CHAPTER 16

VENTILATION AND INFILTRATION

PROVIDING a comfortable and healthy indoor environment for building occupants is the primary concern of HVAC engineers. Comfort and indoor air quality (IAQ) depend on many factors, including thermal regulation; control of internal and external sources of pollutants; supply of acceptable air; removal of unacceptable air; occupants' activities and preferences; and proper construction, operation, and maintenance of building systems. Ventilation and infiltration are only part of the acceptable indoor air quality and thermal comfort problem. HVAC designers, occupants, and building owners must be aware of and address other factors as well. Further information on indoor environmental health may be found in Chapter 10. Changing ventilation and infiltration rates to solve thermal comfort problems and reduce energy consumption can affect indoor air quality and may be against code, so any changes should be approached with care and be under the direction of a registered professional engineer with expertise in HVAC analysis and design.

HVAC design engineers and others concerned with building ventilation and indoor air quality should obtain a copy of ASHRAE *Standard* 62.1 or 62.2. These standards are reviewed regularly and contain ventilation design and evaluation requirements for commercial (62.1) and low-rise residential (62.2) buildings, respectively. In design of a new building or analysis of an existing building, the version of *Standard* 62 that has been adopted by the local code authority must be determined. An existing building may be required to meet current code, or allowed to comply with an older code. If a project involves infiltration in residences, then ASHRAE *Standards* 119 and 136 should be consulted. The last chapter of each year's *ASHRAE Handbook* (Chapter 39 of this volume) has a list of current standards.

This chapter addresses commercial and institutional buildings, where ventilation concerns usually dominate (though infiltration should not be ignored), and single- and multifamily residences, where infiltration has always been considered important but ventilation issues have received increased attention in recent years. Basic concepts and terminology for both are presented before more advanced analytical and design techniques are given. Ventilation of industrial buildings is covered in Chapter 29 of the 2007 *ASHRAE Handbook—HVAC Applications*. However, many of the fundamental ideas and terminology covered in this chapter can also be applied to industrial buildings.

Sustainability Rating Systems

Good indoor air quality is necessary for maintaining health and high productivity. Consequently, green and sustainable building rating systems, such as the U.S. Green Building Council's (USGBC) Leadership in Energy and Environmental Design (LEED®) program, place great importance on creating and maintaining acceptable IAQ. In fact, the LEED rating system was first developed to address IAQ concerns, and roughly one-third of the available credit points for new commercial buildings are still IAQ-related. Preparers of such rating systems, like others, have struggled with how to characterize complex ventilation and infiltration issues; many portions of this chapter; separate ASHRAE designer's guides, manuals, books, and standards; and the references cited address these issues in detail and provide methods for demonstrating the effectiveness of various HVAC systems and techniques in providing good IAQ in residential, commercial, and other buildings.

BASIC CONCEPTS AND TERMINOLOGY

Outdoor air that flows through a building is often used to dilute and remove indoor air contaminants. However, the energy required to condition this outdoor air can be a significant portion of the total space-conditioning load. The magnitude of outdoor airflow into the building must be known for proper sizing of the HVAC equipment and evaluation of energy consumption. For buildings without mechanical cooling and dehumidification, proper ventilation and infiltration airflows are important for providing comfort for occupants. ASHRAE *Standard* 55 specifies conditions under which 80% or more of the occupants in a space will find it thermally acceptable. Chapter 9 of this volume also addresses thermal comfort. Additionally, airflow into buildings and between zones affects fires and the movement of smoke. Smoke management is addressed in Chapter 52 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Ventilation and Infiltration

Air exchange of outdoor air with air already in a building can be divided into two broad classifications: ventilation and infiltration.

Ventilation is intentional introduction of air from the outside into a building; it is further subdivided into natural and mechanical ventilation. **Natural ventilation** is the flow of air through open windows, doors, grilles, and other planned building envelope penetrations, and it is driven by natural and/or artificially produced pressure differentials. **Mechanical** (or **forced**) ventilation, shown in Figure 1, is the intentional movement of air into and out of a building using fans and intake and exhaust vents.

Infiltration is the flow of outdoor air into a building through cracks and other unintentional openings and through the normal use of exterior doors for entrance and egress. Infiltration is also known as **air leakage** into a building. **Exfiltration**, depicted in Figure 1, is leakage of indoor air out of a building through similar types of openings. Like natural ventilation, infiltration and exfiltration are driven by natural and/or artificial pressure differences. These forces are discussed in detail in the section on Driving Mechanisms for Ventilation and Infiltration. **Transfer air** is air that moves from one interior space to another, either intentionally or not.

The preparation of this chapter is assigned to TC 4.3, Ventilation Requirements and Infiltration.

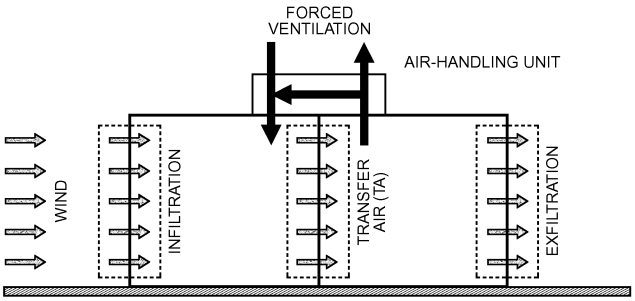


Fig. 1 Two-Space Building with Mechanical Ventilation, Infiltration, and Exfiltration

Ventilation and infiltration differ significantly in how they affect energy consumption, air quality, and thermal comfort, and they can each vary with weather conditions, building operation, and use. Although one mode may be expected to dominate in a particular building, all must be considered in the proper design and operation of an HVAC system.

Ventilation Air

Ventilation air is air used to provide acceptable indoor air quality. It may be composed of mechanical or natural ventilation, infiltration, suitably treated recirculated air, transfer air, or an appropriate combination, although the allowable means of providing ventilation air varies in standards and guidelines.

Modern commercial and institutional buildings normally have mechanical ventilation and are usually pressurized somewhat to reduce or eliminate infiltration. Mechanical ventilation has the greatest potential for control of air exchange when the system is properly designed, installed, and operated; it should provide acceptable indoor air quality and thermal comfort when ASHRAE *Standard* 55 and 62.1 requirements are followed. Mechanical ventilation equipment and systems are described in Chapters 1, 4, and 9 of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment*.

In commercial and institutional buildings, natural ventilation (e.g., through operable windows) may not be desirable from the point of view of energy conservation and comfort. In commercial and institutional buildings with mechanical cooling and ventilation, an air- or water-side economizer may be preferable to operable windows for taking advantage of cool outdoor conditions when interior cooling is required. Infiltration may be significant in commercial and institutional buildings, especially in tall, leaky, or partially pressurized buildings and in lobby areas.

In most of the United States, residential buildings have historically relied on infiltration and natural ventilation to meet their ventilation air needs. Neither is reliable for ventilation air purposes because they depend on weather conditions, building construction, and maintenance. However, natural ventilation, usually through operable windows, is more likely to allow occupants to control airborne contaminants and interior air temperature, but it can have a substantial energy cost if used while the residence's heating or cooling equipment is operating.

In place of operable windows, small exhaust fans should be provided for localized venting in residential spaces, such as kitchens and bathrooms. Not all local building codes require that the exhaust be vented to the outside. Instead, the code may allow the air to be treated and returned to the space or to be discharged to an attic space. Poor maintenance of these treatment devices can make nonducted vents ineffective for ventilation purposes. Condensation in attics should be avoided. In northern Europe and in Canada, some building codes require general mechanical ventilation in residences, and heat recovery heat exchangers are popular for reducing energy consumption. Low-rise residential buildings with low rates of infiltration and natural ventilation, including most new buildings, require mechanical ventilation at rates given in ASHRAE *Standard* 62.2.

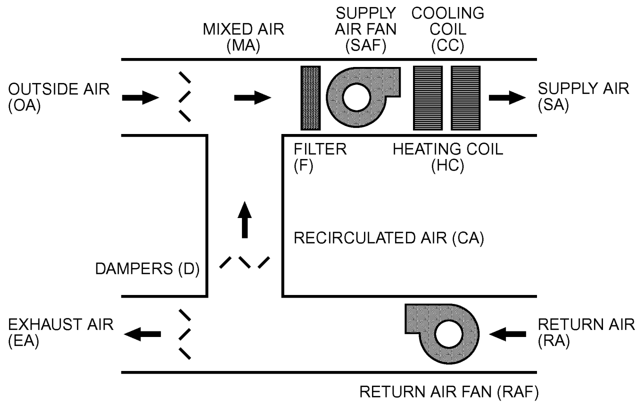


Fig. 2 Simple All-Air Air-Handling Unit with Associated Airflows

Forced-Air Distribution Systems

Figure 2 shows a simple **air-handling unit (AHU)** or **air handler** that conditions air for a building. Air brought back to the air handler from the conditioned space is **return air (RA)**. The return air either is discharged to the environment [**exhaust air (EA)**] or is reused [**recirculated air (CA)**]. Air brought in intentionally from the environment is **outdoor** or **outside air (OA)**. Because outdoor air may need treatment to be acceptable for use in a building, it should not be called "fresh air." Outside and recirculated air are combined to form **mixed air (MA)**, which is then conditioned and delivered to the thermal zone as **supply air (SA)**. Any portion of the mixed air that intentionally or unintentionally circumvents conditioning is **bypass air (BA)**. Because of the wide variety of air-handling systems, the airflows shown in Figure 2 may not all be present in a particular system as defined here. Also, more complex systems may have additional airflows.

Outside Air Fraction

The outside airflow introduced to a building or zone by an air-handling unit can also be described by the **outside air fraction X_{oa}**, which is the ratio of the volumetric flow rate of outside air brought in by the air handler to the total supply airflow rate:

$$X_{oa} = \frac{Q_{oa}}{Q_{sa}} = \frac{Q_{oa}}{Q_{ma}} = \frac{Q_{oa}}{Q_{oa} + Q_{ca}} \tag{1}$$

When expressed as a percentage, the outside air fraction is called the **percent outside air**. The design outside airflow rate for a building's or zone's ventilation system is found by applying the requirements of ASHRAE *Standard* 62.1 to that specific building. The supply airflow rate is that required to meet the thermal load. The outside air fraction and percent outside air then describe the degree of recirculation, where a low value indicates a high rate of recirculation, and a high value shows little recirculation. Conventional all-air air-handling systems for commercial and institutional buildings have approximately 10 to 40% outside air.

100% outside air means no recirculation of return air through the air-handling system. Instead, all the supply air is treated outside air, also known as **makeup air (KA)**, and all return air is discharged directly to the outside as **relief air (LA)**, via separate or centralized exhaust fans. An air-handling unit that provides 100% outside air to offset air that is exhausted is typically called a **makeup air unit (MAU)**.

