## CHAPTER 34

# CHIMNEY, VENT, AND FIREPLACE SYSTEMS

A PROPERLY designed chimney or vent system provides and controls draft to convey flue gas from an appliance to the outdoors. This chapter describes the design of chimneys and vent systems that discharge flue gas from appliances and fireplace systems.

**Sustainability.** Good chimney and vent design is not only a safety issue, but also can enhance a building's sustainability. This chapter explains how to design vent systems to optimize and minimize the materials used to construct fuel-burning appliance vents and chimneys for low cost and long reliability, reducing the need for vent or chimney replacement, thus saving natural resources. Also, systems designed to bring outdoor air directly into the appliance space for combustion and vent gas dilution, instead of relying on air infiltration into the building, reduce heat load and conserve fuel.

## TERMINOLOGY

In this chapter, **appliance** refers to any furnace, boiler, or incinerator (including the burner). Unless the context indicates otherwise, the term **chimney** includes specialized vent products such as masonry, metal, and factory-built chimneys; single-wall metal pipe; type B gas vents; special gas vents; or masonry chimney liner systems. (NFPA *Standard* 211). **Draft** is negative static pressure, measured relative to atmospheric pressure; thus, positive draft is negative static pressure. **Flue gas** is the mixture of gases discharged from the appliance and conveyed by the chimney or vent system.

Appliances can be grouped by draft conditions at the appliance flue gas outlet as follows (Stone 1971):

1. Those that require draft applied at the appliance flue gas outlet to induce air into the appliance
2. Those that operate without draft applied at the appliance flue gas outlet (e.g., a gas appliance with a draft hood in which the combustion process is isolated from chimney draft variations)
3. Those that produce positive pressure at the appliance flue gas outlet collar so that no chimney draft is needed; appliances that produce some positive outlet pressure but also need some chimney draft

In the first two configurations, hot flue gas buoyancy, induced-draft chimney fans, or a combination of both produces draft. The third configuration may not require chimney draft, but it should be considered in the design if a chimney is used. If the chimney system is undersized, draft inducers in the connector or chimney may supply draft needs. If the connector or chimney pressure requires control for proper operation, draft control devices must be used.

Vented gas-fired appliances have been grouped by draft and flue gas conditions as follows by installation codes in Canada (CSA B149.1) and in the United States (ANSI/NFPA 54/ANSI/AGA Z223.1):

1. Category I appliances operate with nonpositive vent static pressure and a vent gas temperature that avoids excessive condensate production in the vent.
2. Category II appliances operate with nonpositive vent static pressure and a vent gas temperature that may cause condensate production in the vent.
3. Category III appliances operate with positive vent static pressure and a vent gas temperature that avoids excessive condensate production in the vent.
4. Category IV appliances operate with positive vent static pressure and a vent gas temperature that may cause condensate production in the vent.

Category I venting systems are typically sized using venting tables for unobstructed vent systems, as listed in the installation codes; they are provided for fan-assisted appliances and natural-draft appliances as well as multiappliance system vent arrangements. Although these categories are intended for gas-fired appliances, they could apply to other appliances (e.g., oil- or coal-fired).

## DRAFT OPERATING PRINCIPLES

**Available draft $D_a$** is the draft supplied by the vent system, available at the appliance flue gas outlet. It can be shown as

$$D_a = D_t - \Delta p - D_p + D_b \tag{1}$$

where

$D_a$ = available draft, in. of water
$D_t$ = theoretical draft, in. of water
$\Delta p$ = flow losses, in. of water
$D_p$ = depressurization, in. of water
$D_b$ = boost (increase in static pressure by fan), in. of water

This equation can account for a nonneutral (nonzero) pressure difference between the space surrounding the appliance or fireplace and the atmosphere. If the surrounding space is at a lower pressure than the atmosphere (space depressurized), the pressure difference $D_p$ should also be subtracted from $D_t$ when calculating available draft $D_a$, and vice versa. This equation applies to all three appliance draft conditions at the vent system inlets; for example, in the second condition with zero draft requirement at the appliance outlet, available draft required is zero, so theoretical draft of the chimney equals the flow resistance, if no depressurization or boost is present.

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Operational consequences of various values of $D_a$ are described as follows:

- **Positive available draft (negative vent pressure); $D_a$ is positive.** Category I fan-assisted and draft hood-equipped appliances and category II appliances can operate satisfactorily when served by venting systems having positive available draft at the appliance flue gas outlet, if the positive draft is sufficient to convey all flue gas from the appliance flue gas outlet to the outdoors and if the positive available draft does not aspirate excessive excess air to cause flame lifting or other detriments to combustion performance. Category III and IV appliances can operate satisfactorily when served by venting systems having positive available draft if the appliance flue gas discharge pressure plus positive draft is sufficient to convey all flue gas from the appliance flue gas outlet to the outdoors. If positive available draft and/or appliance flue gas discharge pressure is insufficient to convey all flue gas from the appliance flue gas outlet outdoors, incomplete combustion, flame rollout, and/or flue gas spillage can occur at the appliance.
- **Zero available draft (neutral vent pressure); $D_a = 0$.** Category I fan-assisted and draft hood-equipped appliances and Category II appliances can operate satisfactorily when served by venting systems having zero available draft (neutral draft) at the appliance flue gas outlet, if the venting system creates sufficient theoretical draft to convey all flue gas from the appliance flue gas outlet to the outdoors. If the venting system creates insufficient theoretical draft to convey all flue gas from the appliance flue gas outlet to the outdoors, incomplete combustion, flame rollout, and/or flue gas spillage can occur at the appliance. Category III and IV appliances can operate satisfactorily when served by venting systems having zero available draft, if appliance flue gas discharge pressure is great enough to overcome the vent flow pressure-drop loss.
- **Negative available draft (positive vent pressure); $D_a$ is negative.** Category I and II appliances cannot operate satisfactorily when served by venting systems having negative available draft. Category III and IV appliances can operate satisfactorily when served by venting systems having negative draft, if appliance flue gas discharge pressure plus the theoretical draft created by the venting system is sufficient to overcome the vent negative draft at the vent inlet and the vent flow pressure-drop loss.
- If chimney height and flue gas temperatures provide **surplus available draft $D_a$** (excessive excess air), draft control is required.

**Theoretical draft $D_t$** is the natural draft produced by the buoyancy of hot gases in the chimney relative to cooler gases in the surrounding atmosphere. It depends on chimney height, local barometric pressure, and the **mean chimney flue gas temperature difference $\Delta t_m$**, which is the difference in temperature between the flue gas and atmospheric gases. Therefore, cooling by heat transfer through the chimney wall is a key variable in chimney design. Precise evaluation of theoretical draft is not necessary for most design calculations because of the availability of chimney design charts, computer programs, capacity tables in the references, building codes, and vent and appliance manufacturers' data sheets.

Chimney temperatures and acceptable combustible material temperatures must be known in order to determine safe clearances between the chimney and combustible materials. Safe clearances for some chimney systems, such as type B gas vents, are determined by standard tests and/or specified in building codes.

**Losses from Flow $\Delta p$** represent the friction losses imposed on the flue gas by flow resistance through the chimney.

**Depressurization $D_p$** is negative pressure in the space surrounding the appliance with respect to the atmosphere into which the chimney discharges. $D_p$ can be caused by other appliances and fans operating in the building that remove or add air to the space surrounding the appliance, by building stack effect, by outdoor atmospheric effects such as wind impacting the chimney exit or the side of the building facing or leeward to the wind, and other building phenomena.

**Boost $D_b$** is the pressure boost from a mechanical-draft fan. A chimney with an forced-draft fan at the inlet of the chimney would have positive boost (increased static pressure). A chimney with an induced-draft fan at the outlet of the chimney would have negative boost (decreased static pressure).

The following sections cover the basis of chimney design for average steady-state category I and III appliance operating conditions. For other appliance operating conditions, a rigorous cyclic evaluation of the flue gas and material surface temperatures in the chimney vent system can be obtained using the VENT-II computer program (Rutz and Paul 1991). For oil-fired appliances, chimney flue gas and material surface temperature evaluations can be obtained using the OHVAP computer program (Krajewski 1996).

## CHIMNEY FUNCTIONS

The proper chimney can be selected by evaluating factors such as draft, configuration, size, and operating conditions of the appliance; construction of surroundings; appliance usage classification; residential, low, medium, or high heat (NFPA *Standard* 211); and building height. The chimney designer should know the applicable codes and standards to ensure acceptable construction.

In addition to chimney draft, the following factors must be considered for safe and reliable operation: adequate air supply for combustion; building depressurization effects; draft control devices; chimney materials (corrosion and temperature resistance); flue gas temperatures, composition, and dew point; wind eddy zones; and particulate dispersion. Chimney materials must resist oxidation and condensation at both high and low fire levels at all design temperatures.

### Start-Up

The equations and design charts in this chapter may be used to determine vent or chimney size for average category I and III vent system operating conditions based on steady-state operating conditions. The equations and charts do not consider modulation, cycling, or time to achieve equilibrium flow conditions from a cold start. Whereas mechanical draft systems can start gas flow, gravity systems rely on the buoyancy of hot flue gases as the sole force to displace the cold air in the chimney. Priming follows Newton's laws of motion. The time to fill a system with hot flue gases, displace the cold air, and start flow is reasonably predictable and is usually a minute or less; however, unfavorable thermal differentials, building/chimney interaction, mechanical equipment (e.g., exhaust fans), or wind forces that oppose the normal flow of vent gases can overwhelm the buoyancy force. Then, rapid priming cannot be obtained solely from correct system design. The VENT-II computer program contains detailed analysis of gas vent and chimney priming and other cold-start considerations and allows for appliance cycling and pressure differentials that affect performance (Rutz and Paul 1991). A copy of the solution methodology (Rutz 1991) for VENT-II, including equations, may be ordered from ASHRAE Customer Service.

### Air Intakes

All rooms or spaces containing fuel-burning appliances must have a constant supply of combustion air from outdoors (either directly or indirectly) at adequate static pressure to ensure proper combustion. In addition, air (either directly or indirectly) is required to replace the air entering chimney systems through draft hoods and barometric draft regulators and to ventilate closely confined boiler and furnace rooms.

The U.S. *National Fuel Gas Code* (ANSI/NFPA 54/ANSI/AGA Z223.1) and the Canadian *Natural Gas and Propane Installation Code* (CSA *Standard* B149.1), along with appliance manufacturers, provide requirements for air openings. Any design must consider flow resistance of the air supply, including register-louver resistance, air duct resistance, and air inlet terminations. Compliance with these codes or the appliance manufacturers' instructions accounts for the air supply flow resistance.

### Vent Size

Small residential and commercial natural-draft gas appliances need vent diameters of 3 to 12 in. U.S. and Canadian codes recommend sizes or input capacities for most acceptable gas appliance venting materials. These sizes also apply to gas appliances with integral automatic vent dampers, as well as to appliances with field-installed automatic vent dampers. Field-installed automatic vent dampers should be listed for use with a specific appliance by a recognized testing agency and installed by qualified installers.

### Draft Control

Pressure, temperature, and other draft controls have replaced draft hoods in many residential furnaces and boilers to attain higher steady-state and seasonal efficiencies. Appliances that use pulse combustion or forced- or induced-draft fans, as well as those designed for sealed combustion or direct venting, do not have draft hoods but may require special venting and special vent terminals. If fan-assisted burners deliver fuel and air to the combustion chamber and also overcome the appliance flow resistance, draft hoods or other control devices may be installed, depending on the design of the appliance. Vent category II, III, and IV and some category I appliances do not use draft hoods; in such cases, the listed appliance manufacturer's vent system design requirements should be followed. The section on Vent and Chimney Accessories has information on draft hoods, barometric regulators, draft fans, and other draft control devices.

Frequently, a chimney must produce excess flow or draft. For example, dangerously high flue gas outlet temperatures from an incinerator (or normal noncondensing appliance flue gas temperatures) may be reduced by diluting the flue gas with air in the chimney (or PVC vent pipe) by applying excess draft.

### Pollution Control

Where control of pollutant emissions is impossible, the chimney should be tall enough to ensure dispersion over a wide area to prevent objectionable ground-level concentrations. The chimney can also serve as a passageway to carry flue gas to pollution control equipment. This passageway must meet the building code requirements for a chimney, even at the exit of pollution control equipment, because of possible exposure to heat and corrosion. A bypass chimney should also be provided to allow continued exhaust in the event of pollution control equipment failure, repair, or maintenance.

### Equipment Location

Chimney materials may allow installation of appliances at intermediate or all levels of a high-rise building without imposing weight penalties. Some gas vent systems allow individual apartment-by-apartment heating systems.

### Wind Effects

Wind and eddy currents affect discharge of gases from vents and chimneys. A vent or chimney must expel flue gas beyond the cavity or eddy zone surrounding a building to prevent reentry through openings and fresh air intakes. A chimney and its termination can stabilize the effects of wind on appliances and their equipment rooms. In many locations, the equipment room air supply is not at neutral pressure under all wind conditions. Locating the chimney outlet well into the undisturbed wind stream and away from the cavity and wake zones around a building both counteracts wind effects on the air supply pressure and prevents reentry through openings and contamination of fresh air intakes.

Chimney outlets below parapet or eaves level, nearly flush with the wall or roof surface, or in known regions of stagnant air may be subjected to downdrafts and are undesirable. Caps for downdraft and rain protection must be installed according to either their listings and the cap manufacturer's instructions or the applicable building code.

Wind effects can be minimized by locating the chimney terminal and the combustion air inlet terminal close together in the same pressure zone while taking care to minimize recirculation of combustion products into the combustion air inlet.

See Chapter 16 of the 2005 *ASHRAE Handbook—Fundamentals*, for more information on wind effects.

### Safety Factors

Safety factors allow for uncertainties of vent and chimney operation. For example, flue gas must not spill from a draft hood or barometric regulator, even when the chimney has very low available draft. The Table 2 design condition for natural gas vents (i.e., vent gas at 300°F rise and 5.3% $CO_2$ concentration) allows gas vents to operate with reasonable safety above or below the suggested temperature and $CO_2$ limits.

Safety factors may also be added to the system friction coefficient to account for a possible extra fitting, soot accumulation, and air supply resistance. The specific gravity of flue gas can vary depending on the fuel burned. Natural gas flue gas, for example, has a density as much as 5% less than air, whereas coke flue gas may be as much as 8% greater. However, these density changes are insignificant relative to other uncertainties, so no compensation is needed.

## STEADY-STATE CHIMNEY DESIGN EQUATIONS

Chimney design balances forces that produce flow against those that retard flow (e.g., friction). **Theoretical draft** is the pressure that produces flow in gravity or natural-draft chimneys. It is defined as the static pressure resulting from the difference in densities between a stagnant column of hot flue gas and an equal column of ambient air. In the design or balancing process, theoretical draft may not equal friction loss because the appliance is frequently built to operate with some specific pressure (positive or negative) at the appliance flue gas exit. This exit pressure is added to the **available draft**, which depends on chimney conditions, appliance operating characteristics, fuel, and type of draft control.

Flow losses caused by friction may be estimated by several formulas for flow in pipes or ducts, such as the equivalent length method or the loss coefficient (velocity head) method. Chapter 34 of the 2005 *ASHRAE Handbook—Fundamentals* covers computation of flow losses. This chapter emphasizes the loss coefficient method because fittings usually cause the greater portion of system pressure drop in chimney systems, and conservative loss coefficients (which are almost independent of piping size) provide an adequate basis for design.

Rutz and Paul (1991) developed a computer program entitled VENT-II: An Interactive Personal Computer Program for Design and Analysis of Venting Systems for One or Two Gas Appliances, which dynamically predicts cyclic flows, temperatures, and pressures in venting systems. Similarly, Krajewski (1996) developed a computer program entitled OHVAP: Oil-Heat Vent Analysis Program.

For large gravity chimneys, steady-state available draft $D_a$ may be calculated from Equation (2) or (3). Both equations use the equivalent length approach, as indicated by the symbol $L_e$ in the flow-loss term (ASHVE 1941). These equations permit consideration of the density difference between chimney gases and ambient air and compensation for shape factors. Mean flue gas temperature $T_m$ must be estimated separately. Starting with Equation (1), the following equations are based on zero depressurization at the appliance flue gas outlet and zero chimney draft boost.

For a cylindrical chimney,

$$D_a = 2.96HB\left(\frac{\rho_o}{T_o} - \frac{\rho_c}{T_m}\right) - \frac{0.000315w^2T_mfL_e}{1.3\times10^7 d_f^5 B\rho_c} \quad (2)$$

For a rectangular chimney,

$$D_a = 2.96HB\left(\frac{\rho_o}{T_o} - \frac{\rho_c}{T_m}\right) - \frac{0.000097w^2T_mfL_e(x+y)}{1.3\times10^7 (xy)^3 B\rho_c} \quad (3)$$

*where*

$H$ = height of vent or chimney system above grade or system inlet, ft
$B$ = existing or local barometric pressure, in. Hg
$\rho_o$ = density of ambient air at design temperature and local barometric pressure conditions
$\rho_c$ = density of chimney flue gas at average temperature and local barometric pressure, lb/ft$^3$
$T_o$ = ambient air design temperature, °R
$T_m$ = mean flue gas temperature at average conditions in system, °R
$w$ = mass flow of flue gas, lb/h
$f$ = Darcy friction factor
$L_e$ = total piping length $L$ plus equivalent length of elbows, ft
$d_f$ = inside diameter, ft
$x$ = length of one internal side of rectangular chimney cross section, ft
$y$ = length of other internal side of rectangular chimney cross section, ft

In these equations, the first term determines theoretical draft, based on applicable gas and ambient density. The second term defines draft loss based on the factors for flow in a circular or rectangular duct system. To use these equations, the mass flow $w$ for a variety of fuels and situations and the available draft needs of various types of appliances must be determined.

Equations (2) and (3) may be derived and expressed in a form that is more readily applied to the problems of chimney design, size, and capacity by considering the following factors, which are the steps used to solve the problems in the section on Chimney Capacity Calculation Examples.

1. Mass flow of combustion products in chimney
2. Chimney gas temperature and density
3. Theoretical draft
4. System pressure loss caused by flow
5. Available draft
6. Chimney gas velocity
7. System resistance coefficient
8. Final input-volume relationships

For application to system design, the chimney gas velocity step is eliminated; however, actual velocity can be found readily, if needed.

## 1. Mass Flow of Combustion Products in Chimneys and Vents

Mass flow in a chimney or venting system may differ from that in the appliance, depending on the type of draft control or number of appliances operating in a multiple-appliance system. Mass flow is preferred to volumetric flow because it remains constant in any continuous portion of the system, regardless of changes in temperature or pressure. For the chimney gases resulting from any combustion process, mass flow can be expressed as

$$w = IM \quad (4)$$

*where*

$w$ = mass flow rate, lb/h
$I$ = appliance heat input, Btu/h
$M$ = ratio of mass flow to heat input, lb of combustion products per 1000 Btu of fuel burned [$M$ depends on fuel composition and percentage excess air (or CO$_2$) in the chimney]

**Table 1 Mass Flow Equations for Common Fuels**

| Fuel | Ratio $M$ of Mass Flow to Input[a]<br>$M = \frac{\text{lb Total Combustion Products}^b}{\text{1000 Btu Fuel Input}}$ |
|---|---|
| Natural gas | $0.705\left(0.159 + \frac{10.72}{\%CO_2}\right)$ |
| LPG (propane, butane, or mixture) | $0.706\left(0.144 + \frac{12.61}{\%CO_2}\right)$ |
| No. 2 oil (light) | $0.72\left(0.12 + \frac{14.4}{\%CO_2}\right)$ |
| No. 6 oil (heavy) | $0.72\left(0.12 + \frac{15.8}{\%CO_2}\right)$ |
| Bituminous coal (soft) | $0.76\left(0.11 + \frac{18.2}{\%CO_2}\right)$ |
| Type 0 waste or wood | $0.69\left(0.16 + \frac{19.7}{\%CO_2}\right)$ |

[a] Percent CO$_2$ is determined in combustion products with water condensed (dry basis).
[b] Total combustion products include combustion products and excess air.

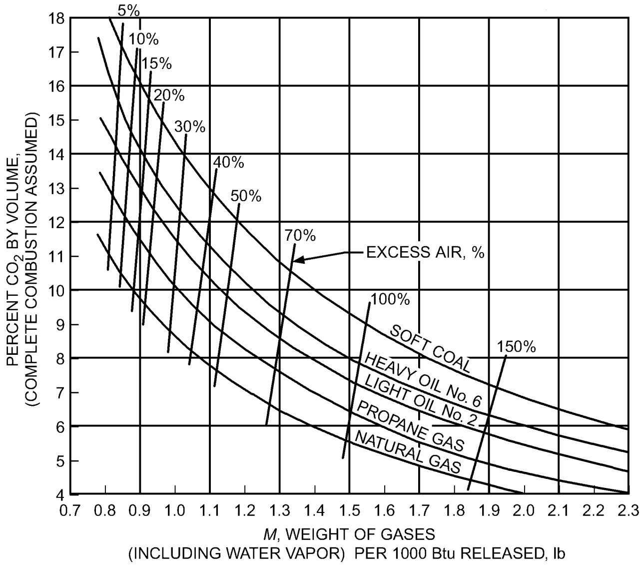


**Fig. 1 Graphical Evaluation of Rate of Vent Gas Flow from Percent CO$_2$ and Fuel Rate**

Volumetric flow rate (ft$^3$/min) $Q$ can be found as follows;

$$Q = \frac{w}{60\rho_m} \quad (5)$$

where $\rho_m$ is gas density (lb/ft$^3$).

In chimney system design, the composition and flow rate of the flue gas must be assumed to determine the ratio $M$ of mass flow to input. When combustion conditions are given in terms of excess air, Figure 1 can be used to estimate CO$_2$. The mass flow equations in Table 1 illustrate the influence of fuel composition; however, additional guidance is needed for system design. The information provided with many heat-producing appliances is limited to whether they have been tested, certified, listed, or approved to comply with applicable standards. From this information and from the type of fuel and draft control, certain inferences can be drawn regarding the flue gas.

Table 2 suggests typical values for the vent or chimney systems for gaseous and liquid fuels when specific outlet conditions for the

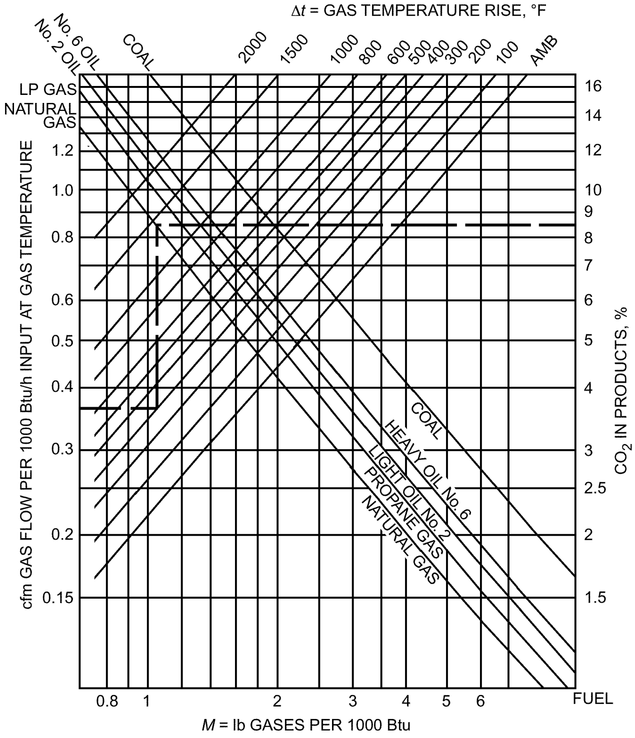
**Table 2 Typical Chimney and Vent Design Conditions**[a]

| Fuel | Appliance | % $CO_2$ | Temperature Rise, °F | *M* (Mass Flow/Input), lb Total Flue Gas[b] per 1000 Btu Fuel Input | Flue Gas Density,[c] lb/ft³ | Flow Rate/Unit Heat Input,[c] cfm per 1000 Btu/h at Flue Gas Temperature |
|---|---|---|---|---|---|---|
| Natural gas | Draft hood | 5.3 | 300 | 1.54 | 0.0483 | 0.530 |
| Propane gas | Draft hood | 6.0 | 300 | 1.59 | 0.0483 | 0.547 |
| Natural gas | | | | | | |
| Low-efficiency | No draft hood | 8.0 | 400 | 1.06 | 0.0431 | 0.409 |
| High-efficiency | No draft hood | 7.0 | 240 | 1.19 | 0.0522 | 0.381 |
| No. 2 oil | Residential | 9.0 | 500 | 1.24 | 0.0389 | 0.531 |
| Oil | Forced-draft, over 400,000 Btu/h | 13.5 | 300 | 0.85 | 0.0483 | 0.295 |
| Waste, Type 0 | Incinerator | 9.0 | 1340 | 1.62 | 0.0213 | 1.267 |

[a] Values are for appliances with flue losses of 17% or more. For appliances with lower flue losses (high-efficiency types), see appliance installation instructions or ask manufacturer for operating data.

[b] Total flue gas includes combustion products and excess air.

[c] At sea level and 60°F ambient temperature.



*EXAMPLE*: Natural gas at 8.5% $CO_2$ and 300°F rise
read *M* = 1.05 lb gases per 1000 Btu
Gas flow = 0.36 cfm per 1000 Btu/h input

**Fig. 2 Flue Gas Mass and Volumetric Flow**

**Table 3 Mass Flow for Incinerator Chimneys**

| Type of Waste | Heat Value of Waste,[a] Btu/lb | Auxiliary Fuel[a] per Unit Waste, Btu/lb | Combustion Products | | |
|---|---|---|---|---|---|
| | | | cfm/lb Waste at 1400°F[b] | lb per lb Waste[c] | *M*, lb Products per 1000 Btu |
| 0 | 8500 | 0 | 10.74 | 13.76 | 1.62 |
| 1 | 6500 | 0 | 8.40 | 10.80 | 1.66 |
| 2 | 4300 | 0 | 5.94 | 7.68 | 1.79 |
| 3 | 2500 | 1500 | 4.92 | 6.25 | 2.50 |
| 4 | 1000 | 3000 | 4.14 | 5.33 | 5.33 |

[a] Auxiliary fuel may be used with any type of waste, depending on incinerator design.

[b] Specialized units may produce higher or lower outlet gas temperatures, which must be considered in chimney sizing, using Equation (14), Equation (23), or Figure 7.

[c] Multiply these values by pounds of waste burned per hour to establish mass flow.

appliance are not known. If a gas appliance with draft hood is used, Table 2 recommends that dilution air through the draft hood reduce the $CO_2$ percentage to 5.3%. For appliances using draft regulators, the dilution and temperature reduction is a function of the draft regulator gate opening, which depends on excess draft. If the chimney system produces the exact draft necessary for the appliance, little dilution takes place.

For manifolded gas appliances that have draft hoods, dilution through draft hoods of inoperative appliances must be considered in precise system design. However, with forced-draft appliances having wind box or inlet air controls, dilution through inoperative appliances may be unimportant, especially if pressure at the outlet of inoperative appliances is neutral (atmospheric level).

Figure 2 can be used to estimate mass and volumetric flow. Flow conditions in the chimney connector, manifold, vent, and chimney vary with configuration and appliance design and are not necessarily the same as boiler or appliance outlet conditions.

Mass flow in incinerator chimneys must account for the probable heat value of the waste, its moisture content, and use of additional fuel to initiate or sustain combustion. Classifications of waste and corresponding values of *M* in Table 3 are based on recommendations from the Incinerator Institute of America. Combustion data given for types 0, 1, and 2 waste do not include any additional fuel. Where constant burner operation accompanies waste combustion, the additional quantity of products should be considered in the chimney design.

