**6000 ft/min**
1	0.4088	0.3959	0.3870	0.3802	0.3748
1.5	0.2489	0.2410	0.2356	0.2315	0.2282
2	0.1750	0 1695	0.1657	0.1628	0.1605
2.5	0.1332	0.1290	0.1261	0.1239	0.1221
3	0.1065	0.1032	0.1009	0.0991	0.0977
3.5	0.0882	0.0854	0.0835	0.0821	0.0809
4	0.0749	0.0726	0.0709	0.0697	0.0687
4.5	0.0649	0.0628	0.0614	0.0603	0.0595
5	0.0570	0.0552	0.0540	0.0530	0.0523
5.5	0.0507	0.0491	0.0480	0.0472	0.0465
6	0.0456	0.0442	0.0432	0.0424	0.0418
6.5	0.0417	0.0404	0.0395	0.0388	0.0382
7	0.0378	0.0366	0.0358	0.0351	0.0346
7.5	0.0350	0.0339	0.0331	0.0324	0.0320
8	0.0321	0.0311	0.0304	0.0298	0.0294
8.5	0.0300	0.0290	0.0284	0.0278	0.0274
9	0.0278	0.0269	0.0263	0.0258	0.0255
9.5	0.0261	0.0253	0.0247	0.0243	0.0240
10	0.0244	0.0236	0.0231	0.0227	0.0224
11	0.0217	0.0210	0.0206	0.0202	0.0199
12	0.0195	0.0189	0.0185	0.0182	0.0179
13	0.0177	0.0171	0.0168	0.0165	0.0162
14	0.0162	0.0157	0.0153	0.0150	0.0148
15	0.0149	0.0144	0.0141	0.0138	0.0136
16	0.0137	0.0133	0.0130	0.0128	0.0126
17	0.0127	0.0123	0.0121	0.0119	0.0117
18	0.0119	0.0115	0.0113	0.0111	0.0109
19	0.0111	0.0108	0.0105	0.0103	0.0102
20	0.0104	0.0101	0.0099	0.0097	0.0096
22	0.0093	0.0090	0.0088	0.0086	0.0085
24	0.0084	0.0081	0.0079	0.0078	0.0077
26	0.0076	0.0073	0.0072	0.0070	0.0069
28	0.0069	0.0067	0.0066	0.0064	0.0063
30	0.0064	0.0062	0.0060	0.0059	0.0058
35	0.0053	0.0051	0.0050	0.0049	0.0048
40	0.0045	0.0043	0.0042	0.0042	0.0041
45	0.0039	0.0038	0.0037	0.0036	0.0036
50	0.0034	0.0033	0.0032	0.0032	0.0031
55	0.0031	0.0030	0.0029	0.0029	0.0028
60	0.0027	0.0026	0.0026	0.0025	0.0025

**Source:* ACGIH *Industrial Ventilation Manual.*[1]

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

where: VP = velocity pressure, inches of H_2O

$$VP = \left(\frac{3500}{4005}\right)^2 = (0.87)^2 = 0.76 \text{ in. } H_2O$$

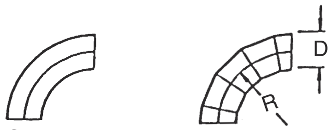
$$\text{Pressure loss } = 1.08\ VP_d \times \frac{0.76 \text{ in. } H_2O}{VP_d}$$

$$= 0.82 \text{ in. } H_2O$$

Elbow Losses

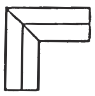
Each elbow causes pressure loss due to turbulence and friction. The magnitude of the *turbulence* loss depends on the construction of the elbow (how many joints) and the severity of the elbow radius of curvature. The elbow radius of curvature is expressed as the ratio of the elbow curvature to the duct diameter. A more gradual bend (larger radius:diameter (R/D) ratio) causes less pressure loss than does a tight bend. For design calculations, a full 90° elbow is considered "1.0 elbow," a 45° bend is considered as 0.5 elbow; a 60° bend is considered

Elbow Loss, VP_d Loss per Elbow



Elbow Radius to Duct Diameter (R/D)	Smooth (Stamped)	5-Piece Construction
0.75	0.33	0.46
1.00	0.22	0.33
1.50	0.15	0.24
2.00	0.13	0.19
2.50	0.12	0.17

Other Elbow Configurations:



- 90° (mitered) – no turning vanes: 1.2 VP_d per elbow
- 90° (mitered) – with turning vanes: 0.6 VP_d per elbow

Note: These loss factors represent only turbulent loss. To account for friction loss, determine "duct length" from intersection of duct centerlines at elbow.