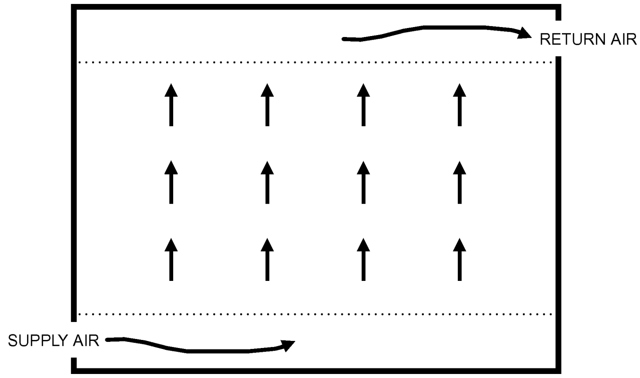


Fig. 3 Displacement Flow Within a Space

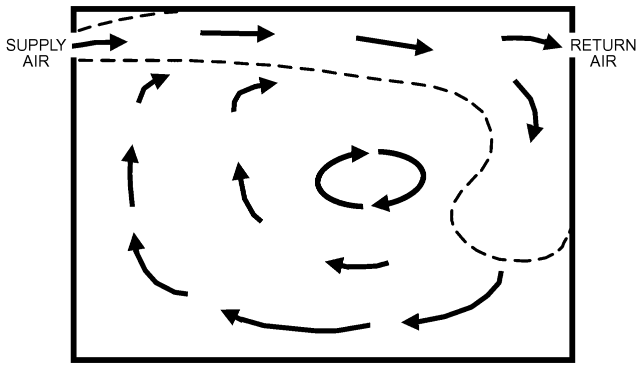


Fig. 4 Entrainment Flow Within a Space

When outside air via mechanical ventilation is used to provide ventilation air, as is common in commercial and institutional buildings, this outside air is usually delivered to spaces as all or part of the supply air. With a variable-air-volume (VAV) system, the outside air fraction of the supply air may need to be increased when supply airflow is reduced to meet a particular thermal load. In some HVAC systems, such as the dedicated outside air system (DOAS), conditioned outside air may be delivered separately from the way the spaces' loads are handled (Mumma and Shank 2001).

Room Air Movement

Air movement within spaces affects the diffusion of ventilation air and, therefore, indoor air quality and comfort. Two distinct flow patterns are commonly used to characterize air movement in rooms: displacement flow and entrainment flow. **Displacement flow**, shown in Figure 3, is the movement of air within a space in a piston- or plug-type motion. Ideally, no mixing of the room air occurs, which is desirable for removing pollutants generated within a space. A laminar-flow air distribution system that sweeps air across a space may produce displacement flow.

Entrainment flow, shown in Figure 4, is also known as **conventional mixing**. Systems with ceiling-based supply air diffusers and return air grilles are common examples of air distribution systems that produce entrainment flow. Entrainment flow with very poor mixing in the room has been called *short-circuiting flow* because much of the supply air leaves the room without mixing with room air. There is little evidence that properly designed, installed, and operated air distribution systems exhibit short-circuiting, although poorly designed, installed, or operated systems may short-circuit, especially ceiling-based systems in heating mode (Offermann and Int-Hout 1989).

Perfect mixing occurs when supply air is instantly and evenly distributed throughout a space. Perfect mixing is also known as

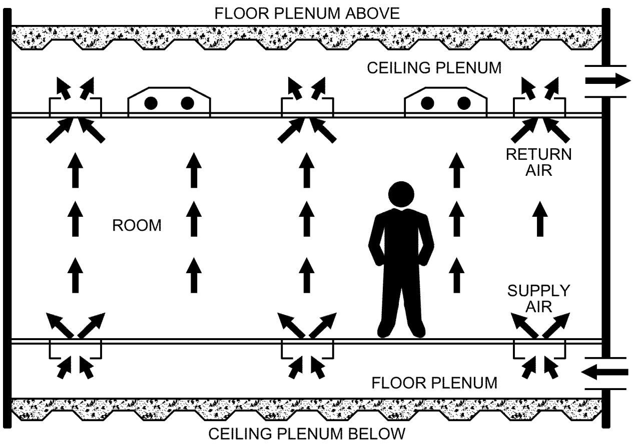


Fig. 5 Underfloor Air Distribution to Occupied Space Above
(Rock and Zhu 2002)

complete or **uniform mixing**; the air may be called **well stirred** or **well mixed**. This theoretical performance is approached by entrainment flow systems that have good mixing and by displacement flow systems that allow too much mixing (Rock et al. 1995). The outdoor air requirements given in Table 6.1 of ASHRAE *Standard* 62.1 assume delivery of ventilation air with perfect mixing within spaces. For more detailed information on space air diffusion, see Chapter 20.

Underfloor air distribution (UFAD or **UAD)**, as shown in Figure 5, is a hybrid method of conditioning and ventilating spaces (Bauman and Daly 2003). Air is introduced through a floor plenum, with or without branch ductwork or terminal units, and delivered to a space by floor-mounted diffusers. These diffusers encourage air mixing near the floor to temper the supply air. The combined air then moves vertically through the space, with reduced mixing, toward returns or exhausts placed in or near the ceiling. This vertical upward movement of the air is in the same direction as the thermal and contaminant plumes created by occupants and common equipment. Ventilation performance for UFAD systems is thus between floor-to-ceiling displacement flow and perfect mixing.

Supply air that enters a space through a diffuser is also known as **primary air**. A **jet** is formed as this primary air leaves the diffuser. **Secondary air** is the room air entrained into the jet. **Total air** is the combination of primary and secondary air at a specific point in a jet. The term *primary air* is also used to describe supply air provided to fan-powered mixing boxes by a central air-handling unit.

For evaluation of indoor air quality and thermal comfort, rooms are often divided into two portions: the **occupied zone** and the remaining volume of the space. Often, this remaining volume is solely the space above the occupants and is referred to as the **ceiling zone**. The occupied zone is usually defined as the lowest 6 ft of a room, although layers near the floor and walls are sometimes deducted from it. Ceiling and floor plenums are not normally included in the occupied or ceiling zones. **Thermal zones** are different from these room air zones, and are defined for HVAC subsystems and their controls.

Air Exchange Rate

The **air exchange** (or **change**) **rate** $\boldsymbol{I}$ compares airflow to volume and is

$$I = Q/V \tag{2}$$

where

Q = volumetric flow rate of air into space, cfm
V = interior volume of space, ft^3

The air exchange rate has units of 1/time, usually h^{-1}. When the time unit is hours, the air exchange rate is also called **air changes per hour (ach)**. The air exchange rate may be defined for several different situations. For example, the air exchange rate for an entire building or thermal zone served by an air-handling unit compares the amount of outside air brought into the building or zone to the total interior volume. This **nominal air exchange rate I_N** is

$$I_N = Q_{oa}/V \tag{3}$$

where Q_{oa} is the outdoor airflow rate including ventilation and infiltration. The nominal air exchange rate describes the outside air ventilation rate entering a building or zone. It does not describe recirculation or the distribution of the ventilation air to each space within a building or zone.

For a particular space, the **space air exchange rate I_S** compares the supply airflow rate Q_{sa} to the volume of that space:

$$I_S = Q_{sa}/V \tag{4}$$

The space air exchange rate for a particular space or zone includes recirculated as well as outside air in the supply air, and it is used frequently in the evaluation of supply air diffuser performance and space air mixing.

Time Constants

Time constants τ, which have units of time (usually in hours or seconds), are also used to describe ventilation and infiltration. One time constant is the time required for one air change in a building, zone, or space if ideal displacement flow existed. It is the inverse of the air exchange rate:

$$\tau = 1/I = V/Q \tag{5}$$

The **nominal time constant** compares the interior volume of a building or zone to the volumetric outdoor airflow rate:

$$\tau_N = V/Q_{oa} \tag{6}$$

Like the nominal air exchange rate, the nominal time constant does not describe recirculation of air within a building or zone. It also does not characterize the distribution of the outside air to individual spaces within a building or zone.

The **space time constant** compares the interior volume of a particular space to the total supply airflow rate to that space. The space time constant is the inverse of the space air exchange rate:

$$\tau_S = V/Q_{sa} \tag{7}$$

The space time constant includes the effect of recirculated air, if present, as well as that of outside air introduced to the space through the supply air. If infiltration is significant in a space, then the infiltration flow rate should be included when determining both the space air exchange rate and the space time constant.

Averaging Time-Varying Ventilation

When assessing time-varying ventilation in terms of controlling indoor air quality, the quantity of interest is often the temporal average rather than the peak. The concept of **effective ventilation** (Sherman and Wilson 1986; Yuill 1986, 1991) describes the proper ventilation rate averaging process. In this concept, the average (effective) rate is the steady-state rate that yields the same average contaminant concentration over the period of interest in the occupied space as does the actual sequence of time-varying discrete ventilation rates over the same period and in the same space. This effective rate is only equal to the simple arithmetic average rate when the discrete ventilation rates are constant over the period of interest and the contaminant concentration has reached its steady-state value. Simple arithmetic averaging of instantaneous ventilation rates or concentrations cannot generally be used to determine these averages because of the nonlinear response of indoor concentrations to ventilation rate variations.

An important constraint in the effective ventilation concept is that the contaminant source strength F must be constant over the period of interest or must be uncorrelated with the ventilation rate. These conditions are satisfied in many residential and commercial buildings because the emission rates of many contaminants that are controlled by whole-building ventilation vary slowly. Sherman and Wilson (1986) describe how to deal with pollutants that have step-wise constant emission rates. Pollutants such as carbon monoxide, radon, and formaldehyde, whose emission rates can be affected by ventilation, cannot be analyzed with this concept and require more complex analyses. For constant-source-strength pollutants, the relationship between effective air exchange rate, effective ventilation rate, volumetric flow, source strength, average concentration, and time-averaged effective turnover time is given by

$$I_m = \frac{\overline{Q}}{V} = \frac{F}{V\overline{C}} = \frac{1}{\overline{\tau}_e} \tag{8}$$

The time-averaged effective turnover time $\overline{\tau}_e$ in Equation (8) represents the characteristic time for the concentration in the occupied space to approach steady state over the period of interest. It can be determined from a sequence of discrete, instantaneous ventilation air change rates I_i using the following (Sherman and Wilson 1986):

$$\overline{\tau}_e = \frac{1}{N}\sum_{i=1}^{N}\tau_{e,\,i} \tag{9}$$

$$\text{for } I_i > 0, \tau_{e,\,i} = \frac{1 - \exp(-I_i\Delta t)}{I_i} + \tau_{e,\,i-1}\exp(-I_i\Delta t) \tag{10}$$

$$\text{for } I_i = 0, \tau_{e,\,i} = \Delta t + t_{e,\,i-1} \tag{11}$$

where

Δt = length of each discrete time period
$\overline{\tau}_e$ = time-averaged effective turnover time
$\overline{\tau}_{e,\,i}$ = instantaneous turnover time in period i
$\overline{\tau}_{e,\,i-1}$ = instantaneous turnover time in previous period

ASHRAE *Standard* 136 provides a set of factors to help calculate the annual effective air exchange rate.

Age of Air

The **age of air** θ_{age} (Sandberg 1981) is the length of time t that some quantity of outside air has been in a building, zone, or space. The "youngest" air is at the point where outside air enters the building by mechanical or natural ventilation or through infiltration (Grieve 1989). The "oldest" air may be at some location in the building or in the exhaust air. When the characteristics of the air distribution system are varied, age of air is inversely correlated with quality of outside air delivery. Units are of time, usually in seconds or minutes, so it is not a true efficiency or effectiveness measure. The age of air concept, however, has gained wide acceptance in Europe and is used increasingly in North America.