The system designer should obtain exact outlet conditions for the maximum rate operation of the specific appliance. This information can reduce chimney construction costs. For appliances with higher seasonal or steady-state efficiencies, however, special attention should be given to the manufacturer's venting recommendations because the flue gas may differ in composition and temperature from conventional values.

## 2. Mean Chimney Gas Temperature and Density

Chimney gas temperature depends on the fuel, appliance, draft control, chimney size, and configuration.

Density of gas within the chimney and theoretical draft both depend on gas temperature. Although the gases flowing in a chimney system lose heat continuously from entrance to exit, a single mean gas temperature must be used in either the design equation or the chart. Mean chimney gas density is virtually the same as air density at the same temperature. Thus, density may be found as

$$\rho_m = \rho_a\left(\frac{T_s}{T_m} \times \frac{B}{B_o}\right) = 1.325\frac{B}{T_m} \tag{6}$$

*where*

$\rho_m$ = chimney gas density, lb/ft³
$T_s$ = standard temperature = 518.67°R

**Table 4 Mean Chimney Gas Temperature for Various Appliances**

| Appliance Type | Mean Temperature $t_m$* in Chimney, °F |
|---|---|
| Natural gas-fired heating appliance with draft hood (low-efficiency) | 360 |
| LP gas-fired heating appliance with draft hood (low-efficiency) | 360 |
| Gas-fired heating appliance, no draft hood | |
| Low-efficiency | 460 |
| High-efficiency | 300 |
| Oil-fired heating appliance (low-efficiency) | 560 |
| Conventional incinerator | 1400 |
| Controlled air incinerator | 1800 to 2400 |
| Pathological incinerator | 1800 to 2800 |
| Turbine exhaust | 900 to 1400 |
| Diesel exhaust | 900 to 1400 |
| Ceramic kiln | 1800 to 2400 |

*Subtract 60°F ambient to obtain temperature rise for use with Figure 7.

**Table 5 Overall Heat Transfer Coefficients of Various Chimneys and Vents**

| Material | $U$, Btu/h · ft$^2$ · °F* Observed | Design | Remarks |
|---|---|---|---|
| Industrial steel stacks | — | 1.3 | Under wet wind |
| Clay or iron sewer pipe | 1.3 to 1.4 | 1.3 | Used as single-wall material |
| Asbestos-cement gas vent | 0.72 to 1.42 | 1.2 | Tested per UL *Standard* 441 |
| Black or painted steel stove pipe | — | 1.2 | Comparable to weathered galvanized steel |
| Single-wall galvanized steel | 0.31 to 1.38 | 1.0 | Depends on surface condition and exposure |
| Single-wall unpainted pure aluminum | — | 1.0 | No. 1100 or other bright-surface aluminum alloy |
| Brick chimney, tile-lined | 0.5 to 1.0 | 1.0 | For gas appliances in residential construction per NFPA *Standard* 211 |
| Double-wall gas vent, 1/4 in. air space | 0.37 to 1.04 | 0.6 | Galvanized steel outer pipe, pure aluminum inner pipe; tested per UL *Standard* 441 |
| Double-wall gas vent, 1/2 in. air space | 0.34 to 0.7 | 0.4 | |
| Insulated prefabricated chimney | 0.34 to 0.7 | 0.3 | Solid insulation meets UL *Standard* 103 when chimney is fully insulated |

*U-factors based on inside area of chimney.

$\rho_a$ = air density at $T_s$ and $B_o$ = 0.0765 lb/ft$^3$
$B$ = local barometric pressure, in. Hg
$T_m$ = mean chimney gas temperature at average system conditions, °R
$B_o$ = standard pressure = 29.92 in. Hg

The density $\rho_a$ in Equation (6) is a compromise value for typical humidity. The subscript $m$ for density and temperature requires that these properties be calculated at mean gas temperature or vertical midpoint of a system (inlet conditions can be used where temperature drop is not significant).

Using a reasonably high ambient (such as 60°F) for design ensures improved operation of the chimney when ambient temperatures drop because temperature differentials and draft increase.

A design requires assuming an initial or inlet chimney gas temperature. In the absence of specific data, Table 4 provides a conservative temperature. For appliances that can operate over a range of temperatures, size should be calculated at both extremes to ensure an adequate chimney.

The drop in vent gas temperature from appliance to exit reduces capacity, particularly in sizes of 12 in. or less. In gravity type B gas vents, which may be as small as 3 in. in diameter, and in other systems used for venting gas appliances, capacity is best determined from the ANSI/NFPA 54/ANSI/AGA Z223.1 or CSA *Standard* B149.1. In these codes, the tables compensate for the particular characteristics of the chimney material involved, except for very high single-wall metal pipe. Between 12 and 18 in. diameters, the effect of heat loss diminishes greatly because there is greater gas flow relative to system surface area. For 20 in. and greater diameters, cooling has little effect on final size or capacity.

A straight vertical vent or chimney directly off the appliance requires little compensation for cooling effects, even with smaller sizes. However, a horizontal connector running from the appliance to the base of the vent or chimney has enough heat loss to diminish draft and capacity. Figure 3 is a plot of temperature correction $C_u$, which is a function of connector size, length, and material for either conventional single-wall metal connectors or double-wall metal connectors.

To use Figure 3, estimate connector size and length and read the temperature multiplier. For example, 16 ft of 7 in. diameter single-wall connector has a multiplier of 0.61. If inlet temperature rise $\Delta t_e$ above ambient is 300°F, operating mean temperature rise $\Delta t_m$ will be 0.61 × 300 = 183°F. This factor adequately corrects the temperature at the midpoint of the vertical vent for heights up to 100 ft.

The correction procedure includes the assumption that the overall heat transfer coefficient of a vertical chimney is approximately 0.6 Btu/h·ft$^2$·°F or less (the value for double-wall metal). This procedure does not correct for cooling in very high stacks constructed entirely of single-wall metal, especially those exposed to cold ambient temperatures. For severe exposures or excessive heat loss, a trial calculation assuming a conservative operating temperature shows whether capacity problems will be encountered.

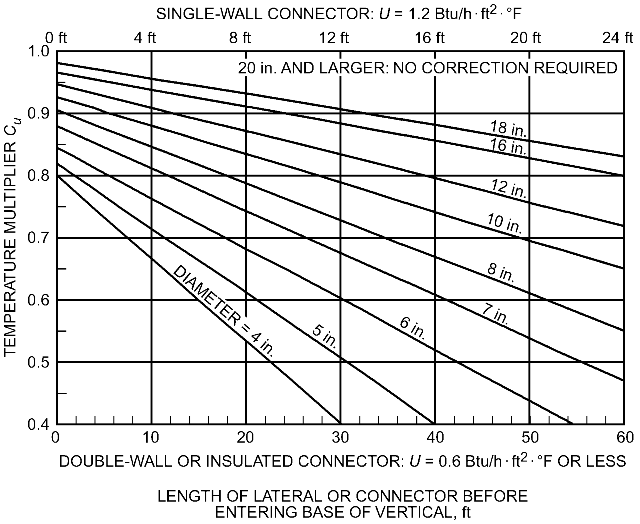


**Fig. 3 Temperature Multiplier $C_u$ for Compensation of Heat Losses in Connector**

For more precise heat loss calculations, Table 5 suggests overall heat transfer coefficients for various constructions installed in typical environments at usual flue gas flow velocities (Segeler 1965). For masonry, any additional thickness beyond the single course of brick plus tile liner used in residential chimneys decreases the coefficient.

Equations (7) and (8) describe flow and heat transfer, respectively, within a venting system with $D_a = 0$.

$$q_m = 3600 T_s \sqrt{2g}\, c_p \frac{B}{B_o} \times \frac{A}{T_m}\left(\frac{H}{kT_o}\right)^{0.5} \Delta t_m^{1.5} \rho_m \tag{7}$$

$$\frac{q}{q_m} = \frac{\Delta t_e}{\Delta t_m} = \exp\left(\pi \bar{U} d_f \frac{L_m \Delta t_m}{q_m}\right) \tag{8}$$

*where*

$A$ = area of passage cross section, ft$^2$
$c_p$ = specific heat
$d_f$ = inside diameter, ft
$\exp x$ = $e^x$
$g$ = gravitational constant = 32.1740 ft/s$^2$
$H$ = height of chimney above grade or inlet, ft
$k$ = fixed, dimensionless system flow resistance coefficient for pipe and fittings
$L_m$ = length from inlet to location (in the vertical) of mean gas temperature, ft
$q$ = heat flow rate at vent inlet, Btu/h
$q_m$ = heat flow rate at midpoint of vent, Btu/h
$\Delta t_e$ = $(T - T_o)$ = temperature difference entering system, °F
$\Delta t_m$ = $(T_m - T_o)$ = chimney gas mean temperature rise, °F
$T$ = flue gas temperature at vent inlet, °F
$T_m$ = chimney mean flue gas temperature, °R
$T_o$ = design ambient temperature, 520°R
$T_s$ = 518.67°F
$\bar{U}$ = heat transfer coefficient, Btu/h·ft$^2$·°F
$\rho_m$ = mean flue gas density from Equation (6)

Assuming reasonable constancy of $\bar{U}$, the overall heat transfer coefficient of the venting system material, Equations (7) and (8) provide a solution for maximum vent gas capacity. They can also be used to develop cooling curves or calculate the length of pipe where internal moisture condenses. Kinkead (1962) details methods of solution and application to both individual and combined gas vents.

## 3. Theoretical Draft

The theoretical draft of a gravity chimney or vent is the difference in weight between a given column of warm (light) chimney gas and an equal column of cold (heavy) ambient air. Chimney gas density or temperature, chimney height, and barometric pressure determine theoretical draft; flow is not a factor. The equation for theoretical draft assumes chimney gas density is the same as that of air at the same temperature and pressure; thus,

$$D_t = 0.2554 BH\left(\frac{1}{T_o} - \frac{1}{T_m}\right) \tag{9}$$

*where*

$B$ = local barometric pressure, in. Hg
$D_t$ = theoretical draft, in. of water
$H$ = height of chimney above grade or inlet, ft
$T_m$ = mean flue gas temperature at average conditions in system, °R
$T_o$ = ambient temperature, °R

Theoretical draft thus increases directly with height and with the difference in density between the hot and cold columns.

Theoretical draft $D_t$ is always positive (unless chimney gases are colder than ambient air). Equation (9) for theoretical draft is the basis for Figure 4, which can be used up to 1000°F and 7000 ft elevation. For example, using Figure 4, ambient air temperature at 40°F, 4000 ft above sea level (25.85 in. Hg), and mean chimney gas temperature at 350°F provides $D_{100}$ draft of 0.5 in. of water per 100 ft of height. For $D_{100}$ draft at 25 in. Hg barometric pressure or 4900 ft above sea level, follow the intersection to the pivot line and read the new $D_{100}$ draft of 0.48 in. of water.

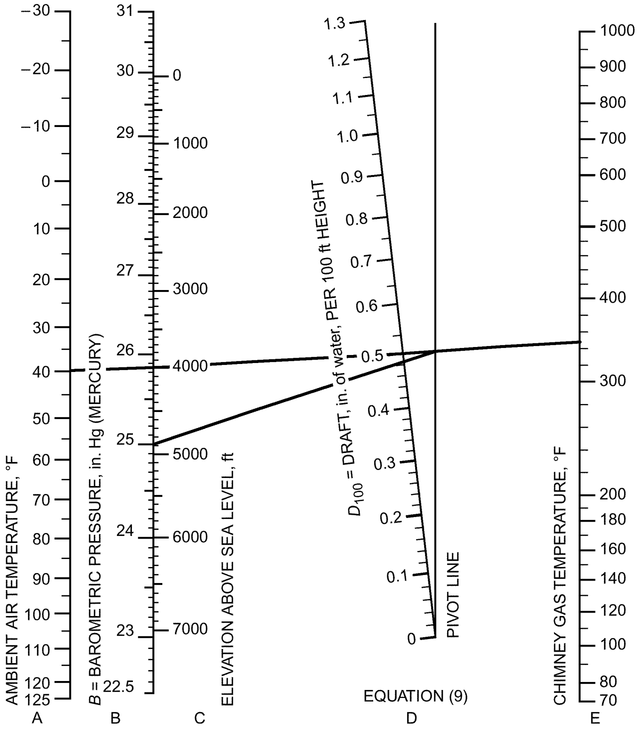


**Fig. 4 Theoretical Draft Nomograph**

**Table 6 Approximate Theoretical Draft of Chimneys**

| Vent Gas Temperature Rise, °F | $D_t$ per 100 ft, in. of water |
|---|---|
| 100 | 0.2 |
| 150 | 0.3 |
| 200 | 0.4 |
| 300 | 0.5 |
| 400 | 0.6 |
| 500 | 0.7 |
| 600 | 0.8 |
| 800 | 0.9 |
| 1100 | 1.0 |
| 1600 | 1.1 |
| 2400 | 1.2 |

*Notes*: Ambient temperature = 60°F = 520°R
Chimney gas density = air density
Sea-level barometric pressure = 29.92 in. Hg
Equation (9) may be used to calculate exact values for $D_t$ at any altitude.

Theoretical draft should be estimated and included in system calculations, even for appliances producing considerable positive outlet static pressure, to achieve the economy of minimum chimney size. Equation (9) may be used directly to calculate exact values for theoretical draft at any altitude. For ease of application and consistency with Figure 7, Table 6 lists approximate theoretical draft for typical gas temperature rises above 60°F ambient.

Appliances with fixed fuel beds, such as hand-fired coal stoves and furnaces, require positive available draft (negative gage pressure). Small oil heaters with pot-type burners, as well as residential furnaces with pressure-atomizing oil burners, need positive available draft, which can usually be set by following the manufacturer's instructions for setting the draft regulator. Available draft requirements for larger packaged boilers or appliances assembled from components may be negative, zero (neutral), or positive.

**Table 7 Input Altitude Factor for Equation (21) Theoretical Draft**

| Altitude, ft | Barometric Pressure $B$, in. Hg | Factor* |
|---|---|---|
| Sea level | 29.92 | 1.00 |
| 2,000 | 27.82 | 1.08 |
| 4,000 | 25.82 | 1.16 |
| 6,000 | 23.98 | 1.25 |
| 8,000 | 22.22 | 1.35 |
| 10,000 | 20.58 | 1.45 |

*Multiply operating input by factor to obtain design input.

Compensation of theoretical draft for altitude or barometric pressure is usually necessary for appliances and chimneys functioning at elevations greater than 2000 ft. Depending on the design, one of the following approaches to pressure or altitude compensation is necessary for chimney sizing.

1. Figure 7: Use sea-level theoretical draft.
2. Equation (21): Use local theoretical draft with actual energy input, or use sea level theoretical draft with energy input multiplied by ratio of sea level to local barometric pressure (Table 7 factor).
3. Equation (23): Use local theoretical draft and barometric pressure with volumetric flow at the local density.

The altitude correction multiplier for input (Table 7) is the only method of correcting to other elevations. Reducing theoretical draft imposes an incorrect compensation on the chart.

Gas appliances with draft hoods, for example, are usually derated 4% per 1000 ft of elevation above sea level when they are operated at 2000 ft altitude or above (see ANSI/NFPA 54/ANSI/AGA Z223.1 or CSA *Standard* B149.1). The altitude correction factor derates the design input so that the vent size at altitude for derated gas appliances is effectively the same as at sea level. For other appliances where burner adjustments or internal changes might be used to adjust for reduced density at altitude, the same factors produce an adequately compensated chimney size. For example, an appliance operating at 6000 ft elevation at $10 \times 10^6$ Btu/h input, but requiring the same draft as at sea level, should have a chimney selected on the basis of 1.244 times the operating input, or $12.5 \times 10^6$ Btu/h.

## 4. System Pressure Loss Caused by Flow

In any chimney system, flow losses, expressed as pressure drop $\Delta p$ in inches of water, are the difference between theoretical and available draft:

$$\Delta p = D_t - D_a \tag{10}$$

where $\Delta p$ is always positive.

In any chimney or vent system, flow losses resulting from velocity and resistance can be determined from the Bernoulli equation:

$$\Delta p = \frac{k \rho_m V^2}{5.2(2g)} \tag{11}$$

*where*

$k$ = dimensionless system resistance coefficient of piping and fittings
$V$ = system gas velocity at mean conditions, fps
$g$ = gravitational constant = 32.1740 ft/s$^2$
$\rho_m$ = mean flue gas density, lb/ft$^3$

Pressure losses are thus directly proportional to the resistance factor and to the square of the velocity.

**Table 8 Pressure Equations for $\Delta p$**

| Required Appliance Outlet Pressure or Available Draft $D_a$ | $\Delta p$ Equation[a] Gravity Only | Gravity plus Inducer[b] |
|---|---|---|
| Negative, needs positive draft | $\Delta p = D_t - D_a$ | $\Delta p = D_t - D_a + D_b$ |
| Zero, vent with draft hood or balanced forced draft | $\Delta p = D_t$ | $\Delta p = D_t + D_b$ |
| Positive, causes negative draft | $\Delta p = D_t + D_a$ | $\Delta p = D_t + D_a + D_b$ |

[a] Equations use absolute pressure for $D_a$.
[b] $D_b$ = static pressure boost of inducer at flue gas temperature and rated flow.

## 5. Available Draft

Starting with Equation (1), the difference between theoretical draft and flow losses without depressurization or boost is:

$$D_a = D_t - \Delta p$$

Available draft $D_a$ can therefore be defined as:

$$D_a = 0.2554BH\left(\frac{1}{T_o} - \frac{1}{T_m}\right) - \frac{k\rho_m V^2}{5.2(2g)} \tag{12}$$

See Equations (2) and (3) for geometrically defined versions.

Available draft $D_a$ can be negative, zero, or positive. The pressure difference $\Delta p$, or theoretical minus available draft, overcomes the flow losses. Table 8 lists the pressure components for three draft configurations. Table 8 applies to still air (no wind) conditions and a neutral (zero) pressure difference between the space surrounding the appliance or fireplace and the atmosphere. Columns with and without boost are provided.

The effect of a nonneutral pressure difference on capacity or draft may be included by imposing a static pressure (either positive or negative). One way to circumvent a space-to-atmosphere pressure difference is to use a sealed-combustion system or direct-vent system (i.e., all combustion air is taken directly from the outdoor atmosphere, and all flue gas is discharged to the outdoor atmosphere with no system openings such as draft hoods). The effect of wind on capacity or draft may be included by imposing a static pressure (either positive or negative) or by changing the vent terminal resistance loss. However, a properly designed and located vent terminal should cause little change in $\Delta p$ at typical wind velocities.

Although small static draft pressures can be measured at the entrance and exit of gas appliance draft hoods, $D_a$ at the appliance is effectively zero. Therefore, all theoretical draft energy produces chimney flow velocity and overcomes chimney flow resistance losses.

## 6. Chimney Gas Velocity

Velocity in a chimney or vent varies inversely with flue gas density $\rho_m$ and directly with mass flow rate. The equation for flue gas velocity at mean gas temperature in the chimney is

$$V = \frac{144 \times 4w}{3600\pi\rho_m d_i^2} \tag{13}$$

*where*

$V$ = flue gas velocity, fps
$d_i$ = inside diameter, in.
$\rho_m$ = mean flue gas density, lb/ft$^3$
$w$ = mass flow rate, lb/h

To express velocity as a function of input and chimney gas composition, $w$ in Equation (13) is replaced by using Equation (4):

$$V = \frac{144 \times 4IM}{3600\pi\rho_m d_i^2} = \frac{0.0509IM}{\rho_m d_i^2} \tag{14}$$

Thus, chimney velocity depends on the product of heat input $I$ and the ratio $M$ of mass flow to input.

The input capacity or diameter of a chimney may usually be found without determining flow velocity. Internal or exit velocity must occasionally be known, to ensure effluent dispersal or avoid flow noise. Also, the flow velocity of incinerator chimneys, turbine exhaust systems, and other appliances with high outlet pressures or velocities is needed to estimate piping loss coefficients.

Equations (11), (13), (14), and (22) can be applied to find velocity. The velocity may also be calculated by dividing volumetric flow rate $Q$ by chimney area A. A similar calculation may be performed when the energy input is known. For example, a 34 in. chimney serving a $10 \times 10^6$ Btu/h natural gas appliance, at 8.5% $CO_2$ and 300°F above ambient in the chimney, produces (from Figure 2) 0.36 cfm per 1000 Btu/h. Chimney gas flow rate is

$$Q = 10 \times 10^6 \left(\frac{0.36}{1000}\right) = 3600 \text{ cfm} \tag{15}$$

Dividing by area to obtain velocity, $V = 3600/6.305 = 571$ fpm

Chimney gas velocity affects the piping friction factor $k_L$ and also the roughness correction factor. The section on Resistance Coefficients has further information, and Example 3 illustrates how these factors are used in the velocity equations.

Chimney systems can operate over a wide range of velocities, depending on modulation characteristics of the burner system or the number of appliances in operation. Typical velocity in vents and chimneys ranges from 300 to 3000 fpm. A chimney design developed for maximum input and maximum velocity should be satisfactory at reduced input because theoretical draft is roughly proportional to flue gas temperature rise, while flow losses are proportional to the square of the velocity. Thus, as input is reduced, flow losses decrease more rapidly than system motive pressures.

Effluent dispersal may occasionally require a minimum upward chimney outlet velocity, such as 3000 fpm. A tapered exit cone can best meet this requirement. For example, to increase the outlet velocity from the 34 in. chimney ($A = 6.305$ ft$^2$ area) from 1600 to 3000 fpm, the cone must have a discharge area of $6.305 \times 1600/3000 = 3.36$ ft$^2$ and a 24.8 in. diameter. An exit cone avoids excessive flow losses because the entire system operates at the lower velocity, and a resistance factor is only added for the cone. In this case, the added resistance for a gradual taper approximates the following (see Table 9):

$$k = \left(\frac{d_{i1}}{d_{i2}}\right)^4 - 1 = \left(\frac{34}{24.8}\right)^4 - 1 = 2.53 \tag{16}$$

Noise in chimneys may be caused by turbulent flow at high velocity or by combustion-induced oscillations or resonance. Noise is seldom encountered in gas vent systems or in systems producing positive available draft, but it may be a problem with forced-draft appliances. Turbulent flow noise can be avoided by designing for lower velocity, which may entail increasing the chimney size above the minimum recommended by the appliance manufacturer. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on noise control.

## 7. System Resistance Coefficient

The velocity head method of determining resistance losses assigns a fixed numerical coefficient (independent of velocity) or $k$ factor to every fitting or turn in the flow circuit, as well as to piping.

The resistance coefficient $k$ that appears in Equations (21) and (23) and Figure 7 summarizes the friction loss of the entire chimney system, including piping, fittings, and configuration or interconnection factors. Capacity of the chimney varies inversely with the square root of $k$, whereas diameter varies as the fourth root of $k$. The insensitivity of diameter and input to small variations in $k$ simplifies design. Analyzing details such as pressure regain, increasers and reducers, and gas cooling junction effects is unnecessary if slightly high resistance coefficients are assigned to any draft diverters, elbows, tees, terminations, and, particularly, piping.

**Table 9 Resistance Loss Coefficients**

| Component | Suggested Design Value, Dimensionless* | Estimated Span and Notes |
|---|---|---|
| Inlet acceleration ($k_1$) | | |
| Gas vent with draft hood | 1.5 | 1.0 to 3.0 |
| Barometric regulator | 0.5 | 0.0 to 0.5 |
| Direct connection | 0.0 | Also dependent on blocking damper position |
| Round elbow ($k_2$) | | |
| 90° | 0.75 | 0.5 to 1.5 |
| 45° | 0.3 | — |
| Tee or 90° connector ($k_3$) | 1.25 | 1.0 to 4.0 |
| Y connector | 0.75 | 0.5 to 1.5 |
| Cap, top ($k_4$) | | |
| Open straight | 0.0 | — |
| Low-resistance (UL) | 0.5 | 0.0 to 1.5 |
| Other | — | 1.5 to 4.5 |
| Spark screen | 0.5 | — |
| Converging exit cone | $(d_{i1}/d_{i2})^4 - 1$ | System designed using $d_{i1}$ |
| Tapered reducer ($d_{i1}$ to $d_{i2}$) | $1 - (d_{i2}/d_{i1})^4$ | System designed using $d_{i2}$ |
| Increaser | — | See Chapter 2, 2005 *ASHRAE Handbook—Fundamentals*. |
| Piping ($k_L$) | $0.4\dfrac{L\text{, ft}}{d_i\text{, in.}}$ | Numerical coefficient (friction factor $F$) varies from 0.2 to 0.5; see Figure 13, Chapter 2, 2005 *ASHRAE Handbook—Fundamentals* for size, roughness, and velocity effects. |

*Initial assumption when size is unknown: $k = 5.0$ for entire system, for first trial; $k = 7.5$ for combined gas vents only

*Note*: For combined gravity gas vents serving two or more appliances (draft hoods), multiply total $k$ [components + piping—see Equations (17), (18), and (19)] by 1.5 to obtain gravity system design coefficient. (This rule does not apply to forced- or induced-draft vents or chimneys.)

The flow resistance of a fitting such as a tee with flue gases entering the side and making a 90° turn is assumed to be constant at $k = 1.25$, independent of size, velocity, orientation, inlet or outlet conditions, or whether the tee is located in an individual vent or in a manifold. Conversely, if flue gases pass straight through a tee, as in a manifold, assumed resistance is zero, regardless of any area changes or flow entry from the side branch. For any chimney with fittings, the total flow resistance is a constant plus variable piping resistance—the latter being a function of centerline length divided by diameter. Table 9 suggests moderately conservative resistance coefficients for common fittings. Elbow resistance may be lowered by long-radius turns; however, corrugated 90° elbows may have resistance values at the high end of the scale. Table 9 shows resistance as a function of inlet diameter $d_{i1}$ and outlet diameter $d_{i2}$.

System resistance $k$ may be expressed as follows:

$$k = k_f + k_L \tag{17}$$

with

$$k_f = k_1 + n_2 k_2 + n_3 k_3 + k_4 + \cdots \tag{18}$$

and

$$k_L = \frac{FL}{d_i} \tag{19}$$

where

$k_f$ = fixed fitting loss coefficient
$k_L$ = piping resistance loss function (Figure 13, Chapter 2 of the 2005 *ASHRAE Handbook—Fundamentals*, adjusted for units)
$k_1$ = inlet acceleration coefficient
$k_2$ = elbow loss coefficient, $n_2$ = number of elbows
$k_3$ = tee loss coefficient, $n_3$ = number of tees
$k_4$ = cap, top, or exit cone loss coefficient
$F$ = friction factor
$L$ = length of all piping in chimney system, ft
$d_i$ = inside diameter, ft

For combined gas vents using appliances with draft hoods, the summation $k$ must be multiplied by a diversity factor of 1.5 (see Table 9 note and Example 5). This multiplier does not apply to forced- or induced-draft vents or chimneys.

When size is unknown, the following $k$ values may be used to run a first trial estimate:

$k$ = 5.0 for the entire system
$k$ = 7.5 for combined gas vents only

The resistance coefficient method adapts well to systems in which the fittings cause significant losses. Even for extensive systems, an initial assumption of $k = 5.0$ gives a tolerably accurate vent or chimney diameter in the first trial solution. Using this diameter with the piping resistance function [Equation (19)] in a second trial normally yields the final answer.

The minimum system resistance coefficient in a gas vent with a draft hood is always 1.0 because all gases must accelerate through the draft hood from almost zero velocity to vent velocity.

