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Chapter 10

HOODS, DUCTWORK, and STACKS

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10.1 Introduction

Most control devices are located some distance from the emission sources they control. This separation may be needed for several reasons. For one thing, there may not be enough room to install the control device close to the source. Or, the device may collect emissions from several sources located throughout the facility and, hence, must be sited at some convenient, equidistant location. Or, it may be that required utility connections for the control device are only available at some remote site. Regardless of the reason, the waste gas stream must be conveyed from the source to the control device and from there to a stack before it can be released to the atmosphere.

The kinds of equipment needed to convey the waste gas are the same for most kinds of control devices. These are: (1) *hoods*, (2) *ductwork*, (3) *stacks*, and (4) *fans*. Together, these items comprise a *ventilation system*. A hood is used to capture the emissions at the source; ductwork, to convey them to the control device; a stack, to disperse them after they leave the device; and a fan, to provide the energy for moving them through the control system. This chapter covers the first three kinds of equipment. However, because they constitute such a broad and complex subject, fans will be dealt with in a future *Manual* chapter. Also, the kinds of stacks covered are short stacks (100-120 feet high or less). Typically, these are included with packaged control systems or added to them. So-called "tall stacks" ("chimneys"), used at power plants or other sources where the exhaust gases must be dispersed over great distances, will *not* be discussed in this chapter.

This chapter presents all the information one would need to develop study ($\pm$ 30%-accurate) cost estimates for hoods, ductwork, and stacks. Accordingly, the following sections include: (1) descriptions of the types of equipment used in air pollution control ventilation systems, (2) procedures for sizing (designing) this equipment, and (3) methodologies and data for estimating their capital and annual costs. Also, sprinkled throughout the chapter are several illustrations (example problems) that show the reader how to apply the various sizing and costing methodologies.

10.2 Equipment Description

In this section, the kinds of hoods, ductwork, and stacks used in air pollution control systems are described, each in a separate subsection. These descriptions have been based on information obtained from standard ventilation and air pollution control references, journal articles, and equipment vendors.

10.2.1 Hoods

Of the several components of an air pollution control system, the capture device is the most important. This should be self-evident, for if emissions are not efficiently captured at the source they cannot be conveyed to and removed by a control device. There are two general categories of capture devices: (1) *direct exhaust connections* (DEC) and (2) *hoods*. As the name implies, a DEC is a section of duct (typically an elbow) into which the emissions directly flow. These connections often are used when the emission source is itself a duct or vent, such as a *process vent* in a chemical manufacturing plant or petroleum refinery. (See discussion below on "Ductwork".)

Hoods comprise a much broader category than DECs. They are used to capture particulates, gases, and/or mists emitted from a variety of sources, such as basic oxygen steelmaking furnaces, welding operations, and electroplating tanks. The hooded processes are generally categorized as either "hot" or "cold", a delineation that, in turn, influences hood selection, placement, and design.

The source conditions also influence the materials from which a hood is fabricated. Mild (carbon) steel is the material of choice for those applications where the emission stream is noncorrosive and of moderate temperature. However, where corrosive substances (e.g., acid gases) are present in high enough concentrations, stainless steels or plastics (e.g., fiberglass-reinforced plastic, or FRP) are needed. As most hoods are custom-designed and built, the vendor involved would determine which material would be optimal for a given application.

10.2.1.1 Types of Hoods

Although the names of certain hoods vary, depending on which ventilation source one consults, there is general agreement as to how they are classified. There are four types of hoods: (1) *enclosures*, (2) *booths*, (3) *captor (capture) hoods*, and (4) *receptor (receiving) hoods*.[1,2]

Enclosures are of two types: (1) those that are completely closed to the outside environment and (2) those that have openings for material input/output. The first type is only used when handling radioactive materials, which must be handled by remote manipulators. They are also dust- and gas-tight. These kinds of enclosures are rarely used in air pollution control.

Total enclosures, the second type, have applications in several areas, such as the control of emissions from electric arc furnaces and from screening and bin filling operations. They are equipped with small wall openings (natural draft openings—"NDO's") that allow for material to be moved in or out and for ventilation. However, the area of these openings must be small compared with the total area of the enclosure walls (typically, 5% or less).

Another application of total enclosures is in the measurement of the *capture efficiency* of VOC (volatile organic compound) control devices. Capture efficiency is that fraction of all VOC's generated at, and released by, an affected facility that is directed to the control device. In this application, a total enclosure is a temporary structure that completely surrounds an emitting process so that all VOC emissions are captured for discharge through ducts or stacks. The air flow through the total enclosure must be high enough to keep the concentration of the VOC mixture inside the enclosure within both the Occupational Safety and Health Administration (OSHA) health requirement limits and the vapor explosive limits. (The latter are typically set at 25% of the lower explosive limit (LEL) for the VOC mixture in question.) In addition, the overall face velocity of air flowing through the enclosure must be at least 200 ft/min.[3]

The surfaces of temporary total enclosures are usually constructed either of plastic film or of such rigid materials as insulation panels or plywood. Plastic film offers the advantages of being lightweight, transparent, inexpensive, and easy to work with. However, it is flimsy, flammable, and has a relatively low melting point. In addition, the plastic must be hung on a framework of wood, plastic piping, or scaffolding.

Although rigid materials are more expensive and less workable than plastic, they are more durable and can withstand larger pressure differentials between the enclosure interior and exterior. Total enclosure design specifications (which have been incorporated into several EPA emission standards) are contained in the EPA report, *The Measurement Solution: Using a Temporary Total Enclosure for Capture Testing.*[4]

Booths are like enclosures, in that they surround the emission source, except for a wall (or portion thereof) that is omitted to allow access by operators and equipment. Like enclosures, booths must be large enough to prevent particulates from impinging on the inner walls. They are used with such operations (and emission sources) as spray painting and portable grinding, polishing, and buffing operations.

Captor Hoods: Unlike enclosures and booths, *captor hoods* (also termed *active* or *external hoods*) do not enclose the source at all. Consisting of one to three sides, they are located at a distance from the source and draw the emissions into them via fans. Captor hoods are further classified as *side-draft/back-draft, slot, downdraft,* and *high-velocity, low-volume (HVLV) hoods.* A *side-draft/back-draft hood* is typically located to the side/behind of an emission source, but as close to it as possible, as air velocities decrease inversely (and sharply) with distance. Examples of these include snorkel-type welding hoods and side shake-out hoods.

A *slot hood* operates in a manner similar to a side-draft/back-draft. However, the inlet opening (face) is much smaller, being long and narrow. Moreover, a slot hood is situated at the periphery of an emission source, such as a narrow, open tank. This type of hood is also employed with bench welding operations.

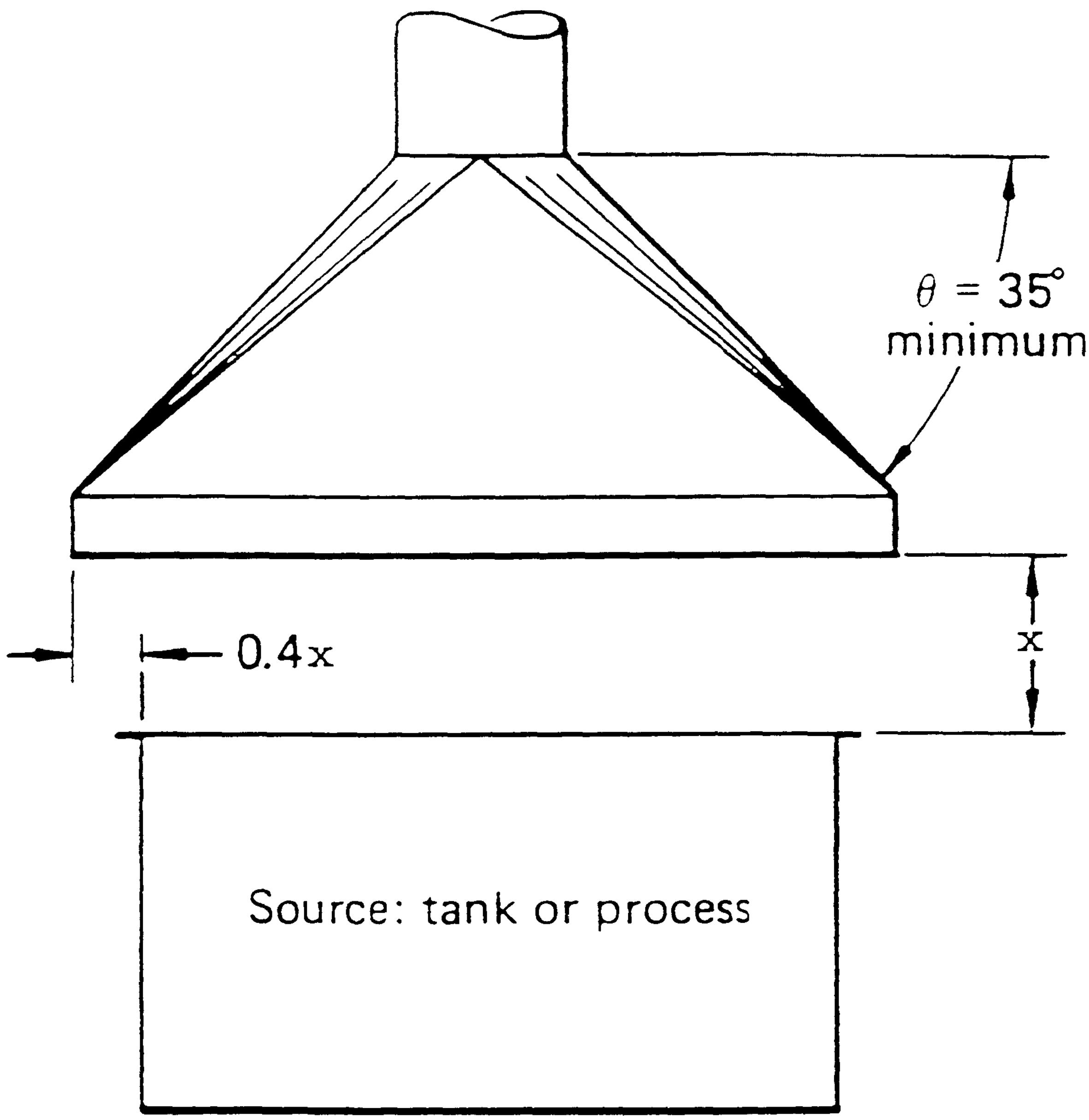
While slot and side-draft/back-draft hoods are located beside/behind a source, a *downdraft hood* is situated immediately beneath it. It draws pollutant-laden air down through the source and, thence, to a control device. Applications of down-draft hoods include foundry shake-out and bench soldering and torch cutting operations.

HVLV hoods are characterized by the use of extremely high velocities (*capture velocities*) to collect contaminants at the source, and by the optimal distribution of those velocities across the hood face. To maintain a low volumetric flow rate, these hoods are located as close to the source as possible, so as to minimize air entrainment.

Receptor hoods: The last category is *receptor hoods* (a.k.a. *passive* or *canopy hoods*). A receptor hood typically is located above or beside a source, to collect the emissions, which are given momentum by the source. For example, a *canopy* hood might be situated directly above an open tank containing a hot liquid (a *buoyant source*). With entrained air, vapors emitted from the liquid would rise into the hood. Here, the canopy hood would function as a passive collector, as the rising gases would be drawn into the hood via natural draft. (See Figure 10.1.)

Receptor hoods are also used with *nonbuoyant sources*, sources from which emissions do not rise. However, the emissions can be "thrown off" from a process, such as a swing grinder. The initial velocity of the emissions typically is high enough to

Figure 10.1 Typical Canopy Hood Installation



convey them into a receiving hood.[5]

10.2.2 Ductwork

Once the emission stream is captured by either a hood or a direct exhaust connection, it is conveyed to the control device via ductwork. The term "ductwork" denotes all of the equipment between the capture device and the control device. This includes: (1) straight duct; (2) fittings, such as elbows and tees; (3) flow control devices (e.g., dampers); and (4) duct supports. These components are described in Section 10.2.2.1.)

In air pollution control systems, the fan is usually located immediately before or after the control device. Consequently, most of the ductwork typically is under a negative static pressure, varying from a few inches to approximately 20 inches of water column. These pressure conditions dictate the type of duct used, as well as such design parameters as the wall thickness (gauge). For instance, welded duct is preferable to spiral-wound duct in vacuum applications.[6]

Ductwork is fabricated from either metal or plastic, the choice of material being dictated by the characteristics of the waste gas stream, structural considerations, purchase and installation costs, aesthetics, and other factors. Metals used include carbon steel (bare or galvanized), stainless steel, and aluminum. The most commonly used plastics are PVC (polyvinyl chloride) and FRP (fiberglass-reinforced plastic), although polypropylene (PP) and linear polyethylene (LPE) also have been applied. However, one serious drawback to PP and LPE is that both are *combustible.*[7]

PVC and other plastic ductwork are resistant to a variety of corrosive substances, from aqua regia to 95% sulfuric acid. But plastic ductwork cannot tolerate environmental temperatures above 150°F.[8] Metal ductwork can handle temperatures up to approximately 1000°F, but only certain alloys can tolerate corrosive streams.