The age of air can be evaluated for existing buildings using tracer gas methods. Using either the decay (step-down) or growth (step-up) tracer gas method, the zone average or **nominal age of air** $\theta_{age,N}$ can be determined by taking concentration measurements in the exhaust air. The **local age of air** $\theta_{age,L}$ is evaluated through tracer gas measurements at any desired point in a space, such as at a worker's desk. When time-dependent data of tracer gas concentration are available, the age of air can be calculated from

$$\theta_{age} = \int_{\theta=0}^{\infty} \frac{C_{in} - C}{C_{in} - C_o} \, d\theta \quad (12)$$

where C_{in} is the concentration of tracer gas being injected.

Because evaluation of the age of air requires integration to infinite time, an exponential tail is usually added to the known concentration data (Farrington et al. 1990).

Air Change Effectiveness

Ventilation effectiveness is a description of an air distribution system's ability to remove internally generated pollutants from a building, zone, or space. **Air change effectiveness** is a description of an air distribution system's ability to deliver ventilation air to a building, zone, or space. The HVAC design engineer usually does not have knowledge or control of actual pollutant sources within buildings, so Table 6.1 of ASHRAE *Standard* 62.1 defines outdoor air requirements for typical, expected building uses. For most projects, therefore, air change effectiveness is of more relevance to HVAC system design than ventilation effectiveness. Various definitions for air change effectiveness have been proposed. The specific measure that meets local code requirements must be determined, if any is needed at all.

Air change effectiveness measures ε_I are nondimensional gages of ventilation air delivery. One common definition of air change effectiveness is the ratio of a time constant to an age of air:

$$\varepsilon_I = \tau/\theta_{age} \quad (13)$$

The **nominal air change effectiveness** $\varepsilon_{I,N}$ shows the effectiveness of outside air delivery to the entire building, zone, or space:

$$\varepsilon_{I,\,N} = \tau_N/\theta_{age,\,N} \quad (14)$$

where the nominal time constant τ_N is usually calculated from measured airflow rates.

The **local air change effectiveness** $\varepsilon_{I,L}$ shows the effectiveness of outside air delivery to one specific point in a space:

$$\varepsilon_{I,\,L} = \tau_N/\theta_{age,\,L} \quad (15)$$

where τ_N is found either through airflow measurements or from tracer gas concentration data. An $\varepsilon_{I,L}$ value of 1.0 indicates that the air distribution system delivers air equivalent to that of a system with perfectly mixed air in the spaces. A value less than 1.0 shows less than perfect mixing with some degree of stagnation. A value of $\varepsilon_{I,L}$ greater than 1.0 suggests that a degree of plug or displacement flow is present at that point (Rock 1992).

An HVAC design engineer often assumes that a properly designed, installed, operated, and maintained air distribution system provides an air change effectiveness of about 1. However, Table 6.1 of ASHRAE *Standard* 62.1 provides some estimates of effectiveness for operating in heating or cooling mode, and with various air distribution techniques. These values are then adjusted for commercial and institutional building design when the ventilation rate procedure is used. If the indoor air quality procedure of *Standard* 62.1 is used, then actual pollutant sources and the air change effectiveness must be known for the successful design of HVAC systems that have fixed ventilation airflow rates.

ASHRAE *Standard* 129 describes a method for measuring air change effectiveness of mechanically vented spaces and buildings with limited air infiltration, exfiltration, and air leakage with surrounding indoor spaces.

TRACER GAS MEASUREMENTS

The only reliable way to determine an existing building's air exchange rate is to measure it. Several tracer gas measurement procedures exist (including the ASTM *Standard* E741 test method), all involving an inert or nonreactive gas used to label the indoor air (Charlesworth 1988; Dietz et al. 1986; Fisk et al. 1989; Fortmann et al. 1990; Harrje et al. 1981, 1990; Hunt 1980; Lagus 1989; Lagus and Persily 1985; Persily 1988; Persily and Axley 1990; Sherman 1989a, 1989b, 1990; Sherman et al. 1980). The tracer is released into the building in a specified manner, and the concentration of the tracer in the building is monitored and related to the building's air exchange rate. Various tracer gases and associated concentration detection devices have been used. Desirable qualities of a tracer gas are detectability, nonreactivity, nontoxicity, neutral buoyancy, relatively low concentration in ambient air, and low cost (Hunt 1980).

All tracer gas measurement techniques are based on a mass balance of the tracer gas in the building. Assuming the outdoor concentration is zero and the indoor air is well mixed, this total balance takes the following form:

$$V\left(\frac{dC}{d\theta}\right) = F(\theta) - Q(\theta)C(\theta) \quad (16)$$

where

V = volume of space being tested, ft^3
$C(\theta)$ = tracer gas concentration at time θ
$dC/d\theta$ = time rate of change of concentration, min^{-1}
$F(\theta)$ = tracer gas injection rate at time θ, cfm
$Q(\theta)$ = airflow rate out of building at time θ, cfm
θ = time, min

In Equation (16), density differences between indoor and outdoor air are generally ignored for moderate climates; therefore, Q also refers to the airflow rate into the building. Although Q is often referred to as the infiltration rate, any measurement includes both mechanical and natural ventilation in addition to infiltration. The ratio of Q to the volume V being tested has units of 1/time (often converted to ach) and is the air exchange rate I.

Equation (16) is based on the assumptions that (1) no unknown tracer gas sources exist, (2) airflow out of the building is the dominant means of removing the tracer gas from the space (i.e., the tracer gas does not react chemically in the space and/or is not adsorbed onto or by interior surfaces), and (3) the tracer gas concentration within the building can be represented by a single value (i.e., the tracer gas is uniformly mixed within the space). In such tracer gas experiments, box-type fans are often placed and operated within rooms to enhance mixing.

Three different tracer gas procedures are used to measure air exchange rates: (1) decay or growth, (2) constant concentration, and (3) constant injection.

Decay or Growth

Decay. The simplest tracer gas measurement technique is the decay method (also known as the step-down method). A small amount of tracer gas is injected into the space and is allowed to mix with the interior air. After the injection, $F = 0$ and then the solution to Equation (16) is

$$C(\theta) = C_o e^{-I\theta} \quad (17)$$

where C_o is the concentration of the tracer in the space at $\theta = 0$.

Equation (17) is generally used to solve for I by measuring the tracer gas concentration periodically during the decay and fitting the data to the logarithmic form of Equation (17):

$$\ln C(\theta) = \ln C_o - I\theta \quad (18)$$

Like all tracer gas techniques, the decay method has advantages and disadvantages. One advantage is that, because logarithms of concentration are taken, only relative concentrations are needed, which can simplify calibration of concentration-measuring equipment. Also, the tracer gas injection rate need not be measured, although it must be controlled so that the tracer gas concentrations are within the range of the concentration-measuring device. The concentration-measuring equipment can be located on site, or building samples can be collected in suitable containers, such as grab bags, and analyzed elsewhere.

The most serious problem with the decay technique is imperfect mixing of tracer gas with interior air, both at initial injection and during decay. Equations (16) and (17) assume that the tracer gas concentration within the building is uniform. If the tracer is not well mixed, this assumption is not appropriate and the determination of I is subject to errors. It is difficult to estimate the magnitude of errors caused by poor mixing, and there has been little analysis of this problem. Sometimes a two-zone model is applied to a room, and a mixing coefficient selected, to estimate the effect of poor mixing (e.g., Rock 1992).

Growth. The growth or step-up method is similar to the decay method except that the initial tracer gas concentration is low and the injected tracer gas is increased suddenly during the test.

Constant Concentration

In the constant concentration technique, the tracer gas injection rate is adjusted to maintain a constant concentration within the building. If the concentration is truly constant, then Equation (16) reduces to

$$Q(\theta) = F(\theta)/C \tag{19}$$

There is less experience with this technique than with the decay procedure, but an increasing number of applications exist (Bohac et al. 1985; Collet 1981; Fortmann et al. 1990; Kumar et al. 1979; Walker and Forest 1995; Walker and Wilson 1998; Wilson and Walker 1993).

Because tracer gas injection is continuous, no initial mixing period is required. Another advantage is that tracer gas injection into each zone of the building can be separately controlled; thus, the amount of outdoor air flowing into each zone can be determined. This procedure is best suited for longer-term continuous monitoring of fluctuating infiltration rates. One disadvantage is that it requires measurement of absolute tracer concentrations and injection rates. Also, imperfect mixing of the tracer and interior air causes a delay in the response of the concentration to changes in the injection rate.

Constant Injection

In the constant-injection procedure, the tracer is injected at a constant rate, and the solution to Equation (16) becomes

$$C(\theta) = (F/Q)(1 - e^{-I\theta}) \tag{20}$$

After sufficient time, the transient term reduces to zero, the concentration attains equilibrium, and Equation (20) reduces to

$$Q = F/C \tag{21}$$

Equation (21) is valid only when air exchange rate I and airflow rate Q are constant; thus, this technique is only appropriate for systems at or near equilibrium. It is particularly useful in spaces with mechanical ventilation or with high air exchange rates. Constant injection requires measurement of absolute concentrations and injection rates.

Dietz et al. (1986) used a special case of the constant-injection technique, using permeation tubes as a tracer gas source. The tubes release the tracer at an ideally constant rate into the building being tested, and a sampling tube packed with an adsorbent collects the tracer from the interior air at a constant rate by diffusion. After a sampling period of one week or more, the sampler is removed and analyzed to determine the average tracer gas concentration within the building during the sampling period.

Solving Equation (16) for C and taking the time average gives

$$\langle C\rangle = \langle F/Q\rangle = F\langle 1/Q\rangle \tag{22}$$

where $\langle \ . \ . \ . \ \rangle$ denotes time average. Note that the time average of dC/dt is assumed to equal zero.

Equation (22) shows that the average tracer concentration $\langle C\rangle$ and injection rate F can be used to calculate the average of the inverse airflow rate. The average of the inverse is less than the inverse of the actual average, with the magnitude of this difference depending on the distribution of airflow rates during the measurement period. Sherman and Wilson (1986) calculated these differences to be about 20% for one-month averaging periods. Differences greater than 30% have been measured when occupant airing of houses caused large changes in air exchange rate; errors from 5 to 30% were measured when the variation was caused by weather effects (Bohac et al. 1987). Longer averaging periods and large changes in air exchange rates during the measurement periods generally lead to larger differences between the average inverse exchange rate and the inverse of the actual average rate.

Multizone Air Exchange Measurement

Equation (16) is based on the assumption of a single, well-mixed enclosure, and the techniques described are for single-zone measurements. Airflow between internal zones and between the exterior and individual internal zones has led to the development of multizone measurement techniques (Fortmann et al. 1990; Harrje et al. 1985; Harrje et al. 1990; Sherman and Dickerhoff 1989). These techniques are important when considering the transport of pollutants from one room of a building to another. A theoretical development is provided by Sinden (1978a). Multizone measurements typically use either multiple tracer gases for the different zones or the constant-concentration technique. A proper error analysis is essential in all multizone flow determination (Charlesworth 1988; D'Ottavio et al. 1988).