For a system connected directly to the outlet of a boiler or other appliance where the capacity is stated as full-rated heat input against a positive static pressure at the chimney connection, minimum system resistance is zero, and no value is added for existing velocity head in the system.

For simplified design, a value of 0.4 for $F$ in Equation (19) applies for all sizes of vents or chimneys and for all velocities and temperatures. As diameter increases, this function becomes increasingly conservative, which is desirable because larger chimneys are more likely to be made of rough masonry construction or other materials with higher pressure losses. The 0.4 constant also introduces an increasing factor of safety for flow losses at greater lengths and heights.

Figure 5 is a plot of friction factor $F$ versus velocity and diameter for commercial iron and steel pipe at a flue gas temperature of 300°F above ambient (Lapple 1949). The figure shows, for example, that a 48 in. diameter chimney with a flue gas velocity of 80 fps may have a friction factor as low as 0.2. In most cases, $k_L = 0.3\ L/d_i$ gives reasonable design results for chimney sizes 18 in. and larger because systems of this size usually operate at flue gas velocities greater than 10 fps.

At 1000°F or over, the factors in Figure 5 should be multiplied by 1.2. Because Figure 5 is for commercial iron and steel pipe, an additional correction for greater or less surface roughness may be imposed. For example, the factor for a very rough 12 in. diameter pipe may be doubled at a velocity as low as 2000 fpm.

For most chimney designs, a friction factor $F$ of 0.4 gives a conservative solution for diameter or input for all sizes, types, and operating conditions of prefabricated and metal chimneys; alternately, $F = 0.3$ is reasonable if the diameter is 18 in. or more. Because neither input nor diameter is particularly sensitive to the total friction factor, the overall value of $k$ requires little correction.

Masonry chimneys, including those lined with clay flue tile, may have rough surfaces, tile shape variations that cause misalignment, and joints at frequent intervals with possible mortar protrusions. In addition, the inside cross-sectional area of liner shapes may be less than expected because of local manufacturing variations, as well as differences between claimed and actual size. To account for these characteristics, the estimate for $k_L$ should be on the high side, regardless of chimney size or velocity.

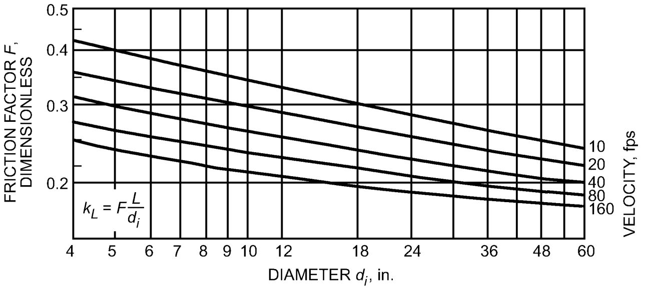


**Fig. 5 Friction Factor for Commercial Iron and Steel Pipe**
(Lapple 1949)

Computations should be made by assuming smooth surfaces and then adding a final size increase to compensate for shape factor and friction loss. Performance or capacity of metal and prefabricated chimneys is generally superior to that of site-constructed masonry.

## Configuration and Manifolding Effects

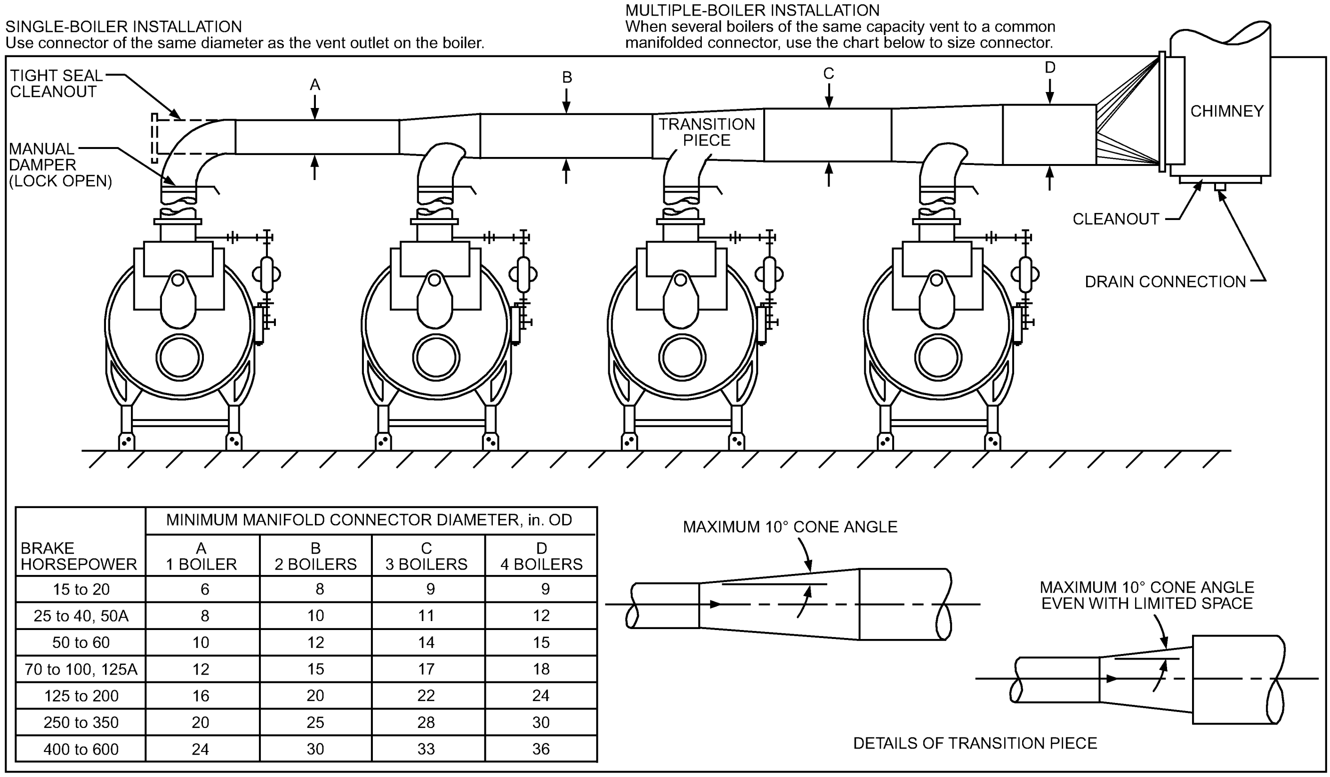
The most common configuration is the individual vent, stack, or chimney, in which one continuous system carries products from appliance to terminus. Other configurations include the combined vent serving a pair of appliances, the manifold serving several, and branched systems with two or more lateral manifolds connected to a common vertical system. As the number of appliances served by a common vertical vent or chimney increases, the precision of design decreases because of diversity factors (variation in the number of appliances in operation) and the need to allow for maximum and minimum input operation (Stone 1957). For example, the vertical common vent for interconnected gas appliances must be larger than for a single appliance of the same input to allow for operating diversity and draft hood dilution effects. Connector rise, headroom, and configuration in the equipment room must be designed carefully to avoid draft hood spillage and related oxygen depletion problems.

For typical combined vents, the diversity effect must be introduced into Figure 7 and the equations by multiplying system resistance loss coefficient $k$ by 1.5 (see Table 9 note and Example 5). This multiplier compensates for junction effect and part-load operation.

Manifolds for appliances with barometric draft regulators can be designed without allowing for dilution air through inoperative appliances. In this case, because draft regulators remain closed until regulation is needed, dilution under part load is negligible. In addition, airflow through any inoperative appliance is negligible because the combustion air inlet dampers are closed and the multiple-pass heat exchanger has a high internal flow resistance.

Manifold systems of oil-burning appliances, for example, have a lower flow velocity and, hence, lower losses. As a result, they produce reasonable draft at part load or with only one of several appliances in operation. Therefore, diversity of operation has little effect on chimney design. Some installers set each draft regulator at a slightly different setting to avoid oscillations or hunting possibly caused by burner or flow pulsations.

Calculation of the resistance coefficient of any portion of a manifold begins with the appliance most distant from the vertical portion. All coefficients are then summed from its outlet to the vent terminus. The resistance of a series of tee joints to flow passing horizontally straight through them (not making a turn) is the same as that of an equal length of piping (as if all other appliances were off). This assumption holds whether the manifold is tapered (to accommodate increasing input) or of a constant size large enough for the accumulated input.



| BRAKE HORSEPOWER | MINIMUM MANIFOLD CONNECTOR DIAMETER, in. OD<br>A<br>1 BOILER | B<br>2 BOILERS | C<br>3 BOILERS | D<br>4 BOILERS |
|---|---|---|---|---|
| 15 to 20 | 6 | 8 | 9 | 9 |
| 25 to 40, 50A | 8 | 10 | 11 | 12 |
| 50 to 60 | 10 | 12 | 14 | 15 |
| 70 to 100, 125A | 12 | 15 | 17 | 18 |
| 125 to 200 | 16 | 20 | 22 | 24 |
| 250 to 350 | 20 | 25 | 28 | 30 |
| 400 to 600 | 24 | 30 | 33 | 36 |

**Fig. 6 Connector Design**

Coefficients are assigned only to inlet and exit conditions, to fittings causing turns, and to the piping running from the affected appliance to the chimney exit. Initially, piping shape (round, square, or rectangular) and function (for connectors, vertical piping, or both) are irrelevant.

Some high-pressure, high-velocity packaged boilers require special manifold design to avoid turbulent flow noise. In such cases, manufacturers' instructions usually recommend increaser Y fittings, as shown in Figure 6. The loss coefficients listed in Table 9 for standard tees and elbows are higher than necessary for long-radius elbows or Y entries. Occasionally, on appliances with high chimneys augmenting boiler outlet pressure, it may appear feasible to reduce the diameter of the vertical portion to below that recommended by the manufacturer. However, any reduction may cause turbulent noise, even though all normal design parameters have been considered.

The manufacturers' sizing recommendations shown in Figure 6 apply to the specific appliance and piping arrangement shown. The values are conservative for long-radius elbows or Y entries. Frequently, the boiler room layout forces use of additional elbows, requiring larger sizes to avoid excessive flow losses.

With the simplifying assumption that the maximum velocity of the flue gas (which exists in the smaller of the two portions) exists throughout the entire system, the design chart (Figure 7) and Equations (20) and (21) can be used to calculate the size of a vertical portion smaller in area than the manifold or of a chimney connector smaller than the vertical. This assumption leads to a conservative design because true losses in the larger area are lower than assumed. Further, if the size change is small, either as a contraction or expansion, the added loss coefficient for this transition fitting (see Table 9) is compensated for by reduced losses in the enlarged part of the system.

These comments on size changes apply more to individual than to combined systems because it is undesirable to reduce the vertical area of the combined type, and, more frequently, it is desirable to enlarge it. If an existing vertical chimney is slightly undersized for the connected load, the complete chart method must be applied to determine whether a pressure boost is needed, because size is no longer a variable.

Sectional gas appliances with two or more draft hoods do not pose any special problems if all sections fire simultaneously. In this case, the designer can treat them as a single appliance. The appliance installation instructions either specify the size of manifold for interconnecting all draft hoods or require a combined area equal to the sum of all attached draft hood outlet areas. Once the manifold has been designed and constructed, it can be connected to a properly sized chimney connector, vent, or chimney. If the connector and chimney size is computed as less than manifold size (as may be the case with a tall chimney), the operating resistance of the manifold will be lower than the sum of the assigned component coefficients because of reduced velocity.

The general rule for conservative system design, in which manifold, chimney connector, vent, or chimney are different sizes, can be stated as follows: Always assign full resistance coefficient values to all portions carrying combined flow, and determine system capacity from the smallest diameter carrying the combined flow. In addition, horizontal chimney connectors or vent connectors should pitch upward toward the stack at a minimum of 1/4 in. per foot of connector length.

## 8. Input, Diameter, and Temperature Relationships

To obtain a design equation in which all terms are readily defined, measured, or predetermined, the gas velocity and density terms must be eliminated. Using Equation (6) to replace $\rho_m$ and Equation (14) to replace $V$ in Equation (11) gives

$$\Delta p = \frac{k\rho_m V^2}{5.2(2g)} = \frac{k}{5.2(2g)}\left(\frac{T_m}{1.325B}\right)\left(\frac{144\times 4\,IM}{3.6\times 10^6\pi d_i^2}\right)^2 \qquad (20)$$

Rearranging to solve for $I$ and including the values of $\pi$ and $2g$ gives

$$I = 4.13 \times 10^5 \frac{d_i^2}{M}\left(\frac{\Delta pB}{kT_m}\right)^{1/2} \tag{21}$$

Solving for input using Equation (21) is a one-step process, given the diameter and configuration of the chimney. More frequently, however, input, available draft, and height are given and the diameter $d_i$ must be found. Because system resistance is a function of the chimney diameter, a trial resistance value must be assumed to calculate a trial diameter. This method allows for a second (and usually accurate) solution for the final required diameter.

## 9. Volumetric Flow in Chimney or System

Volumetric flow $Q$ may be calculated in a chimney system for which Equation (21) can be solved by solving Equation (11) for velocity at mean density (or temperature) conditions:

$$V = 18.3\sqrt{\frac{\Delta p}{k\rho_m}} \tag{22}$$

This equation can be expressed in terms of the same variables as Equation (21) by using the density value $\rho_m$ of Equation (6) in Equation (22) and then substituting Equation (22) for velocity $V$. Area is expressed in terms of $d_i$. Multiplying area and velocity and adjusting for units,

$$Q = 5.2d_i^2\left(\frac{\Delta pT_m}{kB}\right)^{1/2} \tag{23}$$

where $Q$ = volumetric flow rate, cfm. The volumetric flow obtained from Equation (23) is at mean gas temperature $T_m$ and local barometric pressure $B$. Equations (21) and (23) do not account for the effects of heat transfer or cooling on flow, draft, or capacity.

Equation (23) is useful in the design of forced-draft and induced-draft systems because draft fans are usually specified in terms of volumetric flow rate at some standard ambient or selected gas temperature. An induced-draft fan is necessary for chimneys that are undersized, that are too low, or that must be operated with draft in the manifold under all conditions.

## 10. Graphical Solution of Chimney or Vent System

Figure 7 is a graphic solution for Equations (21) and (23) that is accurate enough for most problems. However, to use either the equations or the design chart, the details in the following sections should be understood so that proper choices can be made for mass flow, pressure loss, and heat transfer effects. The equations and Figure 7 are based on the fuel combustion products and temperatures in the chimney system. Figure 7 may also be used to determine flue gas velocity; the right-hand scale of grid E reads directly in velocity for any combination of flow and diameter. For example, at 10,000 cfm and 34 in. diameter (6.305 ft$^2$ area), the indicated velocity is about 1600 fpm. The velocity may also be calculated by dividing volumetric flow rate $Q$ by chimney area $A$. The right-hand scale of grid E, Figure 7, may also be multiplied by the cfm per 1000 Btu/h to find velocity. For the same chimney design conditions, the scale velocity value of 1600 fpm is multiplied by 0.36 to yield a velocity of 576 fpm in the chimney.

## STEADY-STATE CHIMNEY DESIGN GRAPHICAL SOLUTIONS

Design ambient temperature is 60°F (520°R), and all temperatures given are in terms of rise above this ambient. Thus, the 300°F line indicates a 360°F observed vent gas temperature.

Figure 1: Use sea-level theoretical draft. Theoretical draft may be corrected for altitude or reduced air density by multiplying the operating input by the factor in Table 7.

The resistance coefficient $k$ in Figure 7 summarizes the friction loss of the entire chimney system, including piping, fittings, and configuration or interconnection factors.

Figure 2 can be used with Figure 7 to estimate mass and volumetric flow.

For ease of application and consistency with Figure 7, Table 6 lists approximate theoretical draft for typical gas temperature rises above 60°F ambient.

The equations and design chart (Figure 7) are based on the fuel combustion products and temperatures in the chimney system.

When using a temperature multiplier for inclusion of horizontal sections, it must be applied to grids A and C as follows:

1. In grid A, the entering temperature rise $\Delta t_e$ must be multiplied by 0.61. For an appliance with an outlet temperature rise above ambient of 300°F, flow in the vent is based on $\Delta t_m = 300 \times 0.61 =$ 183°F rise.
2. This same 183°F rise must be used in grid C.
3. Determine $p$ using a 183°F rise for theoretical draft to be consistent with the other two temperatures. (It is incorrect to multiply theoretical draft pressure by the temperature multiplier.)

The first trial solution for diameter, using Figure 7 or Equations (21) and (23), need not consider the cooling temperature multiplier, even for small sizes. A first approximate size can be used for the temperature multiplier for all subsequent trials because capacity is insensitive to small changes in temperature.

Neither Figure 7 nor the equations contain the same number or order of steps as the derivation; for example, a step disappears when theoretical and available drafts are combined into $\Delta p$. Similarly, the examples selected vary in their sequence of solution, depending on which parameters are known and on the need for differing answers, such as diameter for a given input, diameter versus height, or the amount of pressure boost $D_b$ from a forced-draft fan. To compare the use of equations with using Figure 7 the following sample is provided. First the sample calculation is made using the provided tables and equations with variations in the original order of steps. It illustrates the direct solution for input, velocity, and volume. Secondly a calculation for input is shown that is derived from Figure 7. It differs from the first solution because of the chart arrangement.

**Example 1.** Find the input capacity (Btu/h) of a vertical, double-wall type B gas vent, 24 in. in diameter, 100 ft high at sea level. This vent is used with draft hood natural-gas-burning appliances.

**Solution:**

1. Mass flow from Table 2. $M$ = 1.54 lb/1000 Btu for natural gas, if no other data are given.
2. Temperature from Table 4. Temperature rise = 300°F and $T_m$ = 360 + 460 = 820°R for natural gas.
3. Theoretical draft from Table 6 or Equation (9). For 100 ft height at 300°F rise, $D_t$ = 0.5 in. of water.
4. Available draft for draft hood appliances: $D_a = 0$.
5. Flow losses from Table 8. $\Delta p = D_t = 0.5$; flow losses for a gravity gas vent equal theoretical draft at mean gas temperature.
6. Resistance coefficients from Table 9. For a vertical vent,

| | |
|---|---|
| Draft hood | $k_1 = 1.5$ |
| Vent cap | $k_4 = 1.0$ |
| 100 ft piping | $k_L = 0.4(100/24) = 1.67$ |
| System total | $k = 4.17$ |

7. Solution for input.

Altitude: Sea level, $B$ = 29.92 from Table 7. $d_i$ = 24 in. These values are substituted into Equation (21) as follows:

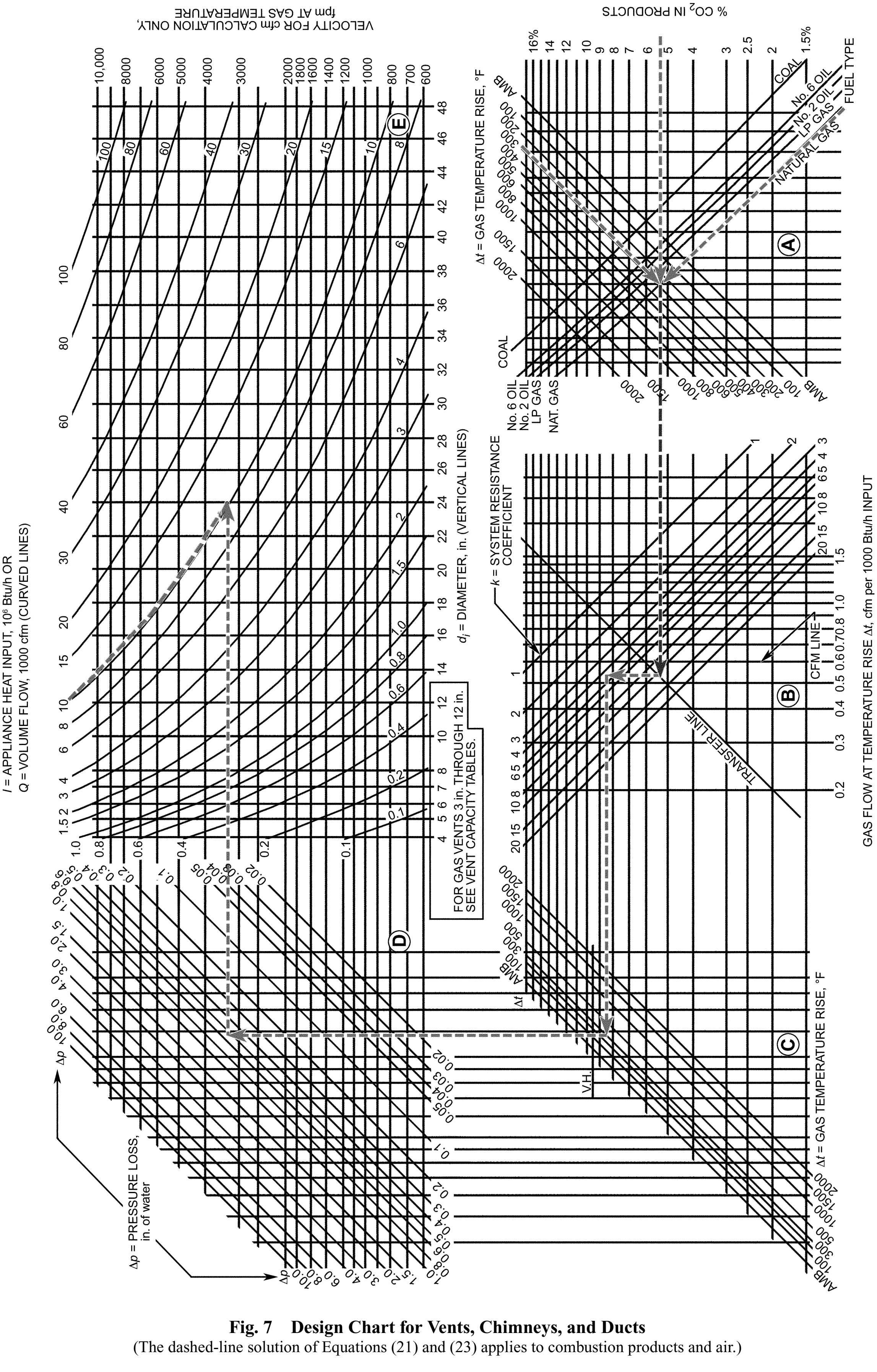


**Fig. 7 Design Chart for Vents, Chimneys, and Ducts**

(The dashed-line solution of Equations (21) and (23) applies to combustion products and air.)

$$I = 4.13 \times 10^5 \frac{(24)^2}{1.54}\left(\frac{(0.5)(29.92)}{(4.17)(820)}\right)^{1/2}$$

$$I = 10.2 \times 10^6 \text{ Btu/h input capacity}$$

8. A solution for velocity requires a prior solution for input to apply to Equation (14). First, using Equation (6),

$$\rho_m = 1.325\left(\frac{29.92}{820}\right) = 0.0483 \text{ lb/ft}^3$$

From Equation (14),

$$V = \frac{0.0509}{(0.0483)(24)^2} \times \frac{\left(10.2 \times 10^6\right)(1.54)}{1000} = 28.7 \text{ ft/s}$$

9. Volume flow can now be found because velocity is known. The flow area of 24 in. diameter is 3.14 ft$^2$, so

$$Q = (60\text{s/min})(3.14 \text{ ft}^2)(28.7 \text{ ft/s}) = 5410 \text{ cfm}$$

No heat loss correction is needed to find the new flue gas temperature because the size is greater than 20 in., and this vent is vertical with no horizontal connector.

For the same problem, Figure 7 requires a different sequence of solution. The ratio of mass flow to input for a given fuel (with parameter $M$) is not used directly; the chart requires selecting a CO$_2$ percentage in the chimney, either from Table 2 or from operating data on the appliances. Then, the temperatures are entered only as rise above ambient. Finally, calculated or estimated resistance coefficients are used to connect the conditions to the vent diameter. At this intersection, an input or flow can be derived. The solution path is as follows:

1. Enter grid A at 5.3% CO$_2$, and construct line horizontally to left intersecting natural gas.
2. From natural gas intersection, construct vertical line to $\Delta t = 300$°F.
3. From 300°F intersection in grid A, go horizontally left to transfer line in grid B.
4. From transfer line go vertically to $k = 4.17$.
5. From $k = 4.17$ run horizontally left to $\Delta t = 300$°F in grid C.
6. From 300°F go up to $\Delta p = 0.54$ in. of water in grid D.
7. From $\Delta p = 0.54$ in. of water in grid $D$, go horizontally right to intersect $d_i = 24$ in. in grid E.
8. Read capacity or input at 24 in. intersection in grid E as $10.2 \times 10^6$ Btu/h between curved lines.

If input is known and diameter must be found, the procedure is the same as with Equation (21). A preliminary $k$, usually 5.0 for an individual vent or chimney, must be estimated to find a trial diameter. This diameter is used to find a corrected $k$, and the chart is solved again for diameter.

## VENT AND CHIMNEY CAPACITY CALCULATION EXAMPLES

Figures 8 to 11 show chimney capacity for individually vented appliances computed by the methods presented. These capacity curves may be used to estimate input or diameter for design chart (Figure 7), Equation (21) or Equation (23). These capacity curves apply primarily to individually vented appliances with a lateral chimney connector; systems with two or more appliances or additional fittings require a more detailed analysis. Figures 8 to 11 assume the length of the horizontal connector is (1) at least 10 ft and (2) no longer than 50% of the height or 50 ft, whichever is less. For chimney heights of 10 to 20 ft, a fixed 10 ft long connector is assumed. Between 20 and 100 ft, the connector is 50% of the height. If the chimney height exceeds 100 ft, the connector is fixed at 50 ft long.

For a chimney of similar configuration but with a shorter connector, the size indicated in the figures is slightly larger than necessary. In deriving the data for Figures 8 to 11, additional conservative assumptions were used, including the temperature correction $C_u$ for double-wall laterals (see Figure 3) and a constant friction factor (0.4) for all sizes.

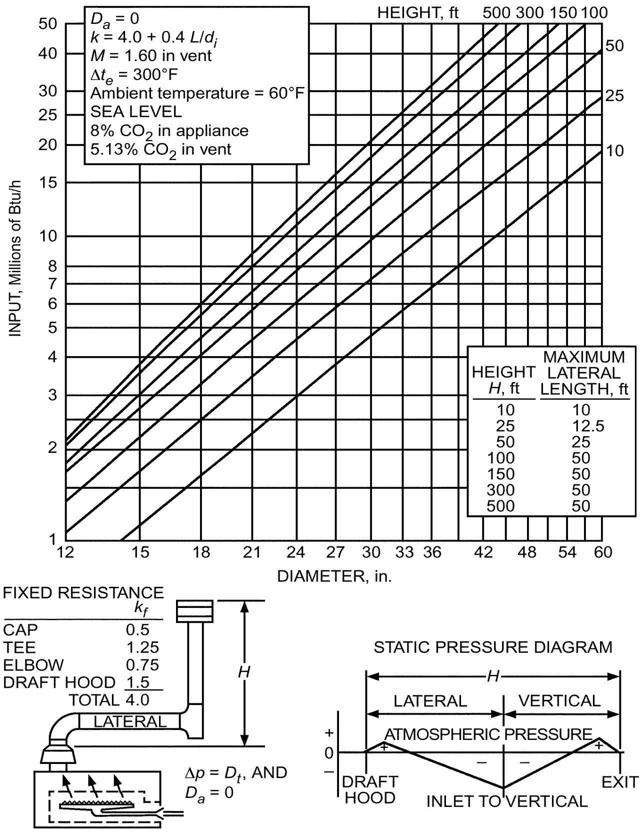


**Fig. 8 Gas Vent with Lateral**

The loss coefficient $k_4$ for a low resistance cap is included in Figures 8, 9, and 10. If no cap is installed, these figures indicate a larger size than needed.