FIGURE 8-1. Elbow pressure loss for round duct elbows. *(Source:* ACGIH *Industrial Ventilation Manual*[1])

as 0.67 elbow, etc. Figure 8-1 shows the elbow turbulence loss for common sheet metal elbow construction. The elbow R/D ratio for the system is chosen before design calculations begin; a ratio of 2.0 is typical for industrial exhaust systems.

The elbow *friction* loss is included in the friction loss calculation for the ducts. The duct length is measured from the intersection of ducts meeting at an elbow.

Example: The 8-in. duct described above has two 90° elbows and a 45° bend. If the elbows are smooth (stamped) and have a radius that is twice the duct diameter, what is the elbow turbulence loss?

Answer*:* From Figure 8-1, the loss per elbow is 0.13 duct velocity pressure for a smooth elbow with R/D = 2.0. Counting the 45° bend as 0.5 elbow:

$$2.5 \text{ elbows} \times 0.13 \frac{VP_d}{\text{elbow}} = 0.33\ VP_d$$

From the earlier calculation, the velocity pressure in the duct is 0.76 in. of water, so the actual pressure loss is:

$$0.33\ VP_d \times \frac{0.76 \text{ in. } H_2O}{VP_d} = 0.25 \text{ in. } H_2O$$

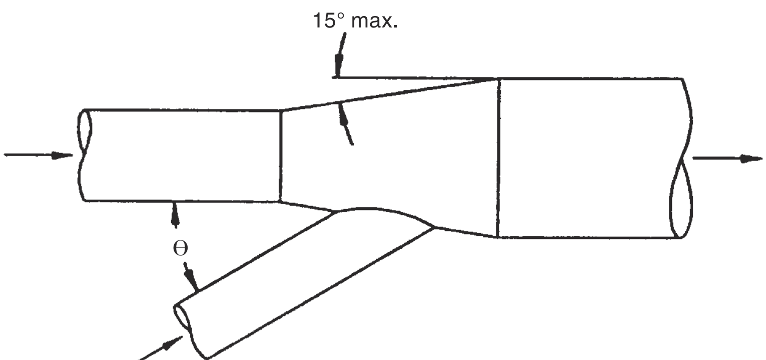
Branch Entry Loss

In a multi-hood system, a pressure loss occurs when a branch duct enters the main duct. The loss is due to turbulent flow where the two airstreams merge. The entry loss is assumed to occur in the branch duct rather than the main duct. As shown in Figure 8-2, the entry should occur at a duct enlargement section so the duct velocity before and after the junction is approximately equal. The magnitude of the loss depends on the angle between the main and branch ducts; an entry angle of 30° or 45° is common. Figure 8-2 also shows the pressure loss factors for branch duct entries.

Example: The 8-in. duct with VP_d of 0.76 in. H_2O in the previous examples is a branch duct that enters a main duct at a 30° angle. What is the branch entry loss?

Answer: From Figure 8-2, a pressure loss of 0.18 velocity pressure occurs in the 8-in. branch duct. The pressure loss is:

$$0.18\ VP_d \times \frac{0.76 \text{ in. } H_2O}{VP_d} = 0.14 \text{ in. } H_2O$$



Angle Θ, Degrees	Loss, Fraction of VP in Branch
10	0.06
15	0.09
20	0.12
25	0.15
30	0.18
35	0.21
40	0.25
45	0.28
50	0.32
60	0.44
90	1.00

FIGURE 8-2. Branch duct entry pressure loss. This loss occurs in the branch duct that is entering a main duct. *(Source:* ACGIH *Industrial Ventilation Manual*[1])

Sometimes the duct velocity after a junction is significantly higher than the velocity in the ducts leading to the junction. This can occur because the standard duct sizes that are available do not permit selection of a diameter that will keep the velocity approximately equal before and after the junction. When this occurs, additional energy will have to be added in the form of suction (i.e., static pressure) to accelerate the air to the higher velocity. The air coming together at the junction has a certain value of velocity pressure, which is the "weighted" average of the velocity pressures in the two ducts coming into the junction. This is called the *resultant velocity pressure* (VP_r) and is calculated from Equation 5.11:

$$VP_r = \left(\frac{Q_1}{Q_3}\right)VP_1 + \left(\frac{Q_2}{Q_3}\right)VP_2$$

where: VP_r = resultant velocity pressure of the combined airflow after the junction, in. H_2O
Q_3 = flowrate in duct after junction, ft³/min
$_{1,2}$ = ducts coming into junction