In terms of construction, ductwork can be either *rigid* or *flexible.* As the name implies, rigid ductwork, whether metal or plastic, has a fixed shape. Conversely, flexible ductwork can be bent to accomodate situations where space is limited or where the layout is so convoluted that rigid fittings cannot meet construction requirements. Usually circular in cross-sectional shape, flexible duct can be fabricated from metals or plastic and can be either insulated or uninsulated.

Rigid ductwork is fabricated into circular, flat oval, or square/rectangular cross-sectional shapes. Of these, circular duct is most commonly used in air pollution control systems.

Although square/rectangular duct is advantageous to use when space is limited, round duct offers several advantages. It resists collapsing, provides better transport conditions, and uses less metal than square/rectangular or flat oval shapes of equivalent cross-sectional area.[9] Unless otherwise noted, the following discussion will pertain to rigid, circular duct, as this is the type most commonly used in air pollution control.

Rigid metal circular duct is further classified according to method of fabrication. *Longitudinal seam* duct is made by bending sheet metal into a circular shape over a mandrel, and butt-welding the two ends together. *Spiral seam* duct is constructed from a long strip of sheet metal, the edges of which are joined by an interlocking helical seam that runs the length of the duct. This seam is either raised or flush to the duct wall surface.

Fabrication method and cross-sectional shape are not the only considerations in designing ductwork, however. One must also specify the diameter; wall thickness; type, number, and location of fittings, controllers, and supports; and other parameters. Consequently, most ductwork components are custom-designed and fabricated, so as to optimally serve the control device. Some vendors offer prefabricated components, but these are usually common fittings (e.g., 90° elbows) that are available only in standard sizes (e.g., 3- to 12-inch diameter)[10,11].

If either the gas stream temperature or moisture content is excessive, the ductwork may need to be insulated. Insulation inhibits heat loss/gain, saving energy (and money), on the one hand, and prevents condensation, on the other. Insulation also protects personnel who might touch the ductwork from sustaining burns. There are two ways to insulate ductwork. The first is to install insulation on the outer surface of the ductwork and cover it with a vapor barrier of plastic or metal foil. The type and thickness of insulation used will depend on several heat transfer-related parameters. For instance, one vendor states that 4 inches of mineral wool insulation is adequate for maintaining a surface ("skin") temperature of 140°F (the OSHA workplace limit) or lower, provided that the exhaust gas temperature does not exceed 600°F.[12]

The second way to insulate ductwork is by using *double-wall, insulated duct and fittings.* Double-wall ductwork serves to reduce both heat loss and noise. One vendor constructs it from a solid sheet metal outer pressure shell and a sheet metal inner liner with a layer of fiberglass insulation sandwiched between. The insulation layer is typically 1-inch, although 2- and 3-inch thicknesses are available for more extreme applications. The thermal conductivities of these thicknesses are 0.27, 0.13, and 0.09 Btu/hr-ft^2-°F, respectively.[13]

10.2.2.1 Ductwork Components

As discussed above, a ductwork system consists of straight duct, fittings, flow control devices, and supports. Straight duct is self-explanatory and easy to visualize. The "fittings" category, however, encompasses a range of components that perform one or more of the following functions: change the direction of the ducted gas stream, modify the stream velocity, tie it to another duct(s), facilitate the connection of two or more components, or provide for expansion/contraction when thermal stresses arise.

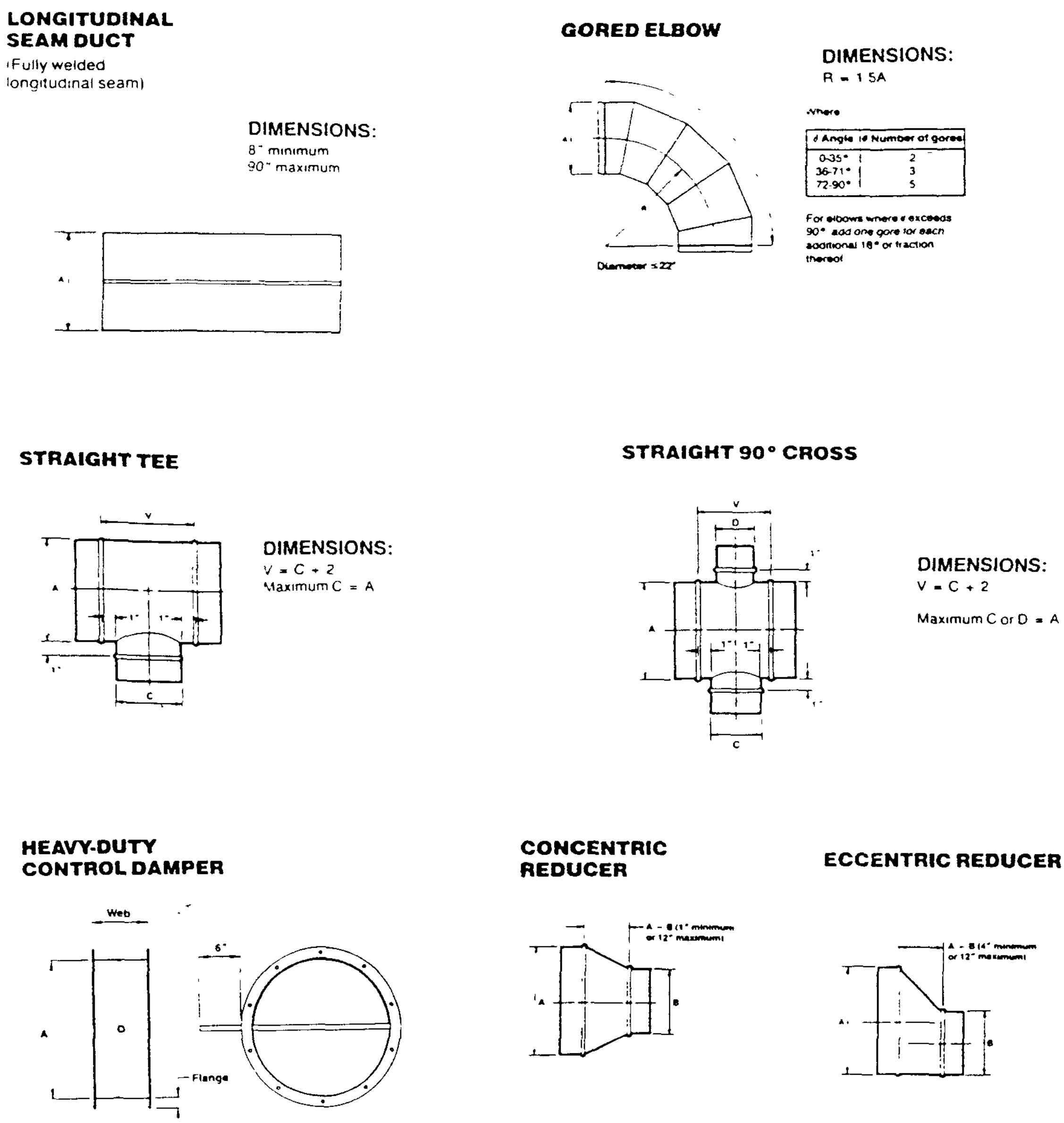
The most commonly used fittings are *elbows* ("ells"). These serve to change the gas stream direction, typically by 30°, 45°, 60°, or 90°, though they may be designed for other angles as well. The elbow centerline radius determines the rate at which this directional change occurs. (See Figure 10.2.) The standard centerline radius (R_{cl}) is 1.5 x the elbow cross-sectional diameter (D_c). However, in "long-radius" elbows, in which the directional change is more gradual than in standard elbows, R_{cl} = $\geq 2D_c$.[14]

Tees are used when two or more gas streams must be connected. In *straight tees*, the streams converge at a 90° angle, while in *angle tees* ("laterals", "wyes") the connection is made at 30°, 45°, 60°, or some other angle. (See Figure 10.2.) Tees may have one "tap" (connection) or two, and may have either a straight or a "conical" cross-section at either or both ends. *Crosses* are also used to connect duct branches. Here, the two branches intersect each other at a right angle.

Reducers (commonly called "expansions" or "contractions") are required whenever ducts of different diameter must be joined. Reducers are either *concentric* or *eccentric* in design. In concentric reducers, the diameter tapers gradually from the larger to smaller cross section. However, in eccentric reducers, the diameter decreases wholly on one side of the fitting.

To control the volumetric flowrate through ventilation systems, *dampers* are used. Dampers are usually delineated according to the flow control mechanism (*single blade* or *multiblade*), pressure rating (low/light or high/heavy), and means of control (manual or automatic). In *single blade* dampers, a circular plate is fastened to a rod, one end of which protrudes outside the duct. In the most commonly used type of single blade damper (*butterfly* type), this rod is used to control the gas flow by rotating the plate in the damper. Fully closed, the damper face sits perpendicular to the gas flow direction; fully open, the face is parallel to the gas flow lines. Several single blade "control" dampers are depicted in Figure 10.2.

Figure 10.2 Selected Circular Ductwork Components†



θ Angle	θ Number of gores
0-35°	2
36-71°	3
72-90°	5

† Reference: "Single-Wall Round and Flat Oval Duct and Fittings." In: *Sheet Metal Division Catalog*. Groveport, OH: United McGill Corporation. 1990.

With *blast gate dampers*, a second type, the flow is controlled by sliding the damper blade in and out of the duct. Blast gates are often used to control the flow of air streams containing suspended solids, such as in pneumatic conveyors. In these respects, butterfly dampers and blast gates are analogous, respectively, to the globe valves and quick-opening gate valves that are used to regulate liquid flow in pipes.

Multiblade (*louvered*) dampers operate by means of the same principal. However, instead using a single blade or plate to control the gas flow, multiblade dampers employ slats that open and close like venetian blinds.[15] Louvered dampers typically are used in very large ducts where a one-piece damper blade would be too difficult to move.

Manually-controlled dampers simply have a handle attached to the control rod which is used to adjust the gas flow by hand. If automatic control is needed, a pneumatic or electronic actuator is used. The actuator receives a pneumatic (pressurized air) or electrical signal from a controller and converts it to mechanical energy which is used, in turn, to open/close the damper via the damper rod. In this respect, an actuated damper is analogous to an automatic control valve.[16] For example, an automatic damper may be used to control the dilution air flow rate to an incinerator combustion chamber. This flow rate, in turn, would depend on the combustibles concentration (i.e., percentage of lower explosive limit—%LEL) in the inlet waste gas stream. If this concentration deviates from a predetermined amount ("set point"), a signal is sent from the measuring device via the controller to the automatic damper to increase/decrease the dilution air flow rate so as to maintain the desired %LEL.

Expansion joints are installed, especially in longer metal duct runs, to allow the ductwork to expand or contract in response to thermal stresses. These fittings are of several designs. One type, the bellows expansion joint, consists of a piece of flexible metal (e.g., 304 stainless steel) that is welded to each of two duct ends, connecting them. As the temperature of the duct increases, the bellows compresses; as the duct temperature decreases, the bellows expands.

Another commonly used expansion joint consists of two flanges between which is installed a section of fabric. Like the bellows expansion joint, it compresses as the duct temperature increases, and vice-versa. The temperature dictates the type of fabric used. For instance, silicone fiberglass and aramid fiber cloth can be used for duct temperatures of up to 500°F., while coated fiberglass cloth is needed to accommodate temperatures of 1,000°F.[17]

The last component to consider is the *ductwork support system*. However, it is far from being the least important. As the SMACNA (Sheet Metal and Air Conditioning Contractors' National Association) *HVAC Duct Construction Standards* manual states, "The selection of a hanging system should not be taken lightly, since it involves not only a significant portion of the erection labor, but also because [the erection of] an inadequate hanging system can be disastrous." As a rule, a support should be provided for every 8 to 10 feet of duct run.[18] Ductwork can be suspended from a ceiling or other overhead structure via hangers or supported from below by girders, pillars, or other supports.

A suspension arrangement typically consists of an upper attachment, a hanger, and a lower attachment. The upper attachment ties the hanger to the ceiling, etc. This can be a concrete insert, an eye bolt, or a fastener such as a rivet or nailed pin. The hanger is generally a strap of galvanized steel, round steel rod, or wire that is anchored to the ceiling by the upper attachment. The type of hanger used will be dictated by the duct diameter, which is proportional to its weight per lineal foot. For instance, wire hangers are only recommended for duct diameters up to 10 inches. For larger diameters (up to 36 inches), straps or rods should be used. Typically, a strap hanger is run from the upper attachment, wrapped around the duct, and secured by a fastener (the lower attachment). A rod hanger also extends down from the ceiling. Unlike strap hangers, they are fastened to the duct via a band or bands that are wrapped around the circumference. Duct of diameters greater than 3 feet should be supported with two hangers, one on either side of the duct, and be fastened to two circumferential bands, one atop and one below the duct.[19] Moreover, supports for larger ductwork should also allow for both axial and longitudinal expansion and contraction, to accomodate thermal stresses.[20]

10.2.3 Stacks

Short stacks are installed after control devices to disperse the exhaust gases above ground level and surrounding buildings. As opposed to "tall" stacks, which can be up to 1000 feet high, short stacks typically are no taller than 120 feet.