Figure 8 applies to a gas vent with draft hood and a lateral that runs to the vertical section. Maximum static draft is developed at the base of the vertical, but friction reduces the observed value to less than the theoretical draft. Areas of positive pressure may exist at the elbow above the draft hood and at the inlet to the cap. The height of the system is the vertical distance from the draft hood outlet to the vent cap.

Figure 9 applies to a typical boiler system requiring both negative combustion chamber pressure and negative static outlet pressure. The chimney static pressure is below atmospheric pressure, except for the minor outlet reversal caused by cap resistance. Height of this system is the difference in elevation between the point of draft measurement (or control) and the exit. (Chimney draft should not be based on the height above the boiler room floor.)

Figure 10 illustrates the use of a negative static pressure connector serving a forced-draft boiler. This system minimizes flue gas leakage in the equipment room. The draft is balanced or neutral, which is similar to a gas vent, with zero draft at the appliance outlet and pressure loss $\Delta p$ equal to theoretical draft.

Figure 11 applies to a forced-draft boiler capable of operating against a positive static outlet pressure of up to 0.50 in. of water. The chimney system has no negative pressure, so outlet pressure may be combined with theoretical draft to get minimum chimney size. For chimney heights or system lengths less than 100 ft, the effect of adding 0.50 in. of water positive pressure to theoretical draft causes all curves to fall into a compressed zone. An appliance that can produce 0.50 in. of water positive forced draft (negative draft) is adequate for venting any simple arrangement with up to 100 ft of

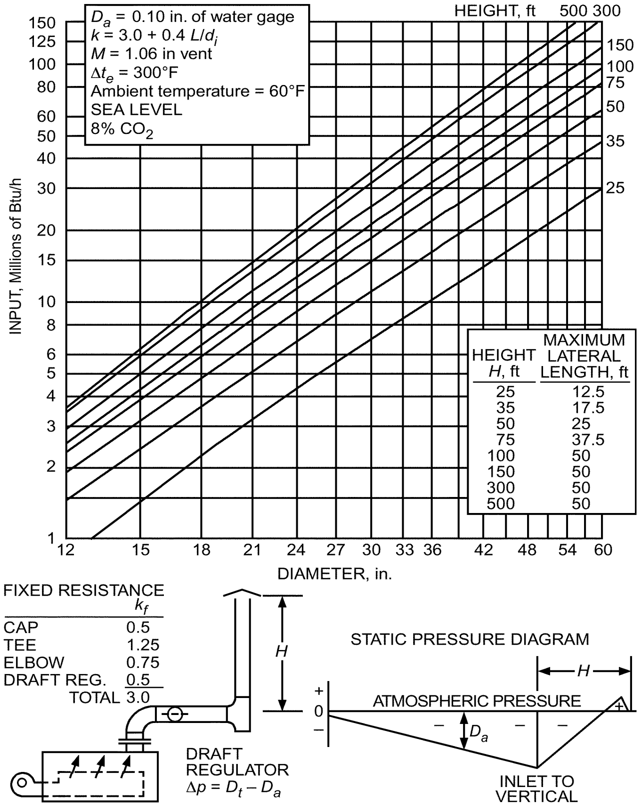


**Fig. 9 Draft-Regulated Appliance with 0.10 in. of water Available Draft Required**

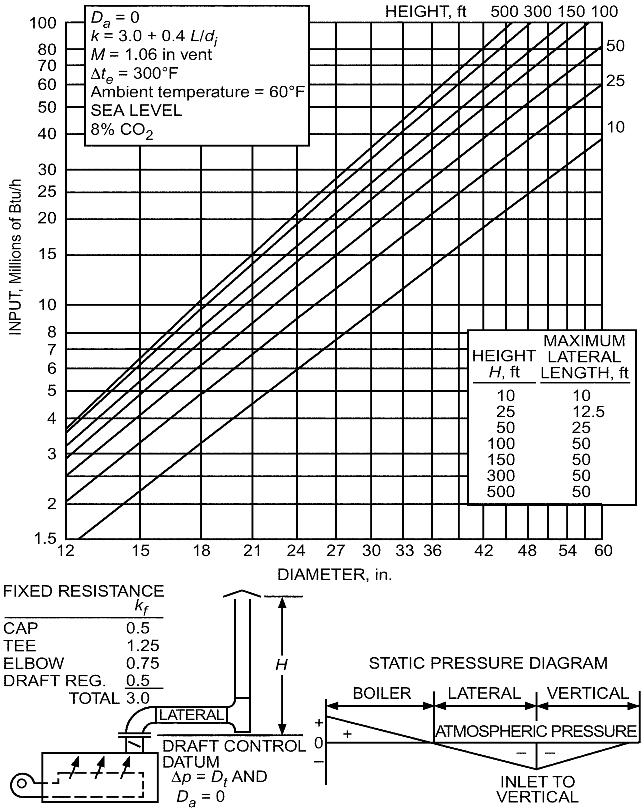


**Fig. 10 Forced-Draft Appliance with Neutral (Zero) Draft (Negative Pressure Lateral)**

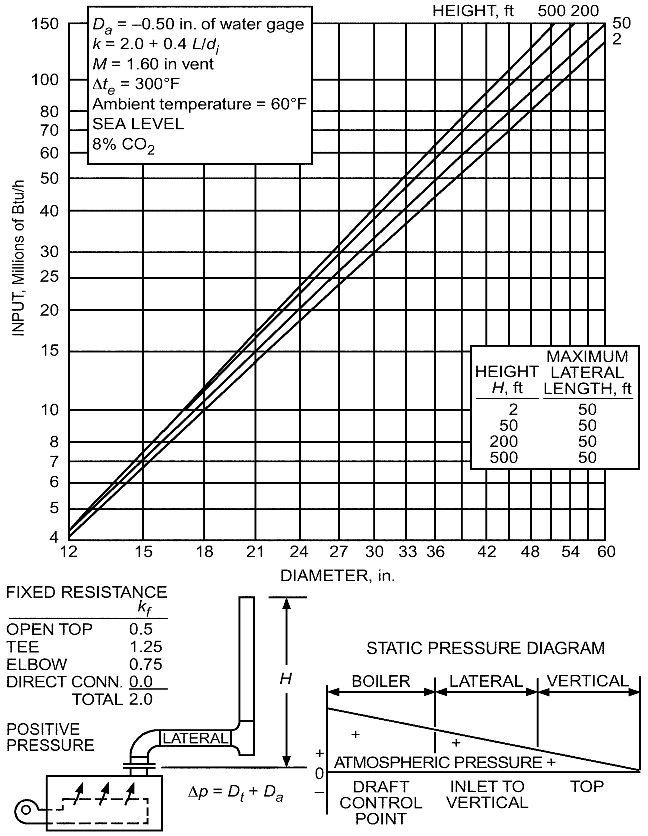


**Fig. 11 Forced-Draft Appliance with Positive Outlet Pressure (Negative Draft)**

flow path and no wind back pressure, for which additional forced draft is required.

The following examples illustrate the use of the design chart (Figure 7) and the corresponding equations.

**Example 2.** Individual gas appliance with draft hood (see Figure 12). The natural gas appliance is located at sea level and has an input of 980,000 Btu/h. The double-wall vent is 80 ft high with 40 ft lateral. Find the vent diameter.

**Solution:** Assume $k = 5.0$. The following factors are used successively in grids A through D of Figure 7: $CO_2 = 5.3\%$ for natural gas; flue gas temperature rise = 300°F; Transfer line: $k = 5.0$; $\Delta p = 0.537(80/100) = 0.43$. See Example 1 for the solution path.

**Preliminary solution:** From grid E at $I = 0.98 \times 10^6$, read diameter $d_i = 8.5$ in. Use next largest diameter, 9 in., to correct temperature rise and theoretical draft; compute new system $k$ and $\Delta p$. From Figure 3, $C_u = 0.7$, $\Delta t = (0.7)(300) = 210$°F. This temperature rise determines the new value of theoretical draft, found by interpolation between 200 and 300°F in Table 6: $D_t$ = (0.41 in. of water/100 ft)(80 ft) = 0.33 in. of water. The four fittings have a total fixed $k_f = 4.0$. Adding $k_L$, the piping component for 120 ft of 9 in. diameter, to $k_f$ gives $k = 4.0 + 0.4(120/9) = 9.3$. System losses $\Delta p = D_t = 0.33$ in. of water.

**Final solution:** Returning to Figure 7, the factors are $CO_2 = 5.3\%$ for natural gas; gas temperature rise = 210°F; Transfer line: $k = 9.3$; $\Delta p = 0.33$ in. of water. At $I = 0.98 \times 10^6$, read $d_i = 10$ in., which is the correct answer. Had system resistance been found using 10 in. rather than 9 in., the final size would be less than 10 in. based on a system $k$ of less than 9.3.

**Example 3.** Gravity incinerator chimney (see Figure 13).

Located at 8000 ft elevation, the appliance burns 600 lb/h of type 0 waste with 100% excess air at 1400°F outlet temperature. Ambient temperature $T_o$ is 60°F. Outlet pressure is zero at low fire, +0.10 in. of water at high fire. The chimney will be a prefabricated medium-heat type with a 60 ft connector and a roughness factor of 1.2. The incinerator outlet is

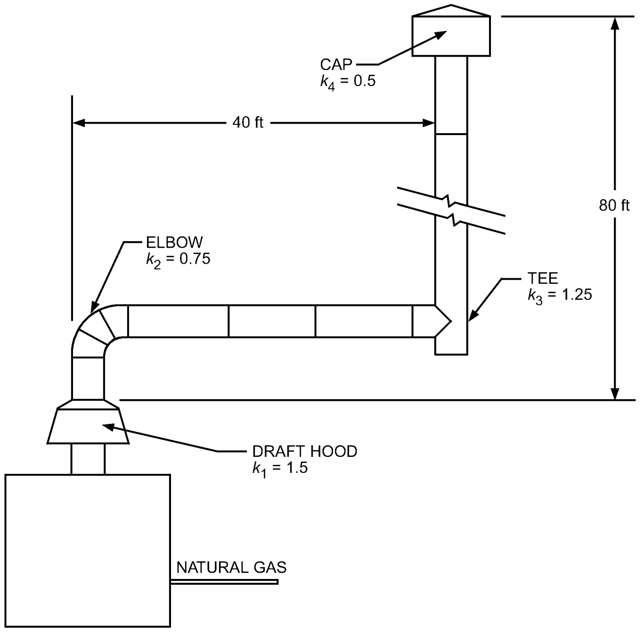


**Fig. 12 Illustration for Example 2**

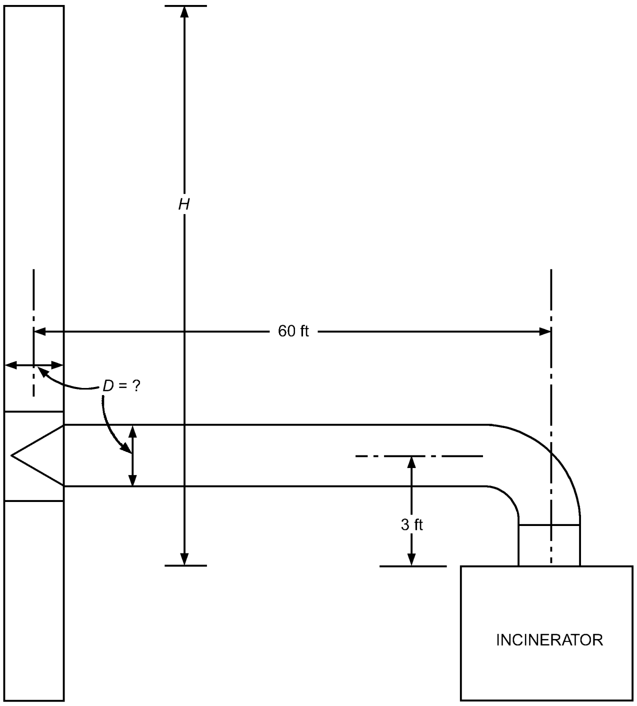


**Fig. 13 Illustration for Example 3**

18 in. in diameter, and it normally uses a 20 ft vertical chimney. Find the diameter of the chimney and the connector and the height required to overcome flow and fitting losses.

**Solution:**

1. Find mass flow from Table 3 as 13.76 lb combustion products per pound of waste, or $w = 600(13.76) = 8256$ total lb/h.
2. Find mean chimney gas temperature. Based on 60 ft length of 18 in. diameter double-wall chimney, $C_u = 0.83$ (see Figure 3). Temperature rise $\Delta t_e = 1400 - 60 = 1340$°F; thus, $\Delta t_m = 1340(0.83) = 1112$°F rise above 60°F ambient. $T_m = 1112 + 60 + 460 = 1632$°R. Use this temperature in Equation (6) to find flue gas density at 8000 ft elevation (from Table 7, $B = 22.22$ in. Hg):

$$\rho_m = 1.325\left(\frac{22.22}{1632}\right) = 0.0180 \text{ lb/ft}^3$$

3. Find the required height by finding theoretical draft per foot from Equation (9) or Figure 5 (Table 6 applies only to sea level).

$$\frac{D_t}{H} = (0.2554)(22.22)\left(\frac{1}{520} - \frac{1}{1632}\right)$$
$$= 0.0074 \text{ in. of water per ft of height}$$

4. Find allowable pressure loss $\Delta p$ in the incinerator chimney for a positive-pressure appliance having an outlet pressure of +0.1 in. of water. From Table 8, $\Delta p = D_t + D_a$, where $D_t = 0.0074H$, $D_a = 0.10$, and $\Delta p = 0.0074H + 0.1$ in. of water.
5. Calculate flow velocity at mean temperature from Equation (13) to balance flow losses against diameter/height combinations:

$$V = \frac{(0.0509)(8256)}{(0.0180)(18)^2} = 72 \text{ fps}$$

This velocity exceeds the capability of a gravity chimney of moderate height and may require a draft inducer if an 18 in. chimney must be used. Verify velocity by calculating resistance and flow losses by the following steps.

6. From Table 9, resistance coefficients for fittings are

| | |
|---|---|
| 1 Tee | $k_3 = 1.25$ |
| 1 Elbow | $k_2 = 0.75$ |
| Spark screen | $k_4 = 0.50$ |
| Fitting total | $k_f = 2.50$ |

The piping resistance, adjusted for length, diameter, and a roughness factor of 1.2, must be added to the total fitting resistance. From Figure 6, find the friction factor $F$ at 18 in. diameter and 72 fps as 0.22. Assuming 20 ft of height with a 60 ft lateral, piping friction loss is

$$k_L = \frac{1.2FL}{d_i} = \frac{(1.2)(0.22)(60+20)}{18} = 1.17$$

and total $k = 2.50 + 1.17 = 3.67$

Use Equation (11) to find $\Delta p$, to determine whether this chimney height and diameter are suitable.

$$\Delta p = \frac{(3.67)(0.0180)(72)^2}{(5.2)(64.4)} = 1.02 \text{ in. of water flow loss}$$

For these operating conditions, theoretical draft plus available draft yields

$$\Delta p = 0.0074(20) + 0.1 = 0.248 \text{ in. of water driving force}$$

Flow losses of 1.02 in. of water exceed the 0.248 in. of water driving force; thus, the selected diameter, height, or both are incorrect, and this chimney will not work. This can also be shown by comparing draft per foot with flow losses per foot for the 18 in. diameter configuration:

Flow losses per foot of 18 in. chimney = 1.02/80 = 0.0128 in. of water

Draft per foot of height = 0.0074 in. of water

Regardless of how high the chimney is, losses caused by a 72 fps velocity build up faster than draft.

7. A draft inducer could be selected to make up the difference between losses of 1.02 in. of water and the 0.248 in. of water driving force. Operating requirements are

$$Q = \frac{w}{(60\rho_m)} = \frac{8256}{(60 \times 0.0180)} = 7644 \text{ cfm}$$

$\Delta p = 1.02 - 0.248 = 0.772$ in. of water at 7644 cfm and 1112°F rise

If the inducer selected (see Figure 27C) injects single or multiple air jets into the gas stream, it should be placed only at the chimney top or outlet. This location requires no compensation for additional air introduced by an enlargement downstream from the inducer.

Because 18 in. is too small, assume that a 24 in. diameter may work at a 20 ft height and recalculate with the new diameter.

1. As before, $w = 8256$ lb/h.
2. At 24 in. diameter, no temperature correction is needed for the 60 ft connector. Thus, $T_m = 1400 + 460 = 1860°\text{R}$ (see Table 4), and density is

$$\rho_m = 1.325\left(\frac{22.22}{1860}\right) = 0.0158 \text{ lb/ft}^3$$

3. Theoretical draft per foot of chimney height is

$$\frac{D_t}{H} = 0.2554(22.22)\left(\frac{1}{520} - \frac{1}{1860}\right) = 0.00786 \text{ in. of water per ft}$$

4. Velocity is

$$V = \frac{(0.0509)(8256)}{(0.0158)(24^2)} = 46.2 \text{ fps}$$

5. From Figure 6, the friction factor is 0.225, which, when multiplied by a roughness factor of 1.2 for the piping used, becomes 0.27. For the entire system with $k_f = 2.50$ and 80 ft of piping, find $k = 2.5 + 0.27(80/24) = 3.4$. From Equation (11),

$$\Delta p = \frac{(3.4)(0.0158)(46.2)^2}{(5.2)(64.4)} = 0.342 \text{ in. of water flow losses}$$

$D_t + D_a = (20)(0.00786) + 0.1 = 0.257$ in. of water, so driving force is less than losses. The small difference indicates that, although 20 ft of height is insufficient, additional height may solve the problem. The added height must make up for 0.085 in. of water additional draft. As a first approximation,

$$\begin{aligned}\text{Added height} &= \text{Additional draft/draft per foot}\\ &= 0.085/0.00786 = 10.8 \text{ ft}\\ \text{Total height} &= 20 + 10.8 = 30.8 \text{ ft}\end{aligned}$$

This is less than the actual height needed because resistance changes have not been included. For an exact solution for height, the driving force can be equated to flow losses as a function of $H$. Substituting $H = +60$ for $L$ in Equation (17) to find $k$ for Equation (11), the complete equations are

$$k_L = \frac{(1.2 \times 0.225 \times 60)}{24} = 0.675 \text{ connector}$$

$$k_L = \frac{(1.2 \times 0.225 \times H)}{24} = 0.01125H \text{ chimney}$$

$$k = 2.5 + 0.675 + 0.01125H = 3.175 + 0.01125H$$

$$0.10 + 0.00786H = \frac{(3.175 + 0.01125H)(0.0158)(46.2)^2}{(5.2)(64.4)}$$

$H = 32.66$ ft at 24 in. diameter.

Checking by substitution, total driving force = 0.357 in. of water and total losses, based on a system with 92.66 ft of piping, equal 0.357 in. of water.

The value of $H = 32.66$ ft is the minimum necessary for proper system operation. Because of the great variation in fuels and firing rate with incinerators, greater height should be used to ensure adequate draft and combustion control. An acceptable height would be from 40 to 50 ft.

**Example 4.** Two forced-draft boilers (see Figure 14).

This example shows how multiple-appliance chimneys can be separated into subsystems. Each boiler is rated 100 boiler horsepower on No. 2 oil. The manufacturer states flue gas operation at 13.5% $CO_2$ and 300°F temperature rise against 0.50 in. positive static pressure at the outlet. The 50 ft high chimney has a 20 ft single-wall manifold and is at sea level. Find the size of connectors, manifold, and vertical.

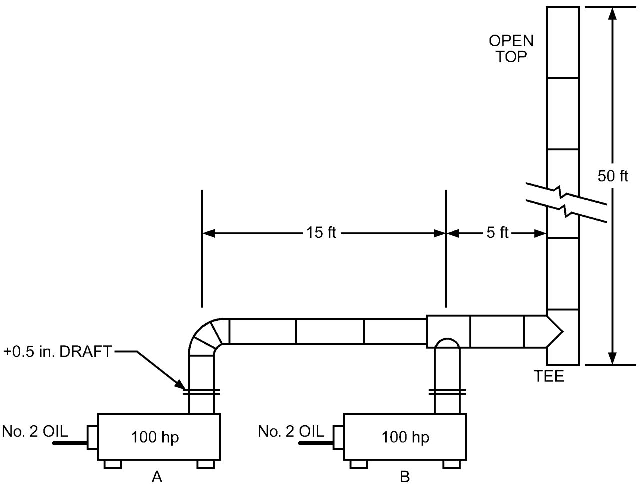


**Fig. 14 Illustration for Example 4**

**Solution:** First, find the capacity or size of the piping and fittings from boiler A to the tee over boiler B. Then, size the boiler B tee and all subsequent portions to carry the combined flow of A and B. Also, check the subsystem for boiler B; however, because its shorter length compensates for greater fitting resistance, its connector may be the same size as for boiler A.

Find the size for combined flow of A and B either by assuming $k = 5.0$ or by estimating that the size will be twice that found for boiler A operating by itself. Estimate system resistance for the combined portion by including those fittings in the B connector with those in the combined portion.

Data needed for the solution of Equation (21) for boiler A for No. 2 oil at 13.5% $CO_2$ include the following:

$$M = 0.72\left(0.12 + \frac{14.4}{13.5}\right) = 0.854 \text{ lb/1000 Btu} \qquad \text{(from Table 1)}$$

For a temperature rise of 300°F and an ambient of 60°F,

$$T_m = 300 + 60 + 460 = 820°\text{R}$$

From Table 6, theoretical draft = 0.5 in. of water for 100 ft of height; for 50 ft height, $D_t = (0.5)50/100 = 0.25$ in. of water.

Using $D_a = 0.5$ in. of water, $\Delta p = 0.25 + 0.5 = 0.75$ in. of water.

Assume $k = 5.0$. Assuming 80% efficiency, input is 41,800 times boiler horsepower (see the section on Conversion Factors):

$$I = 100(41{,}800) = 4.18 \times 10^6 \text{ Btu/h}$$

Substitute in Equation (21):

$$I = 4.18 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{0.854}\left[\frac{(0.75)(29.92)}{(5.0)(820)}\right]^{0.5}$$

Solving, $d_i = 10.81$ in. as a first approximation. From Table 9, find correct $k$ using next largest diameter, or 12 in.:

| | | |
|---|---|---|
| Inlet acceleration (direct connection) | $k_1$ | = 0.0 |
| 90° Elbow | $k_2$ | = 0.75 |
| Tee | $k_3$ | = 1.25 |
| 70 ft piping | $k_L$ | = 0.4(70/12) = 2.33 |
| System total | $k$ | = 4.33 |

*Note*: Assume tee over boiler B has $k = 0$ in subsystem A.

Corrected temperature rise (see Figure 3 for 20 ft single-wall connector) = 0.75(300) = 225°F, and $T_m = 225 + 60 + 460 = 745°\text{R}$. This corrected temperature rise changes $D_t$ to 0.425 per 100 ft [Table 6 or Equation (9)], or 0.21 for 50 ft. Thus, $\Delta p$ becomes 0.21 + 0.50 = 0.71 in. of water.

$$4.18 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{0.854}\left[\frac{(0.71)(29.92)}{(4.33)(745)}\right]^{0.5}$$

So $d_i$ = 10.35 in., or a 12 in. diameter is adequate.

For size of manifold and vertical, starting with the tee over boiler B, assume 16 in. diameter (see also Figure 11).

System $k$ = 3.9 for the 55 ft of piping from B to outlet:

| | | |
|---|---|---|
| Inlet acceleration | $k_1$ = | 0.0 |
| Two tees (boiler B subsystem) | $k_3$ = | 2.5 |
| 55 ft piping | $k_L$ = 0.4(55/16) = | 1.4 |
| System total | $k$ = | 3.9 |

Temperature and $\Delta p$ will be as corrected (a conservative assumption) in the second step for boiler A. Having assumed a size, find input.

$$I = 4.13 \times 10^5 \frac{16^2}{0.854}\left[\frac{(0.71)(29.92)}{(3.9)(745)}\right]^{0.5} = 10.59 \times 10^6 \text{ Btu/h}$$

A 16 in. diameter manifold and vertical is more than adequate. Solving for the diameter at the combined input, $d_i$ = 14.2 in.; thus, a 15 or 16 in. chimney must be used.

*Note*: Regardless of calculations, do not use connectors smaller than the appliance outlet size in any combined system. Applying the temperature correction for a single-wall connector has little effect on the result because positive forced draft is the predominant motive force for this system.

**Example 5.** Six gas boilers manifolded at 6000 ft elevation (see Figure 15).

Each boiler is fired at 1.6 × $10^6$ Btu/h, with draft hoods and an 80 ft long manifold connecting into a 400 ft high vertical. Each boiler is controlled individually. Find the size of the constant-diameter manifold, vertical, and connectors with a 2 ft rise. All are double-wall.

**Solution:** Simultaneous operation determines both the vertical and manifold sizes. Assume the same appliance operating conditions as in Example 1: $CO_2$ = 5.3%, natural gas, flue gas temperature rise = 300°F. Initially assume $k$ = 5.0 is multiplied by 1.5 for combined vent (see note at bottom of Table 9); thus, design $k$ = 7.5. For a gas vent at 400 ft height, $\Delta p = D_t = H \times D_t/m = 4 \times 0.5 = 2.0$ in. of water at rise 300°F in Table 6; $D_t$ = 0.5 in. of water/ft. At 6000 ft elevation, operating input must be multiplied by an altitude correction (Table 7) of 1.25. Total design input is 1.6 × $10^6$(6)(1.25) = 12 × $10^6$ Btu/h. From Table 2, $M$ = 1.54 lb/Btu at operating conditions. $T_m$ = 300 + 60 + 460 = 820°R, and $B$ = 29.92 in. Hg because the 1.25 input multiplier corrects back to sea level. From Equation (21),

$$12 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{2.0 \times 29.92}{7.5 \times 820}\right)^{0.5}$$

$$d_i = 21.3 \text{ in.}$$

Because the diameter is greater than 20 in., no temperature correction is needed.

Recompute $k$ for 400 + 80 = 480 ft of 22 in. diameter (Table 9):

| | | |
|---|---|---|
| Draft hood inlet acceleration | $k_1$ = | 1.5 |
| Two tees (connector and base of chimney) | $2k_3$ = | 2.5 |
| Low-resistance top | $k_4$ = | 0.5 |
| 480 ft piping | $k_L$ = 0.4(480/ 22) = | 8.7 |
| System total | | $k$ = 13.2 |

Combined gas vent design $k$ = multiple vent factor 1.5 × 13.2 = 19.8. Substitute again in Equation (21):

$$12 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{2.0 \times 29.92}{19.8 \times 820}\right)^{0.5}$$

$d_i$ = 27.1 in.; thus use 28 in.

$$\begin{aligned} k_f &= 4.5 \\ k_L = 0.4(480/28) &= 6.9 \\ \hline & \quad 11.4 \end{aligned}$$

Design $k$ = 11.4(1.5) = 17.1

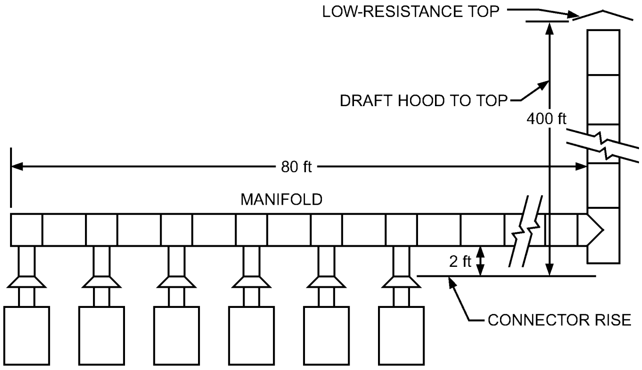


**Fig. 15 Illustration for Example 5**

Substitute in Equation (21) to obtain the third trial:

$$12 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{2.0 \times 29.92}{17.1 \times 820}\right)^{0.5}$$

$$d_i = 26.2 \text{ in.}$$

The third trial is less than the second and again shows the manifold and vertical chimney diameter to be between 26 and 28 in.