If the actual velocity pressure (based on actual duct velocity) in the duct after the junction is greater than VP_r, the additional acceleration loss must be calculated for this duct segment. For practical purposes, differences between VP_r and VP_{actual} of 0.1 in. H_2O or less can be ignored. If the VP_{actual} of the combined airflow after the junction is lower than VP_r, static pressure regain occurs in the duct after the junction. This means that some of the acceleration loss originally expended at the hood is "given back" as the air slows at the junction. Theoretically this allows the designer to reduce the amount of suction required at the junction, but in practice this regain is usually ignored if it is a small value. Larger values (greater than about 0.3 in. H_2O) are included in the duct calculations and serve to reduce the suction requirement in the duct segment where the regain occurs. In most systems, the preferred approach is to maintain approximately the same velocity in the main duct before and after the junction.

Exhaust Stacks

An exhaust stack on a ventilation system serves two purposes: it helps disperse contaminants in the exhaust stream by discharging the exhausted air above roof level; and it improves fan performance because the stack allows the uneven velocity distribution that exists at the fan outlet to stabilize into a more uniform velocity profile. This helps improve fan performance since the uneven velocity profile causes a high velocity pressure at the outlet, which causes high pressure losses as described in Chapter 9.

All systems should have at least a short straight stack on the fan. A high stack discharge velocity (3000 ft/min or higher) helps to disperse contaminants because the air jet action can increase the effective stack height except under severe wind conditions.

If rain entering the stack is a problem, a vertical discharge cap that does not block the stack opening or elbows to offset the stack (Figure 8-3) are effective in keeping rain out of the system. In some cases downward and horizontal fan discharges have been used to prevent rain from entering the stack, but such configurations can impede contaminant dispersion and should be avoided where this is an issue. While the fan is operating, rain cannot enter a vertical stack with stack velocity of about 2600 ft/min or greater.

As discussed later in this chapter, exhaust discharges should be located so that the discharged air will not be drawn back into the building through air inlets for general ventilation or other openings.

Air Density Corrections

The duct pressure loss data are based on "standard air" with a density of 0.075 lb/ft³, which is the density of air at 70°F, 29.92 in. of mercury barometric pressure, and 50% relative humidity. If operating conditions cause a significant deviation in air density from this value, the pressure loss through the system will be different than calculated with these design figures. Usually the causes of nonstandard density are temperature, elevation, and/or elevated moisture content. It is usually easier to calculate the pressure losses through the system assuming standard air and then adjust the fan static pressure value to reflect actual operating conditions as described in Chapter 9. Corrections for density are usually not needed for temperatures between 40° and 100°F and/or elevations of ± 1000 ft relative to sea level. For other density corrections, such as high humidity exhaust streams, see the ACGIH *Industrial Ventilation Manual*.

Fan Size

The fan size in a ventilation system must be specified by both volumetric airflow (ft³/min) and fan static pressure (in. H_2O). Together these two parameters specify how much air the fan is to move against the system static pressure, or resistance. *Fan static pressure* is a term used by fan manufacturers in their catalogs, and it is defined according to the standardized fan test procedure that is used by all manufacturers for consistent fan selection data. It represents the amount of static pressure the fan must develop to