DRIVING MECHANISMS FOR VENTILATION AND INFILTRATION

Natural ventilation and infiltration are driven by pressure differences across the building envelope caused by wind and air density differences because of temperature differences between indoor and outdoor air (buoyancy, or the stack effect). Mechanical air-moving systems also induce pressure differences across the envelope through operation of appliances, such as combustion devices, leaky forced-air thermal distribution systems, and mechanical ventilation systems. The indoor/outdoor pressure difference at a location depends on the magnitude of these driving mechanisms as well as on the characteristics of the openings in the building envelope (i.e., their locations and the relationship between pressure difference and airflow for each opening).

Stack Pressure

Stack pressure is the hydrostatic pressure caused by the weight of a column of air located inside or outside a building. It can also occur within a flow element, such as a duct or chimney that has vertical separation between its inlet and outlet. The hydrostatic pressure in the air depends on density and the height of interest above a reference point.

Air density is a function of local barometric pressure, temperature, and humidity ratio, as described in Chapter 1. As a result, standard conditions should not be used to calculate the density. For example, a building site at 5000 ft has air density that is about 20% less than if the building were at sea level. An air temperature increase from –20 to 70°F causes a similar air density difference. Combined, these elevation and temperature effects reduce air density about 45%. Moisture effects on density are generally negligible, so dry air density can be used instead, except in hot, humid climates when air is hot and close to saturation. For example, saturated air at 105°F has density about 5% less than that of dry air.

Assuming temperature and barometric pressure are constant over the height of interest, the stack pressure decreases linearly as the separation above the reference point increases. For a single column of air, the stack pressure can be calculated as

$$p_s = p_r - 0.00598\rho g H \tag{23}$$

where

p_s = stack pressure, in. of water
p_r = stack pressure at reference height, in. of water
g = gravitational acceleration, 32.2 ft/s^2
ρ = indoor or outdoor air density, lb$_m$/ft^3
H = height above reference plane, ft
0.00598 = unit conversion factor, (in. of water)·ft·s^2/lb$_m$

For tall buildings or when significant temperature stratification occurs indoors, Equation (23) should be modified to include the density gradient over the height of the building.

Temperature differences between indoors and outdoors cause stack pressure differences that drive airflows across the building envelope; the **stack effect** is this buoyancy phenomenon. Sherman (1991) showed that any single-zone building can be treated as an equivalent box from the point of view of stack effect, if its leaks follow the power law as described in the section on Residential Air Leakage. The building is then characterized by an effective stack height and neutral pressure level (NPL) or leakage distribution, as described in the section on Neutral Pressure Level. Once calculated, these parameters can be used in physical, single-zone models to estimate infiltration.

Neglecting vertical density gradients, the stack pressure difference for a horizontal leak at any vertical location is given by

$$\begin{aligned}\Delta p_s &= 0.00598(\rho_o - \rho_i)g(H_{NPL} - H)\\ &= 0.00598\rho_o\left(\frac{T_i - T_o}{T_i}\right)g(H_{NPL} - H)\end{aligned} \tag{24}$$

where

T_o = outdoor temperature, °R
T_i = indoor temperature, °R
ρ_o = outdoor air density, lb/ft^3
ρ_i = indoor air density, lb/ft^3
H_{NPL} = height of neutral pressure level above reference plane without any other driving forces, ft

Chastain and Colliver (1989) showed that, when there is stratification, the average of the vertical distribution of temperature differences is more appropriate to use in Equation (24) than the localized temperature difference near the opening of interest.

By convention, stack pressure differences are positive when the building is pressurized relative to outdoors, which causes flow out of the building. Therefore, absent other driving forces and assuming no stack effect within the flow elements themselves, when indoor air is warmer than outdoors, the base of the building is depressurized and the top is pressurized relative to outdoors; when indoor air is cooler than outdoors, the reverse is true.

Absent other driving forces, the location of the NPL is influenced by leakage distribution over the building exterior and by interior compartmentation. As a result, the NPL is not necessarily located at the mid-height of the building; with effective horizontal barriers in tall buildings, it is also possible to have more than one NPL. NPL location and leakage distribution are described later in the section on Combining Driving Forces.

For a penetration through the building envelope for which (1) there is vertical separation between its inlet and outlet and (2) air inside the flow element is not at the indoor or outdoor temperature, such as in a chimney, more complex analyses than Equation (24) are required to determine the stack effect at any location on the building envelope.

Wind Pressure

When wind impinges on a building, it creates a distribution of static pressures on the building's exterior surface that depends on the wind direction, wind speed, air density, surface orientation, and surrounding conditions. Wind pressures are generally positive with respect to the static pressure in the undisturbed airstream on the windward side of a building and negative on the leeward sides. However, pressures on these sides can be negative or positive, depending on wind angle and building shape. Static pressures over building surfaces are almost proportional to the velocity head of the undisturbed airstream. The wind pressure or velocity head is given by the Bernoulli equation, assuming no height change or pressure losses:

$$p_w = 0.0129 C_p \rho \frac{U^2}{2} \tag{25}$$

where

p_w = wind surface pressure relative to outdoor static pressure in undisturbed flow, in. of water
ρ = outside air density, lb$_m$/ft^3 (about 0.075 at or near sea level)
U = wind speed, mph
C_p = wind surface pressure coefficient, dimensionless
0.0129 = unit conversion factor, (in. of water)·ft^3/lb$_m$·mph^2

C_p is a function of location on the building envelope and wind direction. Chapter 24 provides additional information on values of C_p.

Most pressure coefficient data are for winds normal to building surfaces. Unfortunately, for a real building, this fixed wind direction rarely occurs, and when the wind is not normal to the upwind wall, these pressure coefficients do not apply. Walker and Wilson (1994) developed a harmonic trigonometric function to interpolate between the surface average pressure coefficients on a wall that were measured with the wind normal to each of the four building surfaces. This function was developed for low-rise buildings three stories or less in height. For each wall of the building, C_p is given by

$$\begin{aligned}C_p(\phi) = 1/2\{&[C_p(1) + C_p(2)](\cos^2\phi)^{1/4}\\ &+ [C_p(1) - C_p(2)](\cos\phi)^{3/4}\\ &+ [C_p(3) - C_p(4)](\sin^2\phi)^2\\ &+ [C_p(3) - C_p(4)]\sin\phi\}\end{aligned} \tag{26}$$

where

$C_p(1)$ = pressure coefficient when wind is at 0°
$C_p(2)$ = pressure coefficient when wind is at 180°
$C_p(3)$ = pressure coefficient when wind is at 90°
$C_p(4)$ = pressure coefficient when wind is at 270°
ϕ = wind angle measured clockwise from the normal to wall 1

Because the cosine term in Equation (26) can be negative, its sign must be tracked. When cos(ϕ) is negative, subtract the value of the absolute of cos(ϕ) to the 3/4 power.

The measured data used to develop the harmonic function from Akins et al. (1979) and Wiren (1985) show that typical values for the pressure coefficients are $C_p(1) = 0.6$, $C_p(2) = -0.3$, $C_p(3) = C_p(4) = -0.65$. Because of geometry effects on flow around a building, application of this interpolation function is limited to low-rise buildings of rectangular plan form (i.e., not L-shaped) with the longest wall less than three times the length of the shortest wall. For less regular buildings, simple correlations are inadequate and building-specific pressure coefficients are required. Chapter 24 discusses wind pressures for complex building shapes and for high-rise buildings in more detail.

The wind speed most commonly available for infiltration calculations is that measured at the local weather station, typically the nearest airport. This wind speed needs to be corrected for reductions caused by the difference between the height where the wind speed is measured and the height of the building, and reductions caused by shelter effects.

The reference wind speed used to determine pressure coefficients is usually the wind speed at the eaves height for low-rise buildings and the building height for high-rise buildings. However, meteorological wind speed measurements are made at a different height (typically 33 ft) and at a different location. The difference in terrain between the measurement station and the building under study must also be accounted for. Chapter 24 shows how to calculate the effective wind speed U_H from the reference wind speed U_{met} using boundary layer theory and estimates of terrain effects.

In addition to the reduction in wind pressures caused by reduced wind speed, the effects of local shelter also act to reduce wind pressures. The shielding effects of trees, shrubbery, and other buildings within several building heights of a particular building produce large-scale turbulence eddies that not only reduce effective wind speed but also alter wind direction. Thus, meteorological wind speed data must be reduced carefully when applied to low buildings.

Ventilation rates measured by Wilson and Walker (1991) for a row of houses showed reductions in ventilation rates of up to a factor of three when the wind changed direction from perpendicular to parallel to the row. They recommended estimating wind shelter for winds perpendicular to each side of the building and then using the interpolation function in Equation (27) to find the wind shelter for intermediate wind angles:

$$s = \frac{1}{2}\left\{\begin{array}{l}[s(1)+s(2)]\cos^2\phi + [s(1)-s(2)]\cos\phi \\ \quad + [s(3)+s(4)]\sin^2\phi + [s(3)-s(4)]\sin\phi\end{array}\right\} \tag{27}$$

where

s = shelter factor for the particular wind direction ϕ
$s(i)$ = shelter factor when wind is normal to wall i (i = 1 to 4, for four sides of a building)

Although this method gives a realistic variation of wind shelter effects with wind direction, estimates for numerical values of wind shelter factor s for each of the four cardinal directions must be provided. Table 8 in the section on Residential Calculation Examples lists typical shelter factors. The wind speed used in Equation (25) is then given by

$$U = sU_H \tag{28}$$

The magnitude of pressure differences found on the surfaces of buildings varies rapidly with time because of turbulent fluctuations in the wind (Etheridge and Nolan 1979; Grimsrud et al. 1979). However, using average wind pressures to calculate pressure differences is usually sufficient to calculate average infiltration values.

Mechanical Systems

Operation of mechanical equipment, such as supply or exhaust systems and vented combustion devices, affects pressure differences across the building envelope. Interior static pressure adjusts such that the sum of all airflows through openings in the building envelope plus equipment-induced airflows balance to zero. To predict these changes in pressure differences and airflow rates caused by mechanical equipment, the location of each opening in the envelope and the relationship between pressure difference and airflow rate for each opening must be known. The interaction between mechanical ventilation system operation and envelope airtightness has been discussed for low-rise buildings (Nylund 1980) and for office buildings (Persily and Grot 1985a; Tamura and Wilson 1966, 1967a).

Air exhausted from a building by a whole-building exhaust system must be balanced by increasing airflow into the building through other openings. As a result, airflow at some locations changes from outflow to inflow. For supply fans, the situation is reversed and envelope inflows become outflows. Thus, the effects of a mechanical system on a building must be considered. Depressurization caused by an improperly designed exhaust system can increase the rate of radon entry into a building and interfere with proper operation of combustion device venting or other exhaust systems. Depressurization can also force moist outdoor air through the building envelope; for example, during the cooling season in hot, humid climates, moisture may condense within the building envelope and cause rust, rot, or mold. A similar phenomenon, but in reverse, can occur during the heating (and potentially humidifying) season in cold climates if the building is pressurized.

The interaction between mechanical systems and the building envelope also pertains to systems serving zones of buildings. Performance of zone-specific exhaust or pressurization systems is affected by leakage in zone partitions as well as in exterior walls.