For connector size, see the *National Fuel Gas Code* for double-wall connectors of combined vents. The height limit of the table is 100 ft; do not extrapolate and read the capacity of 18 in. connector as 1,740,000 Btu/h at 2 ft rise. Use 18 in. connector or draft hood outlet, whichever is larger. No altitude correction is needed for connector size; the draft hood outlet size considers this effect.

*Note*: Equation (21) can also be solved at local elevation for exact operating conditions. At 6000 ft, the local barometric pressure is 23.98 in. Hg (Table 7), and assumed theoretical draft must be corrected in proportion to the reduction in pressure:

$D_t$ = 2.0(23.98/29.92) = 1.60 in. of water. Operating input of 6(1.6 × $10^6$) = 9.6 × $10^6$ Btu/h is used to find $d_i$, again taking final $k$ = 17.1:

$$9.6 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{1.60 \times 23.98}{17.1 \times 820}\right)^{0.5}$$

$d_i$ = 26.2 (same as above)

This example illustrates the equivalence of the chart method of solution with solution by Equation (21). Equation (21) gives the correct solution using either method 1, with only the fuel input corrected back to sea level condition, or method 2, correcting $\Delta p$ for local barometric pressure and using operating input at altitude. Method 1, correcting input only, is the only choice with Figure 1 because the design chart cannot correct to local barometric pressure.

**Example 6.** Pressure boost for undersized chimney (not illustrated).

A natural gas boiler at sea level (no draft hood) is connected to an existing 12 in. diameter chimney. Input is 4 × $10^6$ Btu/h with flue gas at 10% $CO_2$ and 300°F temperature rise above ambient. System resistance loss coefficient $k$ = 5.0 with 20 ft vertical chimney. The appliance operates with neutral outlet static draft, so, $D_a$ = 0 in. of water.

*a*. How much draft boost is needed at operating temperature?
*b*. What fan rating is required at 60°F ambient temperature?
*c*. Where in the system should the fan be located?

**Solution:** Combustion data: from Figure 2, 10% $CO_2$ at 300°F temperature rise indicates 0.31 cfm per 1000 Btu/h.

Total flow rate $Q$ = (0.31/1000)(4 × $10^6$) = 1240 cfm at chimney gas temperature. Then, Equation (23) can be solved for the only unknown, $\Delta p$:

$$1240 = 5.2(12)^2\left[\frac{\Delta p(300 + 60 + 460)}{(5.0)(29.92)}\right]^{0.5}$$

$\Delta p$ = 0.50 in. of water needed at 300°F temperature rise. For 20 ft of height at 300°F temperature rise [Table 6 or Equation (9)],

$D_t = 0.5(20/100) = 0.10$ in. of water

*a*. Pressure boost $D_b$ supplied by the fan must equal $\Delta p$ minus theoretical draft (Table 8) when available draft is zero. $D_b = \Delta p - D_t = 0.5 - 0.10 = 0.40$ in. of water at operating temperature.

*b*. Draft fans are usually rated for standard ambient (60°F) air. Pressure is inversely proportional to absolute flue gas temperature. Thus, for ambient air,

$$D_b = 0.40\frac{T_m}{T_o} = 0.40\left(\frac{300 + 60 + 460}{60 + 460}\right) = 0.63 \text{ in. of water}$$

This pressure is needed to produce 0.40 in. of water at operating temperature. In specifying power ratings for draft fan motors, a safe policy is to select one that operates at the required flow rate at ambient temperature and pressure (see Example 7).

*c*. A fan can be located anywhere from boiler outlet to chimney outlet. Regardless of location, the amount of boost is the same; however, chimney pressure relative to atmosphere will change. At boiler outlet, the fan pressurizes the entire connector and chimney. Thus, the system should be gastight to avoid leaks. At the chimney outlet, the system is below atmospheric pressure; any leaks flow into the system and seldom cause problems. With an ordinary sheet metal connector attached to a tight vertical chimney, the fan may be placed close to the vertical chimney inlet. Thus, the connector leaks safely inward, while the vertical chimney is under pressure.

**Example 7.** Draft inducer selection (see Figure 16).

A third gas boiler must be added to a two-boiler system at sea level with an 18 in. diameter, 15 ft horizontal, and 75 ft of total height of connector and chimney system. Outlet conditions for natural gas draft hood appliances are 5.3% $CO_2$ at 300°F temperature rise. Boilers are controlled individually, each with $1.6 \times 10^6$ Btu/h, for $4.8 \times 10^6$ Btu/h total input. The system is currently undersized for gravity full-load operation. Find capacity, pressure, size, and power rating of a draft inducer fan installed at the outlet.

**Solution:** Using Equation (23) requires evaluating two operating conditions: (1) full input at 300°F rise and (2) no input with ambient air. Because the boilers are controlled individually, the system may operate at nearly ambient temperature (100°F flue gas temperature rise or less) when only one boiler operates at part load. Use the system resistance $k$ for boiler 3 as the system value for simultaneous operation. It needs no compensating increased draft, as with gravity multiple venting, because a fan induces flow at all flue gas temperatures. From Table 9, the resistance summation is

| | |
|---|---|
| Inlet acceleration (draft hoods) | $k_1 = 1.50$ |
| Tee above boiler | $k_3 = 1.25$ |
| Tee at base of vertical | $k_3 = 1.25$ |
| 90 ft of 18 in. diameter pipe | $k_L = 0.4(90/18) = 2.00$ |
| System total | $k = 6.00$ |

At full load, $T_m = 300 + 60 + 460 = 820$°R; $B = 29.92$ in. Hg. Flow rate $Q$ must be found for operating conditions of 1.54 lb per 1000 Btu (Table 2) at density $\rho_m$ and full input $(4.8 \times 10^6)/60$ Btu/min.

From Equation (6),

$$\rho_m = 1.325B/T_m = 1.325(29.92/820) = 0.0483 \text{ lb/ft}^3$$

Volumetric flow rate is

$$Q = \left(\frac{1.54}{1000}\right)\left(\frac{1}{0.0483}\right)\left(\frac{4.8 \times 10^6}{60}\right) = 2551 \text{ cfm}$$

For 300°F flue gas temperature rise, $D_t = 0.5(75/100) = 0.375$ in. of water theoretical draft in the system (Table 6). Solving Equation (23) for $\Delta p$,

$$\Delta p = \frac{(6.00)(29.92)(2551)^2}{(5.2)^2(18)^4(820)} = 0.502 \text{ in. of water}$$

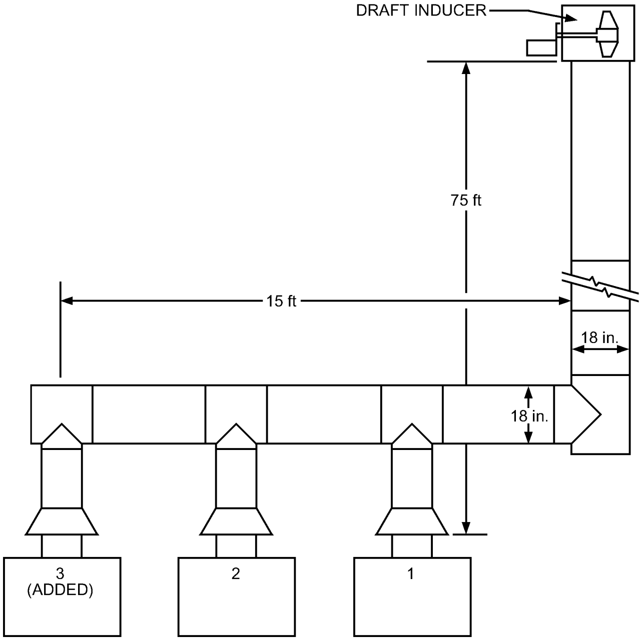


**Fig. 16 Illustration for Example 7**

Thus, a fan is needed because $\Delta p$ exceeds $D_t$. Required static pressure boost (Table 8) is

$$D_b = 0.502 - 0.375 = 0.127 \text{ in. of water at 300°F flue gas temperature rise or a flue gas density of } 0.0483 \text{ lb/ft}^3$$

Fans are rated for ambient or standard air (60 to 70°F) conditions. Flue gas pressure is directly proportional to density or inversely proportional to absolute temperature. Moving 2551 cfm of flue gas at 300°F temperature rise against 0.127 in. of water pressure requires the static pressure boost with standard air to be

$$D_b = 0.127(820/520) = 0.20 \text{ in. of water}$$

Thus, a fan that delivers 2551 cfm of flue gas at 0.20 in. of water at 60°F is required. Figure 17 shows the operating curves of a typical fan that meets this requirement. The exact volume flow rate and pressure developed against a system $k$ of 6.0 can be found for this fan by plotting airflow rate versus $\Delta p$ from Equation (23) on the fan curve. The solution, at point C, occurs at 1950 cfm, where both the system $\Delta p$ and fan static pressure equal 0.46 in. of water.

Although some fan manufacturers' ratings are given at standard air conditions, the motors selected will be overloaded at temperatures below 300°F air temperature rise. Figure 17 shows that power required for two conditions with ambient air is as follows:

1. 1950 cfm at 0.46 in. of water static pressure requires 0.51 hp
2. 2650 cfm at 0.25 in. of water static pressure requires 0.50 hp

Thus, the minimum size motor will be 0.5 hp and run at 1590 rpm.

Manufacturers' literature must be analyzed carefully to discover whether the sizing and selection method is consistent with appliance and chimney operating conditions. Final selection requires both a thorough analysis of fan and system interrelationships and consultation with the fan manufacturer to verify the fan and motor capacity and power ratings.

## GAS APPLIANCE VENTING

In much of the United States, gas-burning appliances requiring venting of combustion products are installed and vented in accordance with the *National Fuel Gas Code* (ANSI/NFPA 54/ANSI/AGA Z223.1), and in Canada with the *Natural Gas and Propane Installation Code* (CAN/CSA *Standard* B149.1). This

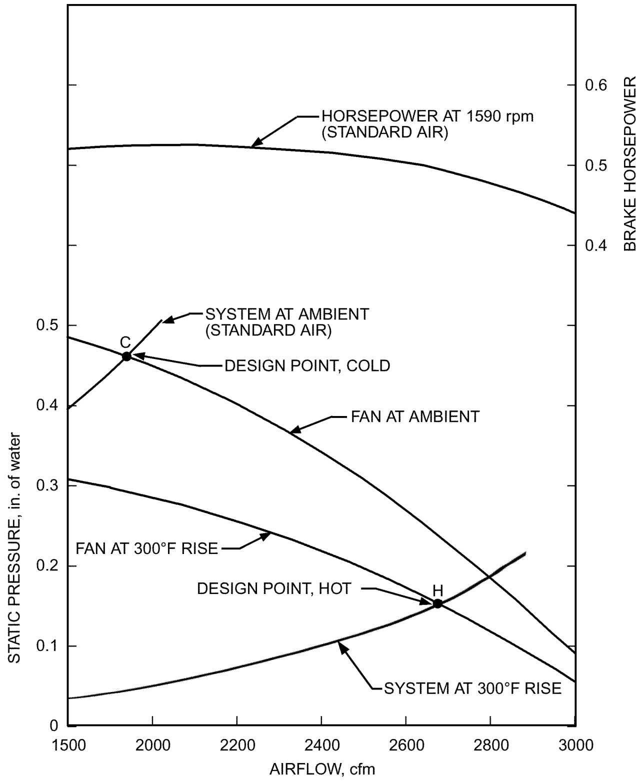


**Fig. 17 Typical Fan Operating Data and System Curves**

standard includes capacity data and definitions for commonly used gas vent systems.

Traditionally, gas appliances were designed with a draft hood or draft diverter and depended on natural buoyancy to vent products of combustion. The operating characteristics of many different types of appliances were similar, allowing generic venting guidelines to be applied to any gas appliance. These guidelines consisted of tables of maximum capacities, representing the largest appliance input rating that could safely be vented using a certain vent diameter, height, lateral, and material.

Small gas-fired appliance design changed significantly, however, with the increase of furnace minimum efficiency requirements to 78% annual fuel utilization efficiency (AFUE) isolated combustion system (ICS) imposed by the U.S. National Appliance Energy Conservation Act (NAECA) of 1987. Many manufacturers developed appliances with fan-assisted combustion systems (FACSs) to meet efficiency standards, which greatly changed the venting characteristics of gas appliances.

Because FACS appliances do not entrain dilution air, a FACS appliance connected to a given vent system can have a larger maximum capacity than a draft-hood-equipped appliance connected to the same vent system. The dew point of vent gases from a FACS appliance, however, remains high in the absence of dilution air, and the potential for condensation to form and remain in the vent system is much greater than with draft-hood-equipped appliances. Excessive condensate dwell time (wet time) in the vent system can cause corrosion failure of the vent system or problems with condensate runoff, either into the appliance or into the structure surrounding the vent.

New venting guidelines (*National Fuel Gas Code* and *Natural Gas and Propane Installation Code*) were developed to address the differences between draft-hood-equipped and FACS category I appliances. In addition to new maximum capacity values for FACS appliances, the guidelines include minimum capacity values for FACS appliances to ensure that vent system wet times do not reach a level that would lead to corrosion or drainage problems. Because corrosion is the principal effect of condensation in the vent connector, whereas drainage is the principal concern in the vertical portion of the vent, different wet time limits were established for the vent connector and the vertical portion of the vent.

Wet time values in the vent system were determined using the VENT-II computer program (Rutz and Paul 1991) to perform the transient analysis with the appliance(s) cycling. Gas-fired central furnace cycle times of 3.87 min on, 13.3 min off, 17.17 min total, were determined based on a design outdoor ambient of 42°F with an oversize factor of 1.7. The vent connector is required to dry before the end of the appliance *on* cycle, whereas the vertical portion of the vent is required to dry out before the end of the total cycle.

The new venting guidelines for FACS appliances severely restrict use of single-wall metal vent connectors. Also, because of excessive condensation, tile-lined masonry chimneys are not recommended in most typical installations when a FACS appliance is the only appliance connected to the venting system. FACS appliances can be used with a typical masonry chimney, however, if (1) the FACS appliance is common-vented with a draft-hood-equipped appliance or (2) a liner listed for use with gas appliances is installed in the masonry chimney.

The *National Fuel Gas Code* (NFGC) and *Natural Gas and Propane Installation Code* list both the minimum and maximum vent capacities for FACS gas appliances and the previous maximum vent capacities for gas appliances equipped with draft hoods. NFGC sections 11.2.9 and 11.3.18 of the 1999 revision, sections 13.1.9 and 13.2.20 of the 2002 revision, and sections 13.1.11 and 13.2.23 of the 2006 revision provide criteria for permitting use of masonry chimneys (either exposed or not exposed to the outdoors below the roof line) and allow alternative venting designs and installation of vents serving listed appliances in accordance with the appliance manufacturer's instructions and the terms of listing.

## Vent Connectors

Vent connectors connect gas appliances to the gas vent, chimney, or single-wall metal pipe, except when the appliance flue outlet or draft hood outlet is connected directly to one of these vent systems. Materials for vent connectors for conversion burners or other appliances without draft hoods must have resistance to corrosion and heat not less than that of galvanized 0.0276 in. (24 gage) thick sheet steel. Where a draft hood is used, the connector must have resistance to corrosion and heat not less than that of galvanized 0.0187 in. (28 gage) thick sheet steel or type B vent material.

## Masonry Chimneys for Gas Appliances

A masonry chimney serving a gas-burning appliance should have a tile liner and should comply with applicable building codes such as NFPA *Standard* 211. *National Fuel Gas Code*, sections 11.2.9 and 11.3.18 of the 1999 revision, sections 13.1.9 and 13.2.20 of the 2002 revision, and sections 13.1.11 and 13.2.23 of the 2006 revision have other provisions pertaining to masonry chimneys. An additional chimney liner may be needed to avoid slow priming and/or condensation, particularly for an exposed masonry chimney with high mass and low flue gas temperature. A low-temperature chimney liner may be a single-wall passage of pure aluminum or stainless steel or a double-wall type B vent.

## Type B and Type L Factory-Built Venting Systems

Factory-prefabricated vents are listed by Underwriters Laboratories for use with various types of fuel-burning appliances. These vents should be installed according to the manufacturer's instructions and the appliance's listing.

Type B gas vents are listed for vented gas-burning appliances. They should not be used for incinerators, appliances readily converted to the use of solid or liquid fuel, or combination gas-oil burning appliances. They may be used in multiple gas appliance venting systems.

Type BW gas vents are listed for vented wall furnaces certified as complying with the pertinent ANSI standard.

Type L venting systems are listed by Underwriters Laboratories in 3 to 6 in. sizes and may be used for those oil- and gas-burning appliances (primarily residential) certified or listed as suitable for this type of venting. Under the terms of the listing, a single-wall connector may be used in open accessible areas between the appliance's outlet and the type L material in a manner analogous to type B. Type L piping material is recognized in the *National Fuel Gas Code* and NFPA *Standard* 211 for certain connector uses between appliances such as domestic incinerators and chimneys.

### Gas Appliances Without Draft Hoods

Figure 1 or the equations may be used to calculate chimney size for nonresidential gas appliances with the draft configurations listed as 1 and 3 at the beginning of this chapter. Draft conditions 1 and 3 (see draft conditions 1, 2, and 3 under Terminology and Table 8) for residential gas appliances, such as boilers and furnaces, may require special vent systems. The appliance test and certification standards include evaluation of manufacturers' appliance installation instructions (including the vent system) and of operating and application conditions that affect venting. The instructions must be followed strictly.

### Conversion to Gas

Installation of conversion burner equipment requires evaluating for proper chimney draft and capacity by the methods indicated in this chapter or by conformance to local regulations. The physical condition and suitability of an existing chimney must be checked before it is converted from a solid or liquid fuel to gas. For masonry chimneys, local experience may indicate how well the construction will withstand the lower temperature and higher moisture content of natural or liquefied petroleum gas combustion products. The section on Masonry Chimneys for Gas Appliances has more details.

The chimney should be relined, if required, with corrosion-resistant masonry or metal to prevent deterioration. The liner must extend beyond the top of the chimney. The chimney drop-leg (bottom of the chimney) must be at least 4 in. below the bottom of the connection to the chimney. The chimney should be inspected and, if needed, cleaned. The chimney should also have a cleanout at the base.

## OIL-FIRED APPLIANCE VENTING

Oil-fired appliances requiring venting of combustion products must be installed and vented in accordance with ANSI/NFPA *Standard* 31. The standard offers recommendations for metal relining of masonry chimneys.

Recommendations for minimum chimney areas for oil-fired natural-draft appliances are offered in Tables 3 and 4 in the HYDI (1989).

Implementation of the U.S. National Appliance Energy Conservation Act (NAECA) of 1987 brought attention to heating appliance efficiency and the effect of NAECA on existing chimney systems. Oil-fired appliances maintained a steady growth in efficiency since the advent of the retention-head oil burner and its broad application in both new appliances and the replacement of older burners in existing appliances. Higher appliance efficiencies brought about lower flue gas temperatures. Reduced firing rates became more common as heating appliances are more closely matched to the building heating load. Burner excess air levels also dropped, which resulted in lower mass flows through the chimney and additional reductions in the flue gas temperature. However, the improvements in overall appliance efficiency have not been accompanied by upgrades in existing chimney systems, and upgraded systems are not commonly applied in new construction. An upgrade in a vent system probably involves application of corrosion-resistant materials and/or the reduction in heat loss from the vent system to maintain draft and reduce condensation on interior surfaces of the vent system.

### Condensation and Corrosion

Condensation and corrosion within the vent system are of growing concern as manufacturers of oil-fired appliances strive to improve equipment efficiencies. The conditions for condensation of the corrosive components of the flue gas produced in oil-fired appliances involve a complex interaction of the water formed in the combustion process and the sulfur trioxide formed from small quantities of sulfur in the fuel oil. The sulfur is typically less than 0.5% by mass for no. 2 fuel oil. The dew point of the two-component system (sulfuric acid and water) in the flue gas resulting from the combustion of this fuel is about 225 to 240°F. This is similar to the effect on dew point of fuel gas sulfur (see Figure 4 in Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals*). In determining the proper curve for fuel oil in that figure, a value of 18 may be used. This value applies to no. 2 fuel oil with 0.5% sulfur by mass.

The effect of post-combustion air dilution on the dew point characteristics of flue gas from oil-fired appliances is highly dependent on the presence of sulfur in the fuel. Verhoff and Banchero (1974) developed an equation relating flue gas dew point to the partial pressures of both water and sulfuric acid present in the flue gas. Predictions obtained using this equation are in good agreement with experimental data. Applying this equation to the flue gas from the combustion of no. 2 fuel oil without sulfur and with varying sulfur contents reveals that a broad range of dew point temperatures is possible.

For a fuel oil with zero sulfur and 20% excess combustion air, dew point of the combustion products is calculated as approximately 114°F, typical for the presence of water formed in combustion. With the addition of post-combustion dilution air to a level equivalent to increasing the excess air in the flue gas to 100%, the apparent water dew point is reduced to 99°F. For fuel oil with 0.25% sulfur and 20% excess combustion air, the dew point is elevated to 229°F because of formation of dilute sulfuric acid in the flue gas. With the addition of post-combustion dilution air, the dew point is 237°F at 100% excess air and 234°F at 200% excess air. The implication of these calculations is that, for combustion of fuel oil, the flue gas dew-point temperature can range from a low at the apparent water dew point (calculated with no fuel sulfur) up to some elevated dew point caused by the presence of sulfur in the fuel. With no sulfur in the fuel, adding post-combustion dilution air to the flue gas has some effect on depressing the apparent water dew-point temperature. Adding post-combustion dilution air has no significant effect on the elevated flue gas dew point when sulfur is present in the fuel.

It is difficult to meet the flue-gas-side material surface temperatures required to exceed the elevated dew point at all points in the venting system of an oil-fired appliance. Even if the dew-point temperature is not reached at these surfaces, however, a surface temperature approaching 200°F allows condensation of sulfuric acid at higher concentrations and results in lower rates of corrosion.

The condensed acid concentration is critical to the applicability of plain carbon and stainless steels in vent connectors and chimney liners. The flue-gas-side surface temperatures of conventional connectors and masonry chimney tile liners are often at or below the dew point for some portion of the burner *on* period. During cooldown (burner *off* period), these surface temperatures can drop to below the apparent water dew point and, in some cases, to ambient conditions. This is of great concern because, while system surfaces are below the apparent water dew point during burner operation, the condensed sulfuric acid is formed in concentrations well below limits acceptable for steel connectors. This can be seen from an interpretation of a sulfuric acid/water phase diagram presented by Land (1977). An estimate of the condensed liquid sulfuric acid concentration at a condensing surface temperature of 120°F, for example, shows a concentration for the condensed liquid acid of about 10 to 20%.

According to Fontana and Greene (1967), the relative corrosion of plain carbon steel rises rapidly at sulfuric acid concentrations below 65%. According to Land (1977), for the condensed liquid acid concentration to rise above 65%, the condensing surface temperature must be above 200°F. However, according to Fontana and Greene, at acid concentrations above 65%, corrosion rates increase at metal surface temperatures above 175°F. This presents the designer with a restrictive operating range for steel surfaces (i.e., between 175 and 200°F). This is a compromise that does not completely satisfy either the acid concentration or the metal temperature criterion, but should minimize the corrosion rates induced by each.

Another important phenomenon is that when vent system surfaces are at or below the apparent water dew point, a large amount of water condensation occurs on these surfaces. This condensate contains, in addition to sulfuric acid, quantities of sulfurous, nitrous, carbonic, and hydrochloric acids. Under these conditions, the corrosion rate of commonly used vent materials is severe. Koebel and Elsener (1989) found that corrosion rates increase by a factor of 10 when material temperatures on the flue gas side are allowed to fall below the apparent water dew point.

The applicability of ordinary stainless steels is very limited and generally follows that of plain carbon steel. Nickel-molybdenum and nickel-molybdenum-chromium alloys show good corrosion resistance over a wide range of sulfuric acid concentrations for surface temperatures up to 220°F. This is also true for high-silicon cast iron, used in heat exchangers for oil-fired appliances, and which might find application as a liner system for masonry and metal chimneys.

## Connector and Chimney Corrosion

Water and acid condensation can each result in corrosion of the connector wall and deterioration of the chimney material. Although there is little documentation of specific failures, concern in the industry is growing. The volume of anecdotal information regarding corrosive failures is significant and well supported by findings from the heating industry.

For oil-fired appliances, the rate of acid corrosion in the connector and chimney is a function of two groups of contributing factors: combustion factors and operational factors. The sulfur content of the fuel and the percent excess combustion air are the major combustion-related factors. The frequency and duration of equipment *on* and *off* periods, draft control dilution air, and rate of heat loss from the vent system are the major operational factors. In terms of combustion, Butcher and Celebi (1993) found a direct correlation of acid deposition (condensation) rate and subsequent corrosion to fuel sulfur content and excess combustion air. In general, reductions in fuel sulfur and excess combustion air reduce the amount of sulfuric acid produced in combustion and delivered to condensing surfaces in the appliance heat exchanger and carried over into the vent system. From an operational standpoint, long equipment *on* and short *off* periods and low vent system heat loss result in shorter warm-up transients and higher end-point temperatures for surfaces exposed to the flue gas. Within the limits of frictional loss, reduced vent sizes increase flue gas velocities and vent surface temperatures.

## Vent Connectors

An oil-fired appliance is commonly connected to a chimney through a connector pipe. Generally, a draft control device in the form of a barometric damper is included as a component part of the connector assembly. With the advent of new power burners having high static pressure capability, draft control devices in the vent system have become less important, although many local codes still require their use. The portion of the connector assembly between the appliance flue collar and the draft control is called the flue connector; the portion between the draft control and the chimney is called the stack or chimney connector.

Chimney connectors are usually of single-wall galvanized steel. The required wall thickness for these connectors varies as a function of pipe diameter. For example, in accordance with Table 5-2.2.3 in NFPA *Standard* 211, the material thickness for galvanized steel pipe connectors between 6 and 10 in. in diameter is set at 0.023 in. (24 gage).

In accordance with the 2006 *National Fuel Gas Code*, Paragraph 12.11.2.4(2), vent, chimney, stack, and flue connectors should not be covered with insulation except listed insulated connectors installed according to the terms of their listing. Because single-wall connectors must remain uninsulated for inspection, substantial cooling of the connector wall and the flue gas can occur, especially with long connector runs through spaces with low ambient temperatures. Close examination of the connector joints, seams, and surfaces is essential whenever the heating appliance is serviced. If the connector is left unrepaired, corrosion damage can cause a complete separation failure of the connector and leakage of flue gas into the occupied space.

Where corrosion in the connector has proven to be a chronic problem, consider replacing the connector with a type L vent pipe or its listed equivalent. One product configuration consists of connector pipe with a double wall (stainless steel inner and galvanized steel outer with 0.25 in. gap). The insulated gap of this type of double-wall connector elevates inner wall surface temperature and reduces the overall connector heat loss.