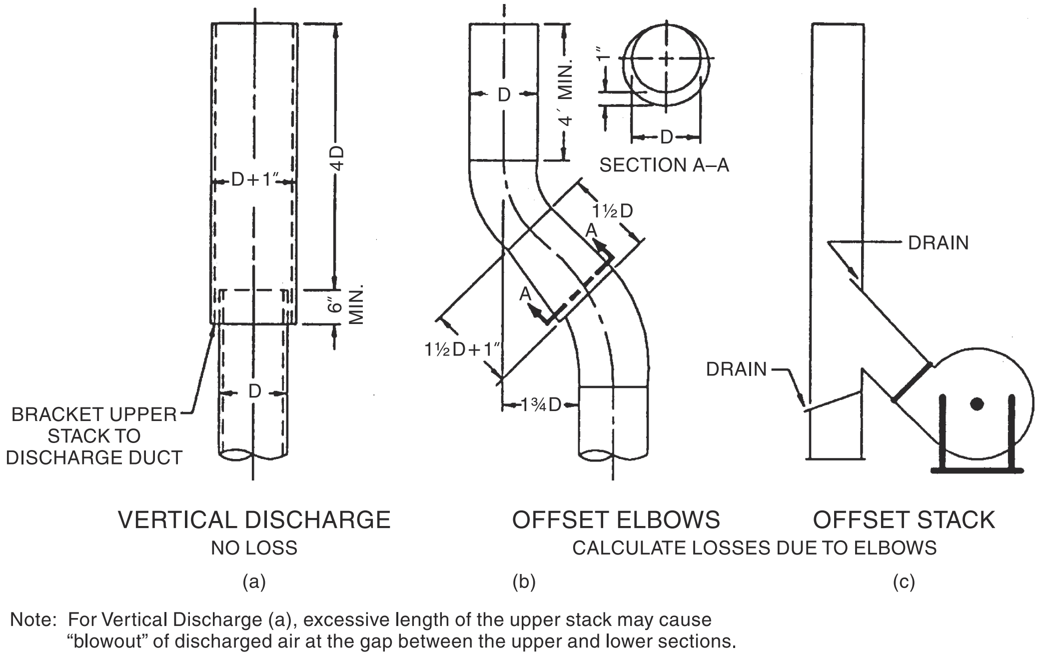


FIGURE 8-3. Discharge stack designs that prevent rain from entering the ventilation system. *(Source:* ACGIH *Industrial Ventilation Manual*[1]*)*

move the required amount of air through the system, i.e., to accelerate room air, overcome hood entry and duct losses, and the losses in the discharge stack after the fan. The fan static pressure can be calculated as follows:

$$FSP = |SP_{inlet}| + |SP_{outlet}| - VP_{inlet} \quad \textbf{(8.1)}$$

where: FSP = fan static pressure, in. H_2O

$|SP_{inlet}|$ = absolute value (disregarding the sign) of static pressure at the fan inlet (representing all losses in the system up to the fan), in. H_2O

$|SP_{outlet}|$ = absolute value of static pressure losses at the fan outlet (representing all losses in the system after the fan), in. H_2O

VP_{inlet} = velocity pressure at fan inlet, in. H_2O

The absolute values of static pressure are used in the calculation because the static pressure on the inlet side of the fan is negative while it is positive on the discharge side of the fan.

Once the fan volume and fan static pressure are known, the type of fan best suited for the system can be selected according to information in Chapter 9.

Design Aids

In addition to the design data described above, it is helpful to have a calculation worksheet and other design aids.

Calculation Worksheet

The Calculation Worksheet shows how to “account” for each loss in a system. It expresses the loss in each hood, elbow, and most other system components as the number of “VP_d equivalents” lost. After these individual losses are summed, the resulting pressure loss in units of “inches of water” can be calculated by multiplying the number of VP equivalents lost times the value of VP_d. The value of VP_d, of course, is related to the duct velocity by Equation 5.3:

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

where: VP = velocity pressure, in. H_2O

V = velocity, ft/min

A blank Calculation Worksheet is printed at the back of this book to be photocopied for use in following the illustrations or designing other systems; Figure 8-4 shows an explanation for each element on the worksheet. Each vertical column represents one branch duct (from hood to the junction with another duct), a section of main duct, or the exhaust stack. Where possible, all pressure losses are expressed in terms of duct velocity pressure. However, not all losses can be expressed in terms of VP_d: if a duct has a slotted hood (a narrow slot with high air velocity is used to distribute airflow across the hood opening), the slot entry loss is proportional to the *slot* velocity pressure. Some losses (e.g., air cleaners) are most easily expressed