Mechanical systems can also create infiltration-driving forces in single-zone buildings. Specifically, some single-family houses with central forced-air duct systems have multiple supply registers, yet only a central return grille. When internal doors are closed in these houses, large positive indoor/outdoor pressure differentials are created for rooms with only supply registers, whereas the room or hallway with the return grille tends to depressurize relative to outside. This is caused by the resistance of internal door undercuts, often partially blocked by carpeting, to flow from the supply register to the return (Modera et al. 1991). The magnitudes of the indoor/outdoor pressure differentials created have been measured to average 0.012 to 0.024 in. of water (Modera et al. 1991). Balanced airflow systems, with ducted air return and distributed grilles, or adequately sized transfer grilles (where allowed by fire code) reduce this significantly.

Building envelope airtightness and interzonal airflow resistance can also affect performance of mechanical systems. The actual airflow rate delivered by these systems, particularly ventilation systems, depends on the pressure they work against. This effect is the same as the interaction of a fan with its associated ductwork, which is discussed in Chapter 21 of this volume and Chapter 20 of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment*. The building envelope and its leakage must be considered part of the ductwork in determining the pressure drop of the system.

Duct leakage can cause similar problems. Supply leaks to the outside tend to depressurize the building; return leaks to the outside tend to pressurize it. Keeping ducts within the conditioned buildings, and sealing ducts well with durable materials and high-quality construction methods, significantly reduces this problem.

Combining Driving Forces

Pressure differences caused by wind, stack effect, and mechanical systems are considered in combination by adding them together and then determining the resulting airflow rate through each building envelope. The airflows must be determined in this manner, as opposed to adding the airflow rates due to the separate driving forces, because the airflow rate through each opening is not linearly related to pressure difference.

For uniform indoor air temperatures, the total pressure difference across each leak can be written in terms of a reference wind parameter P_U and stack effect parameter P_T common to all leaks:

$$P_U = \rho_o \frac{U_H^2}{2} \tag{29}$$

$$P_T = g\rho_o[(T_i - T_o)/T_i] \tag{30}$$

where T is air temperature, in °R.

The pressure difference across each leak, with positive pressures for flow into the building, is then given by

$$\Delta p = 0.0129 s^2 C_p P_U + HP_T + \Delta p_I \tag{31}$$

where Δp_I is the pressure that acts to balance inflows and outflows, including mechanical system flows. Equation (31) can then be applied to every leak for the building with appropriate values of C_p, s, and H. Thus, each leak is defined by its pressure coefficient, shelter, and height. Where indoor pressures are not uniform, more complex analyses are required.

Neutral Pressure Level

The neutral pressure level (NPL) is that location or locations in the building envelope where there is no indoor-to-outdoor pressure difference. Internal partitions, stairwells, elevator shafts, utility ducts, chimneys, vents, operable windows, and mechanical supply and exhaust systems complicate the prediction of NPL location. An opening with a large area relative to the total building leakage causes the NPL to shift toward the opening. In particular, chimneys and openings at or above roof height raise the NPL in small buildings. Exhaust systems increase the height of the NPL; outdoor air supply systems lower it.

Figure 6 qualitatively shows the addition of driving forces for a building with uniform openings above and below mid-height and without significant internal resistance to airflow. The slopes of the pressure lines are a function of the densities of the indoor and outdoor air. In Figure 6A, with inside air warmer than outside and pressure differences caused solely by thermal forces, the NPL is at mid-height, with inflow through lower openings and outflow through higher openings. Direction of flow is always from the higher to the lower pressure region.

Figure 6B presents qualitative uniform pressure differences caused by wind alone, with opposing effects on the windward and leeward sides. When temperature difference and wind effects both exist, the pressures caused by each are added together to determine the total pressure difference across the building envelope. In Figure 6B, there is no NPL because no locations on the building envelope have zero pressure difference. Figure 6C shows the combination, where the wind force of Figure 6B has just balanced the thermal force of Figure 6A, causing no pressure difference at the top windward or bottom leeward side.

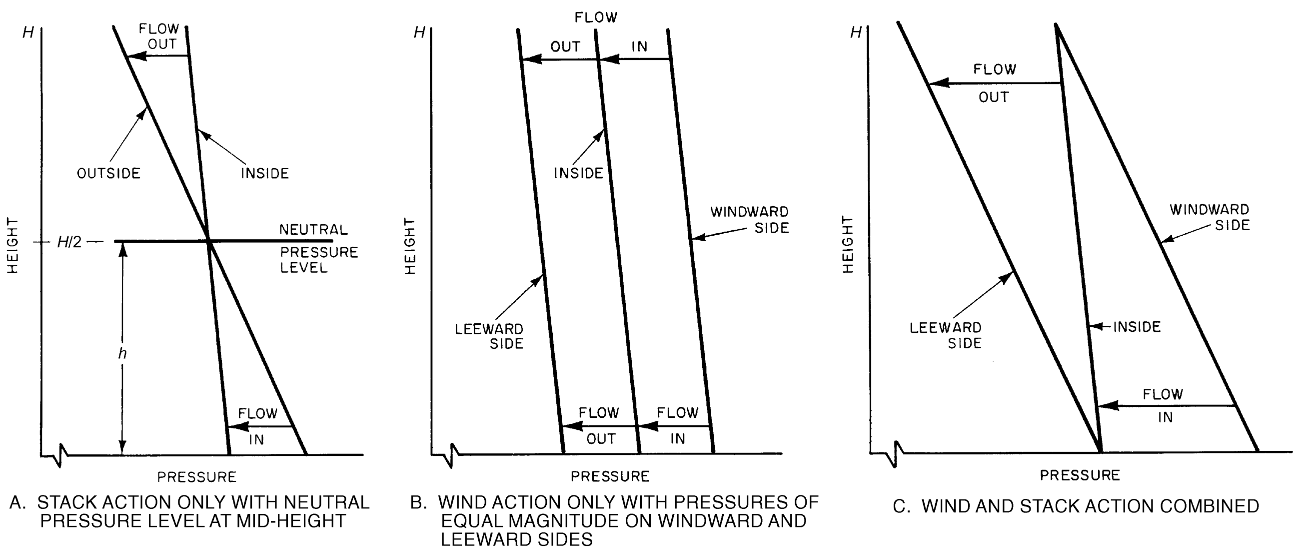
The relative importance of wind and stack pressures in a building depends on building height, internal resistance to vertical airflow, location and flow resistance characteristics of envelope openings, local terrain, and the immediate shielding of the building. The taller the building and the smaller its internal resistance to airflow, the stronger the stack effect. The more exposed a building is, the more susceptible it is to wind. For any building, there are ranges of wind speed and temperature difference for which the building's infiltration is dominated by stack effect, wind, or the driving pressures of both (Sinden 1978b). These building and terrain factors determine, for specific values of temperature difference and wind speed, in which regime the building's infiltration lies.

The effect of mechanical ventilation on envelope pressure differences is more complex and depends on both the direction of ventilation flow (exhaust or supply) and the differences in these ventilation flows among the zones of the building. If mechanically supplied outdoor air is provided uniformly to each story, the change in the exterior wall pressure difference pattern is uniform. With a nonuniform supply of outdoor air (for example, to one story only), the extent of pressurization varies from story to story and depends on internal airflow resistance. Pressurizing all levels uniformly has little effect on pressure differences across floors and vertical shaft enclosures, but pressurizing individual stories increases the pressure drop across these internal separations. Pressurizing the ground level is often used in tall buildings in winter to reduce negative air pressures across entries.

Available data on the NPL in various kinds of buildings are limited. The NPL in tall buildings varies from 0.3 to 0.7 of total building height (Tamura and Wilson 1966, 1967b). For houses, especially houses with chimneys, the NPL is usually above mid-height. Operating a combustion heat source with a flue raises the NPL further, sometimes above the ceiling (Shaw and Brown 1982).

Thermal Draft Coefficient

Compartmentation of a building also affects the NPL location. Equation (24) provides a maximum stack pressure difference, given no internal airflow resistance. The sum of pressure differences across the exterior wall at the bottom and top of the building, as calculated by these equations, equals the total theoretical draft for the building. The sum of actual top and bottom pressure differences, divided by the total theoretical draft pressure difference, equals the **thermal draft coefficient**. The value of the thermal draft coefficient depends on the airflow resistance of exterior walls relative to the airflow resistance between floors. For a building without internal partitions, the total theoretical draft is achieved across the exterior walls (Figure 7A), and the thermal draft coefficient equals 1. In a building with airtight separations at each floor, each story acts



A. STACK ACTION ONLY WITH NEUTRAL PRESSURE LEVEL AT MID-HEIGHT

B. WIND ACTION ONLY WITH PRESSURES OF EQUAL MAGNITUDE ON WINDWARD AND LEEWARD SIDES

C. WIND AND STACK ACTION COMBINED

Fig. 6 Distribution of Inside and Outside Pressures over Height of Building

independently, its own stack effect being unaffected by that of any other floor (Figure 7B). The theoretical draft is minimized in this case, and each story has an NPL.

Real multistory buildings are neither open inside (Figure 7A), nor airtight between stories (Figure 7B). Vertical air passages, stairwells, elevators, and other service shafts allow airflow between floors. Figure 7C represents a heated building with uniform openings in the exterior wall, through each floor, and into the vertical shaft at each story. Between floors, the slope of the line representing the inside pressure is the same as that shown in Figure 7A, and the discontinuity at each floor (Figure 7B) represents the pressure difference across it. Some of the pressure difference maintains flow through openings in the floors and vertical shafts. As a result, the pressure difference across the exterior wall at any level is less than it would be with no internal flow resistance.

Maintaining airtightness between floors and from floors to vertical shafts is a way to control indoor/outdoor pressure differences because of the stack effect and, therefore, infiltration. Good separation is also conducive to proper operation of mechanical ventilation and smoke management systems. However, care is needed

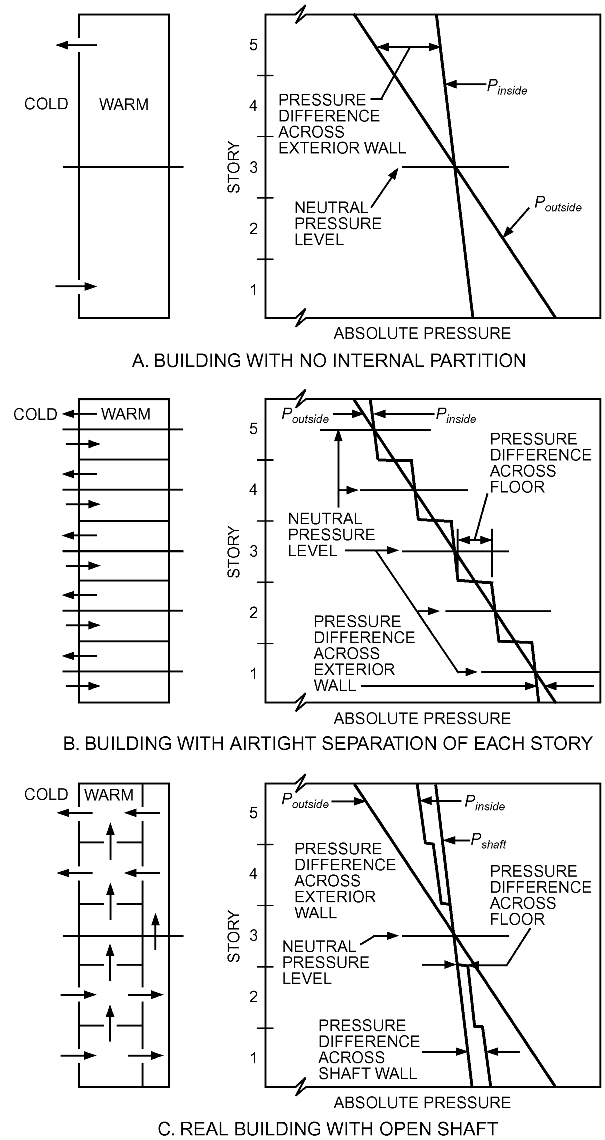


Fig. 7 Compartmentation Effect in Buildings

to avoid pressure differences that could prevent door opening in an emergency. Tamura and Wilson (1967a) showed that when vertical shaft leakage is at least two times envelope leakage, the thermal draft coefficient is almost one and the effect of compartmentation is negligible. Measurements of pressure differences in three tall office buildings by Tamura and Wilson (1967b) indicated that the thermal draft coefficient ranged from 0.8 to 0.9 with ventilation systems off.