## Masonry Chimneys for Oil-Fired Appliances

A masonry chimney serving an oil-fired appliance should have a tile liner and should comply with applicable building codes such as NFPA *Standard* 211. An additional listed chimney liner may be needed to improve thermal response (warm-up) of the inner chimney surface, thereby reducing transient low draft during start-up and acid/water condensation during cyclic operation. This is particularly true for exposed exterior high-mass chimneys but does not exclude cold interior chimneys serving oil-fired appliances that produce relatively low flue gas temperatures. Application of insulation around tile liners within masonry chimneys is common in Europe and may be worth considering in chimney replacement or new construction.

A computational analysis by Krajewski (1996) using OHVAP (Version 3.1) to analyze a series of masonry chimney systems with various firing rates and exit temperatures revealed that current applications of modern oil-fired heating appliances may have problems with acid/water condensation during winter operation. For residential oil-fired heating appliance firing rates below 1.25 to 1.5 gph with flue-loss steady-state efficiencies of 82% or higher, exterior masonry chimneys may need special treatment to reduce condensation. For conservative design, listed chimney liners and listed type L connectors may be required for some exterior chimneys serving equipment operating under these conditions.

## Replacement of Appliances

The physical condition and suitability of an existing chimney must be checked before installation of a new oil-fired appliance. The chimney should be inspected and, if necessary, cleaned. In accordance with NFPA *Standard* 211, Section 3-2.6, the chimney drop-leg (bottom of the chimney flue) must be at least 8 in. below the bottom of the appliance connection to the chimney. The liner must be continuous, properly aligned, and intact and must extend beyond the top of the chimney. The chimney should also have a properly installed, reasonably airtight clean-out at the base.

For masonry chimneys, local experience may indicate how well the construction has withstood the lower temperatures produced by a modern oil-fired appliance. Evidence of potential or existing chimney damage can be procured by visual examination. Exterior indicators such as missing or loose mortar/bricks, white deposits (efflorescence) on brickwork, a leaning chimney, or water stains on interior building walls should be investigated further. Interior chimney examination with a mirror or video camera can reveal damaged or missing liner material. Any debris collected in the chimney base, drop-leg, or connector should be removed and examined. If any doubt exists about the chimney's condition, examination by an experienced professional is recommended. Kam et al. (1993) offer specific guidance on the examination and evaluation of existing masonry chimneys in the field.

## FIREPLACE CHIMNEYS

This section is condensed from an *ASHRAE Journal* article; for more details, please see Stone (1969).

Fireplaces with natural-draft chimneys follow the same gravity fluid flow law as gas vents and thermal flow ventilation systems. All thermal or buoyant energy is converted into flow, and no draft exists over the fire or at the fireplace inlet. Formulas have been developed to study a wide range of fireplace applications, but the material in this section covers general cases only.

Mass flow of hot flue gases through a vertical pipe is a function of rate of heat release and the chimney area, height, and system pressure loss coefficient $k$. The flow induced in a vertical pipe has a limiting value. A fireplace may be considered as a gravity duct inlet fitting with a characteristic entrance-loss coefficient and an internal heat source. A fireplace functions properly (does not smoke) when adequate intake or face velocity across those critical portions of the frontal opening nullifies external drafts and internal convection effects.

In a fireplace-chimney system, the equations assume that all potential buoyant energy is converted into flow as controlled by various losses. This system is analogous to a gas venting system with a draft hood, thereby allowing use of similar concepts as a starting point for size or capacity analysis. The amount of available draft ahead of the fireplace opening is insignificant and need not be considered. Because chimney efficiency, by one definition, equals available draft divided by theoretical draft, the numerical efficiency value approaches zero. Thus, the flow conversion basis is preferable for design over the efficiency approach.

System parameters for preventing flue gas spillage from a draft hood or similar collection fitting, can be computed with considerable certainty when heat input is constant or cannot exceed a predictable limiting value. Fireplaces, however, can be fired at extremes ranging from smoldering embers to flash fires of newspapers or dry kindling. Normal opening width and length allow greater access of combustion air to fires than a typical chimney can carry away; thus, combustion overloading occasionally leads to some smoking. At low rates of combustion, airflow velocities into the fireplace face are less than the velocity of natural convection currents induced at side walls by heat stored in the brick, allowing wisps of smoke to stray away from the combustion products' main flow path. Smoking tendencies are compounded at low firing rates by indoor/outdoor pressure differentials caused by winds, thermal forces, or fans, because the accompanying thermal force (buoyancy) of low combustion products temperature is insufficient to overcome strong wind or fan effects.

In the following analysis, note that fireplaces are primarily air-collecting hoods, diluting a small amount of combustion products with large amounts of air. Maximum mass flow of air into any given fireplace chimney not only is limited, but actually diminishes past a certain maximum. Thus, as combustion rate increases, combustion product temperatures rise to the point where masonry cracks; metals overexpand, warp, and oxidize; and steady smoking can occur because heated flue gases evolve beyond the limited capacity of the chimney.

An inoperative fireplace is completely at the mercy of indoor/outdoor pressure differences caused by winds, building stack effects, and operation of forced-air heating systems or mechanical ventilation. Thus, the complaint of smoking during start-up can have complex causes seldom related to the chimney. Increasingly in new homes and especially in high-rise multiple family construction, fireplaces of normal design cannot cope with mechanically induced reverse flow or shortages of combustion air. It is mandatory in these circumstances to treat and design a fireplace as a constantly operating mechanical exhaust system, with induced-draft blowers (mechanical-draft systems) that can overpower other mechanized air-consuming systems, and can develop sufficient flow to avoid smoking and excessive flue temperatures.

The gravity-flow capacity equation (Kinkead 1962) of a fireplace-chimney system equates mass flow with the resultant system driving forces and losses.

$$w = A_c\left(\frac{2gH}{k}\right)^{1/2}[\rho_m(\rho_o - \rho_m)]^{1/2} \qquad (24)$$

*where*

$w$ = flue gas flow rate, lb/s
$A_c$ = chimney flue cross-sectional area, ft$^2$
$g$ = gravitational constant, 32.1740 ft/s$^2$
$H$ = height of chimney above lintel, ft
$k$ = system equivalent resistance coefficient, dimensionless
$\rho_m$ = flue gas density at mean temperature, lb/ft$^3$
$\rho_o$ = air density of ambient temperature, lb/ft$^3$

From Equation (24), it is possible to develop a relationship for average frontal velocity, maximum chimney capacity, and variation of gas temperature with changes in fuel input rate.

Using resistance coefficients in these compact systems is preferable to the usual method of equivalent lengths. The summation term $k$ in a vertical chimney is the total of four individual resistance factors:

- Acceleration $k_a$ of ambient air to flue gas velocity, a constant value that must always be included in the total; $k_a = 1.0$
- Inlet loss coefficient $k_i$ for fireplace configuration, including smoke shelf

| | |
|---|---|
| Cone-type fireplaces | $k_i = 0.5$ |
| Masonry (damper throat = 2 × flue area) | $k_i = 1.0$ |
| Masonry (damper throat = flue area) | $k_i = 2.5$ |

- Chimney flue pipe friction $k_c$ at a typical Reynolds number of 10,000 and roughness of 0.001, where $r_h$ = hydraulic radius, ft

$$k_c = 0.0083\left(\frac{H}{r_h}\right)$$

- Termination coefficient $k_t$

| | |
|---|---|
| For open top pipe or tile, same size as chimney | $k_t = 0$ |
| Disk or cone cap at $D$/2 above outlet | $k_t = 0.5$ |
| Manufactured caps | $k_t = 0$ to 4.0 |

For a 12 ft high open top chimney, 12 in. in diameter on a typical fireplace, the system resistance is

$$\begin{array}{ll} k_a = & 1.0 \\ k_i = & 2.5 \\ k_c = 0.0083\ (12/0.25) = & 0.4 \\ k_t = & \underline{0.0} \\ & k = 3.9 \text{ summation} \end{array}$$

Note that in a short chimney, the wall friction coefficient $k_c$ is only 0.4 and has a minor effect on system flow. Greater or lesser chimney roughness, or a change from low to high heat loss materials will have little bearing on fireplace effectiveness in short chimneys. In some situations, it may be necessary, for completeness, to include a $k$ term for air supply resistance.

To determine the frontal velocity $V_F$ of ambient air, the term $w$ is replaced using the substitution

$$w = \rho_o A_F V_F \tag{25}$$

*where*

$A_F$ = fireplace frontal opening area, ft$^2$
$V_F$ = fireplace mean frontal velocity, ft/s

Accordingly, mean frontal velocity $V_F$ becomes

$$V_F = \frac{A_c}{A_F}\left(\frac{2gH}{k}\right)^{1/2}\left[\frac{\rho_m(\rho_o - \rho_m)}{\rho_o}\right]^{1/2} \tag{26}$$

For present purposes, the molecular weight or specific gravity of dilute combustion products is practically the same as that of air, and both can be expressed with adequate accuracy in terms of absolute gas temperature by the same relationship:

$$\rho_o = 1.325\frac{B_o}{T_o} \tag{27}$$

$$\rho_m = 1.325\frac{B_o}{T_m} \tag{28}$$

*where*

$B_o$ = existing barometric pressure, in. Hg
$T_o$ = ambient air temperature, °R
$T_m$ = mean combustion products temperature, °R

Substitution of the density/temperature relationships [Equations (27) and (28)], into Equation (26) allows further simplification, leading to the general frontal velocity expression:

$$V_F = \frac{A_c}{A_F}\left(\frac{2gH}{k}\right)^{1/2}\frac{[T_o(T_m - T_o)]^{1/2}}{T_m} \tag{29}$$

Here, frontal velocity is a function of the product of three terms:

- Dimensionless area ratio $A_c/A_F$
- Height/resistance term $(2gH/k)^{1/2}$
- Dimensionless temperature effects $[T_o(T_m - T_o)]^{1/2}/T_m$

For the 12 ft high example chimney, assume ambient temperature (for calculation purposes) is 70°F, or $T_o$ = 530°R indoors and outdoors, with no air supply resistance. Equation (29) expresses variation in $V_F$ with gas temperature as shown in Figure 19. Assume also $A_c/A_F = 0.10$, so that frontal opening is ten times chimney area.

The mean airflow velocity into a fireplace frontal opening is nearly constant from 300°F flue gas temperature rise up to any higher temperature. Local velocities vary within the opening. Depending on design, air typically enters horizontally along the hearth and then flows into the fire and upward, clinging to the back wall (see Figure 18). A recirculating eddy forms just inside the upper front half of the opening, induced by the high velocity of flow along the back. Restrictions or poor construction in the throat area between the lintel and damper also increase the eddy. Because the eddy moves smoke out of the zone of maximum velocity, the tendency of this smoke to escape must be counteracted by some minimum inward air movement over the entire front of the fireplace, particularly under the lintel.

Construction of a fireplace, its internal configuration, damper location, height and location of lintel, slope of back and sides, and so on all affect minimum frontal velocity to prevent smoking with ordinary fires. It seems desirable to maintain a smooth, gradual tapering transition between the hearth or flame region up into the damper location. A sudden transition, unless it is well above the lintel, induces velocity components that tend to increase eddying. With a shallow chamber between lintel and damper zone, there is insufficient volume for convection currents, and some flue gases may be diverted

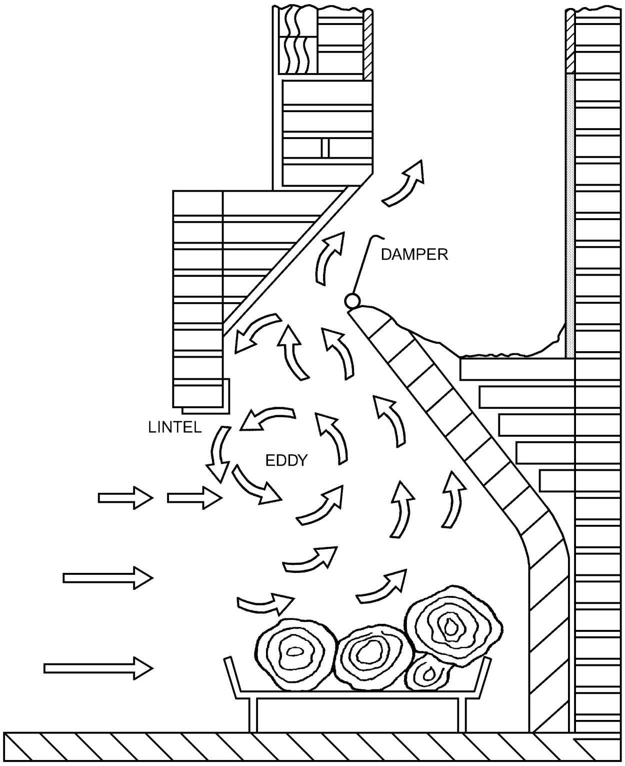


**Fig. 18 Eddy Formation**

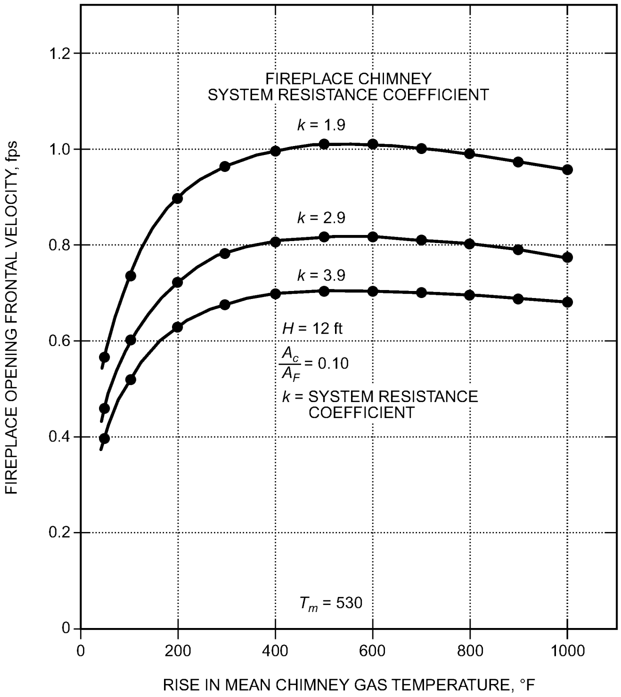


**Fig. 19 Effect of Chimney Gas (Combustion Products) Temperature on Fireplace Frontal Opening Velocity**

horizontally before being captured by the main flow. Masons following the dimensional parameters recommended by Ramsey and Sleeper (1956) or in damper literature can avoid these design flaws. With prefabricated fireplace systems, which often tend to be unconventional, careful testing is essential to ensure safe, smoke-free performance at minimum chimney height.

A minimum mean frontal inlet velocity of 0.8 fps, in conjunction with a chimney flue gas temperature of at least 300 to 500°F above ambient, should control smoking in a well-constructed conventional masonry fireplace. As noted in Figure 19, this velocity can be achieved even in low chimneys by system resistance of 2.9 or less, in conjunction with rates of combustion producing flue gas temperatures above a certain minimum level.

Figure 19 also illustrates why increases in flue gas temperature rise greater than 300°F have no perceptible effect on fireplace smoking, because combustion air mass flow and face velocity actually decrease at flue gas temperature rises higher than 500°F. For practical purposes, chimney flue gas temperature has little influence on fireplace performance, if flue gas temperature is at least 300°F above ambient. Fireplace performance analysis thus can be continued assuming a constant flue gas temperature of 500°F rise above ambient:

$$\frac{[T_o(T_m - T_o)]^{1/2}}{T_m} = 0.5$$

where $T_m$ = 1030°R and $T_o$ = 530°R.

Assuming 70°F ambient ($T_o$ = 530°R), and factoring out 2$g$, Equation (29) becomes roughly

$$V_F = 4.0\frac{A_c}{A_F}\left(\frac{H}{k}\right)^{1/2} \tag{30}$$

This expression states that relative fireplace performance is purely a matter of geometry. It permits evaluation of the opening size to maintain minimum frontal velocity as a function of height, while permitting quick analysis of effect of frontal area on velocity.

Figure 20 shows that 0.8 fps face velocity requires a relatively small $A_F/A_c$ area ratio at 8 ft chimney height, whereas the fireplace frontal opening area can be nearly twice as large with a 40 ft high chimney. To compute this curve, system resistance is assumed as

$$k = 2.5 + 0.033H/D$$

$$D = 1.0 \text{ ft diameter}$$

This expression of resistance assumes the fully open free area of the damper throat to be twice chimney flue area.

A corollary application of Equation (30) assumes fixed chimney size and height, and explores variation in frontal velocity with changes in area ratio. Figure 21 shows that a 12 in. diameter, 15 ft high chimney cannot produce adequate velocities for frontal area ratios greater than 11. These curves point out the possibility of further simplification to yield a fireplace-chimney design equation for a constant face velocity of 0.8 fps.

$$A_F = 5.0A_c\left(\frac{H}{k}\right)^{1/2} \tag{31}$$

This equation allows permissible frontal area $A_F$ to be determined as a function of chimney area, height, and system resistance. It can also be used to determine $A_c$, if $A_F$ is known. However, in this latter case, Equations (32) and (33) are preferable, because chimney size ($D_c$, $A_c$, and $R_h$) appears twice in the right-hand side of each equation.

*For circular flues*

$$A_F = 3.93D_c^2\left(\frac{H}{2.5 + 0.033\dfrac{H}{D_c}}\right)^{1/2} \tag{32}$$

*For other flue shapes*

$$A_F = 5A_c\left(\frac{H}{2.5 + 0.0083\dfrac{H}{R_h}}\right)^{1/2} \tag{33}$$

where $D_c$ is chimney inside diameter, ft, $R_h$ is inside hydraulic radius, ft, frontal velocity $V_F$ is 0.8 fps at maximum combustion air mass flow rate, and damper opening free area is twice chimney flue area combustion air $A_F$.

These relationships clearly reveal the origin of rules of thumb specifying opening area as 8, 10, or 12 times chimney area. Some

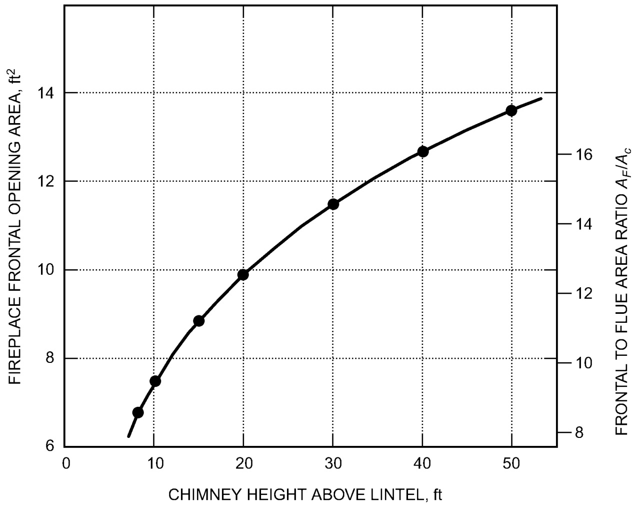


**Fig. 20 Permissible Fireplace Frontal Opening Area for Design Conditions (0.8 fps mean frontal velocity with 12 in. inside diameter round flue)**

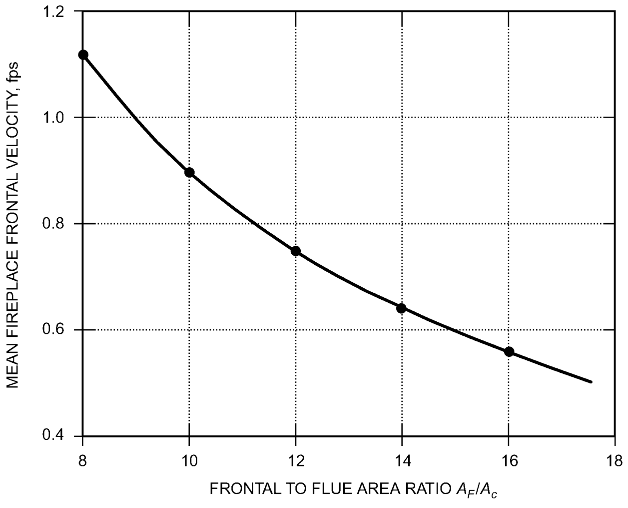


**Fig. 21 Effect of Area Ratio on Frontal Velocity (for chimney height of 15 ft with 12 in. inside diameter round flue)**

design guides go beyond and classify chimneys by height groups so that short ones serve smaller fireplace openings. Using a mean face velocity of 0.8 fps yields ratios and heights that fall well within the limits of such rules, as well as providing a unifying concept for computing design charts.

The previous relationships primarily apply to masonry single-face fireplaces of conventional construction, but are applicable to other types with considerable validity, if face or opening area is properly treated. Corner or double-face designs and many free-standing types embody conventional smokeshelf construction with similar resistance coefficients.

The preceding equations can be applied to illustrate the effect of excessive firing rates on chimney flue gas temperature. Masonry fireplaces are highly inefficient as heating devices, and tests show that, over a wide range of controlled fuel inputs using a drilled-port nonaerated gas burner, 75 to 80% of the gross heating value goes up the chimney. With constant flue heat loss of 80%, 70% of heat loss is sensible heat, which produces the rise in flue gas temperature. This heat input/temperature relationship may be developed by expressing the system flow relationship in terms of heat input:

$$w = \frac{q}{c_p(T_m - T_o)} \tag{34}$$

*where*

$q$ = heat content of chimney flue gases, Btu/s

$c_p$ = specific heat of chimney gases, assumed to be approximately 0.25 Btu/lb·°F

Equating Equations (34) and (24), and eliminating $w$,

$$\frac{q}{c_p(T_m - T_o)} = A_c\left(\frac{2gH}{k}\right)^{1/2}[\rho_m(\rho_o - \rho_m)]^{1/2} \tag{35}$$

Substituting Equations (27) and (28) for density and solving for $q$,

$$q = \frac{1.325B_oA_cc_p}{(T_o)^{0.5}}\left(\frac{2gH}{k}\right)^{1/2}\frac{(T_m - T_o)^{3/2}}{T_m} \tag{36}$$

For any given system, all terms except $(T_m - T_o)^{3/2}/T_m$ may be considered fixed; therefore, gas temperature rise for a fireplace can be obtained as a concise function of heat input:

$$q = \overline{C}\frac{(T_m - T_o)^{1.5}}{T_m} \tag{37}$$

where $\overline{C}$ is a constant.

The flow-temperature function of a carefully controlled experimental fireplace [Equation (36) or (37)] in which $\overline{C}$ as fixed is plotted in Figure 22 as chimney flue gas temperature versus heat input rate. Data taken on a fireplace-chimney system in this manner may readily be evaluated to determine system resistance coefficient $k$.

Figure 23 shows fireplace and chimney dimensions for the specific conditions of circular flues at 0.8 fps frontal velocity. This chart solves readily for maximum frontal opening for a given chimney, as well as for chimney size and height with a predetermined opening. For example, a 30 in. high by 42 in. wide opening for a 12 ft high chimney (measured from the highest point of front opening) requires a 12.5 in. flue. Figure 23 assumes no wind or air supply difficulties. For other face velocities, $A_F$ is found by multiplying frontal area (center scale) by velocity ratio $0.8/V_F$. To confirm the example results, calculate from Equation (32) as follows:

$$A_F = 3.93D_c^2{\frac{H}{(2.5 + 0.033H)/D_c}}^{1/2}$$

where $H$ is 12 ft and $D_c$ is 14 in.

$$A_F = 8.75\text{ ft}^2 = \frac{30\text{ in.} \times 42\text{ in.}}{12\text{ in.}/1\text{ ft}}$$

Although derived specifically for circular flues, $A_F$ applies with negligible sacrifice in performance to chimney flue cross sections such as squared or rounded ovals, because flue area is a much more important factor than friction caused by changes in hydraulic radius. For example, in a 20 ft high chimney, assuming a square flue section equal in area to an 8 in. circle, frontal area is reduced from 4.16 ft² with the round, to 4.09 ft² with the square, or about 2%, a difference

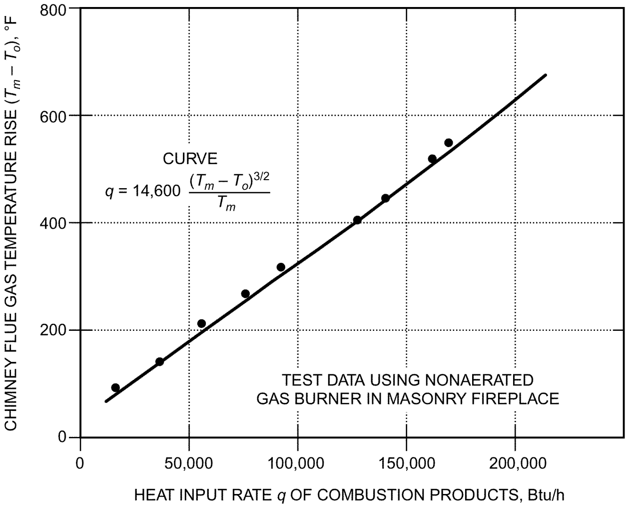


**Fig. 22 Variation of Chimney Flue Gas Temperature with Heat Input Rate of Combustion Products**

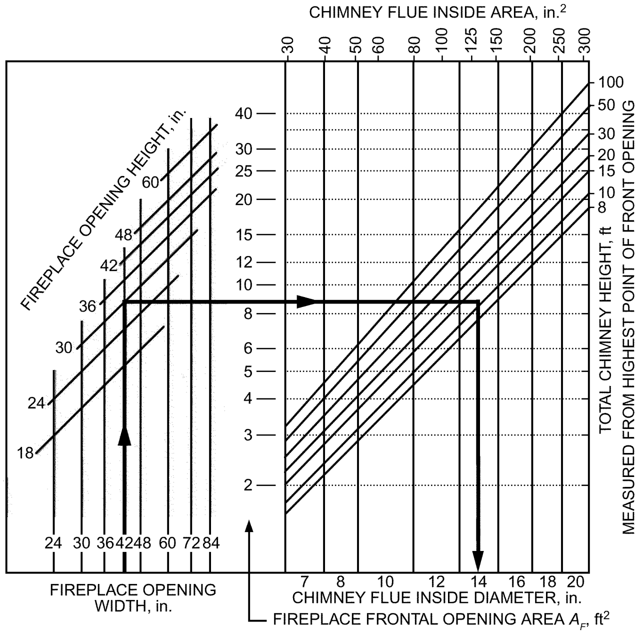


**Fig. 23 Chimney Sizing Chart for Fireplaces**

Mean Face Velocity = 0.8 fps

(Stone 2005)

that is hardly observable. For some typical constructions, Figure 24 suggests methods of estimating frontal area.

These relationships apply to steady-state conditions, which are obtained only after warm-up. Igniting a rapidly flammable charge in a cold system creates pulses of expanding hot combustion products, which frequently escape from the fireplace. In a typical chimney, priming time (time to accelerate from no flow to full upward velocity) is around 5 to 10 s. Thus, initial or intermittent smoking caused by momentarily excessive combustion rates occurs because of system inability to increase flue gas velocity in pace with combustion surges.

Flue or chimney material is of little relevance to fireplace-chimney operation. Materials to which these equations and charts apply include the very hazardous uninsulated single-wall metal, through conventional masonry, as well as the various constructions of lightweight, insulated, factory-built low-heat-space-appliance chimneys. (Safety standards classify fireplaces as low-heat appliances.) Because most fireplace chimneys are short and vertical, neither heat loss nor wall roughness has any important effect on flow. The governing factors in chimney selection for fireplaces are mainly safety, installation, convenience, and esthetics.