Certain packaged control devices come equipped with short ("stub") stacks, with heights ranging from 30 to 50 feet. But if such a stack is neither provided nor adequate, the facility must erect a separate stack to serve one or more devices. Essentially, this stack is a vertical duct erected on a foundation and supported in some manner. For structural stability, the diameter of the stack bottom is slightly larger than the top diameter, which typically ranges from 1 to 7 feet.[21]

A short stack may be fabricated of steel, brick, or plastic (e.g., fiberglass-reinforced plastic, or FRP). A stack may be lined or unlined. The material selection depends on the physical and chemical properties of the gas stream, such as corrosiveness and acidity, as well as the temperature differential between the gas stream and the ambient air. Liners of stainless steel, brick, or FRP usually are used to protect the stack against damage from the gas stream. They are much easier and less expensive to replace than the entire stack. Alternatively, the interior of an unlined stack may be coated with zinc (galvanized), aluminum, or another corrosion-resistant material, but a coating does not provide the same protection as a liner and does not last as long.[22]

Short stacks are either self-supporting (free-standing), supported by guy wires, or fastened to adjacent structures. The type of support used depends on the stack diameter, height and weight, the wind load, local seismic zone characteristics, and other factors.

Auxiliary equipment for a typical stack includes an access door, a sampling platform, ladders, lightning protection system, and aircraft warning lights. The access door allows for removal of any accumulated materials at the bottom of the stack and provides access to the liner for repair or replacement. Local and state air pollution control regulations also may require the permanent installation of sampling platforms for use during periodic compliance tests, while ladders are used both during stack sampling and maintenance procedures. The lightning protection system is needed to prevent damage to the stack and immediate surroundings during electrical storms. Lastly, aircraft warning lights are required by local aviation authorities.[23] Altogether, these auxiliaries can add a large amount to the base stack cost.

10.3 Design Procedures

As stated above, a hood, ductwork, and a stack are key elements in any air pollution control system. Because each of these elements is different, both in appearance and function, each must be designed separately. But at the same time, these elements comprise a *system*, which is governed by certain physical laws that serve to unite these elements in "common cause". Thus, before the individual design procedures for hoods, ductwork, and stacks are described, ventilation fundamentals will be presented. These fundamentals will cover basic fluid flow concepts and how they may be applied to air pollution control ventilation systems. Nonetheless, these concepts will be given as straightforwardly as possible, with the aim of making the design parameters easy to understand and compute.

10.3.1 Design Fundamentals

10.3.1.1 The Bernoulli Equation

The flow of fluids in any hood, duct, pipe, stack, or other enclosure is governed by a single *relationship*, the *familiar Bernoulli equation.* Put simply and ideally, the Bernoulli equation states that the total mechanical energy of an element of flowing fluid is constant throughout the system. This includes its potential energy, kinetic energy, and pressure energy. However, as no system is ideal, the Bernoulli equation must be adjusted to take into account losses to the surroundings due to friction. Gains due to the energy added by fans, pumps, etc., also must be accounted for. For a pound mass (lb_m) of fluid flowing in a steady-state system the adjusted Bernoulli equation is:[24]

$$\int v\,dp + \Delta z(g/g_c) + \Delta(u^2)/2g_c = W - F \qquad (10.1)$$

where: v = specific volume of fluid (ft^3/lb_m)
p = static pressure—gauge (lb_f/ft^2)
z = height of fluid above some reference point (ft)
u = fluid velocity through duct, hood, etc. (ft/sec)
g = gravitational acceleration (ft/sec^2)
g_c = gravitational constant (32.174 ([lb_m-ft/sec^2]/lb_f)
W = work added by fan, etc. (ft-lb_f/lb_m)
F = energy lost due to friction (ft-lb_f/lb_m)

Each of the terms on the left hand side of equation 10.1 represents an energy change to a pound mass of fluid between two locations in the system—points "1" and "2". The work (W) and friction (F) terms denote the amounts of energy added/lost between points 1 and 2.

Note that the units of each term in equation 10.1 are "ft-lb_f/lb_m," energy per unit mass. In the English system of units, "lb_f" and "lb_m" are, for all intents, numerically equivalent, since the ratio of the gravitational acceleration term (g) to the gravitational constant (g_c) is very close to 1. In effect, therefore, the equation units are "feet of fluid" or "fluid head in feet". In air pollution control situations, the fluid often has the properties of air. That is because the contaminants in the waste gas stream are present in such small amounts that the stream physical properties approximate those of pure air.

Because air is a "compressible" fluid, its specific volume is much more sensitive to changes in pressure and temperature than the specific volume of such "incompressible" fluids as water. Hence, the "vdp" term in the equation has to be integrated between points 1 and 2. However, in most air pollution control ventilation systems neither the pressure nor the temperature changes appreciably from the point where the emissions are captured to the inlet of the control device. Consequently, the specific volume is, for all practical purposes, constant throughout the ventilation system, and one does not have to integrate the vdp term. With this assumption, the first term in equation 10.1 becomes simply:

$$\int v\,dp = v\int dp = v\Delta p \tag{10.2}$$

Illustration: VOC emitted by an open tank is captured by a hood and conveyed, via a blower, through 150 feet of 12-inch diameter ductwork to a refrigerated condenser outdoors. The blower, which moves the gas through the hood, ductwork, and condenser, is located immediately before the inlet to the condenser. Thus, the entire ventilation system is under vacuum. The stream temperature and absolute pressure are 100°F and approximately 1 atmosphere (14.696 lb_f/in^2), respectively. The elevation of the refrigerated condenser inlet is 30 feet below that of the tank. The air velocity at the source is essentially zero, while the duct transport velocity is 2,000 ft/min. The static gauge pressure increases from -0.50 in. w.c. (water column) at the source to 4.5 in. w.c. at the blower outlet. Finally, the calculated friction loss through the ductwork and hood totals 1.25 in. w.c. Calculate the amount of mechanical energy that the blower adds to the gas stream. Assume that the gas temperature remains constant throughout.

Solution:

☞ First, develop a factor to convert "inches of water" to "feet of air":

$$\text{Feet of air} = (\text{Inches of water})(1\text{ ft}/12\text{ in})(v_{a100}/v_{w100}) \tag{10.3}$$

where: v_{w100} = specific volume of water @ 100°F = 0.01613 ft^3/lb_m
v_{a100} = specific volume of air @ 100°F, 1 atmosphere

Because the system absolute pressure is close to atmospheric, the waste gas behaves as an *ideal gas*. Thus, the specific volume can be calculated from the ideal gas law:

$$v_a = RT/pM \tag{10.4}$$

where: R = ideal gas constant = 1,545 ft-lb_f/(lb_m-mole)(°R)
T = absolute temperature of gas = 100 + 460 = 560°R
M = molecular weight of gas (air) = 28.85 lb_m/lb_m-mole
p = absolute pressure = 2,116 lb_f/ft^2

Substituting, we obtain:

$v_a = 14.17\ ft^3/lb_m$

Finally, substitution of these values for v_a and v_w into equation 10.3 yields:

Feet of air (@ 100°F, 1 atm.) = 73.207 x Inches of water

☞ Compute the *changes* in the mechanical energy terms and the friction losses between the hood inlet (point 1) and the blower outlet/condenser inlet (point 2):

Pressure: $v\Delta p$ = (4.5 - [-0.50] in. w.c.)(73.207 ft air/in. w.c.)
= 366.0 ft air

Potential: Δz = -30 ft air (point 2 is below point 1)

Kinetic: $\Delta u^2/2g_c$ = $([2{,}000\ ft/min]/[60\ ft/min/1\ ft/sec])^2$ x
$(1/2)(32.174[lb_m\text{-}ft/sec^2]/lb_f)^{-1}$
= 17.3 ft air

Friction losses: F = 1.25 in. w.c. x 73.207
= 91.5 ft air

☞ Substitute above results into equation 10.1 and solve for W, the fan energy added:

366.0 + (-30) + 17.1 = W - 91.5, or
W = 444.6 $ft\text{-}lb_f/lb_m$ air = 6.07 in. w.c.

To convert the fan energy input, W, to horsepower (hp_f), we would have to multiply it by the air mass flow rate (lb_m/sec), and divide the result by the horsepower conversion factor, 550 $ft\text{-}lb_f$/sec-hp. However, the mass flow rate is just the volume flow rate (Q, ft^3/sec) divided by the specific volume:

$$hp_f = W(Q/v_a)(1/550) = 0.001818WQ/v_a \qquad (10.5)$$

(The reader may wish to compare this equation to the fan horsepower equation in Chapter 3 [page 3-55] of this manual.)

In turn, Q is a function of the duct velocity (u_t, ft/sec) and duct diameter (D_d, ft):

$$Q = u_t(\pi D_d^2/4) \qquad (10.6)$$

Equation 10.6 applies, of course, only to *circular* ducts.

If we combine equations 10.5 and 10.6 and substitute the inputs for this illustration, we obtain:

$$hp_f = (444.6)(2{,}000/60)(\pi/4)(1)^2(1/14.17)(1/550)$$
$$= 1.49 \text{ hp}$$

* * *

Some observations about this illustration:

☞ Recall that the precise units for W and the other terms in equation 10.1 are "ft-lb_f/lb_m air," which, for convenience, have been shortened to "ft air". Thus, they measure *energy*, not length.

☞ Compared to the pressure energy and friction terms, the potential and kinetic energy terms are small. Had they been ignored, the results would not have changed appreciably.

☞ The large magnitude of the pressure and friction terms clearly illustrates the importance of keeping one's units straight. As shown in step (1), one inch of water is equivalent to over 73 feet of air. However, as equation 10.3 indicates, the *pressure* corresponding to equivalent heights of air and water columns would be the same.

☞ The fan power input depends not just on the total "head" (ft air) required, but also on the gas flow rate. Also, note that the horsepower computed via equation 10.5 is a *theoretical* value. It would have to be adjusted to account for the efficiencies of the fan and fan motor. As mentioned in Chapter 3, the fan efficiency ranges from 40 to 70 percent, while the motor efficiency is typically 90 percent. These efficiencies are usually combined into a single efficiency (ϵ, fraction), by which the theoretical horsepower is divided to obtain the actual horsepower requirement.

10.3.1.2 Pressure: Static, Velocity, and Total

Although it is more rigorous and consistent to express the Bernoulli equation terms in terms of feet of air (or, precisely, ft-lb_f/lb_m of air), industrial ventilation engineers prefer to use the units "inches of water column (in. w.c.)." These units were chosen because, as the above illustration shows, results expressed in "feet of air" are often large numbers that are cumbersome to use. In addition, the total pressure changes in ventilation systems are relatively small, compared to those in liquid flow systems. Total pressure changes expressed in inches of mercury would be small numbers which are just as awkward to work with as large numbers. Hence, "inches of water" is a compromise, as values expressed in this measurement unit typically range from only 1 to 10. Moreover, practical measurement of pressure changes is done with water-filled manometers.

In the previous paragraph, a new quantity was mentioned, *total pressure* (TP). Also known as the "impact pressure", the total pressure is the sum of the static gauge (SP) and velocity pressures (VP) at any point within a duct, hood, etc., all expressed in in. w.c.[25] That is:

$$TP = SP + VP \tag{10.7}$$

where: $SP = (cf)vp$
$VP = (cf)u^2/2g_c$

The "cf" in the expressions for SP and TP is the factor for converting the energy terms from "ft air" to "in. w.c.", both at standard temperature and absolute pressure (70°F, 1 atmosphere). (Again, keep in mind that, regardless of what units SP or VP are expressed in, the actual units are "energy per unit mass".) This conversion factor would be obtained via rearranging equation 10.3:

$$cf = \text{in. w.c./ft. air} = 12(v_{w70}/v_{a70}) \tag{10.8}$$

where: v_{w70} = specific volume of water at 70°F = 0.01605 (ft^3/lb_m)
v_{a70} = specific volume of air at 70°F = 13.41 (ft^3/lb_m)

Thus: cf = 0.01436 in. w.c./ft air

Clearly, "cf" varies as a function of temperature and pressure. For instance, at 100°F and 1 atmosphere, cf = 1/73.207 = 0.01366. *Nevertheless, unless noted otherwise, all quantities henceforth in this chapter will reflect conditions at 70°F and 1 atmosphere.*

Conspicuously absent from equation 10.7 is the potential energy term, "$z(g/g_c)$". This omission was not inadvertent. In ventilation systems, the potential energy (P.E.) is usually small compared to the other terms. (For example, see illustration above.) The P.E. is, of course, a function of the vertical distance of the measurement point in question from some datum level, usually the ground. At most, that distance would amount to no more than 20 or 30 feet, corresponding to a P.E. of approximately 0.3 to 0.4 in. w.c. Consequently, we can usually ignore the P.E. contribution in ventilation systems without introducing significant error.