Line	Section	Item	Units	Explanation	Note
1.		Duct Segment Identification		From system layout drawing.	
2.		Target Volumetric Flowrate (Q)	ft^3/min	From hood design data or other source.	
3.		Minimum Transport Velocity	ft/min	For dust systems, to avoid material settling in ducts	
4.		Maximum Duct Area	ft^2	To give minimum velocity with target flowrate.	
5.		Selected Duct Diameter (Table 8-6)	inches	A standard duct diameter with Area equal to or smaller than Line 4.	
6.		Duct Area (Table 8-6)	ft^2	Of selected duct diameter.	
7.		Actual Duct Velocity	ft/min	Line 2 ÷ Line 6	
8.		Duct Velocity Pressure (Eq. 5.3)	in. H_2O	Corresponding to velocity on Line 7.	
9.	HOOD SUCTION – SLOTS	Maximum Slot Area	ft^2	To generate required slot velocity with flowrate on Line 2.	Note: Lines 9 through 16 are completed only for slot hood.
10.	HOOD SUCTION – SLOTS	Selected Slot Area	ft^2	Selected Area equal to or smaller than Line 9.	
11.	HOOD SUCTION – SLOTS	Actual Slot Velocity	ft/min	Line 2 ÷ Line 10	
12.	HOOD SUCTION – SLOTS	Slot Velocity Press. (Eq. 5.3)	in H_2O	Corresponding to velocity on Line 11.	
13.	HOOD SUCTION – SLOTS	Slot Loss Coefficient	VP_s	From hood design plate or other source.	
14.	HOOD SUCTION – SLOTS	Acceleration Factor (0 or 1)	VP_s	Equals 1.0 if $V_{slot} > V_{duct}$, otherwise 0.	
15.	HOOD SUCTION – SLOTS	Slot Loss (13 + 14)	VP_s	Total loss through slot in units of VP_s.	
16.	HOOD SUCTION – SLOTS	Slot Static Press. (12 x 15)	in. H_2O	Total loss through slot in units of in. H_2O.	
17.	HOOD SUCTION	Entry Loss Coeff. (F_d)	VP_d	From hood design plate or other source.	
18.	HOOD SUCTION	Acceleration Factor (0 or 1)	VP_d	Equals 1.0 if $V_{duct} > V_{slot}$ *or* there is no slot, otherwise 0.	
19.	HOOD SUCTION	Duct Entry Loss (17 + 18)	VP_d	Total loss as air enters duct at hood in units of VP_d.	
20.	HOOD SUCTION	Duct Entry Loss (8 x 19)	in. H_2O	Total loss as air enters duct at hood in units of in. H_2O.	
21.	HOOD SUCTION	Other Hood Losses	in. H_2O	Generally zero unless other hood losses are identified.	
22.	HOOD SUCTION	Hood Static Pressure (16+20+21)	in. H_2O	Represents suction needed in duct at hood for proper hood performance.	
23.		Straight Duct Length	ft.	From sketch of system.	
24.		Friction Loss Factor (H_f) (Table 8-5)	VP_d/ft.	An estimate based on values in Table 8-5 is usually sufficiently accurate.	
25.		Duct Friction Loss (23 x 24)	VP_d	(Self-explanatory)	
26.		Number of 90° Elbows (including partial)		A 90° elbow = 1; 60° = 0.67; 45° = 0.5, etc.	
27.		Elbow Loss Coefficient (Fig. 8-1)	VP_d/elbow	Depends on elbow construction and R/D value selected for system.	
28.		Elbow Loss (26 x 27)	VP_d	(Self-explanatory)	
29.		Duct has a Branch Entry? (Yes = 1, No = 0)		Loss at branch entry is assumed to occur in branch, not main duct.	
30.		Branch Entry Loss Coeff. (Fig. 8-2)	VP_d	Depends on angle of entry.	
31.		Branch Entry Loss (29 x 30)	VP_d	(Self-explanatory)	
32.		Special Fitting Loss Coefficients	VP_d	Feature such as a duct enlargement or contraction, etc.	
33.		Duct Loss (25 + 28 + 31 + 32)	VP_d	Total duct loss in units of VP_d.	
34.		Duct Loss (33 x 8)	in. H_2O	Total duct loss in units of in. H_2O.	
35.		Other Losses (Air Cleaner, ΔVP, etc.)	in. H_2O	Other losses expressed as "inches H_2O," including any added Accel. Loss.	
36.		Hood/Duct Segment Loss (22+34+35)	in. H_2O	Sum of Duct, Hood and Other Losses (if any) for this segment of system.	
37.		Cumulative Static Press. at Segment	in. H_2O	Total losses in the system up to the end of this duct segment.	
38.		Governing Static Press. at Junction	in. H_2O	For ducts meeting at a junction, the highest SP value for any of the ducts.	
39.		Corrected Volumetric Flowrate	ft^3/min	New Q for the duct segment when airflow is increased to balance junction.	
40.		Corrected Velocity	ft/min	New duct velocity based on corrected Q on Line 39.	
41.		Corrected Velocity Pressure	in. H_2O	New VP_d based on corrected velocity on Line 40.	
42.		Resultant VP (VP_r) (Eq. 5.11)	in. H_2O	VP resulting from the combined airflow from two ducts entering a junction.	

FIGURE 8-4. Calculation Worksheet for system design with an explanation of the entries. A blank sheet is printed inside the back cover.

directly in "inches of water." The calculation sheet has lines for these different pressure losses.