Indoor/outdoor pressure differences caused by winds, kitchen or bath exhaust fans, building stack effects, and operation of forced-air heating systems or mechanical ventilation affect the operation of a fireplace. Thus, smoking during start-up can be caused by factors unrelated to the chimney. Frequently, in new homes (especially in high-rise multiple-family construction), fireplaces of normal design cannot cope with mechanically induced reverse flow or shortages of combustion air. In these circumstances, a fireplace should include an induced-draft blower able to overpower other mechanized air-consuming systems. An inducer for this purpose is best located at the chimney outlet and should produce 0.8 to 1.0 fps fireplace face velocity of ambient air in an individual flue or 10 to 12 fps chimney velocity.

In conventional fireplaces, the greater the frontal velocity, the more freedom from smoking. The damper free area, together with its resultant resistance coefficient, are thus major factors in obtaining good masonry fireplace performance, especially with short chimneys. Tests show that damper free area need not exceed twice the required flue area, because little further resistance reduction occurs past this limit.

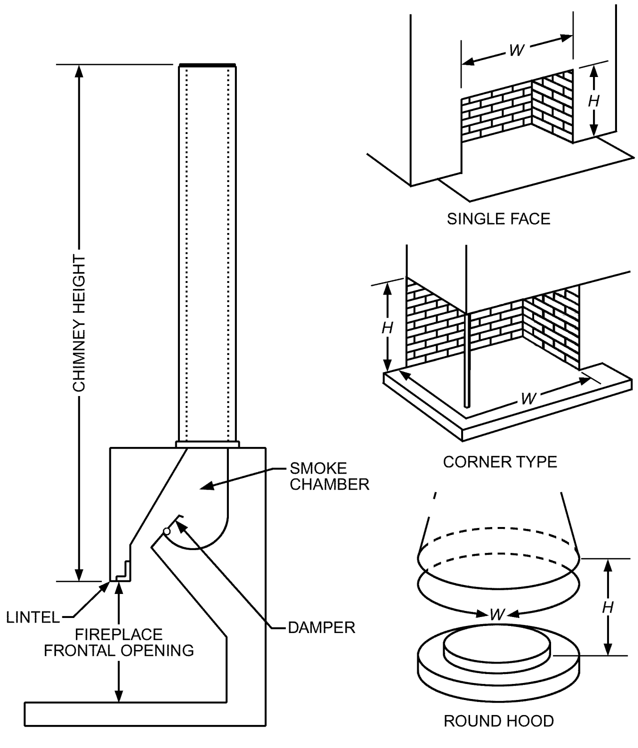


**Fig. 24 Estimation of Fireplace Frontal Opening Area**

If the damper selected has a free area equal to or less than required area, it will be definitely restrictive, despite complete adequacy of other factors. Manufacturers' literature seldom includes damper free area or opening dimensions, and the dimensions may vary further after installation because of interferences with lintels and other parts. It is expedient to select dampers of adequate free area for best results.

Partially closing a damper during a vigorous fire illustrates this point; what is not so obvious is that greater damper openings may be needed in some cases to control smoke by achieving adequate frontal velocities.

Many free-standing fireplaces are built without the usual smokeshelf/throat damper configuration. The same parameters and relationships apply to free-standing fireplaces as to masonry fireplaces; however, in many designs it is difficult to assign a true frontal area for velocity analysis. Where freestanding fireplaces include back outlets, or require a horizontal run to reach the chimney, compensation is necessary for the flow resistance or pressure losses caused by turns. As a rough approximation, increase the system resistance coefficient $k$ about 1.5 for two 90° elbows, or a back outlet with lateral run into a tee.

Losses caused by lateral connectors and tees generally result in a one-size increase (e.g., moving from 7 to 8) being sufficient. Collar sizes generally determine correct chimney size, and instructions are furnished to cover special situations. More sophisticated prefabricated fireplace designs are provided with matching correctly sized chimneys, and are intended for installation in conventional wood-frame residences.

The equations and design charts presented here assume no wind or air supply difficulties. Lack of replacement air, competing ventilation exhaust fans, and negative interior pressures caused by winds are all obvious causes of smoking or poor fireplace priming. Even when fully primed and hot, thermal forces in a fireplace chimney can be overpowered by a combination of adverse influences. In modern high-rise residential apartments, where an effort has been made to provide all amenities, fireplaces may have to cope simultaneously with all of these troubles. Continuous induced draft for the chimney alleviates most of these problems by maintaining a chimney prime at all times. An inducer for this purpose should be able to produce 0.8 to 1.0 fps fireplace frontal velocity of ambient air in any individual flue. Where multiple flues are installed in a chase, a single, larger inducer serving the chase can be sized for the combined fireplace frontal opening area. Even where flues are of different heights and sizes, a draft inducer selection, assuming all flues to be some compromise median height and size, produces far greater user satisfaction than reliance on gravity alone.

In single-family dwellings, fireplace problems are more frequently caused by reduced interior pressures from wind effects than by poor chimney terminal location or characteristics. Efforts to cure smoking, slow priming, or blowback of ashes usually involve one of the myriad forms of stationary or rotating caps, cowls, or chimney pots. This questionable expedient has contradictory effects. Usually, the added still-air resistance of a cap reduces fireplace frontal velocity, which limits combustion airflow rate, and thus may tend to increase smoking. On the other hand, a cap reduces air dilution of smaller fires, raises chimney temperature, and improves stability of flame, thus tending to mitigate wind impulses that cause momentary flow reduction. The usual fireplace damper can also be used to restrict flow, and thus raise temperature.

Remedies for fireplace malfunctions may be analyzed using Equation (30). For example, it is apparent that any change of parameters on the right-hand side that might decrease $V_F$ can increase smoking tendencies. If frontal area $A_F$ or $k$ increases, there will be a corresponding decrease in $V_F$. Similarly, if chimney area $A_c$ or chimney height $H$ are reduced, then $V_F$ decreases. Further, because frontal velocity varies as the square root of the term $H/k$, it is more effective to reduce frontal area, thereby increasing $A_c/A_F$, than to increase $H$ or reduce $k$.

Logical expedients for increasing $V_F$ frontal velocity, and thus improving performance of fireplaces and chimneys, include the following:

- Increase chimney height (using the same flue area) and extend the last tile 6 in. upward, or more.
- Decrease frontal opening by lowering the lintel, or raising the hearth. (Glass doors may help by increasing $V_F$.)
- Increase free area through damper. (Check that it opens fully without interferences.)

CSA *Standard* P.4.1-02 can be used for measuring annual efficiency in Canada.

## AIR SUPPLY TO FUEL-BURNING APPLIANCES

Failure to supply outdoor air for combustion may result in erratic or even dangerous operating conditions. A correctly designed gas appliance with a draft hood can function with short vents (5 ft high) using an outdoor air supply opening as small in area as the vent outlet collar. Such an orifice, when equal to vent area, has a resistance coefficient in the range of 2 to 3. If the air supply opening is as much as twice the vent area, however, the coefficient drops to 0.5 or less.

The following rules may be used as a guide:

1. Residential heating appliances installed in unconfined spaces in buildings of conventional construction do not ordinarily require ventilation other than normal air infiltration. In any residence or building that has been built or altered to conserve energy or minimize infiltration, the heating appliance area should be considered a confined space. The air supply should be installed in accordance with ANSI/NFPA 54/ANSI/AGA Z223.1, CSA *Standard* B149.1, or the following recommendations.
2. Residential heating appliances installed in a confined space having unusually tight construction require two permanent openings to an unconfined space or to the outdoors. An unconfined space has a volume of at least 50 $\text{ft}^3$ per 1000 Btu/h of the total input rating of all appliances installed in that space. Free opening areas must be greater than 1 $\text{in}^2$ per 4000 Btu/h input with vertical ducts or 1 $\text{in}^2$ per 2000 Btu/h with horizontal ducts to the outdoors. The two openings communicating directly with sufficient unconfined space must be greater than 1 $\text{in}^2$ per 1000 Btu/h. Upper openings should be within 12 in. of the ceiling; lower openings should be within 12 in. of the floor.
3. Complete combustion of natural and propane gas or fuel oil requires approximately 1 $\text{ft}^3$ of air, at standard conditions, for each 100 Btu of fuel burned, but excess air is usually required for proper burner operation.
4. The size of these air openings may be modified if special engineering ensures an adequate supply of air for combustion, dilution, and ventilation or if local ordinances apply to boiler and machinery rooms.
5. In calculating free area of air inlets, consider the blocking effect of louvers, grilles, or screens protecting openings. Screens should not be smaller than 1/4 in. mesh. If the free area through a particular louver or grille is known, it should be used in calculating the size opening required to provide the free area specified. If the free area is not known, assume that wood louvers have 20 to 25% free area and metal louvers and grilles have 60 to 75% free area.
6. Mechanical ventilation systems serving the fuel-burning appliance room or adjacent spaces should not be allowed to create negative appliance room air pressure. The appliance room may require tight self-closing doors and provisions to supply air to spaces under negative pressure so fuel-burning appliances and venting operate properly.
7. Fireplaces may require special consideration. For example, a residential attic fan can be hazardous if it is inadvertently turned on while a fireplace is in use.
8. In buildings where large quantities of combustion and ventilation or process air are exhausted, a sufficient supply of fresh uncontaminated makeup air, warmed if necessary to the proper temperature, should be provided. It is good practice to provide about 5 to 10% more makeup air than the amount exhausted.

## VENT AND CHIMNEY MATERIALS

Factors to be considered when selecting chimney materials include (1) the temperature of flue gases; (2) their composition and propensity for condensation of water vapor from combustion products (dew point); (3) presence of sulfur, halogens, and other fuel and air contaminants that lead to corrosion of the chimney vent system; and (4) the appliance's operating cycle (condensate dwell time).

Figure 7 covers materials for vents and chimneys in the 4 to 48 in. size range; these include single-wall metal, various multiwall air- and mass-insulated types, and precast and site-constructed masonry. Each has different characteristics, such as frequency of joints, roughness, and heat loss, but the type of materials used for systems 14 in. and larger is relatively unimportant in determining draft or capacity. This does not preclude selecting a safe product or method of construction that minimizes heat loss and fire hazard in the building.

National codes and standards classify heat-producing appliances as low, medium, and high heat, with appropriate reference to chimney and vent constructions permitted with each. These classifications are primarily based on size, process use, or combustion temperature. In many cases, the appliance classification gives little information about outlet gas temperature or venting needed. The designer should, wherever possible, obtain gas outlet temperature conditions and properties that apply to the specific appliance, rather than going by code classification only.

Where building codes permit engineered chimney systems, chimney material selection based on gas outlet temperature can save space as well as reduce structural and material costs. For example, in some jurisdictions, approved gas-burning appliances with draft hoods operating at inputs over 400,000 Btu/h may be placed in a heat-producing classification that prohibits use of type B gas vents. An increase in input may not cause an increase in outlet temperature or in venting hazards, and most building codes recommend correct matching appliance and vent.

Single-wall uninsulated steel stacks can be protected from condensation and corrosion internally with refractory firebrick liners or by spraying calcium aluminate cement over a suitable interior expanded metal mesh or other reinforcement. Another form of protection applies proprietary silica or other prepared refractory coatings to pins or a support mesh on the steel. The material must then be suitably cured for moisture and heat resistance.

Moisture condensation on interior surfaces of connectors, vents, stacks, and chimneys is a more serious cause of deterioration than heat. Chimney wall temperature and flue gas velocity, temperature, and dew point affect condensation. Contaminants such as sulfur, chlorides, and fluorides in the fuel and combustion air raise the flue gas dew point. Studies by Beaumont et al. (1970), Mueller (1968), Pray et al. (1942-53), and Yeaw and Schnidman (1943) indicate the variety of analytical methods as well as difficulties in predicting the causes and probability of actual condensation.

Combustion products from any fuel containing hydrogen condense onto cold surfaces or condense in bulk if the main flow of flue gas is cooled sufficiently. Because flue gas loses heat through walls, condensation, which first occurs on interior wall surfaces cooled to the flue gas dew point, forms successively a dew and then a liquid film and, with further cooling, causes liquid to flow down into zones where condensation would not normally occur.

Start-up of cold interior chimney surfaces is accompanied by transient dew formation, which evaporates on heating above the dew point. This phenomenon causes little corrosion when very low sulfur fuels are used. Proper selection of chimney dimensions and materials minimizes condensation and thus corrosion.

Experience shows a correlation between the sulfur content of the fuel and the deterioration of interior chimney surfaces. Figure 4 in Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals* illustrates one case, which applies to any fuel gas. The figure shows that the flue gas dew point increases at 40% excess air from 127°F with zero sulfur to 160°F with 15 grains of sulfur per 100 $ft^3$ of fuel having a heat value of 550 $Btu/ft^3$.

The figure can also be used to approximate the effect of fuel oil sulfur content on flue gas dew point. For example, fuel oil with a sulfur content of 0.5% (by mass) contains about 252 grains of sulfur per gallon or 25,200 grains of sulfur per 100 gallons. If the fuel heat value is 140,000 Btu/gal, the ratio that defines the curves in the figure gives a curve value of 18. Estimates for lower percentages of sulfur (0.25, 0.05) can be formed as factors of the value 18.

Because the corrosion mechanism is not completely understood, judicious use of resistant materials, suitably insulated or jacketed to reduce heat loss, is preferable to low-cost single-wall construction. Refractory materials and mortars should be acid-resistant, while steels should be resistant to sulfuric, hydrochloric, and hydrofluoric acids; pitting; and oxidation. Where low flue gas temperatures are expected together with low ambients, an air space jacket or mineral fiber lagging, suitably protected against water entry, helps maintain surface and flue gas temperatures above the dew point. Using low-sulfur fuel, which is required in many localities, reduces both corrosion and air pollution.

Type 1100 aluminum alloy or any other non-copper-bearing aluminum alloy of 99% purity or better provides satisfactory performance in prefabricated metal gas vent products. For chimney service, flue gas temperatures from appliances burning oil or solid fuels may exceed the melting point of aluminum; therefore, steel is required. Stainless steels such as type 430 or 304 give good service in residential construction and are referenced in UL-listed prefabricated chimneys. Where more corrosive substances (e.g., high-sulfur fuel or chlorides from solid fuel, contaminated air, or refuse) are anticipated, type AL 29-4CR® or equivalent stainless steel offers a good match of corrosion resistance and mechanical properties.

As an alternative to stainless steel, porcelain enamel offers good resistance to corrosion if two coats of acid-resistant enamel are used on all surfaces. A single coat, which always has imperfections, allows base metal corrosion, spalling, and early failure.

Prefabricated chimneys and venting products are available that use light corrosion-resistant materials, both in metal and masonry. The standardized, prefabricated, double-wall metal type B gas vent has an aluminum inner pipe and a coated steel outer casing, either galvanized or aluminized. Standard air space from 1/4 to 1/2 in. is adequate for applicable tests and a wide variety of exposures.

Air-insulated all-metal chimneys are available for low-heat use in residential construction. Thermosiphon air circulation or multiple reflective shielding with three or more walls keeps these units cool. Insulated, double-wall residential chimneys are also available. The annulus between metal inner and outer walls is filled with insulation and retained by coupler end structures for rapid assembly.

Prefabricated, air-insulated, double-wall metal chimneys for multifamily residential and larger buildings, classed as building heating appliance chimneys, are available (Figure 25). Refractory-lined prefabricated chimneys (medium-heat type) are also available for this use.

Commercial and industrial incinerators, as well as heating appliances, may be vented by prefabricated metal-jacketed cast refractory chimneys, which are listed in the medium-heat category and are suitable for intermittent flue gas temperatures to 2000°F. All prefabricated chimneys and vents carrying a listing by a recognized testing laboratory have been evaluated for class of service regarding temperature, strength, clearance to adjacent combustible materials, and suitability of construction in accordance with applicable national standards.

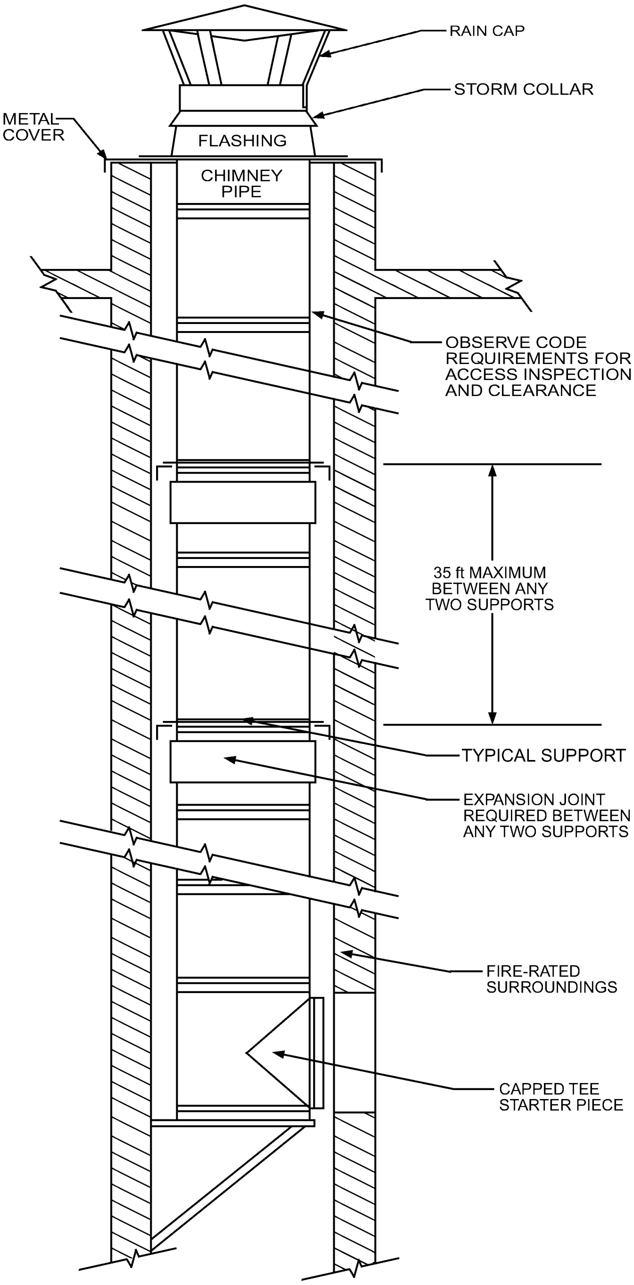


**Fig. 25 Building Heating Appliance, Medium-Heat Chimney**

Underwriters Laboratories (UL) standards, listed in Table 10, describe the construction and temperature testing of various classes of prefabricated vent and chimney materials. Standards for some related parts and appliances are also included in Table 10 because a listed factory-built fireplace, for example, must be used with a specified type of factory-built chimney. The temperature given for the steady-state operation of chimneys is the lowest in the test sequence. Factory-built chimneys under UL *Standard* 103 are also required to demonstrate adequate safety during a 1 h test at 1330°F rise and to

**Table 10 Underwriters Laboratories Test Standards**

| No. | Subject | Steady-State Appliance Flue Gas Temperature Rise, °F | Fuel |
|---|---|---|---|
| 103 | Chimneys, factory-built, residential type (includes building heating appliances) | 930 | All |
| 127 | Fireplaces, factory-built | 930 | Solid or gas |
| 311 | Roof jacks for manufactured homes and recreational vehicles | 930 | Oil, gas |
| 378 | Draft equipment (such as regulators and inducers) | — | All |
| 441 | Gas vents (type B, BW) | 480 | Gas only |
| 641 | Low-temperature venting systems (type L) | 500 | Oil, gas |
| 959 | Chimneys, factory-built, medium-heat | 1730 | All |
| 1738 | Venting systems for gas-burning appliances, categories II, III, and IV | 140 to 480 | Gas only |

withstand a 10 min simulated soot burnout at either 1630 or 2030°F rise.

These product tests determine minimum clearance to combustible surfaces or enclosures, based on allowable temperature rise on combustibles. They also ensure that the supports, spacers, and parts of the product that contact combustible materials remain at safe temperatures during operation. Product markings and installation instructions of listed materials are required to be consistent with test results, refer to types of appliances that may be used, and explain structural and other limitations.

## VENT AND CHIMNEY ACCESSORIES

Vent or chimney system design must consider the existence of or need for accessories such as draft diverters, draft regulators, induced-draft fans, blocking dampers, expansion joints, and vent or chimney terminals. Draft regulators include barometric draft regulators and furnace sequence draft controls, which monitor automatic flue dampers during operation. The design, materials, and flow losses of chimney and vent connectors are covered in previous sections.

### Draft Hoods

The draft hood isolates the appliance from venting disturbances (updrafts, downdrafts, or blocked vent) and allows combustion to start without venting action. Suggested general dimensions of draft hoods are given in ANSI *Standard* Z21.12, which describes certification test methods for draft hoods. In general, the pipe size of the inlet and outlet flues of the draft hood should be the same as that of the appliance outlet connection. The vent connection at the draft hood outlet should have a cross-sectional area at least as large as that of the draft hood inlet.

Draft hood selection comes under the following two categories:

1. Draft hood supplied with a design-certified gas appliance: certification of a gas appliance design under pertinent national standards includes its draft hood (or draft diverter). Consequently, the draft hood should not be altered or replaced without consulting the manufacturer and local code authorities.
2. Draft hoods supplied separately for gas appliances: listed draft hoods for existing vent or chimney connectors should be installed by experienced installers in accordance with accepted practice standards.

Every design-certified gas appliance requiring a draft hood must be accompanied by a draft hood or provided with a draft diverter as an integral part of the appliance. The draft hood is a vent inlet fitting as well as a safety device for the appliance, and assumptions can be made regarding its interaction with a vent. First, when the hood is operating without spillage, the heat content of flue gases (enthalpy relative to dilution air temperature) leaving the draft hood is almost the same as that entering. Second, safe operation is obtained with 40 to 50% dilution air. It is unnecessary to assume 100% dilution air for gas venting conditions. Third, during certification tests, the draft hood must function without spillage, using a vent with not over 5 ft of effective height and one or two elbows. Therefore, if vent heights appreciably greater than 5 ft are used, an individual vent of the same size as the draft hood outlet may be much larger than necessary.

When vent size is reduced, as with tall vents, draft hood resistance is less than design value relative to the vent; the vent tables in the *National Fuel Gas Code* (ANSI/NFPA 54/ANSI/AGA Z223.1) give adequate guidance for such size reductions.

Despite its importance as a vent inlet fitting, the draft hood designed for a typical gas appliance primarily represents a compromise of the many design criteria and tests solely applicable to that appliance. This allows considerable variation in resistance loss; thus, catalog data on draft hood resistance loss coefficients do not exist. The span of draft hood loss coefficients, including inlet acceleration, varies from the theoretical minimum of 1.0 for certain low-loss bell or conical shapes to 3 or 4, where the draft hood relief opening is located within a hot-air discharge (as with wall furnaces) and high resistance is needed to limit sensible heat loss into the vent.

Draft hoods must not be used on appliances having draft configuration 1 or 3 (see conditions 1, 2, and 3 under Terminology and Table 8) that is operated with either power burners or forced venting, unless the appliances have fan-assisted burners that overcome some or most of the appliance flow resistance and create a pressure inversion ahead of the draft hood or barometric regulator.

Gas appliances with draft hoods must have excess chimney draft capacity to draw in adequate draft hood dilution air. Failure to provide adequate combustion air can cause oxygen depletion and spillage of flue gases and flame rollout from the combustion air inlet at the burner(s).

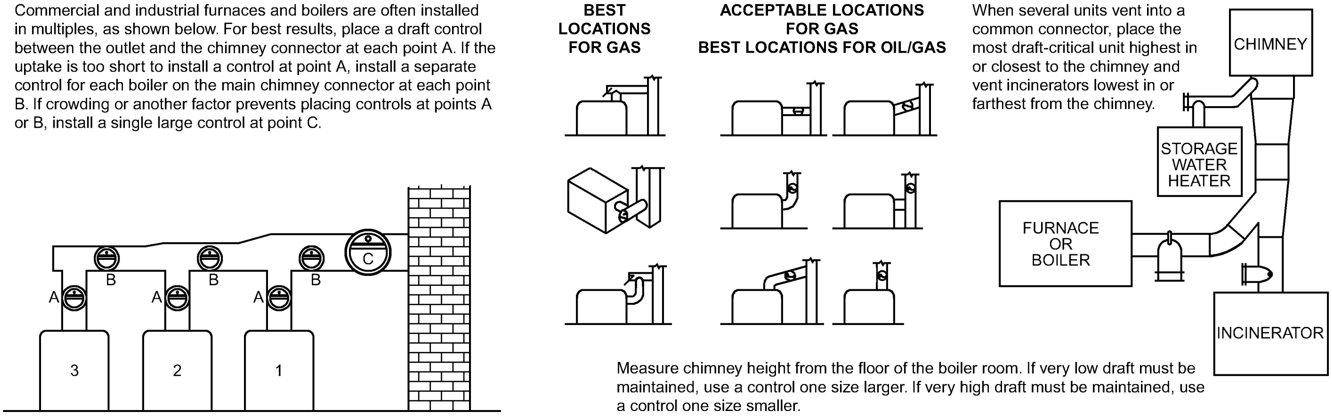
### Draft Regulators

Appliances requiring draft at the appliance flue gas outlet generally use barometric regulators for combustion stability. A balanced hinged gate in these devices bleeds air into the chimney automatically when pressure decreases. This action simultaneously increases vent gas flow and reduces temperature. Well-designed barometric regulators provide constant flue gas static pressure over a span of impressed vent gas draft of about 0.2 in. of water, where impressed vent gas draft is that which would exist without regulation. A regulator can maintain 0.06 in. of water draft for impressed drafts from 0.06 to 0.26 in. of water. If the chimney system is very high or otherwise capable of generating available draft in excess of the pressure span capability of a single regulator, additional or oversize regulators may be used. Figure 26 shows proper locations for regulators in a chimney manifold.

Barometric regulators are available with double-acting dampers, which also swing out to relieve momentary internal pressures or divert continuing downdrafts. In the case of downdrafts, temperature safety switches actuated by hot gases escaping at the regulator sense and limit malfunctions.

### Vent Dampers

Electrically, mechanically, and thermally actuated automatic vent dampers can reduce energy consumption and improve seasonal efficiency of gas- and oil-burning appliances. Vent dampers reduce loss of heated air through gas appliance draft hoods and loss of specific


Commercial and industrial furnaces and boilers are often installed in multiples, as shown below. For best results, place a draft control between the outlet and the chimney connector at each point A. If the uptake is too short to install a control at point A, install a separate control for each boiler on the main chimney connector at each point B. If crowding or another factor prevents placing controls at points A or B, install a single large control at point C.

When several units vent into a common connector, place the most draft-critical unit highest in or closest to the chimney and vent incinerators lowest in or farthest from the chimney.

Measure chimney height from the floor of the boiler room. If very low draft must be maintained, use a control one size larger. If very high draft must be maintained, use a control one size smaller.

Fig. 26 Use of Barometric Draft Regulators

heat from the appliance after the burner stops firing. These dampers may be retrofit devices or integral components of some appliances.

Electrically and mechanically actuated dampers must open before main burner gas ignition and must not close during burner operation. These safety interlocks, which electrically interconnect with existing control circuitry, include an additional main control valve, if called for, or special gas pressure-actuated controls.

Vent dampers that are thermally actuated with bimetallic elements and have spillage-sensing interlocks with burner controls are available for draft hood-type gas appliances. These dampers open in response to gas temperature after burner ignition. Because thermally actuated dampers may exhibit some flow resistance, even at equilibrium operating conditions, carefully follow instructions regarding allowable heat input and minimum required vent or chimney height.