The static gauge pressure in a duct is equal in all directions, while the velocity pressure, a function of the velocity, varies across the duct face. The duct velocity is highest at the center and lowest at the duct walls. However, for air flowing in a long, straight duct, the average velocity (u_t)

approximates the center line velocity (u_{cl}).[26] This is an important point, for the average velocity is often measured by a pitot tube situated at the center of the duct.

By substituting for "cf" in equation 10.7, we can obtain a simple equation that relates velocity to velocity pressure at standard conditions:

$$VP = 0.01436u_t^2/2g_c \qquad (10.9)$$

Solving:

$$u_t \text{ (ft/sec)} = 66.94(VP)^{1/2} \qquad (10.10)$$

Or:

$$u_t \text{ (ft/min)} = 4{,}016(VP)^{1/2} \qquad (10.11)$$

Incidentally, these equations apply to *any* duct, regardless of its shape.

As Burton describes it, static gauge pressure can be thought of as the "stored" energy in a ventilation system. This stored energy is converted to the kinetic energy of velocity and the losses of friction (which are mainly heat, vibration, and noise). Friction losses fall into several categories:[27]

☞ Losses through straight duct

☞ Losses through duct fittings—elbows, tees, reducers, etc.

☞ Losses in branch and control device entries

☞ Losses in hoods due to turbulence, shock, vena contracta

☞ Losses in fans

☞ Losses in stacks

These losses will be discussed in later sections of this chapter. Generally speaking, much more of the static gauge pressure energy is lost to friction than is converted to velocity pressure energy. It is customary to express these friction losses (ΔSP_f) in terms of the velocity pressure:

$$F = \Delta SP_f = kVP \qquad (10.12)$$

where: k = experimentally-determined loss factor (unitless)

Alternatively, equations 10.11 and 10.12 may be combined to express F (in. w.c.) in terms of the average duct velocity, u_t (ft/min):

$$F = (6.200 \times 10^{-8}) k u_t^2 \tag{10.13}$$

10.3.1.3 Temperature and Pressure Adjustments

Equations 10.8 to 10.13 were developed assuming that the waste gas stream was at standard temperature and pressure. These conditions were defined as 70°F and 1 atmosphere (14.696 lb_f/in^2), respectively. While 1 atmosphere is almost always taken as the standard pressure, several different standard temperatures are used in scientific and engineering calculations: 32°F, 68°F, and 77°F, as well as 70°F. The standard temperature selected varies according to the industry or engineering discipline in question. For instance, industrial hygienists and air conditioning engineers prefer 70°F as a standard temperature, while combustion engineers prefer 77°F, the standard temperature used in Chapter 3 ("Thermal and Catalytic Incinerators").

Before these equations can be used with waste gas streams not at 70°F and 1 atmosphere, their variables must be adjusted. As noted above, waste gas streams in air pollution control applications obey the ideal gas law. From this law the following adjustment equation can be derived:

$$Q_2 = Q_1 (T_2/T_1)(P_1/P_2) \tag{10.14}$$

where: Q_2, Q_1 = gas flow rates at conditions 2 and 1, respectively (actual ft^3/min)

T_2, T_1 = absolute temperatures at conditions 2 and 1, respectively (°R)

P_2, P_1 = absolute pressures at conditions 2 and 1, respectively (atm)

However, according to equation 10.6:

$$Q = u_t (\pi D_d^2/4)$$

If equations 10.6 and 10.14 were combined, we would obtain:

$$u_{t2} = u_{t1} (T_2/T_1)(P_1/P_2)(D_{d2}^2/D_{d1}^2) \tag{10.15}$$

This last expression can be used to adjust u_t in any equation, as long as the gas flow is in circular ducts.

10.3.2 Hood Design Procedure

10.3.2.1 Hood Design Factors

When designing a hood, several factors must be considered:[28]

- Hood shape
- Volumetric flow rate
- Capture velocity
- Friction

Each of these factors and their interrelationships will be explained in this section.

As discussed in section 10.2.1, the hood shape is determined by the nature of the source being controlled. This includes such factors as the temperature and composition of the emissions, as well as the dimensions and configuration of the emission stream. Also important are such environmental factors as the velocity and temperature of air currents in the vicinity.

The hood shape partly determines the volumetric flow rate needed to capture the emissions. Because a hood is under negative pressure, air is drawn to it from all directions. Consider the simplest type of hood, a plain open-ended duct. Now, envision an imaginary sphere surrounding the duct opening. The center of this sphere would be at the center of the duct opening, while the sphere radius would be the distance from the end of the duct to the point where emissions are captured. The air would be drawn through this imaginary sphere and into the duct hood. Now, the volume of air drawn through the sphere would be the product of the sphere surface area and the hood capture velocity, u_c:[29]

$$Q = u_c(4\pi x^2) \tag{10.16}$$

where: x = radius of imaginary sphere (ft)

Equation 10.16 applies to a duct whose diameter is small relative to the sphere radius. However, if the duct diameter is larger, the capture area will have to be reduced by the cross-sectional area of the duct (D_d), or:

$$Q = u_c(4\pi x^2 - \pi D_d^2/4) \tag{10.17}$$

Similarly, if a flange were installed around the outside of the duct end, the surface area through which the air was drawn—and the volume flow rate—would be cut in half. That occurs because the flange would, in effect, block the flow of air from points behind it. Hence:

$$Q = u_c(2\pi x^2) \tag{10.18}$$

From these examples, it should be clear that the hood shape has a direct bearing on the gas flow rate drawn into it. But equations 10.16 to 10.18 apply only to hoods with spherical flow patterns. To other hoods, other flow patterns apply—cylindrical, planal, etc. We can generalize this relationship between volumetric flow rate and hood design parameters as follows:

$$Q = f(u_i, x, Sh) \tag{10.19}$$

where: "f(...)" denotes "function of..."
"Sh" indicates hood shape factors
u_i = design velocity—capture, face, slot

Table 10.1 lists design equations for several commonly used hood shapes. As this table shows, Q is a function of x, the hood shape, and, in general, the capture velocity (u_c). But in one case (booth hood), the design velocity is the *hood face velocity* (u_f). And in the case of slotted side-draft and back-draft hoods, the *slot velocity* (u_s) is the design velocity. In reality, both the hood face and slot velocities are the same, as each measures the speed at which the gas passes through the hood inlet opening(s).

When gas enters a hood, there is mechanical energy loss due to friction. This friction loss is calculated using equations 10.1 and 10.2, assuming that the potential energy contribution from gravity, $\Delta z(g/g_c)$, and the work added to the system, W, are both zero. Thus:

$$vp_2 - vp_1 + u_2^2/2g_c - u_1^2/2g_c = -F \tag{10.20}$$

Replacing these terms with the corresponding ones from equations 10.7 and 10.12, we obtain:

$$SP_2 - SP_1 + VP_2 - VP_1 = -H_e = -k_hVP_2 \tag{10.21}$$

where: SP_i = static gauge pressure at point i (in. w.c.)
VP_i = velocity pressure at point i (in. w.c.)
H_e = *hood entry loss* (in. w.c.)
k_h = *hood loss factor* (unitless)

In this equation, subscript 1 refers to a point just outside the hood face. Subscript 2 denotes the point in the duct, just downstream of the hood, where the duct static pressure, SP_2 or SP_h, and the duct transport velocity, u_2 or u_t, are measured. At point 1, the hood velocity pressure, VP_1, is essentially zero, as the air velocity there is negligible. Moreover, the static gauge

Table 10.1 Design Equations, Loss Factors, and Coefficients of Entry for Selected Hood Types‡

Hood Type	Design Equation•	Loss Factor (k_h)	Coefficient of Entry (C_e)
Duct end (round)	$Q = 4\pi x^2 u_c$	0.93	0.72
Flanged duct end (round)	$Q = 2\pi x^2 u_c$	0.50	0.82
Free-standing slot hood	$Q = 2\pi x L u_c$	1.78	0.55
Slot hood w/sides, back	$Q = 0.5\pi x L u_c$	1.78	n.a.§§
Tapered hood	$Q = 2\pi x u_c$	0.06††	0.97
Booth hood with tapered take-off duct (round)	$Q = u_f A_h$	0.25	0.89
Canopy hood	$Q = 1.4 P x u_c$	0.25	0.89
Canopy hood w/insert	$Q = 1.4 P x u_c$	1.0	0.71
Dip tank hood (slotted)	$Q = 125 A_t$	1.78	n.a.
Paint booth hood	$Q = 100 A_b$	0.25	n.a.

‡ Reference: Burton, D. Jeff. *Industrial Ventilation Work Book*. Salt Lake City: DJBA, Inc. 1989.

• In the equations: Q = flow rate drawn into hood (ft^3/min)
x = distance from hood to source (ft)
u_c = hood capture velocity (ft/min)
u_f = hood face velocity (ft/min)
u_s = hood slot velocity (ft/min)
A_h = hood face area (ft^2)
P = perimeter of source (ft)
L = width of hood slot (ft)
A_t = tank + drainboard surface area (ft^2)
A_b = booth cross-sectional area (ft^2)

§§ Not applicable.

†† Both k_h and C_e pertain to round ducts and to hoods with a 45° taper. At other angles, k_h and C_e will differ.

pressure, SP_1, will be zero, as the absolute pressure at point 1 is assumed to be at one atmosphere, the reference pressure. After these simplifications are made, equation 10.21 can be rearranged to solve for the hood loss factor (k_h):

$$k_h = (-SP_h/VP_2) - 1 \qquad (10.22)$$

At first glance, it appears that k_h could be negative, since VP is always positive. However, as the air entering the hood is under a vacuum created by a fan downstream, SP_h must be negative. Thus, the term "$-SP_h/VP_2$" must be *positive*. Finally, because the *absolute value* of SP_h is larger than VP_2, $k_h > 0$.

The hood loss factor varies according to the hood shape. It can range from 0.04 for bell mouth hoods to 1.78 for various slotted hoods. A parameter related to the hood loss factor is the *coefficient of entry* (c_e).[30] This is defined as:

$$c_e = \{1/(1+k_h)\}^{1/2} \qquad (10.23)$$

c_e depends solely on the shape of the hood, and may be used to compute k_h and related parameters. Values of k_h and c_e are listed in Table 10.1.

Illustration: The static gauge pressure, SP_h, is -1.75 in. w.c. The duct transport velocity (u_t) is 3,500 ft/min. Calculate the loss factor and coefficient of entry for the hood. Assume standard temperature and pressure.

Solution: First, calculate the duct velocity pressure. By rearranging equation 10.11 and substituting for u_t, we obtain:

$$VP = (u_t/4{,}016)^2 = (3{,}500/4{,}016)^2 = 0.76 \text{ in. w.c.}$$

Next, substitute for VP in equation 10.22 and solve:

$$k_h = (-SP_h/VP) - 1 = (-[-1.75]/0.76) - 1 = 1.30.$$

Finally, use this value and equation 10.23 to calculate the coefficient of entry:

$$c_e = \{1/(1+1.30)\}^{1/2} = 0.66.$$

* * *

Hood design velocities are listed in Table 10.2. Three kinds of velocities are shown: (1) capture (defined in Section 10.2.1), (2) face, and (3) slot. As stated in Section 10.2.1, the *capture velocity* is the air velocity induced by the hood to capture contaminants emitted at some distance from the hood inlet. The *face velocity* is the average velocity of the air passing through the hood inlet (face). A similar parameter is the *slot velocity*, which is the average air velocity through the hood slot openings, whose area is only a fraction of the entire hood face area. Consequently, the slot velocity is usually much higher than the face velocity.[31]

Note that these velocities range from 50 to 100 ft/min (tank and degreasing hoods) to 2,000 ft/min, the recommended slot velocity for slotted side-draft/back-draft hoods. As a reference point, the velocity of air in industrial operations due to thermal mixing alone is 50 ft/min. Thus, hood design velocities must exceed this value if effective capture is to occur. [32]

Two other velocities are also discussed in the industrial hygiene literature, although they do not have as much bearing on hood design as the capture, face, or slot velocities. These are the *plenum velocity* and the *transport velocity*. The first is the velocity of the gas stream as it passes through the tapered portion of a hood (plenum) between the hood opening and the duct connection. This plenum is a transition area between the hood opening and duct. Consequently, the plenum velocity is higher than the hood face velocity, but lower than the duct (transport) velocity. The transport velocity—the gas velocity through the duct—varies according to the waste gas composition. It is a crucial parameter in determining the duct diameter, the static pressure loss, and the sizes of the system fan and fan motor. (For more on transport velocity, see Section 10.3.3.)

10.3.2.2 Hood Sizing Procedure

As with many control devices and auxiliaries, there are several approaches to sizing hoods. Some of these approaches are quite complex, entailing a series of complex calculations that yield correspondingly accurate results. For instance, one hood sizing method in the literature involves first determining the hood dimensions (length and width for rectangular hoods; diameter, for circular). The next step is to estimate the amount of metal plate area (ft^2) required to fabricate a hood of these dimensions, via parametric curves. (No curves are provided for nonmetal hoods.) This plate area is input to an equation that includes a "pricing factor" and the per-pound price of metal. The cost of labor needed to fabricate this hood is estimated from equations similar to the plate-area relationships. Finally, the metal and labor costs are summed to obtain the total fabricated hood cost.[33]

This method does yield reasonably accurate hood cost—or rather, it *did*. Unfortunately, the labor cost data are outdated—1977 vintage—which makes them unescalatable. (The rule-of-thumb time limit for escalating costs is five years.) Even if the costs were up-to-date, the procedure is cumbersome to use, especially if calculations are made by hand.