The purpose of the Worksheet is to assure that all losses are identified and included in the system design calculations. It may take some practice to use it comfortably; because of limited space, the labels and explanations are cryptic, and many of the entries do not apply to all ventilation systems. The sheet used in this chapter is a simplified version of the worksheet appearing in the ACGIH *Industrial Ventilation Manual*.

Most of the entries on the Worksheet are straightforward and are easy to follow from the background information in Chapters 5 and 6. However, there is one concept that is less obvious and must be considered at several places in the Worksheet: *Acceleration Loss*. Recall that Acceleration Loss represents pressure that must be added to the system to accelerate air to a higher velocity. It is important to check for Acceleration Loss, and account for it if needed, wherever air velocity may increase in the system:

- For ducts with a hood, an Acceleration Loss occurs at the hood. For a hood without a slot, the loss occurs as the room air enters the duct at the hood and is accounted for on Line 18. For a slot hood, the loss can either occur at the slot *or* as the air enters the duct. If the slot velocity > duct velocity, the Acceleration Loss occurs at the slot and is accounted for on Line 14. Otherwise, it is recorded on Line 18. The Acceleration Loss at the hood *always* equals "one VP"—the velocity pressure value corresponding to the air velocity.
- At each junction, it is important to evaluate whether an acceleration has occurred as the combined airflow merges into the next main duct segment. The diameter of the next main duct is usually chosen to keep the velocities about equal before and after the junction. But with only standard size ducts available, sometimes the duct velocity increases after the junction. For each junction, calculate the Resultant VP (VP_r) of the flow entering the junction using Equation 5.11. This is a "weighted" average of the VP from each incoming duct. Record this value on Line 42. Then compare this to the VP in the next main duct segment calculated in the next column on the Worksheet and recorded on Line 8. If the VP in the next main duct segment is greater than VP_r, it is necessary to add the difference as an additional acceleration loss on Line 35. Increases of 0.1 in. H_2O or less may be ignored.
- There are other locations where the duct velocity increases, such as duct contractions or an exhaust stack with a smaller diameter than the fan inlet duct in order to achieve a high discharge velocity to disperse the exhausted air. Again account for any acceleration loss in the segment on Line 35 of the Worksheet.

Cross-sectional Area of Ducts

Ducts are generally available only in certain diameters; the diameters listed in Table 8-6 are the sizes that most sheet metal shops can fabricate using standard patterns. Select a diameter from this table rather than using a non-standard duct size in designs. It may be necessary to check with local suppliers to determine available duct diameters for the duct material you are using. Duct areas are used to calculate duct velocity once the duct diameter and airflow are selected; for consistency of units, the duct area is expressed as ft^2.

When designing a duct with low airflow, it is often difficult to select a duct diameter from available sizes that will yield an actual duct velocity close to the specified minimum velocity. This is because for small ducts there is a relatively large percentage change in area from one size to the next. In these cases, it is prudent to select the higher velocity or make changes in the hood size or design to achieve a more reasonable duct velocity, rather than selecting a larger duct that gives a velocity below the specified minimum.

Example: A duct carries 150 ft^3/min in a system that requires a minimum duct velocity of 3500 ft/min to prevent settling. Find the proper duct diameter to achieve the required velocity.

Answer: Since Q = 150 ft^3/min and $V_{d(min)}$ = 3500 ft/min,

$$A = \frac{Q}{V} = \frac{150\ ft^3/min}{3500\ ft/min} = 0.0429\ ft^2$$

From Table 8-6, a 2.5-inch diameter duct with an area of 0.0341 ft^2 is the largest duct that will not exceed the maximum allowable area.

$$V_{actual} = \frac{Q}{A} = \frac{150\ ft^3/min}{0.0341\ ft^2} = 4399\ ft/min.$$

This is a relatively high value for duct velocity compared to the design criteria of 3500 ft/min, but the next larger standard duct diameter (3-in.) will give a duct velocity of 3055 ft/min, which is below the minimum specified velocity.

Single-Hood System

Designing single-hood ventilation systems involves adding up all the pressure losses in the system after choosing the hood type, airflow rate, and duct layout. A single-hood system is easier to design than a multi-hood system because there is no need to add dampers or adjust duct diameters to distribute the air properly between hoods.

Sample Design 1: Welding Bench System

Design the welding bench system shown in Figure 8-5 to operate at standard conditions. Hood design parameters