Special care must be taken to ensure that safety interlocks with appliance controls are installed according to instructions. Spillage-free gas venting after the damper has been installed must be verified with all damper types.

Energy savings of a vent damper can vary widely. Dampers reduce energy consumption under one or a combination of the following conditions:

- Heating appliance is oversized.
- Chimney is too high or oversized.
- Appliance is located in heated space.
- Two or more appliances are on the same chimney (a damper must be installed on each appliance connected to that chimney).
- Appliance is located in building zone at higher pressure than outdoors. This positive pressure can cause steady flow losses through the chimney.

Energy savings may not justify the cost of installing a vent damper if one or more of the following conditions exist:

- All combustion and ventilation air is supplied from outdoors to direct-vent appliances or to appliances located in an isolated, unheated room.
- Appliance is in an unheated basement that is isolated from the heated space.
- A one-story flat-roof house has a short vent, which is unlikely to carry away a significant amount of heated air.

For vents or chimneys serving two or more appliances, dampers (if used) should be installed on all attached appliances for maximum effectiveness. If only one damper is installed in such systems, loss of heated air through an open draft hood may negate a large portion of the potential energy savings.

### Heat Exchangers or Flue Gas Heat Extractors

Sensible heat available in flue gas of properly adjusted furnaces burning oil or gas is about 10 to 15% of the rated input. Small accessory heat exchangers that fit in the connector between the appliance outlet and the chimney can recover some of this heat for localized use; however, they may cause some adverse effects.

All gas vent and chimney size or capacity tables assume the gas temperature or heat available to create theoretical draft is not reduced by a heat transfer device. In addition, the tables assume flow resistance for connectors, vents, and chimneys, comprising typical values for draft hoods, elbows, tees, caps, and piping with no allowance for added devices placed directly in the flue gas stream. Thus, heat exchangers or flue gas extractors should offer no flow resistance or negligible resistance coefficients when they are installed.

A heat exchanger that is reasonably efficient and offers some flow resistance may adversely affect the system by reducing both flow rate and flue gas temperature. This may cause moisture condensation in the chimney, draft hood spillage, or both. Increasing heat transfer efficiency increases the probability of the simultaneous occurrence of both effects. An accessory heat exchanger in a solid-fuel system, especially a wood stove or heater, may collect creosote or cause its formation downstream.

Retrofitting heat exchangers in gas appliance venting systems requires careful evaluation of heat recovered versus both installed cost and the potential for chimney safety and operating problems. Every heat exchanger installation should undergo the same spillage tests given a damper installation. In addition, the flue gas temperature should be checked to ensure it is high enough to avoid condensation between the exchanger outlet and the chimney outlet.

## DRAFT FANS

The selection of draft fans, blowers, or inducers must consider (1) types and combinations of appliances, (2) types of venting material, (3) building and safety codes, (4) control circuits, (5) gas temperature, (6) permissible location, (7) noise, and (8) power cost. Besides specially designed fans and blowers, some conventional fans can be used if the wheel and housing materials are heat- and corrosion-resistant and if blower and motor bearings are protected from adverse effects of the flue gas stream.

Small draft inducers for residential gas appliance and unit heater use are available with direct-drive blower wheels and an integral device to sense flow (Figure 27A). The control circuit for these applications must provide adequate vent gas flow both before and while fuel flows to the main burner. Other types of small inducers are either saddle-mounted blower wheels (Figure 27B) or venturi ejectors that induce flow by jet action (Figure 27C). An essential

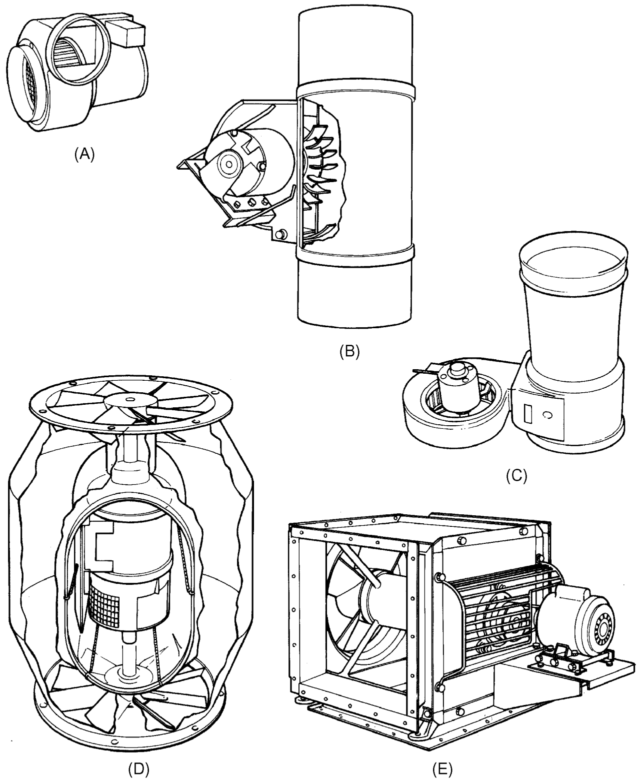


**Fig. 27 Draft Inducers**

safety requirement for inducers serving draft hood gas appliances does not permit appliance interconnections on the discharge or outlet side of the inducer. This requirement prevents backflow through an inoperative appliance.

With prefabricated sheet metal venting products such as type B gas vents, the vent draft inducer should be located at or downstream from the point the vent exits the building. This placement keeps the indoor system below atmospheric pressure and prevents flue gas from escaping through seams and joints. If the inducer cannot be placed on the roof or outdoor wall, metal joints must be reliably sealed in all pressurized parts of the system.

Pressure capability of residential draft inducers is usually less than 1 in. of water at rated flow. Larger inducers of the fan, blower, or ejector type have greater pressure capability and may be used to reduce system size as well as supplement available draft. Figure 27D shows a specialized axial-flow fan capable of higher pressures. This unit is structurally self-supporting and can be mounted in any position in the connector or stack because the motor is in a well, separated from the flue gas stream. A right-angle fan, as shown in Figure 27E, is supported by an external bracket and adapts to several inlet and exit combinations. The unit uses the developed draft and an insulated tube to cool the extended shaft and bearings.

Pressure, volume, and power curves should be obtained to match an inducer to the application. For example, in an individual chimney system (without draft hood) in which a directly connected inducer only handles combustion products, calculation of the power required for continuous operation need only consider volume at operating flue gas temperature. An inducer serving multiple, separately controlled draft hood gas appliances must be powered for ambient temperature operation at full flow volume in the system. At any input, the inducer for a draft hood gas appliance must handle about 50% more standard chimney gas volume than a directly connected inducer. At constant volume with a given size inducer or fan, these demands follow the Fan Laws (see Chapter 20) applicable to power venting as follows:

- Pressure difference developed is directly proportional to gas density
- Pressure difference developed is inversely proportional to absolute gas temperature
- Pressure developed diminishes in direct proportion to drop in absolute atmospheric pressure, as with altitude
- Required power is directly proportional to gas density
- Required power is inversely proportional to absolute gas temperature

Centrifugal and propeller draft inducers in vents and chimneys are applied and installed the same as in any heat-carrying duct system. Venturi ejector draft boosters involve some added consideration. An advantage of the ejector is that motor, bearings, and blower blades are outside the contaminated flue gas stream, thus eliminating a major source of deterioration. This advantage causes some loss of efficiency and can lead to reduced capacity because undersized systems, having considerable resistance downstream, may be unable to handle the added volume of the injected airstream without loss of performance. Ejectors are best suited for use at the chimney or vent exit or where there is an adequately sized chimney or vent to carry the combined discharge.

If total pressure defines outlet conditions or is used for fan selection, the relative amounts of static pressure and velocity pressure must be factored out; otherwise, the velocity head method of calculation does not apply. To factor total pressure into its two components, either the discharge velocity in an outlet of known area or the flow rate must be known. For example, if an appliance or blower produces a total pressure of 0.25 in. of water at 1500 fpm discharge velocity, the velocity pressure component can be found from Figure 7. Enter at the horizontal line marked V.H. in grid C, and move horizontally to ambient operating temperature. Read up from the appropriate temperature rise line to the 1500 fpm horizontal velocity line in grid D. Here the velocity pressure reads 0.14 in. of water (calculates to 0.143 at ambient standard conditions), so static pressure is 0.25 – 0.14 = 0.11 in. of water. Because this static pressure is part of the system driving force, it combines with theoretical draft to overcome losses in the system.

Draft inducer fans can be operated either continuously or on demand. In either case, a safety switch that senses flue gas flow or pressure is needed to interrupt burner controls if adequate draft fails. Demand operation links the thermostat with the draft control motor. Once flow starts, as sensed with a flow or pressure switch, the burner is allowed to start. A single draft inducer, operating continuously, can be installed in the common vent of a system serving several separately controlled appliances. This simplifies the circuitry because only one control is needed to sense loss of draft. However, the single fan increases appliance standby loss (especially in boilers) and heat losses via ambient air drawn through inoperative appliances.

## TERMINATIONS: CAPS AND WIND EFFECTS

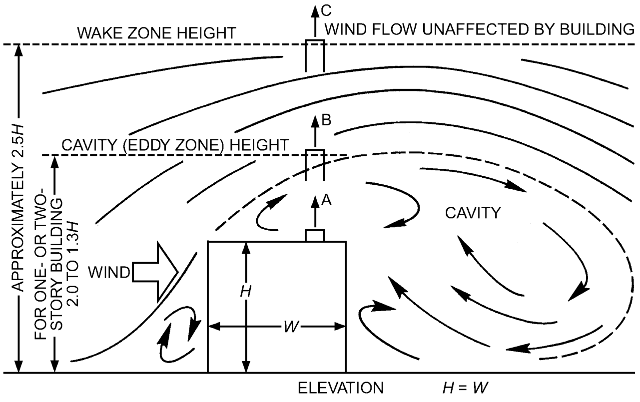
The vent or chimney height and method of termination is governed by a variety of considerations, including fire hazard; wind effects; entry of rain, debris, and birds; and operating considerations such as draft and capacity. For example, the 3 ft height required for residential chimneys above a roof is necessary so that small sparks will burn out before they fall on the roof shingles.

Many vent and chimney malfunctions are attributed to interactions of the chimney termination or its cap with winds acting on the roof or with adjoining buildings, trees, or mountains. Because winds fluctuate, no simple method of analysis or reduction to practice exists for this complex situation. Figures 28 to 30 show some of the complexities of wind flow contours around simple structural shapes.

Figure 28 shows three zones with differing degrees of flue gas dispersion around a rectangular building: the cavity or eddy zone, the wake zone, and the undisturbed flow zone (Clarke 1967). In addition, a fourth flow zone of intense turbulence is located downwind of the cavity. Chimney flue gases discharged into the wind at a point close to the roof surface in the cavity zone may be recirculated locally. Higher in the cavity zone, wind eddies can carry more dilute flue gas to the lee side of the building. Flue gases discharged into the wind in the wake zone do not recirculate into the immediate vicinity, but may soon descend to ground level. Above the wake zone, dispersal into the undisturbed wind flow carries and dilutes the flue gases over a wider area. The boundaries of these zones vary with building configuration and wind direction and turbulence; they are strongly influenced by surroundings.

The possibility of air pollutants reentering the cavity zone because of plume spread or of air pollution intercepting downwind cavities associated with adjacent structures or downwind buildings should be considered. Thus, the design criterion of elevating the stack discharge above the cavity is not valid for all cases. Consult a meteorologist experienced with dispersion processes near buildings for complex cases and for cases involving air contaminants.

As chimney height increases through the three zones, draft performance improves dispersion, while additional problems of gas cooling, condensation, and structural wind load are created. As building height increases (Figure 29), the eddy forming the cavity zone no longer descends to ground level. For a low, wide building (Figure 30), wind blowing parallel to the long roof dimension can reattach to the surface; thus, the eddy zone becomes flush with the roof surface (Evans 1957). For satisfactory dispersion with low, wide buildings, chimney height must still be determined as if $H = W$ (Figure 28).



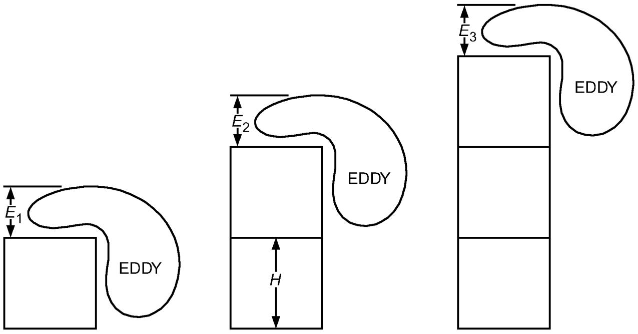
Chimney heights:

A: Discharge into cavity should be avoided because reentry will occur. Dispersion equations do not apply.

B: Discharge above cavity is good. Reentry is avoided, but dispersion may be marginal or poor from standpoint of air pollution. Dispersion equations do not apply.

C: Discharge above wake zone is best; no reentry; maximum dispersion.

**Fig. 28 Wind Eddy and Wake Zones for One- or Two-Story Buildings and Their Effect on Chimney Gas Discharge**



$E$ = EDDY HEIGHT ABOVE ROOF = 0.5$H$

$E_1 = E_2 = E_3$

Studies found for a single cube-shaped building (length equals height) that (1) the height of the eddy above grade is 1.5 times the building height and (2) the height of unaffected air is 2.5 times height above grade. The eddy height above the roof equals 0.5$H$, and does not change as building height increases in relation to building width.

**Fig. 29 Height of Eddy Currents Around Single High-Rise Buildings**

Chien et al. (1951) and Evans (1957) studied pitched roofs in relation to wind flow and surface pressures. Because the typical residence has a pitched roof and probably uses natural gas or a low-sulfur fossil fuel, dispersion is not important because combustion products are relatively free of pollutants. For example, ANSI/NFPA 54/ANSI/AGA Z223.1 requires a minimum distance between the gas vent termination and any air intake, but it does not require penetration above the cavity zone.

Flow of wind over a chimney termination can impede or assist draft. In regions of stagnation on the windward side of a wall or a steep roof, winds create positive static pressures that impede established flow or cause backdrafts in vents and chimneys. Locating a chimney termination near the surface of a low, flat roof can aid draft because the entire roof surface is under negative static pressure. Velocity is low, however, because of the cavity formed as wind sweeps up over the building. With greater chimney height, termination above the low-velocity cavity or negative-pressure zone subjects the chimney exit to greater wind velocity, thereby increasing draft from two causes: (1) height and (2) wind aspiration over an open top. As the termination is moved from the center of the building to the sides, its exposure to winds and pressure also varies.

Terminations on pitched roofs may be exposed to either negative or positive static pressure, as well as to variation in wind velocity and direction. On the windward side, pitched roofs vary from complete to partial negative pressure as pitch increases from approximately flat to 30° (Chien et al. 1951). At a 45° pitch, the windward pitched roof surface is strongly positive; beyond this slope, pressures approach those observed on a vertical wall facing the wind. Wind pressure varies with its horizontal direction on a pitched roof, and on the lee (sheltered) side, wind velocity is very low, and static pressures are usually negative. Wind velocities and pressures vary not only with pitch, but with position between ridge and eaves. Reduction of these observed external wind effects to simple rules of termination for a wide variety of chimney and venting systems requires many compromises.

In the wake zone or any higher location exposed to full wind velocity, an open top can create strong venting updrafts. The updraft effect relative to wind dynamic pressure is related to the Reynolds number. Open tops, however, are sensitive to the wind angle as well as to rain (Clarke 1967), and many proprietary tops have been

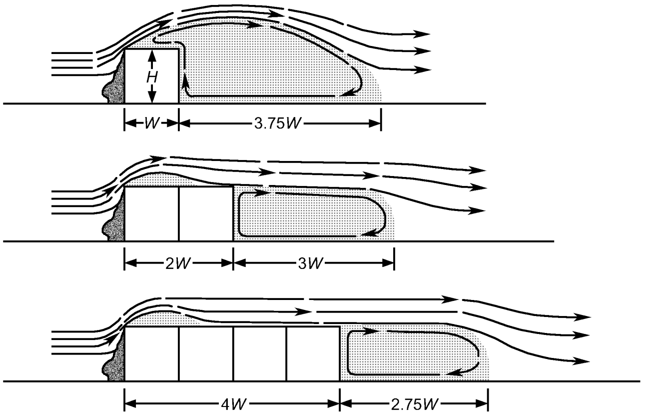


**Fig. 30 Eddy and Wake Zones for Low, Wide Buildings**

**Table 11 List of U.S. National Standards Relating to Installation**[a]

| Subject | Materials Covered | NFPA[b] | ANSI[c] | CSA[f] |
|---|---|---|---|---|
| Oil-burning equipment | Type L listed chimneys, single-wall, masonry | 31 | — | — |
| Gas appliances and gas piping | Type B, L listed chimneys, single-wall, masonry | 54 | Z223.1[d] | B149.1 |
| Chimneys, fireplaces, vents, and solid-fuel-burning appliances | All types | 211 | — | CAN/ULC S605-M91 |
| Recreational vehicles | Roof jacks and vents | 501C | — | — |
| Gas piping and gas equipment on industrial premises | All types | 54 | Z223.1[d] | B149.1 |
| Gas conversion burners | Chimneys | | Z21.8[e] | — |
| | Safe design | | Z21.17[e] | — |
| Draft hoods | Part dimensions | | Z21.12[e] | CAN1-6.2 |
| Automatic vent dampers for use with gas-fired appliances | Construction and performance | | Z21.66[e] | — |

[a] These standards are subject to periodic review and revision to reflect advances in industry, as well as for consistency with legal requirements and other codes.
[b] National Fire Protection Association, Quincy, MA.
[c] American National Standards Institute, New York.
[d] Available from American Gas Association, Washington, D.C.
[e] Available from CSA America, Cleveland, OH.
[f] Canadian Standards Association, Mississauga, ON.

designed to stabilize wind effects and improve the performance. Because of the many compromises made in vent termination design, this stability is usually achieved by sacrificing some of the updraft created by the wind. Further, locating a vent cap in a cavity region frequently removes it from the zone where wind velocity could have a significant effect.

Performance optimization studies of residential vent cap design indicate that the following performance features are important: (1) still-air resistance, (2) updraft ability with no flow, and (3) discharge resistance when vent gases are carried at low velocity in a typical wind (10 fps vent velocity in a 20 mph wind). Tests in UL *Standard* 441 for proprietary gas vent caps consider these three aspects of performance to ensure adequate vent capacity.

Frequently, air supply to an appliance room is difficult to orient to eliminate wind effects. Therefore, the vent outlet must have a certain updraft capability, which can help balance a possible adverse wind. When wind flows across an inoperative vent termination, a strong updraft develops. Appliance start-up reduces this updraft, and in typical winds, the vent cap may develop greater resistance than it would have in still air. Certain vent caps can be made with very low still-air resistance, yet exhibit excessive wind resistance, which reduces capacity. Finally, because the appliance operates whether or not there is a wind, still-air resistance must be low.

Some proprietary air ventilators have excessive still-air resistance and should be avoided on vent and chimney systems unless a considerably oversized vent is specified. Vertical-slot ventilators, for example, have still-air resistance coefficients of about 4.5. To achieve low still-air resistance on vents and chimneys, the vertical-slot ventilator must be 50% larger than the diameter of the chimney or vent unless it has been specifically listed for such use.

Freestanding chimneys high enough to project above the cavity zone require structurally adequate materials or guying and bracing for prefabricated products. The prefabricated metal building heating appliance chimney places little load on the roof structure, but guying is required at 8 to 12 ft intervals to resist both overturning and oscillating wind forces. Various other expedients, such as spiral baffles on heavy-gage freestanding chimneys, have been used to reduce oscillation.

The chimney height needed to carry the effluent into the undisturbed flow stream above the wake zone can be reduced by increasing the effluent discharge velocity. A 3000 fpm discharge velocity avoids downward eddying along the chimney and expels the effluent free of the wake zone. Velocity this high can be achieved only with forced or induced draft.

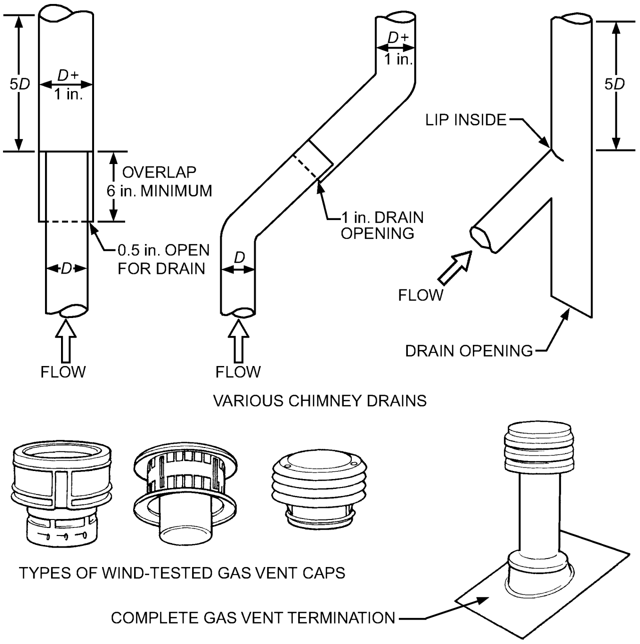


**Fig. 31 Vent and Chimney Rain Protection**

Rain entry is a problem for open, low-velocity, or inoperative systems. Good results have been obtained with drains that divert the water onto a roof or into a collection system leading to a sump. Figure 31 shows several configurations (Clarke 1967; Hama and Downing 1963). Runoff from stack drains contains acids, soot, and metallic corrosion products, which can cause roof staining. Therefore, these methods are not recommended for residential use. An alternative procedure is to allow all water to drain to the base of the chimney, where it is piped from a capped tee to a sump.

Rain caps prevent vertical discharge of high-velocity flue gases. However, caps are preferred for residential gas-burning equipment because it is easier to exclude rain than to risk rainwater leakage at horizontal joints or to drain it. Also, caps keep out debris and bird nests, which can block the chimney. Satisfactory vent cap performance can be achieved in the wind by using one of a variety of standard configurations, including the A cap and the wind band ventilator, or one of the proprietary designs shown in Figure 31.

Where partial rain protection without excessive flow resistance is desired, and either wind characteristics are unimportant or windflow is horizontal, a flat disk or cone cap 1.7 to 2.0 diameters across located 0.5 diameter above the end of the pipe has a still-air resistance loss coefficient of about 0.5.

Consult Chapter 44 of the 2007 *ASHRAE Handbook—HVAC Applications* for additional information on vent and chimney termination and wind effects.

## CODES AND STANDARDS

Building and installation codes and standards prescribe the installation and safety requirements of heat-producing appliances and their vents and chimneys. Chapter 51 lists the major national building codes, one of which may be in effect in a given area. Some jurisdictions either adopt a national building code with varying degrees of revisions to suit local custom or, as in many major metropolitan areas, develop a local code that agrees in principle but shares little common text with the national codes. Familiarity with applicable building codes is essential because of the great variation in local codes and adoption of modern chimney design practice.

The national standards listed in Table 11 give greater detail on the mechanical aspects of fuel systems and chimney or vent construction. Although these standards emphasize safety aspects, especially clearances to combustibles for various venting materials, they also recognize the importance of proper flow, draft, and capacity.

## CONVERSION FACTORS

The following conversion factors have been simplified for chimney design.

$$
\begin{aligned}
\text{Btu/h input} &= 3{,}347{,}500\ \frac{\text{Boiler horsepower}}{\text{Percent efficiency}}\\
&= 44{,}600 \times \text{Boiler horsepower (approx. 75\% eff.)}\\
&= 41{,}800 \times \text{Boiler horsepower (approx. 80\% eff.)}\\
&= 37{,}200 \times \text{Boiler horsepower (approx. 90\% eff.)}\\
&= 140{,}000 \times \text{No. 1 and 2 oil (gph)}\\
&= 150{,}000 \times \text{No. 4, 5, and 6 oil (gph)}\\
&= 13{,}000 \times \text{Coal (lb/h)}\\
&= 1000 \times \text{Natural gas (ft}^3\text{/h)}\\
&= 3.412 \times \text{Watt rating}\\
\text{kW input} &= \text{kW output/efficiency}\\
\text{kW} &= 9.81 \times \text{Boiler horsepower}
\end{aligned}
$$

## SYMBOLS

$A$ = area of passage cross section, ft$^2$
$A_c$ = chimney flue cross-sectionally area, ft$^2$
$A_F$ = fireplace frontal opening area, ft$^2$
$B$ = existing or local barometric pressure, in. Hg
$B_o$ = standard pressure, 29.92 in. Hg
$C_u$ = temperature multiplier for heat loss, dimensionless
$c_p$ = specific heat of gas at constant pressure, Btu/lb · °F
$d_f$ = inside diameter, ft
$d_i$ = inside diameter, in.
$D_a$ = available draft, in. of water
$D_b$ = boost (increase in static pressure by fan), in. of water
$D_p$ = depressurization, in. of water
$D_t$ = *theoretical draft, in. of water*
$F$ = friction factor for $L/d_i$
$f$ = Darcy friction factor from Moody diagram (Figure 13, Chapter 2, 2005 *ASHRAE Handbook—Fundamentals*)
$g$ = gravitational constant, 32.174 ft/s$^2$
$H$ = height of vent or chimney system above grade or system inlet, ft
$I$ = operating heat input, Btu/h
$k$ = system resistance loss coefficient, dimensionless
$k_f$ = fitting friction loss coefficient, dimensionless
$k_L$ = piping friction loss coefficient, dimensionless
$L$ = length of all piping in chimney system from inlet to exit, linear ft
$L_e$ = total equivalent length, $L$ plus equivalent length of elbows, etc. [for use in Equations (2) and (3) only], ft
$L_m$ = length of system from inlet to midpoint of vertical or to location of mean gas temperature, ft
$M$ = ratio of mass flow to heat input, lb of combustion products per 1000 Btu of fuel burned
$\Delta p$ = system flow losses or pressure drop, in. of water
$q$ = sensible heat at a particular point in vent, Btu/h
$q_m$ = sensible heat at average temperature in vent, Btu/h
$Q$ = volumetric flow rate, cfm
$\Delta t$ = temperature difference, °F
$\Delta t_e$ = temperature difference entering system, °F
$\Delta t_m$ = temperature difference at average temperature location in system, °F [$\Delta t_m = T_m - T_o$]
$T$ = absolute temperature, °R
$T_m$ = mean flue gas temperature at average conditions in system, °R
$T_o$ = ambient temperature, °R
$T_s$ = standard temperature, °R (518.67°R)
$U$ = overall heat transfer coefficient of vent or chimney wall material, referred to inside surface area, Btu/h · ft$^2$ · °F
$\bar{U}$ = heat transfer coefficient for Equation (8), Btu/h · ft$^2$ · °F
$V$ = velocity of gas flow in passage, fps
$V_F$ = frontal velocity of ambient air, fps
$w$ = mass flow of gas, lb/h
$x$ = length of one internal side of rectangular chimney cross section, ft
$y$ = length of other internal side of rectangular chimney cross section, ft
$\rho_a$ = density of air at 59°F and 29.92 in. Hg, lb/ft$^3$ (0.0765 lb/ft$^3$)]
$\rho_c$ = density of chimney gas at 0°F and 29.92 in. Hg, lb/ft$^3$ [0.09 lb/ft$^3$ in Equations (2) and (3)]
$\rho_m$ = density of chimney gas at average temperature and local barometric pressure, lb/ft$^3$
$\rho_o$ = density of air at 0°F and 29.92 in. Hg, lb/ft$^3$ (0.0863 lb/ft$^3$)

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