A simpler sizing method—yet one sufficiently accurate for study estimating purposes—involves determining a single dimension, the *hood face area* (A_f). This area, identical to the hood inlet area, can be correlated against the fabricated hood cost to yield a relatively simple cost equation with a single independent variable. To calculate A_f, the following information is needed:

☞ Hood type

☞ Distance of the hood face from source (x)

☞ Capture (u_c), face (u_f), or slot velocity (u_s)

☞ Source dimensions (for some hood types).

As the equations in Table 10.1 indicate, these same parameters are the ones that are used to determine the volumetric flow rate (Q) through the hood and ductwork. With most control devices and auxiliaries being sized, Q is given. For hoods, however, Q usually must be *calculated.*

Illustration: A circular canopy hood is being used to capture emissions from a chromium electroplating tank. The hood face is 6 feet above the tank, an 8-foot diameter circular vessel. The capture velocity for this example is 200 ft/min. Assuming that the tank and surroundings are at standard conditions, calculate the required volumetric flow rate drawn into the hood, the hood face area, and the hood face velocity.

Solution: Obtain the canopy hood equation from Table 10.1:

$$Q = 1.4Pxu_c \qquad (10.24)$$

where: P = perimeter of tank (ft)
x = distance of hood above tank (ft)
u_c = capture velocity (ft/min)

Because the tank is circular, $P = \pi(8) = 25.1$ ft. Therefore:

$$Q = (1.4)(25.1)(6)(200) = 42{,}200 \text{ ft}^3/\text{min}.$$

For this type of canopy hood, the hood diameter is 40% greater than the tank diameter (hence, the "1.4" factor in equation 10.24). Thus:

$$A_f = (\pi/4)([1.4][8])^2 = 98.5 \text{ ft}^2$$

Finally, the hood face velocity (u_f) would be:

Table 10.2 Hood Design Velocities[‡‡]

Operation/Hood Type	Velocity Type	Velocity Range (ft/min)
Tanks, degreasing	Capture	50 - 100
Drying oven	Face	75 - 125
Spray booth	Capture	100 - 200
Canopy hood	Capture	200 - 500
Grinding, abrasive blasting	Capture	500 - 2,000
Slot hood	Slot	2,000

[‡‡] Reference: Burton, D. Jeff. *Industrial Ventilation Work Book*. Salt Lake City: DJBA, Inc. 1989.

$u_f = Q/A_f = 42,200/98.5 = 428$ ft/min.

* * *

In this example, note that the hood face velocity is higher than the capture velocity. This is logical, given the fact that the hood inlet area is smaller than the area through which the tank fumes are being drawn. The face velocity for some hoods is even higher. For example, for slotted hoods it is at least 1,000 ft/min.[34] In fact, one vendor sizes the openings in his slotted hoods so as to achieve a slot velocity equal to the duct transport velocity.[35]

10.3.3 Ductwork Design Procedure

The design of ductwork can be an extremely complex undertaking. Determining the number, placement, and dimensions of ductwork components—straight duct, elbows, tees, dampers, etc.—can be tedious and time-consuming. However, for purposes of making study-level control system cost estimates, such involved design procedures are not necessary. Instead, a much simpler ductwork sizing method can be devised.

10.3.3.1 Two Ductwork Design Approaches

There are two commonly used methods for sizing and pricing ductwork. In the first, the total weight of duct is computed from the number and dimensions of the several components. Next, this weight is multiplied by a single price (in $/lb) to obtain the ductwork equipment cost. To determine the ductwork weight, one needs to know the diameter, length, and wall thickness of every component in the system. As stated above, obtaining these data can be a significant effort.

The second method is a variation of the first. In this technique, the ductwork components are sized and priced individually. The straight duct is typically priced as a function of length, diameter, and wall thickness, as well as, of course, the material of construction. The elbows, tees, and other fittings are priced according to all of these factors, except for length. Other variables, such as the amount and type of insulation, also affect the price. Because it provides more detail and precision, the second method will be used in this chapter.

10.3.3.2 Ductwork Design Parameters

Again, the primary ductwork sizing variable are *length, diameter,* and *wall thickness*. Another parameter is the amount of *insulation* required, if any.

☞ **Length:** The length of ductwork needed with an air pollution control system depends on such factors as the distance of the source from the control device and the number of directional changes required. Without having specific knowledge of the source layout, it is impossible to determine this length accurately. It could range from 20 to 2,000 feet or more. It is best to give the straight duct cost on a \$/ft basis and let the reader provide the length. This length must be part of the specifications of the emission source at which the ductwork is installed.

☞ **Diameter:** As discussed in Section 10.2.2., circular duct is preferred over rectangular, oval, or other duct shapes. Therefore:

$$A_d = \pi D_d^2/4 \qquad (10.25)$$

where: A_d = cross-sectional area of duct (ft^2)
D_d = duct diameter (ft)

The duct cross-sectional area is the quotient of the volumetric flow rate (Q) and the duct transport velocity (u_t):

$$A_d = Q/u_t \qquad (10.26)$$

Combining equations 10.25 and 10.26 and solving for D_d:

$$D_d = 1.128(Q/u_t)^{1/2} \qquad (10.27)$$

As Q is usually known, the key variable in equation 10.27 is the duct transport velocity. This variable must be chosen carefully. If the u_t selected is too low, the duct will be oversized and, more importantly, the velocity will not be high enough to convey the particulate matter in the waste gas stream to the control device. However, if u_t is too high, the static pressure drop (which is proportional to the *square* of u_t) will be excessive, as will be the corresponding fan power consumption.

Cost is also a consideration when determining the optimum duct diameter. The equipment cost increases with increasing duct diameter. However, the fan power cost changes *inversely* with diameter. Nonetheless, for study-estimating purposes, the optimum duct diameter does *not* have to be determined. It is sufficient to calculate the duct diameter merely by using the transport velocity values contained in this section.

The transport velocity typically varies from 2,000 to 6,000 ft/min, depending on the waste gas composition. The lower duct velocity would be adequate for a waste gas containing gaseous pollutants or very fine, light dusts, while the higher velocity would be needed to convey a stream with a large quantity of metals or other heavy or moist materials. The following velocities may be used as general guidance:[36]

Material(s) Conveyed	Minimum Transport Velocity (u_t, ft/min)
Gases; very fine, light dusts	2,000
Fine, dry dusts and powders	3,000
Average industrial dusts	3,500
Coarse dusts	4,000 - 4,500
Heavy or moist dust loading	$\geq$ 4,500

Table 10.3 supplements these values with recommended duct velocities for a variety of conveyed materials.

☞ **Wall thickness:** The wall thickness of a duct depends on several factors—internal pressure, diameter, material of fabrication, and other structural parameters. Nonetheless, duct of a given diameter can be fabricated of a range of wall thicknesses, and vice-versa. For instance, 24-in. diameter 304 stainless steel "fully-welded longitudinal seam duct" is fabricated in thicknesses ranging from 22 to 14 gauge (0.0313 to 0.0781 in.). This same range of gauges is used with duct diameters ranging from 3 to 36 in.[37]

Note that the gauge number *decreases* with increasing wall thickness. This measure, which is traditionally used in the metal fabricating industries, is more convenient to deal with than the thickness expressed in inches, as the latter are usually small numbers less than 0.25. Moreover, the gauge number varies according to the metal used—carbon steel (galvanized or nongalvanized), stainless steel, or aluminum. Gauges for these metals are given in Table 10.4 for a wide range of nominal thicknesses.

The gauge measure is not used with plastic duct, as the wall thickness is typically expressed in inches. In any event, the wall thickness usually does not need to be known to estimate duct cost, as this parameter is already accounted for in the cost equations. (See Section 10.4.)

☞ **Insulation:** As discussed in Section 10.2.2., insulation can be either installed on the outer surface of ductwork or the ductwork itself can be fabricated with built-in insulation. In the first case, the amount of insulation required will depend on several heat transfer variables, such as: the temperature, velocity, composition, and other properties of the waste gas; the ambient temperature; the duct diameter, wall thickness, and thermal conductivity; and the desired surface ("skin") temperature. Determining these variables involves making a series of complex calculations that, while well-established, are beyond the scope of this chapter. Such standard references as *Perry's Chemical Engineers' Handbook* and *Plant Design and Economics for Chemical Engineers* present these calculations, as do heat transfer bibliographies.[38,39]

The second approach is to select pre-insulated ductwork. As mentioned previously, it can be equipped with any type and thickness of insulation. However, 1, 2, or 3 inches is typical. (Prices for these are presented in Section 10.4.)

10.3.3.3 Ductwork Pressure Drop

As mentioned in Section 10.3.1, ventilation system energy losses due to friction are traditionally computed as fractions of the velocity pressure, VP. In most cases, equation 10.12 can be used to estimate these losses. Technically, though, these equations apply only to those regions in the ventilation system where there are no changes in the velocity pressure (i.e., where the duct diameter is constant). These regions would include straight duct, hoods, and such fittings as couplings and simple elbows. But, with tees, wyes, and other *divided flow fittings*, the velocity—and velocity pressure—are *not* constant between the fitting inlet and outlet. The corresponding friction loss (F_b) is a function of both the upstream (inlet) and branch VP's, as the following equation indicates:[40]

$$F_b = VP_u(k_b-1) + VP_b \tag{10.28}$$

where: VP_u, VP_b = upstream and branch velocity pressures, respectively (in. w.c.)

k_b = branch loss coefficient

However, divided flow fittings generally are not used with simple pollution control ventilation systems, except in those cases where a tee might be needed, say, for purposes of adding dilution air.[§]

As any fluid mechanics textbook would attest, the friction loss for ductwork is a complex function of several variables:

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§ Divided flow fittings *are* needed with more-complex control systems that collect waste gases from several emission points. The design of such ventilation systems is beyond the scope of this chapter, however.

Table 10.3 Minimum Duct Velocities for Selected Materials[§]

Material	Minimum Transport Velocity (ft/min)
Aluminum dust (coarse)	4,000
Brass turnings	4,000
Cast iron boring dust	4,000
Clay dust	3,500
Coal dust (powdered)	4,000
Cocoa dust	3,000
Cotton dust	3,000
Flour dust	2,500
Foundry dust	3,000 - 5,000[†]
Grain dust	2,500 - 3,000
Lead dust	4,000
Limestone dust	3,500
Magnesium dust (coarse)	4,000
Metal turnings	4,000 - 5,000
Plastics dust (buffing)	3,000
Rubber dust	2,500 (fine) - 4,000 (coarse)
Silica dust	3,500 - 4,500
Soap dust	3,000
Soapstone dust	3,000
Spray paint	2,000
Starch dust	3,000
Stone dust	3,500
Tobacco dust	3,500

[§] Reference: Burton, D. Jeff. *Industrial Ventilation Work Book*. Salt Lake City: DJBA, Inc. 1989.

[†] Transport velocity varies with foundry operation.

Table 10.4 Wall Thicknesses of Steel and Aluminum Duct[§]

Gauge Number	Nominal Thickness (inches)			
	Carbon Steel		Stainless Steel (304 or 316)	Aluminum 3003-H14[†]
	Galv[‡]	Nongalv[•]		
28	0.0187	0.0149	0.0156	0.025
26	0.0217	0.0179	0.0188	0.032
24	0.0276	0.0239	0.0250	0.040
22	0.0336	0.0299	0.0313	0.050
20	0.0396	0.0359	0.0375	0.063
18	0.0516	0.0478	0.0500	0.080
16	0.0635	0.0598	0.0625	0.090
14	0.0785	0.0747	0.0781	
12	0.1084	0.1046	0.1094	
10	0.1382	0.1345	0.1406	

[§] Reference: *Engineering Design Reference Manual for Supply Air Handling Systems*. Groveport, OH: United McGill Corporation. 1992.

[†] To provide equivalent strength and stiffness, the nominal thickness of aluminum is approximately 150% of the nominal thickness of galvanized carbon steel of the same gauge.

[‡] Galvanized and paintable galvanized carbon steel.

[•] Nongalvanized carbon steel.

duct diameter and length, transport velocity, and gas viscosity and density. Specifically, the Darcy-Weisbach and Colebrook equations are typically used to make this calculation, the latter being used to compute the *Reynolds number.*[41] Traditionally, the friction loss has been obtained from a nomograph or, more recently, computer programs. A typical nomograph is found in Burton.[42] Also, to simplify the calculation, empirical equations have been derived for certain kinds of commerically-available ductwork. For instance, to estimate the friction loss per 100 ft (F_d/100 ft) at standard conditions for round, spiral, galvanized ductwork having 10 joints per 100 ft, use the following equation:[43]

$$F_d/100\ \text{ft} = 0.136(1/D)^{1.18}(u_t/1{,}000)^{1.8} \qquad (10.29)$$

where: D_d = duct diameter (ft), and: $0.25 \leq D_d \leq 5$

Clearly, this equation provides the total friction loss, not the loss factor (k). However, the reader may compute k for a given diameter (D_d) and flow rate (Q) by simply dividing the equation 10.29 results by VP and multiplying by 100.