INDOOR AIR QUALITY

Outdoor air requirements for acceptable indoor air quality (IAQ) have long been debated, and different rationales have produced radically different ventilation standards (Grimsrud and Teichman 1989; Janssen 1989; Klauss et al. 1970; Yaglou et al. 1936; Yaglou and Witheridge 1937). Historically, the major considerations have included the amount of outdoor air required to control moisture, carbon dioxide (CO_2), odors, and tobacco smoke generated by occupants. These considerations have led to prescriptions of a minimum rate of outdoor air supply per occupant. More recently, a major concern has been maintaining acceptable indoor concentrations of various additional pollutants that are not generated primarily by occupants. Engineering experience and field studies indicate that an outdoor air supply of about 20 cfm per person is very likely to provide acceptable perceived indoor air quality in office spaces, whereas lower rates may lead to increased sick building syndrome symptoms (Apte et al. 2000; Mendell 1993; Seppanen et al. 1999). Information on contaminants can be found in Chapter 11, and odors are covered in Chapter 12.

Indoor pollutant concentrations depend on the strength of pollutant sources and the total rate of pollutant removal. Pollutant sources include outdoor air; indoor sources such as occupants, furnishings, and appliances; dirty ventilation system ducts and filters; soil adjacent to the building; and building materials themselves, especially when new. Pollutant removal processes include dilution with outside air, local exhaust ventilation, deposition on surfaces, chemical reactions, and air-cleaning processes. If (1) general building ventilation is the only significant pollutant removal process, (2) indoor air is thoroughly mixed, and (3) pollutant source strength and ventilation rate have been stable for a sufficient period, then the steady-state indoor pollutant concentration is given by

$$C_i = C_o + 10^6 S/Q_{oa} \tag{32}$$

where

C_i = steady-state indoor concentration, ppm
C_o = outdoor concentration, ppm
S = total pollutant source strength, cfm
Q_{oa} = ventilation rate, cfm

Variation in pollutant source strengths (rather than variation in ventilation rate) is considered the largest cause of building-to-building variation in concentrations of pollutants that are not generated by occupants. Turk et al. (1989) found that a lack of correlation between average indoor respirable particle concentrations and whole-building outdoor ventilation rate indicated that source strength, high outdoor concentrations, building volume, and removal processes are important. Because pollutant source strengths are highly variable, maintaining minimum ventilation rates does not ensure acceptable indoor air quality in all situations. The lack of health-based concentration standards for many indoor air pollutants, primarily because of the lack of health data, makes the specification of minimum ventilation rates difficult.

In cases of high contaminant source strengths, such as with indoor sanding, spray painting, or smoking, impractically high rates of dilution ventilation are required to control contaminant levels, and other methods of control are more effective. Removal or reduction of contaminant sources is the most effective means of control. Controlling a localized source by means of local exhaust, such as range hoods or bathroom exhaust fans, as well as filtration and absorption, may also be effective [e.g., Rock (2006)].

Particles can be removed with various types of air filters. Gaseous contaminants with higher molecular weight can be controlled with activated carbon or alumina pellets impregnated with a substance such as potassium permanganate. Chapter 28 of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* has information on air cleaning.

Protection from Extraordinary Events

The design, operation and maintenance of a building's ventilation system, envelope, and other factors can significantly affect the building's potential vulnerability to extraordinary threats, which range from intentional releases of chemical or biological agents inside or outside a building, to releases of chemicals in industrial or transportation accidents, to natural disasters. ASHRAE (2003) addresses several key steps to manage risk from extraordinary incidents, including the following:

- Evaluate the risk to a facility of an extraordinary incident.
- Assess the building's vulnerability.
- Determine the degree of acceptable vulnerability.
- Consider protective measures or options in relation to new, renovated, and existing buildings.

Persily (2004) details how ventilation affects buildings' vulnerability to airborne chemical and biological releases, as well as some strategies for using ventilation (particularly involving airtightness and pressurizing the building interior to protect against outdoor releases) to increase the level of building protection against such incidents. Persily et al. (2007) evaluate retrofit options for building protection from airborne threats; approaches considered include enhanced particle filtration, sorbent-based gaseous air cleaning, ventilation system recommissioning, building envelope airtightening, building pressurization, relocation of outdoor air intakes, shelter-in-place (SIP), isolation of vulnerable spaces such as lobbies, and system shutdown and purge cycles. The filtration and air cleaning options have the advantage of always being operational as long as the systems are properly designed, installed, and maintained. However, the lack of standard test methods is a critical issue in application of some air-cleaning technologies. Building envelope air sealing and pressurization can be quite effective in protecting against outdoor releases as long as effective filtration against the contaminant of concern is also in place. The protection provided by operational changes such as system shutdown and purging depends heavily on timing; if timing is inappropriate, occupant exposure may increase. Isolating vulnerable zones and other system-related modifications depend on building layout and system design, and careful implementation is necessary for effectiveness under the range of conditions that exist in buildings. Finally, many retrofits also increase energy efficiency and improve indoor air quality, which should be included in a life-cycle cost comparison of different options to the degree possible. Chapter 58 of the 2007 *ASHRAE Handbook—HVAC Applications* addresses this topic of extraordinary events further.

THERMAL LOADS

Outdoor air introduced into a building constitutes a large part of the total space-conditioning (heating, cooling, humidification, and dehumidification) load, which is one reason to limit air exchange rates in buildings to the minimum required. Air exchange typically represents 20 to 50% of a building's thermal load. Chapters 17 and 18 cover thermal loads in more detail.

Air exchange increases a building's thermal load in several ways. First, incoming air must be heated or cooled from the outdoor air temperature to the indoor or supply air temperature. The rate of energy consumption by this sensible heating or cooling is given by

$$q_s = 60Q\rho c_p \Delta t \tag{33}$$

where

q_s = sensible heat load, Btu/h
Q = airflow rate, cfm
ρ = air density, lb/ft^3 (about 0.075 at or near sea level)
c_p = specific heat of air, Btu/lb·°F (about 0.24)
Δt = temperature difference between indoors and outdoors, °F

and at or near sea-level air density, with an adjustment for typical room air humidity, this equation is commonly presented for design use as

$$q_s = 1.1Q\Delta t \tag{34}$$

Equation (33) is known as the **sensible heat equation**. HVAC designers typically assume sea-level air pressure for locations with altitudes of 2000 ft or lower. A method to adjust for elevation is provided in Chapter 18.

Air exchange also modifies the moisture content of the air in a building. The rate of energy consumption associated with these latent loads (neglecting the energy associated with any condensate) is given by

$$q_l = 60Q\rho\Delta W(1061 + 0.444t) \tag{35}$$

where

q_l = latent heat load, Btu/h
ΔW = humidity ratio difference between indoors and outdoors, lb$_m$ water/lb$_m$ dry air
t = average of indoor and outdoor temperatures, °F

Equation (35) is known as the **latent heat equation**. When at or near sea level, and for common comfort air temperatures, the right-hand side of Equation (35) is approximately $4840Q\Delta W$.

Example 1. A makeup air unit (MAU) is to condition 5000 cfm of outdoor air in the winter for a building in Atlanta, Georgia. If the air is to be delivered directly to the occupied spaces at 75°F and 30% rh, how much sensible and latent heat must be added to this ventilation air at winter design conditions?

Solution: From the weather data tables provided on the CD included with this volume, Atlanta is at an elevation of about 1000 ft. Because this below the rule-of-thumb cutoff of 2000 ft for assuming sea-level conditions, air density is assumed to be 0.075 lb$_m$/ft^3. Also from the Atlanta data table, the winter 99% design dry-bulb (db) temperature is 23.9°F, but a mean coincident wet bulb is not provided. However, for humidification design, a dew-point temperature of 1.9°F is given along with its 24.9°F mean coincident dry bulb (MCDB). Using these data, a 1°F dew point is assumed as the 99% mean coincident dew point.

From ASHRAE's sea-level psychrometric chart and the winter design conditions, the desired humidity ratio W of the 75°F, 30% rh makeup air is about 0.0056 lb$_m$w/lb$_m$da. For the very dry outside air, with a dew point of 1°F, a problem occurs: the standard sea-level psychrometric chart does not extend below 32°F. Designers often assume that air below this temperature has $W = 0$, and this assumption gives conservative results. However, both high- and low-temperature psychrometric charts are available from ASHRAE, as is a table of moist air properties at standard conditions in Chapter 1. From this table, saturated air at 1°F, which is also its dew point, has a humidity ratio of 0.0008298 lb$_m$w/lb$_m$da. With 5000 cfm of outdoor air to be conditioned, and using the sensible and latent heat equations for sea level, the energy needed to condition this outdoor air is

$$q_s = 1.1Q\Delta T = 1.1 \times 5000 \text{ cfm}(75 - 23.9°\text{F})$$
$$= 281{,}050 \text{ Btu/h} \approx 282{,}000 \text{ Btu/h}$$

and

$$q_l = 4840Q\Delta W = 4840 \times 5000 \text{ cfm}(0.0056 - 0.0008298 \text{ lb}_m\text{w/lb}_m\text{da})$$
$$= 115{,}439 \text{ Btu/h} \approx 115{,}000 \text{ Btu/h}$$

Thus, the MAU's heating coil and humidifier, neglecting fan heat, need to be sized to provide at least a net 282,000 Btu/h of sensible heat, and 115,000 Btu/h of latent heat. Humidification can be provided by cold water, warm water, or steam, so a more precise psychrometric analysis is needed to size the heating coil correctly after the humidification method is selected and it is decided whether the humidifier will be placed before or after the heating coil.

As Example 1 shows, ventilation loads are substantial. They are often 50% or more of the total space conditioning loads in modern, well-insulated commercial buildings in less temperate climates. When cooling outdoor air, substantial moisture usually must be removed from the ventilation air; reheat or regenerative heat recovery may be required in all but dry climates.

Effect on Envelope Insulation

Air exchange also can affect a building's thermal load by altering performance of the envelope insulation system. Airflow through insulation can decrease thermal load through heat exchange between infiltrating or exfiltrating air and the insulation. Conversely, air moving in and out of the insulation from outside can increase the thermal load. Experimental and numerical studies have demonstrated that significant thermal coupling can occur between air leakage and insulation layers, thereby modifying the heat transmission in building envelopes. In particular, research (Bankvall 1987; Berlad et al. 1978; Lecompte 1987; Wolf 1966) has shown that convective airflow through air-permeable insulation in an envelope assembly may degrade its effective thermal resistance. This R-value degradation occurs when outside air moves through and/or around the insulation within the wall cavity and returns to the outdoors without reaching the conditioned space. A literature review by Powell et al. (1989) summarized the findings about air movement effects on the effective thermal resistance of porous insulation under various conditions. The effect of such airflow on insulation system performance is difficult to quantify, but should be considered. Airflow within the insulation system can also decrease the system's performance because of moisture condensation in and on the insulation.

Even if air flows only through cracks instead of through the insulation, the actual heating/cooling load from the combined effect of conduction and airflow heat transfer can be lower than the heating/cooling load calculated by Equation (33). This reduction in total heating/cooling load is a consequence of the thermal coupling between conduction and convection heat transfer and is called **infiltration heat recovery (IHR)**. Using a computer simulation, Kohonen et al. (1987) found that the conduction/infiltration thermal interaction reduced total heating load by 15%. Several experimental studies (e.g., Claridge and Bhattacharyya 1990; Claridge et al. 1988; Liu and Claridge 1992a, 1992b, 1992c, 1995; Timusk et al. 1992), using a test cell under both steady-state and dynamic conditions, found that the actual energy attributed to air infiltration can be 20 to 80% of the values given by Equation (35). Judkoff et al. (1997) measured heat recovery in a mobile home under steady-state conditions, and found that up to 40% heat recovery occurs during exfiltration through the envelope. Buchanan and Sherman (2000) performed two- and three-dimensional computational fluid dynamics (CFD) simulations to study the fundamental physics of the IHR process and developed a simple macro-scale mathematical model based on the steady-state one-dimensional convection-diffusion equation to predict a heat recovery factor. Their results show that the traditional method may overpredict the infiltration energy load. Using physical experiments, ASHRAE research project RP-1169 (Ackerman et al. 2006) showed that thermal resistances are affected by infiltration and exfiltration, but, on a net basis, the IHR effect can be neglected.

Infiltration Degree-Days

Heating and cooling degree-days are a simple way to characterize the severity of a particular climate. Heating and cooling degree-day values are based on sensible temperature data, but infiltration loads are both sensible and latent. **Infiltration degree days (IDDs)** more fully describe a climate and can be used to estimate heat loss or gain from infiltration in residences (Sherman 1986). Total infiltration degree-days is the sum of the heating and cooling infiltration degree-days and is calculated from hour-by-hour weather data and base conditions using weather weighted by infiltration rate. The selection of base conditions is an important part of the calculation of the IDDs. ASHRAE *Standard* 119 lists IDDs for many locations with a particular set of base conditions.

NATURAL VENTILATION

Natural ventilation is the flow of outdoor air caused by wind and thermal pressures through intentional openings in the building's shell. Under some circumstances, it can effectively control both temperature and contaminants in mild climates, but it is not considered practical in hot and humid climates or in cold climates. Temperature control by natural ventilation is often the only means of providing cooling when mechanical air conditioning is not available. The arrangement, location, and control of ventilation openings should combine the driving forces of wind and temperature to achieve a desired ventilation rate and good distribution of ventilation air through the building. However, intentional openings cannot always guarantee adequate temperature and humidity control or indoor air quality because of the dependence on natural (wind and stack) effects to drive the flow (Wilson and Walker 1992). Using night ventilation and the building's thermal mass effect may be effective for reducing conventional cooling energy consumption in some buildings and climates if moisture condensation can be controlled. Axley (2001a) and the Chartered Institute of Building Services Engineers (CIBSE 2005) review natural ventilation in commercial buildings, including potential advantages and problems, natural ventilation components and system designs, and recommended design and analysis approaches.

Natural Ventilation Openings

Natural ventilation openings include (1) windows, doors, dormer (monitor) openings, and skylights; (2) roof ventilators; (3) stacks; and (4) specially designed inlet or outlet openings.

Windows transmit light and provide ventilation when open. They may open by sliding vertically or horizontally; by tilting on horizontal pivots at or near the center; or by swinging on pivots at the top, bottom, or side. The type of pivoting used is important for weather protection and affects airflow rate.

Roof ventilators provide a weather-resistant air outlet. Capacity is determined by the ventilator's location on the roof; the resistance to airflow of the ventilator and its ductwork; the ventilator's ability to use kinetic wind energy to induce flow by centrifugal or ejector action; and the height of the draft.

Natural-draft or gravity roof ventilators can be stationary, pivoting, oscillating, or rotating. Selection criteria include ruggedness, corrosion resistance, stormproofing features, dampers and operating mechanisms, noise, cost, and maintenance. Natural ventilators can be supplemented with power-driven supply fans; the motors need only be energized when the natural exhaust capacity is too low. Gravity ventilator dampers can be manual or controlled by thermostat or wind velocity.

A natural-draft roof ventilator should be positioned so that it receives full, unrestricted wind. Turbulence created by surrounding obstructions, including higher adjacent buildings, impairs a ventilator's ejector action. Inlets can be conical or bell-mouthed to increase their flow coefficients. The opening area at any inlet should be increased if screens, grilles, or other structural members cause flow resistance. Building air inlets at lower levels should be larger than the combined throat areas of all roof ventilators.

Stacks or vertical flues should be located where wind can act on them from any direction. Without wind, stack effect alone removes air from the room with the inlets.

Ceiling Heights

In buildings that rely on natural ventilation for cooling, floor-to-ceiling heights are often increased well beyond the normal 8 to 10 ft. Higher ceilings, as seen in buildings constructed before air conditioning was available, allow warm air and contaminants to rise above the occupied portions of rooms. Air is then exhausted from the ceiling zones, and cooler outside air is provided near the floors; a degree of floor-to-ceiling displacement airflow is thus desirable when using natural ventilation for cooling.

Required Flow for Indoor Temperature Control

The ventilation airflow rate required to remove a given amount of heat from a building can be calculated from Equations (33) and (35) if the quantity of heat to be removed and the indoor/outdoor temperature difference are known.

Airflow Through Large Intentional Openings

The relationship describing the airflow through a large intentional opening is based on the Bernoulli equation with steady, incompressible flow. The general form that includes stack, wind, and mechanical ventilation pressures across the opening is

$$Q = 776 C_D A \sqrt{2\Delta p/\rho} \tag{36}$$

where

Q = airflow rate, cfm
C_D = discharge coefficient for opening, dimensionless
A = cross-sectional area of opening, ft^2
ρ = air density, lb$_m$/ft^3
Δp = pressure difference across opening, in. of water
776 = unit conversion factor

The discharge coefficient C_D is a dimensionless number that depends on the geometry of the opening and the Reynolds number of the flow.

Flow Caused by Wind Only

Aspects of wind that affect the ventilation rate include average speed, prevailing direction, seasonal and daily variation in speed and direction, and local obstructions such as nearby buildings, hills, trees, and shrubbery. Liddament (1988) reviewed the relevance of wind pressure as a driving mechanism. A multiflow path simulation model was developed and used to illustrate the effects of wind on air exchange rate.

Wind speeds may be lower in summer than in winter; directional frequency is also a function of season. Natural ventilation systems are often designed for wind speeds of one-half the seasonal average. Equation (37) shows the rate of air forced through ventilation inlet openings by wind or determines the proper size of openings to produce given airflow rates:

$$Q = 88.0 C_v A U \tag{37}$$

where

Q = airflow rate, cfm
C_v = effectiveness of openings (C_v is assumed to be 0.5 to 0.6 for perpendicular winds and 0.25 to 0.35 for diagonal winds)
A = free area of inlet openings, ft^2
U = wind speed, mph
88.0 = unit conversion factor

Inlets should face directly into the prevailing wind. If they are not advantageously placed, flow will be less than that predicted by Equation (37); if inlets are unusually well placed, flow will be slightly more. Desirable outlet locations are (1) on the leeward side of the building directly opposite the inlet; (2) on the roof, in the low-pressure area caused by a flow discontinuity of the wind; (3) on the side adjacent to the windward face where low-pressure areas occur; (4) in a dormer on the leeward side; (5) in roof ventilators; or (6) by stacks. Chapter 24 gives a general description of the wind pressure distribution on a building. Inlets should be placed in exterior high-pressure regions; outlets should be placed in exterior low-pressure regions.

Flow Caused by Thermal Forces Only

If building internal resistance is not significant, flow caused by stack effect can be expressed by

$$Q = 60 C_D A \sqrt{2g\Delta H_{\mathrm{NPL}}(T_i - T_o)/T_i} \tag{38}$$

where

Q = airflow rate, cfm
C_D = discharge coefficient for opening
ΔH_{NPL} = height from midpoint of lower opening to NPL, ft
T_i = indoor temperature, °R
T_o = outdoor temperature, °R

Equation (38) applies when $T_i > T_o$. If $T_i < T_o$, replace T_i in the denominator with T_o, and replace $(T_i - T_o)$ in the numerator with $(T_o - T_i)$. An average temperature should be used for T_i if there is thermal stratification. If the building has more than one opening, the outlet and inlet areas are considered equal. The discharge coefficient C_D accounts for all viscous effects such as surface drag and interfacial mixing.

Estimation of ΔH_{NPL} is difficult for naturally ventilated buildings. If one window or door represents a large fraction (approximately 90%) of the total opening area in the envelope, then the NPL is at the mid-height of that aperture, and ΔH_{NPL} equals one-half the height of the aperture. For this condition, flow through the opening is bidirectional (i.e., air from the warmer side flows through the top of the opening, and air from the colder side flows through the bottom). Interfacial mixing occurs across the counterflow interface, and the orifice coefficient can be calculated according to the following equation (Kiel and Wilson 1986):

$$C_D = 0.40 + 0.0025|T_i - T_o| \tag{39}$$

If enough other openings are available, airflow through the opening will be unidirectional, and mixing cannot occur. A discharge coefficient of $C_D = 0.65$ should then be used. Additional information on stack-driven airflows for natural ventilation can be found in Foster and Down (1987).

Greatest flow per unit area of openings is obtained when inlet and outlet areas are equal; Equations (38) and (39) are based on this equality. Increasing the outlet area over inlet area (or vice versa) increases airflow but not in proportion to the added area. When openings are unequal, use the smaller area in Equation (38) and add the increase as determined from Figure 8.

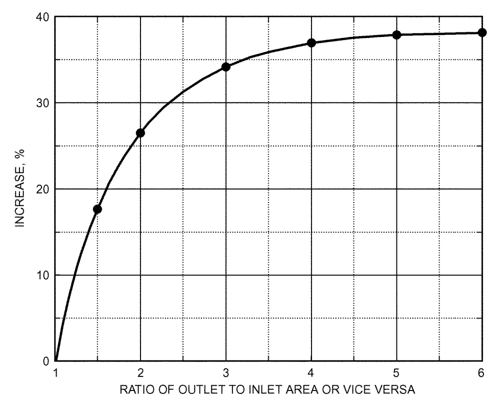


Fig. 8 Increase in Flow Caused by Excess Area of One Opening over the Other

Natural Ventilation Guidelines

Several general guidelines should be observed in designing for natural ventilation. Some of these may conflict with other climate-responsive strategies (such as using orientation and shading devices to minimize solar gain), with building codes that encourage compartmentalization to restrict fire and smoke movement, or with other design considerations.