To estimate the friction loss for other duct materials, multiply the value from equation 10.29 by a roughness correction factor, approximate values of which are:[44]

Material	Roughness Correction Factor
Non-spiral-wound galvanized	0.9
Fiberglass (smooth finish)	0.8
ABS and PVC plastic	0.8
Concrete	1.4
Corrugated flex duct	2.3

Loss factors for fittings have also been compiled, based on experimental data. Mainly of interest are those for 90° elbows, arguably the most commonly used fitting in air pollution control systems. The "k_{90}" values for elbows vary according to the diameter and radius of curvature, which is expressed as a multiple of the elbow diameter. Typical ranges of these values are as follows:[45]

Radius of Curvature	Friction Loss Factor (k_{90})
0.50	0.80
1.00	0.35
1.25	0.30 - 0.55
1.50	0.27 - 0.39
2.00	0.24 - 0.27
2.50	0.22 - 0.24

As these values indicate, the higher the radius of curvature, the lower the friction loss. This stands to reason, as the higher the radius of curvature, the more gradually the gas stream changes direction. For an elbow having of angle less than 90°, multiply the above k_{90} value by an adjustment factor (θ/90), so that:

$$k_\theta = (\theta/90)k_{90} \tag{10.30}$$

where: k_θ = loss factor for $\theta \leq 90°$

Illustration: A control device at a cosmetic factory is connected to a source by 250 feet of round spiral duct. The duct run includes three 90° elbows and two 45° elbows, each with a 1.50 radius of curvature. The volumetric flow rate (Q) of the waste gas (which contains entrained face powder) is 15,000 ft^3/min at standard conditions. Calculate the friction loss for the ductwork.

Solution: Because the material being conveyed in the ductwork (face powder) is light, an appropriate transport velocity (u_t) in this case is 2,000 ft/min. (See text table above.) Upon substituting this value and the volumetric flow rate into equation 10.27 we obtain the duct diameter (D_d):

$$D_d = 1.128(15{,}000/2{,}000)^{0.5} = 3.09 \text{ ft}$$

Next, substitute the diameter and velocity into equation 10.29 to compute the straight duct friction (static pressure) loss, F_d:

$$F_d = 0.136(1/3.09)^{1.18}(2{,}000/1{,}000)^{1.8}(250/100)$$

$$= 0.313 \text{ in. w.c.}$$

The 250/100 factor in this expression adjusts the friction loss from 100 feet (the basis of equation 10.29) to 250 feet (the length of the duct system in this illustration).

The rest of the friction loss occurs through the five elbows (three 90°, two 45°), each with a 1.50 radius of curvature. These losses (F_e) are computed via equation 10.12:

$$F_e = k_\theta VP$$

where: $VP = (2{,}000/4{,}016)^2$ (equation 10.11, rearranged)

$= 0.248$ in. w.c.

For the 90° elbows, $k_\theta = k_{90} = 0.33$ (average of table range), and:

$$F_e = 3 \times 0.33(0.248) = 0.246 \text{ in. w.c.}$$

For the 45° elbows, $k_\theta = (45/90)k_{90} = 0.165$ (equation 10.30), and:

$$F_e = 2 \times 0.165(0.248) = 0.0818 \text{ in. w.c.}$$

The total friction loss is, therefore:

$$F = 0.313 + 0.246 + 0.0818 = 0.641 \text{ in. w.c.}$$

* * *

From this illustration, two observations may be made: (1) the static pressure loss through the straight duct is not large, even at this length (250 ft.) and (2) the losses through the elbows—which total 0.328 in. w.c.—are larger than the straight duct loss. Though it may be tempting to neglect fittings losses for the sake of expediency, doing so can cause a significant underestimation of the ventilation system static pressure loss.

10.3.4 Stack Design Procedures

As with ductwork, the design of stacks involves a number of stream, structural, and site-specific parameters.[46,47] These include:

☞ **Waste gas variables:** inlet volumetric flow rate, temperature, and composition;

☞ **Site-specific data:** elevation above sea level, ambient temperature fluctuations, topographic and seismic data, meteorological records, and building elevations and layout;

☞ **Structural parameters:** thickness of stack wall and liner, location of breeching opening, type of supports, load capacity of foundation, modulus of resistance, and natural vibration frequency.

Fortunately, for study cost-estimating purposes, the only two stack design parameters that need to be determined are: (1) the stack diameter and (2) the stack height. The other variables (e.g., wall thickness) are incorporated into the equipment cost correlations. The stack diameter is relatively easy to determine, as it depends primarily on waste stream conditions. The stack height is more difficult to arrive at, as it is influenced by several site-specific variables. Nonetheless, ample guidance has been developed to allow the estimator to determine an acceptably accurate stack height.

10.3.4.1 Calculating Stack Diameter

Because most stacks have circular cross-sections, the stack diameter (D_s, ft) can be calculated via the duct diameter formula (equation 10.27):

$$D_s = 1.128(Q_c/u_c)^{1/2} \qquad (10.31)$$

where: u_c = stack exit velocity (ft/min)
Q_c = exit volumetric flow rate (actual ft^3/min)

It should be noted that the stack diameter in this formula is measured at the stack *exit*, not at the entrance. That is because, for structural reasons, the diameter at the bottom of the stack typically is larger than the top diameter. Also note that the stack exit velocity does *not* necessarily equal the duct transport velocity. Finally, Q_c may be different from the volumetric flow rate used to size the ductwork. Because the stack always follows the control device, the flow rate entering the device may not equal the flow rate entering the stack, either in standard or actual ft^3/min terms. For instance, in a thermal incinerator, the outlet standard waste gas flow rate is almost always higher than the inlet flow rate due to the addition of supplemental fuel.

The stack exit velocity, u_c, affects the *plume height*, the distance that the plume rises above the top of the stack once it exits. In a well-designed stack, u_c should be 1.5 times the wind speed. Typically, design exit velocities of 3,000 to 4,000 ft/min are adequate.[48] This range corresponds to wind speeds of 34 to 45 mi/hr.

10.3.4.2 Calculating Stack Height

Estimating the stack height is more difficult than calculating the stack exit diameter. The stack height depends on several variables: the height of the source; the stack exit velocity; the stack and ambient temperatures; the height, shape, and arrangement of the nearby structures and terrain; and the composition of the stack outlet gas. Some of these variables are straightforward to determine, while others (such as the dimensions and layout of nearby structures) are difficult to determine without performing on-site modeling and monitoring studies.

This height has two components: the height of the stack itself (H_s) and the plume rise height (H_{pr}). Together these components comprise the *effective stack height* (H_e). That is:

$$H_e = H_s + H_{pr} \tag{10.32}$$

However, the cost of the stack is a function of H_s alone. (See Section 10.4.) As discussed above, the plume rise is a function of the stack exit velocity. It also depends on the temperature differential between the stack gas and the ambient air. Specifically, a 1°F temperature difference corresponds to approximately a 2.5-ft. increase in H_{pr}.[49]

For those sources subject to State Implementation Plans (SIPs), the stack height (H_s) should be determined according to "good engineering practice" (GEP). GEP is defined as "the height necessary to insure that emissions from the stack do not result in excessive concentrations of any air pollutant in the immediate vicinity of the source as a result of atmospheric downwash, eddies, or wakes which may be created by the source itself, nearby structures, or nearby terrain obstacles."[50] In this respect, GEP establishes the *maximum allowable* stack height *credit* for purposes of calculating the ambient air quality impact of the emitting source. A source may build a stack to any height, but only a certain amount of stack height will be allowed in determining environmental impacts.[51]

For stacks constructed after January 12, 1979, the GEP stack height shall be the *greater of*: (1) 65 meters (213 ft); (2) the height demonstrated by an approved fluid model or field study that ensures that stack emissions do not cause excessive pollutant concentrations from atmospheric downwash, wakes, eddy effects., etc; or (3) the height determined by the following equation:[52]

$$H_s = H_b + 1.5L \tag{10.33}$$

where: H_s = GEP stack height, measured from the ground level elevation at the stack base (ft)
H_b = height of nearby structure(s) measured from this ground level elevation (ft)
L = lesser dimension (height or projected width of nearby structure(s))

10.3.4.3 Calculating Stack Draft

As discussed previously, waste gas flowing through hoods and ductwork loses static pressure due to friction. In the case of stacks, however, the gas stream can actually *gain* static pressure, as a result of *stack draft*, which is the draft created by the stack gas-ambient air temperature differential. Stack draft (SP_s, in. w.c.) can be calculated as follows:[53]

$$SP_s = 0.034(H_s - H_{br})\Pi(1/T_{amb} - 1/T_{sa}) \qquad (10.34)$$

where: H_{br} = height of stack breeching (inlet duct connection) above stack base (ft)
Π = barometric pressure (in. w.c.)
T_{amb} = ambient temperature (°R)
T_{sa} = average stack gas temperature (°R)

Illustration: The waste gas from a thermal incinerator has an outlet flow rate and temperature of 21,700 actual ft^3/min. and 550°F, respectively. The maximum wind speed in the vicinity is 42 mi/hr, while the stack exit and ambient temperatures are 450°F and 70°F, in turn. The barometric pressure is 1 atm. (29.92 in. Hg). The incinerator is near a 35-ft tall brick building, while the "projected width" of an adjacent building is 40 ft. For a stack to disperse the incinerator offgas, calculate the required: (1) exit velocity, (2) diameter, (3) height, and (4) draft.

Solution:

☞ <u>Exit velocity</u>: According to the above guideline, the velocity should be 1.5 times the wind speed, or:

$$u_c = 1.5 \text{ x } 42 \text{ mph x } 88 \text{ fpm/mph} = 5{,}540 \text{ ft/min.}$$

☞ <u>Stack diameter</u>: The exit volumetric flow rate is measured at the stack exit temperature, namely 450°F. However, the above flow rate was measured at 550°F, the incinerator outlet temperature. Correcting to the stack exit temperature, we obtain:

$$Q_c = 21{,}700 \text{ x } (450 + 460)/(550 + 460) = 19{,}600 \text{ actual ft}^3\text{/min.}$$

Substituting this value into equation 10.31:

$$D_s = 1.128(19{,}600/5{,}540)^{1/2} = 2.12 \text{ ft.}$$

☞ <u>Stack height</u>: As a first approximation, estimate the GEP stack height from equation 10.33, where the variables H_b and L are 35 ft and 40 ft, respectively:

$H_s = 35 + 1.5(40) = 95$ ft.

Clearly, this H_s is less than the GEP maximum height (213 ft), so it will be used in this example.

☞ Stack draft: All of the inputs needed to compute the stack draft via equation 10.34 are known except the stack breeching height, H_{br}. However, a minimum of 5 ft is recommended for this parameter.[54] This value will be used in this calculation. Also, the average stack temperature is:

$$T_{as} = (450 + 550)/2 + 460 = 960°R.$$

Finally, the barometric pressure expressed in inches of water is:

$$\Pi = 29.92 \text{ in. Hg} \times 13.6 \text{ in. water/in. Hg} = 407 \text{ in. w.c.}$$

Upon substitution, we obtain:

$$SP_s = 0.034(118 - 5)(407)(1/[70 + 460] - 1/960) = 1.32 \text{ w.c.}$$

10.4 Estimating Total Capital Investment

This section presents the information needed for estimating the total capital investment (TCI) for hoods, ductwork, and stacks. The TCI includes the equipment cost (EC) for the hood, ductwork, or stack; taxes; freight charges; instrumentation (if applicable); and direct and installation costs. All costs are presented in **second quarter 1993 dollars**, and are of "study" estimate accuracy (**± 30 percent**). Moreover, the costs are for new facility installations; *no retrofit costs are included.*

The equipment costs are presented in Section 10.4.1, while the installation costs are shown in Section 10.4.2. In each of these sections, the three categories of equipment are covered in separate subsections.

10.4.1 Equipment Costs

Several vendors provided costs (prices) for each of the three equipment categories. Their responses reflected a range of sizes, designs, and materials of construction. These prices have been correlated against some easy-to-determine design (sizing) parameter via least-squares regression analysis. Each of these correlations pertains to a certain type of equipment (e.g., circular canopy hoods) within a specified size range of the parameter in question (e.g., 2 to 200 ft^2 inlet area). For that reason, *a cost correlation should not be extrapolated outside the* *parameter range specified.*

Some of the prices the vendors provided pertain to stock ("off-the-shelf") items, while other costs are for custom-fabricated equipment. Vendors tend to specialize in either stock or custom items. Most hoods and stacks are custom-made, either fabricated in the vendor's factory or erected on-site. Conversely, ductwork components usually are stock items, though larger pieces have to be custom-made. (Of course, there are exceptions to this.) Finally, all prices given in the following section are "F.O.B. (free-on-board) vendor," meaning that they include neither freight nor taxes.

10.4.1.1 Hood Costs

In all, four vendors provided prices for hoods.[55] These prices covered the following types of hoods:

- ☞ Canopy—circular
- ☞ Canopy—rectangular
- ☞ Push-pull
- ☞ Side-draft
- ☞ Back-draft (slotted)

Descriptions and design procedures for these hoods are given in Sections 10.2.1 and 10.3.2, respectively. As explained in Section 10.3.2, hood costs have been found to correlate well with the hood inlet or face area (A_f, ft^2). Furthermore, the functional form that best fits the cost-face area correlation (equation) is the "power function", or:

$$C_h = aA_f^b \qquad (10.35)$$

where: C_h = hood cost (\$)
a,b = equation regression parameters

The values of the equation parameters vary according to hood type and material of construction. These parameters are shown in Table 10.5.

Illustration: What would be the cost of the electroplating tank canopy hood sized for the illustration in Section 10.3.2.? Assume that the hood is fabricated of FRP.

Solution: Recall that the face area (A_f) calculated for that hood was 98.5 ft^2. Because this is a circular canopy hood, the equation parameters from Table 10.5 are: a = 123 and b = 0.575. (Note that

Table 10.5 Parameters for Hood Cost Equation§

Type of Hood	Fabrication Material	Equation Parameter		Equation Range (A_f, ft^2)
		a	b	
Canopy-circular	FRP†	123	0.575	2-200
Canopy-rectangular	FRP	294	0.505	2-200
Push-pull	FRP	595	0.318	2-200
Side-draft	FRP	476	0.332	2-200
Backdraft (slotted)	PVC‡,•	303	1.43	0.6-2.0§§
Backdraft (slotted)	PVC††	789	0.503	1.1-2.1
Backdraft (slotted)	PP‡‡	645	0.714	1.1-2.1
Backdraft (slotted)	FRP	928	0.516	1.1-2.1
Backdraft (slotted)	Galvanized Steel	688	0.687	0.5-1.3

§ Based on data received from hood vendors. (See Reference 52.)

† Fiberglass-reinforced plastic.

‡ Polyvinyl chloride.

• Each hood is equipped with two rows of slots, but no dampers.

§§ For each slotted hood, "equation range" denotes the range in the area of the slot openings, which is much less than the total hood face area.

†† Each hood is equipped with manual slot dampers and four rows of slots.

‡‡ Polypropylene.

this hood area falls within the equation range of 2 to 200 ft^2.) Substituting these parameters into equation 10.35, we obtain:

$$C_h = 123(98.5)^{0.575} = \$1,720.$$

10.4.1.2 Ductwork Costs

Several vendors provided ductwork prices, also for a range of sizes, materials, and designs.[56] These prices covered the following equipment items:

☞ Straight ductwork:
- ♦ Circular
  - ♠ Steel sheet (galvanized carbon, w/ & w/o insulation; 304 stainless;)
  - ♠ Steel plate (coated carbon; 304 stainless)
  - ♠ Plastic (FRP; PVC)
- ♦ Square
  - ♠ Steel (aluminized carbon; w/ & w/o insulation)

☞ Elbows (90°):
- ♦ Steel (galvanized carbon, w/ & w/o insulation; 304 stainless)
- ♦ Plastic (FRP; PVC)

☞ Dampers:
- ♦ Butterfly
  - ♠ Steel (galvanized carbon, w/ & w/o insulation)
  - ♠ Plastic (FRP; PVC, w/ & w/o actuators)
- ♦ Louvered
  - ♠ Steel (aluminized carbon w/ & w/o actuators)
- ♦ Blast gate
  - ♠ Steel (carbon)
  - ♠ PVC

These prices were regressed against the diameter of the equipment item (straight duct, elbow, or damper). The regression correlations were of three forms: power function (primarily), exponential, and linear. Equation 10.35 depicts the power function, while the other forms are:

Exponential: $C_i = ae^{bD}$ (10.36)

Linear: $C_i = a + bD$ (10.37)

where: C_i = cost of equipment item in question
a,b = regression parameters

The regression parameters are listed in Tables 10.6 to 10.8, along with the size applicability ranges for the respective correlations. (**Note:** *The correlations should not be extrapolated outside these ranges.*) The following paragraphs contain additional information about the price data and the correlations:

¶ **Straight duct:** As indicated above, vendors provided prices for steel plate, steel sheet (spiral-wound and longitudinal seam), and plastic straight duct. The major difference between the two steel duct types lies in the wall thickness. Steel plate duct typically has wall thicknesses of from 3/16 in. to 1/2 in., while steel sheet duct wall thicknesses usually range from 28 gauge to 10 gauge. As Table 10.4 shows, this range corresponds to thicknesses of 0.0149 in. to 0.1406 in., respectively, although the exact thicknesses will vary with the type of steel used (e.g., carbon vs. stainless). Also, as discussed in Section 10.3.3.2, each duct diameter can be fabricated with a range of wall thicknesses.

Most of the steel duct vendors supplied prices for a minimum and a maximum wall thickness for a given diameter. However, to simplify matters for cost estimators, these "low" and "high" prices first were averaged, and then the average prices were regressed against the diameters. This averaging was necessary, because those making study cost estimates usually do not have enough information available to predict duct wall thicknesses.

Prices for both circular and square insulated steel sheet duct were among the data received. The insulated circular steel duct is "double-wall, spiral-wound" in construction, wherein the insulation is installed between the inner and outer walls. Costs were provided for both 1-in. and 3-in. fiberglass insulation thicknesses. For the square duct, prices were given for a 4-in. thickness of mineral wool insulation applied to the outer surface of the duct. The correlation parameters in Table 10.6 reflect these specifications.

Prices for both carbon steel (galvanized, painted, or aluminized) and 304 stainless steel duct were received. The carbon steel duct is used in situations where "mild" steel is suitable, while the stainless steel duct is required whenever the gas stream contains high concentrations of corrosive substances.

Vendors gave prices for plastic (FRP and PVC) duct also (Table 10.8). However, for a given diameter this duct is fabricated in a

Table 10.6 Parameters for Straight Steel Ductwork Cost Equations[§]

Duct Type	Material	Insulation Thickness (in.)	Equation Type	Equation Parameter		Equation Range (D, in.)
				a	b	
Circular-spiral[†]	Sheet-galv CS[‡]	None	Power function	0.322	1.21	3 - 84
Circular-spiral	Sheet-304 SS[•]	None	Power function	1.56	1.00	3 - 84
Circular-spiral	Sheet-galv CS	1	Power function	1.55	0.936	3 - 82
Circular-spiral	Sheet-galv CS	3	Power function	2.56	0.937	3 - 82
Circular-longitudinal[§§]	Sheet-galv CS	None	Power function	2.03	0.784	6 - 84
Circular-longitudinal	Sheet-304 SS	None	Power function	2.98	0.930	6 - 84
Circular-longitudinal	Plate-coat CS[††]	None	Power function	2.49	1.15	6 - 84
Circular-longitudinal	Plate-304 SS[‡‡]	None	Power function	6.29	1.23	6 - 84
Square	Sheet-alum CS[••]	None	Linear	0.254	2.21	18 - 48
Square	Sheet-alum CS	4	Linear	21.1	5.81	18 - 48

[§] Based on data from ductwork vendors. (Reference 53.)

[†] Spiral-wound and welded circular duct.

[‡] Galvanized carbon steel sheet.

[•] 304 stainless steel sheet.

[§§] Circular duct welded along the longitudinal seam.

[††] Carbon steel plate with one coat of "shop paint".

[‡‡] 304 stainless steel plate.

[••] Aluminized carbon steel sheet.

Table 10.7 Parameters for Steel Elbows and Dampers Cost Equations§

Ductwork Item	Material	Equation Type	Equation Parameter		Equation Range (D, in.)
			a	b	
Elbows†	Galv CS‡	Exponential	30.4	0.0594	6 - 84
Elbows	304 SS•	Exponential	74.2	0.0668	6 - 60
Elbows-insulated§§	Galv CS	Exponential	53.4	0.0633	3 - 78
Dampers-butterfly††	Galv CS	Exponential	23.0	0.0567	4 - 40
Dampers-butterfly /insulated‡‡	Galv CS	Exponential	45.5	0.0597	4 - 40
Dampers-louvered••	Alum CS§§§	Power function	78.4	0.860	18 - 48
Dampers-louvered w/actuators†††	Alum CS	Power function	208.	0.791	18 - 48
Dampers-blast gates	Carbon steel	Power function	17.2	0.825	3 - 18

§ Based on data received from ductwork vendors. (See Reference 53.)

† Single-wall "gored" 90° elbows, uninsulated.

‡ Galvanized carbon steel sheet.

• 304 stainless steel sheet.

§§ Double-wall "gored" 90° elbows with 1-inch fiberglass insulation.

†† Single-wall "opposed blade" type manual butterfly dampers.

‡‡ Double-wall "opposed blade" butterfly dampers with 1-inch fiberglass insulation.

•• Louvered dampers with 95-98% sealing.

§§§ "Aluminized" carbon steel sheet.

††† Louvered dampers with electric actuators (automatic controls).

Table 10.8 Parameters for Plastic Ductwork Cost Equations§

Ductwork Item	Material	Equation Type	Equation Parameter		Equation Range (D, in.)
			a	b	
Straight duct	PVC†	Power function	0.547	1.37	6 - 48
Straight duct	FRP‡	Exponential	11.8	0.0542	4 - 60
Elbows-90°	PVC	Power function	3.02	1.49	6 - 48
Elbows-90°	FRP	Exponential	34.9	0.0841	4 - 36
Dampers-butterfly	PVC	Power function	10.6	1.25	4 - 48
Dampers-butterfly	FRP	Power function	35.9	0.708	4 - 36
Dampers-butterfly w/actuators•	PVC	Exponential	299.	0.0439	4 - 48
Dampers-blast gate	PVC	Power function	8.14	1.10	4 - 48

§ Based on data received from ductwork vendors. (See Reference 53.)

† Polyvinyl chloride.

‡ Fiberglass-reinforced plastic.

• Butterfly dampers with pneumatic actuators (automatic controls). All other dampers listed in this table are manually-controlled.

single wall thickness, which varies from approximately 1/8 in. to 1/4 in. Consequently, the estimator is not required to select a wall thickness when costing plastic duct.

¶ **Elbows:** Prices for steel sheet and plastic 90° elbows were also submitted. The steel sheet elbows were "gored" (sectioned) elbows fabricated from five pieces of sheet metal welded together. Like the straight duct, the steel elbows were priced at two wall thicknesses: "minimum" and "maximum". These prices were averaged before being regressed against the elbow diameter. Prices were also given for both galvanized carbon steel elbows (with and without 1-in. fiberglass insulation) and 304 stainless steel elbows. Correlation parameters for steel elbows are listed in Table 10.7.

Costs for both PVC and FRP 90° elbows were also given. The PVC ells were fabricated from three sections ("three-piece miter"), while the FRP elbows were one-piece molded fittings. As with the plastic straight duct, each elbow of a given diameter was fabricated in a single wall thickness. Table 10.8 contains correlation parameters for plastic elbows.

¶ **Dampers:** Prices were obtained for three types of dampers: butterfly, louvered, and blast gates. The galvanized carbon steel butterfly dampers were priced with and without 1-in. fiberglass insulation, while prices for the aluminized carbon steel louvered dampers were based on either manual or automatic control (via electric actuators). Similarly, the PVC butterfly dampers were manual or equipped with pneumatic actuators. Both the carbon steel and the PVC blast gates were manual. Correlation parameters for the steel and plastic dampers are shown in Tables 10.7 and 10.8, in turn.

Illustration: A fabric filter handling 16,500 ft^3/min of 200°F. waste gas laden with noncorrosive cocoa dust is located 95 ft across from and 20 ft above, the emission source (a drying oven). Straight duct with four 90° elbows (all fabricated from spiral-wound, galvanized carbon steel sheet) and a butterfly damper (also galvanized CS) will be required to convey the gas from the source to the control device. Assume that the ductwork is insulated to prevent condensation. Estimate the cost of these items.

Solution: First, determine the diameter of the straight duct, elbows, and damper. From Table 10.3, the minimum transport velocity (u_t) for cocoa dust is 3,000 ft/min. Substituting this value and the gas volumetric flow rate into equation 10.27, we obtain:

$$D_d = 1.128(16,500/3,000)^{1/2} = 2.65 \text{ ft} = 31.7 \text{ in.}$$

Next, obtain the costs of the ductwork items as follows:

☞ Straight ductwork: From Table 10.6, select the equation parameters for galvanized circular spiral-wound duct (1-in. insulation) and substitute them and the diameter into the appropriate equation type (power function, equation 10.35).

$$\text{Straight duct cost (\$/ft)} = 1.55(31.7)^{0.936} = \$39.4/\text{ft}.$$

However, a total of 115 ft (95 + 20) of duct is required, so:

$$\text{Straight duct cost} = \$39.4/\text{ft} \times 115 \text{ ft} = \$4,531.$$

☞ Elbows: The Table 10.7 correlation parameters for galvanized carbon steel, insulated elbows are 53.4 (a) and 0.0633 (b). However, the regression correlation form is exponential (equation 10.36). Thus:

$$\text{Elbow cost (\$)} = 53.4e^{0.0633(31.7)} = \$397 \text{ ea.}$$

For four elbows, the cost is: \$397 x 4 = \$1,588.

☞ Damper: Also from Table 10.7, select the correlation parameters for galvanized carbon steel "dampers-butterfly/insulated" and substitute into equation 10.36:

$$\text{Damper cost (\$)} = 45.5e^{0.0597(31.7)} = \$302.$$

After summing the above three costs, we obtain:

$$\text{Total ductwork cost} = \$6,421 \approx \$6,420.$$

10.4.1.3 Stack Costs

Prices for steel and PVC short stacks were obtained from four vendors.[57] The steel stack costs were for those fabricated from carbon and 304 stainless steels, both plate and sheet metal. As with ductwork, the difference between steel sheet and plate lies in the thickness. For these stacks, the sheet steel thickness ranged from 18 to 16 gauge (0.05 to 0.06 in., approximately). Steel plate thicknesses were considerably higher: 0.25 to 0.75 in, a fact that makes them more resistant to wind and other loadings than stacks fabricated of steel sheet. This is especially true for taller stacks. The major drawback is that plate steel stacks are more costly than those fabricated from steel sheet.

Another feature that increases costs is insulation. As the correlation parameters show (Table 10.9), insulated stacks cost as much as three times more per foot than uninsulated. With or without insulation, a typical short (15-ft) steel stack consists of the following components:[58]

- ☞ Longitudinal seam duct (12-ft section)
- ☞ Reducer fitting (3-ft)
- ☞ Drip pan
- ☞ Support plate (1/4-in, welded to stack)
- ☞ Rectangular tap (for connecting to fan discharge)
- ☞ Ring (for attaching guy wires)

Taller stacks may require additional components, such as ladders and platforms, guy wires or other supports, and aircraft warning lights. (See Section 10.2.3.)

Table 10.9 lists the parameters and applicable ranges of the stack cost correlations. The correlations cover short PVC stacks, and taller stacks fabricated from plate steel (carbon and 304 stainless types) and sheet steel (insulated and uninsulated). Except for three double-wall sheet steel designs, these stacks are of *single-wall* construction.

Note that all of the correlations are power functions. Also note that the equations apply to various ranges of stack height. In all but one of these equations the cost is expressed in *\$/ft of stack height*. The exception is the cost equation for insulated carbon steel sheet stacks of heights ranging from 30 to 75 feet. In this equation the cost is expressed in *\$*.

This last cost equation is different in another respect. The other six equations in Table 10.9 correlate stack cost (\$/ft) with stack diameter (D_s, in.). However, this seventh equation correlates stack cost with *stack surface area* (S_s, ft^2), a variable that incorporates both the stack diameter and the stack height (H_s). The surface area is calculated via the following equation:

$$S_s = (\pi/12) D_s H_s \tag{10.38}$$

where: 1/12 = stack diameter (D_s) conversion factor

Illustration: Estimate the cost of the stack sized in the Section 10.3.4.3 illustration.

Solution: Recall that the stack dimensions were: H_s = 95 ft and D_s = 2.12 ft = 25.4 in. Both dimensions fall within the ranges of the cost correlations for steel *plate* stacks. Because the previous illustration did not indicate whether the waste gas was corrosive, we will estimate the prices for both carbon steel and 304 stainless

Table 10.9 Parameters for Stack Cost Equations§

Material	Equation Parameter†		Equation Range	
	a	b	D_s (in)‡	H_s (ft)•
PVC§§	0.393	1.61	12 - 36	$\leq$ 10
Plate-coated CS††	3.74	1.16	6 - 84	20 - 100
Plate-304 SS‡‡	12.0	1.20	6 - 84	20 - 100
Sheet-galv CS••	2.41	1.15	8 - 36	$\leq$ 75
Sheet-304 SS§§§	4.90	1.18	8 - 36	$\leq$ 75
Sheet-insul CS/DW†††	143.	0.402	18 - 48	$\leq$ 15
Sheet-uninsul CS/DW‡‡‡	10.0	1.03	18 - 48	$\leq$ 15
Sheet-insul CS/DW•••	142.	0.794	24 - 48	30 - 75

§ Based on data received from stack vendors. (See Reference 54.)

† All cost equations are power functions. (See equation 10.35.) Except where noted, costs are expressed in terms of $/ft of stack height.

‡ Stack diameter range to which each equation applies.

• Stack height range to which each equation applies.

§§ Polyvinyl chloride.

†† Carbon steel plate with one coat of "shop paint".

‡‡ 304 stainless steel plate.

•• Galvanized carbon steel sheet.

§§§ 304 stainless steel sheet.

††† Aluminized carbon steel sheet covered with 4 inches of fiberglass insulation (double-wall construction).

‡‡‡ Uninsulated aluminized carbon steel sheet (double-wall construction).

••• Costs for these stacks are expressed in $, and are correlated with the stack surface area (S_s, ft^2).

steel plate stacks.

Upon substituting the equation parameters and stack dimensions into equation 10.35, we obtain:

$$\text{Price (carbon steel)} = 3.74(25.4)^{1.16}\ (\$/\text{ft}) \times 95\ \text{ft} = \$15{,}100.$$

$$\text{Price (304 stainless)} = 12.0(25.4)^{1.20}\ (\$/\text{ft}) \times 95\ \text{ft} = \$55{,}300.$$

Notice that the price of the stainless steel stack is nearly four times that of the carbon steel stack. In view of this difference, the estimator needs to obtain more information on the waste gas stream properties, so that he/she can select the most suitable stack fabrication material. Clearly, it would be a poor use of funds to install a stainless steel stack where one is not needed.

10.4.2 Taxes, Freight, and Instrumentation Costs

Taxes (sales, etc.) and freight charges apply to hoods, ductwork, and stacks, as they do to the control devices that these auxiliaries support. As discussed in Chapter 2, these costs vary, respectively, according to the location of the ventilation system and the site's distance from the vendor. Typical values are 3% (taxes) and 5% (freight) of the total equipment cost.

Unlike the control devices, ventilation systems generally are not instrumented. The exception would be an electric or pneumatic actuator for a butterfly or louvered damper. In such a case, however, the cost of the instrument (actuator and auxiliaries) would be included in the damper price. Thus, no supplementary instrumentation cost is included.

10.4.3 Purchased Equipment Cost

With ventilation systems, the purchased equipment cost (PEC_t) is the sum of the equipment, taxes, and freight costs. Incorporating the typical values listed in Section 10.4.2, we obtain:

$$PEC_t = EC_t + 0.03EC_t + 0.05EC_t$$

$$= 1.08(EC_t) \qquad (10.39)$$

where: EC_t = total cost of hood(s), ductwork, and stack(s)

10.4.4 Installation Costs

When making a cost estimate for an air pollution control system according to the procedure in this manual, the estimator first determines the cost of the control device, then estimates the costs of such auxiliaries as the hood, ductwork, stack, fan and motor, and other items. To these items he/she adds the costs of instrumentation, taxes, and freight, to obtain the PEC. Finally, the estimator multiplies the PEC by the installation factor appropriate to the control device (e.g., 2.20 for gas absorbers) to obtain the total capital investment. In these cases, the installation factor incorporates all direct and indirect costs needed to install and start up the control system equipment, including, of course, the hood, ductwork, and stack. (See Chapters 3 to 9 for more information about these factors.)

For this reason, it usually is unnecessary to estimate the installation cost of the ventilation system separately. However, there may be occasions where a hood, a stack, or ductwork has to be installed alone, either as replacement equipment or to augment the existing ventilation system. In those instances, the estimator may want to estimate the cost of installing this item.

As might be imagined, these installation costs vary considerably, according to geographic location, size and layout of the facility, equipment design, and sundry other variables. Nonetheless, some of the vendors (and a peer reviewer[59]) provided factors for hoods and ductwork, which, when multiplied by their respective purchased equipment costs, will yield *approximate* installation costs. These are:

☞ Hoods: 50 to 100%

☞ Ductwork: 25 to 50%

If one or both of the latter factors is used, the total capital investment (TCI) of the hood and/or ductwork would be:

$$TCI = (1 + IF_{h/d}) \times PEC_{h/d} \quad (10.40)$$

where: $IF_{h/d}$ = installation factor for hood(h)/ductwork(d)
$PEC_{h/d}$ = purchased equipment cost of hood(h)/ductwork(d)

10.5 Estimating Total Annual Cost

10.5.1 Direct Annual Costs

Ventilation systems incur few, if any, direct annual costs, as they function to support control devices. There are no costs for operating or supervisory labor, operating materials, or waste treatment/disposal allocated to ventilation systems. Maintenance costs would also be minimal, except for such minor expenses as painting, insulation repair, or calibration of automatic damper controls. The only utilities cost would be the incremental electricity needed for the waste gas stream to overcome the static pressure loss in the hood, ducting, and stack.[13] The incremental electricity cost (C_e, \$/yr) can be calculated as follows:

$$C_e = (1.175 \times 10^{-4}) p_e Q F_d \theta / \epsilon \qquad (10.41)$$

where: p_e = electricity price (\$/kwh)
Q = waste gas flow rate (actual ft^3/min)
F = static pressure drop through ventilation system (in. w.c.)
θ = operating factor (hr/yr)
ϵ = combined fan-motor efficiency

Illustration: In the cosmetic factory ventilation system illustration above (Section 10.3.3.3), what would be the cost of the electricity consumed by the fan needed to convey the gas through the ductwork? Assume an electricity price of \$0.075/kwh, a combined fan-motor efficiency of 0.6, and an 8,000-hr/yr operating factor.

Solution: Recall that the pressure drop and gas flow rate for this illustration were 0.313 in. w.c. and 15,000 actual ft^3/min, respectively. Upon substituting these values and the other parameters into equation 10.40, we obtain:

$$C_e = (1.175 \times 10^{-4})(0.075)(15{,}000)(0.313)(8{,}000)/0.6$$

$$= \$552/yr.$$

[13] Technically, this direct annual cost should be allocated to the ventilation system fan, not to the hood, ductwork, and stack. The fan power cost equation will be included in the *Manual* chapter on fans. However, as the fans chapter has yet to be written, this equation has been provided as a temporary convenience to *Manual* users.

10.5.2 Indirect Annual Costs

The indirect annual costs for ventilation systems include property taxes, insurance, general and administrative (G&A), and capital recovery costs. (Overhead—a fifth indirect annual cost—is not considered, because it is factored from the sum of the operating, supervisory, maintenance labor and maintenance materials costs, which is negligible.) When a ventilation system is part of a control system, these costs are included in the control system indirect annual cost. However, if the ventilation equipment has been sized and costed separately, these costs can be computed from the total capital investment (TCI) via standard factors, as follows:

Indirect Annual Cost	Computation Equation
Property taxes	0.01 x TCI
Insurance	0.01 x TCI
General and Administrative	0.02 x TCI
Capital recovery	CRF x TCI

The "CRF" term in the capital recovery equation is the *capital recovery factor*, which is a function of the economic life of the ventilation system and the interest rate charged to the total capital investment. (See Section 2.3 of this manual for more discussion of the CRF and the formula used for computing it.)

For a ventilation system, the economic life varies from at least 5 to 10 years to 15 to 20 years or more.[60,61] In general, the ventilation equipment should last as long as the control system it supports. As discussed in Section 2.3, the interest rate to use in the CRF computation should be a "pre-tax, marginal (real) rate of return" that is appropriate for the investor. However, for those cost analyses related to governmental regulations, an appropriate "social" interest (discount) rate should be used. For these kinds of analyses, the Office of Management and Budget (OMB) directs that a real annual interest rate of 7% be used.[62] (This replaces the 10% rate OMB previously had mandated.)

10.5.3 Total Annual Cost

The total annual cost (TAC) is calculated by adding the direct (DC) and indirect (IC) annual costs:

$$TAC = DC + IC \tag{10.42}$$

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Abstract (Item #16) for EPA Form 2220-1

This is the third supplement to the *OAQPS Control Cost Manual* (Fourth Edition). The supplement consists of a new *Manual* chapter, Chapter 10 ("Hoods, Ductwork, and Stacks"). Like the other chapters in the *Manual*, Chapter 10 is self-contained. It discusses: (1) the types and applications of hoods used to support add-on air pollution control devices; (2) the theory underlying their operation and design; (3) basic sizing procedures; and (4) procedures and current (1993) data for estimating study-level ($\pm$ 30%-accurate) capital and annual costs. In particular, the chapter contains equipment costs for canopy, push-pull, side-draft, and backdraft hoods; straight ductwork (circular and square); 90° elbows; butterfly, louvered, and blast gate dampers; and short (up to 100-foot) stacks. In addition, the prices of each type of equipment reflect at least two kinds of fabrication materials, such as carbon and 304 stainless steel (plate and sheet types), FRP (fiberglass-reinforced plastic), and PVC (polyvinyl chloride). These prices have been correlated with appropriate sizing parameters (e.g., duct diameter). Finally, Chapter 10 includes several example problems that illustrate the various sizing and costing procedures; a table of contents; and a list of references.