System selection

- In hot, humid climates, use mechanical cooling. If mechanical cooling is not available, air velocities should be maximized in the occupied zones of rooms.
- In hot, arid climates, consider evaporative cooling. Airflow throughout the building should be maximized for structural cooling, particularly at night when the outside air temperature is low.

Building and surroundings characteristics

- Topography, landscaping, and surrounding buildings should be used to redirect airflow and give maximum exposure to breezes. Vegetation can funnel breezes and avoid wind dams, which reduce the driving pressure differential around the building. Site objects should not obstruct inlet openings.
- The building should be shaped to expose maximum shell openings to breezes.
- Architectural elements such as wing walls, parapets, and overhangs should be used to promote airflow into the building interior.
- The long façade of the building and the majority of door and window openings should be oriented with respect to prevailing summer breezes. If there is no prevailing direction, openings should be sufficient to provide ventilation regardless of wind direction.

Opening locations

- Windows should be located in opposing pressure zones. Two openings on opposite sides of a space increase ventilation flow. Openings on adjacent sides force air to change direction, providing ventilation to a greater area. The benefits of the window arrangement depend on the outlet location relative to the direction of the inlet airstream.
- If a room has only one external wall, better airflow is achieved with two widely spaced windows.
- If openings are at the same level and near the ceiling, much of the flow may bypass the occupied level and be ineffective in diluting contaminants there.
- Vertical distance between openings is required to take advantage of stack effect; the greater the vertical distance, the greater the ventilation rate.
- Openings in the vicinity of the NPL are least effective for thermally induced ventilation. If the building has only one large opening, the NPL tends to move to that level, which reduces pressure across the opening.

Opening characteristics

- Greatest flow per unit area of total opening is obtained by inlet and outlet openings of nearly equal areas. An inlet window smaller than the outlet creates higher inlet velocities. An outlet smaller than the inlet creates lower but more uniform airspeed through the room.
- Openings with areas much larger than calculated are sometimes desirable when anticipating increased occupancy or very hot weather.
- Horizontal windows are generally better than square or vertical windows. They produce more airflow over a wider range of wind directions and are most beneficial in locations where prevailing wind patterns shift.
- Window openings should be accessible to and operable by occupants, unless fully automated. For secondary fire egress, operable windows may be required.
- Inlet openings should not be obstructed by indoor partitions. Partitions can be placed to split and redirect airflow but should not restrict flow between the building's inlets and outlets. Vertical airshafts or open staircases can be used to increase and take advantage of stack effects. However, enclosed staircases intended for evacuation during a fire should not be used for ventilation.

Hybrid Ventilation

Application of purely natural ventilation systems may be limited in hot or humid climates, such as in much of the United States, by thermal comfort issues and the need for reliability. However, hybrid (or mixed-mode) ventilation systems or operational strategies offer the possibility of saving energy in a greater number of buildings and climates by combining natural ventilation systems with mechanical equipment (Emmerich 2006). The **air-side economizer** is one form of hybrid ventilation control scheme, and enjoys wide use in commercial, industrial, and institutional buildings in appropriate climates. The report of the International Energy Agency's (IEA) Annex 35 describes the principles of hybrid ventilation technologies, control strategies, design and analysis methods, and case studies (Heiselberg 2002). Integrated multizone airflow and thermal modeling is recommended when designing natural and hybrid ventilation systems (Axley 2001a; Li and Heiselberg 2003).

RESIDENTIAL AIR LEAKAGE

Most infiltration in U.S. residential buildings is dominated by envelope leakage. However, new construction tends toward tighter building envelopes.

Envelope Leakage Measurement

A building's envelope leakage can be measured with **pressurization testing,** commonly called a **blower-door test**. Fan pressurization is relatively quick and inexpensive, and it characterizes building envelope airtightness independent of weather conditions. In this procedure, a large fan or blower is mounted in a door or window and induces a large and roughly uniform pressure difference across the building shell [ASTM *Standards* E779 and E1827; Canadian General Standards Board (CGSB) *Standard* 149.10; ISO *Standard* 9972]. The airflow required to maintain this pressure difference is then measured. The leakier the building is, the more airflow is necessary to induce a specific indoor/outdoor pressure difference. The airflow rate is generally measured at a series of pressure differences ranging from about 0.04 to 0.30 in. of water.

The results of a pressurization test, therefore, consist of several combinations of pressure difference and airflow rate data. An example of typical data is shown in Figure 9. These data points characterize the air leakage of a building and are generally converted to a single value that serves as a measure of the building's airtightness. There are several different measures of airtightness, most of which involve fitting the data to a curve describing the relationship between the airflow Q through an opening in the building envelope and the pressure difference Δp across it. This relationship is called the **leakage function** of the opening. The form of the leakage function depends on the geometry of the opening. Background theoretical material relevant to leakage functions may be found in Chastain et al. (1987), Etheridge (1977), Hopkins and Hansford (1974), Kronvall (1980), and Walker et al. (1997).

Openings in a building envelope are not uniform in geometry and, generally, the flow never becomes fully developed. Each opening in the building envelope can be described by Equation (40), commonly called the **power law equation**:

$$Q = c(\Delta p)^n \tag{40}$$

where

Q = airflow through opening, cfm
c = flow coefficient, cfm/(in. of water)^n
n = pressure exponent, dimensionless

Sherman (1992a) showed how the power law can be developed analytically by looking at developing laminar flow in short pipes. Equation (40) only approximates the relationship between Q and Δp. Measurements of single cracks (Honma 1975; Kreith and Eisenstadt 1957) show that n can vary if Δp changes over a wide range. Additional investigation of pressure/flow data for simple cracks by Chastain et al. (1987) indicated the importance of adequately characterizing the three-dimensional geometry of openings and the entrance and exit effects. Walker et al. (1997) showed that, for the arrays of cracks in a building envelope over the range of pressures acting during infiltration, n is constant. A typical value for n is about 0.65. Values for c and n can be determined for a building by using fan pressurization testing.

Airtightness Ratings

In some cases, the predicted airflow rate is converted to an **equivalent** or **effective air leakage area** as follows:

$$A_L = 0.186 Q_r \frac{\sqrt{\rho / 2\Delta p_r}}{C_D} \tag{41}$$

where

A_L = equivalent or effective air leakage area, in^2
Q_r = predicted airflow rate at Δp_r (from curve fit to pressurization test data), cfm
ρ = air density, lb$_m$/ft^3
Δp_r = reference pressure difference, in. of water
C_D = discharge coefficient
0.186 = unit conversion factor

All openings in the building shell are combined into an overall opening area and discharge coefficient for the building when the equivalent or effective air leakage area is calculated. Some users of the leakage area approach set $C_D = 1$. Others set $C_D \approx 0.6$ (i.e., the discharge coefficient for a sharp-edged orifice). The air leakage area of a building is, therefore, the area of an orifice (with an assumed value of C_D) that would produce the same amount of leakage as the building envelope at the reference pressure.

An airtightness rating, whether based on an air leakage area or a predicted airflow rate, is generally normalized by some factor to account for building size. Normalization factors include floor area, exterior envelope area, and building volume.

With the wide variety of possible approaches to normalization and reference pressure difference, and the use of the air leakage area concept, many different airtightness ratings are used. Reference pressure differences include 0.016, 0.04, 0.10, 0.20, and 0.30 in. of water. Reference pressure differences of 0.016 and 0.04 in. of water are advocated because they are closer to the pressure differences that actually induce air exchange and, therefore, better model the opening's flow characteristics. Although this may be true, they are outside the range of measured values in the test; therefore, predicted airflow rates at 0.016 and 0.04 in. of water are subject to significant uncertainty. This uncertainty and its implications for quantifying airtightness are discussed in Chastain (1987), Modera and Wilson (1990), and Persily and Grot (1985b). Round-robin tests by Murphy et al. (1991) to determine the repeatability and reproducibility of fan pressurization devices found that subtle errors in fan calibration or operator technique are greatly exaggerated when extrapolating the pressure versus flow curve out to 0.016 in. of water, with errors as great as ±40%, mainly because of fan calibration errors at low flow.

Some common airtightness ratings include the effective air leakage area at 0.016 in. of water assuming $C_D = 1.0$ (Sherman and Grimsrud 1980); the equivalent air leakage area at 0.04 in. of water assuming $C_D = 0.611$ (CGSB *Standard* 149.10); and the airflow rate at 0.20 in. of water, divided by the building volume to give units of air changes per hour (Blomsterberg and Harrje 1979).

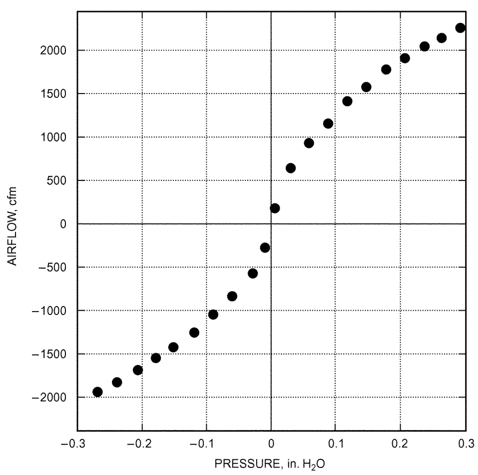


Fig. 9 Airflow Rate Versus Pressure Difference Data from Whole-House Pressurization Test

Conversion Between Ratings

Air leakage areas at one reference pressure difference can be converted to air leakage areas at another reference pressure difference according to

$$A_{r,2} = A_{r,1} \left(\frac{C_{D,1}}{C_{D,2}} \right) \left(\frac{\Delta p_{r,2}}{\Delta p_{r,1}} \right)^{n - 0.5} \tag{42}$$

where

$A_{r,1}$ = air leakage area at reference pressure difference $\Delta p_{r,1}$, in^2
$A_{r,2}$ = air leakage area at reference pressure difference $\Delta p_{r,2}$, in^2
$C_{D,1}$ = discharge coefficient used to calculate $A_{r,1}$
$C_{D,2}$ = discharge coefficient used to calculate $A_{r,2}$
n = pressure exponent from Equation (40)

Air leakage area at one reference pressure difference can be converted to airflow rate at some other reference pressure difference according to

$$Q_{r,2} = 5.39 C_{D,1} A_{r,1} \sqrt{\frac{2}{\rho}} (\Delta p_{r,1})^{0.5 - n} (\Delta p_{r,2})^n \tag{43}$$

where

$Q_{r,2}$ = airflow rate at reference pressure difference $\Delta p_{r,2}$, cfm
5.39 = unit conversion factor

Flow coefficient c in Equation (40) may be converted to air leakage area according to

$$A_L = \frac{c}{5.39 C_D} \sqrt{\frac{\rho}{2}} \Delta p_r^{(n - 0.5)} \tag{44}$$

Finally, air leakage area may be converted to flow coefficient c in Equation (40) according to

$$c = 5.39 C_D A_L \sqrt{\frac{\rho}{2}} (\Delta p_r)^{0.5 - n} \tag{45}$$

Equations (42) to (45) require assumption of a value of n, unless it is reported with the measurement results. When whole-building pressurization test data are fitted to Equation (40), the value of n generally lies between 0.6 and 0.7. Therefore, using a value of n in this range is reasonable.

Building Air Leakage Data

Fan pressurization measures a building property that ideally varies little with time and weather conditions. In reality, unless wind

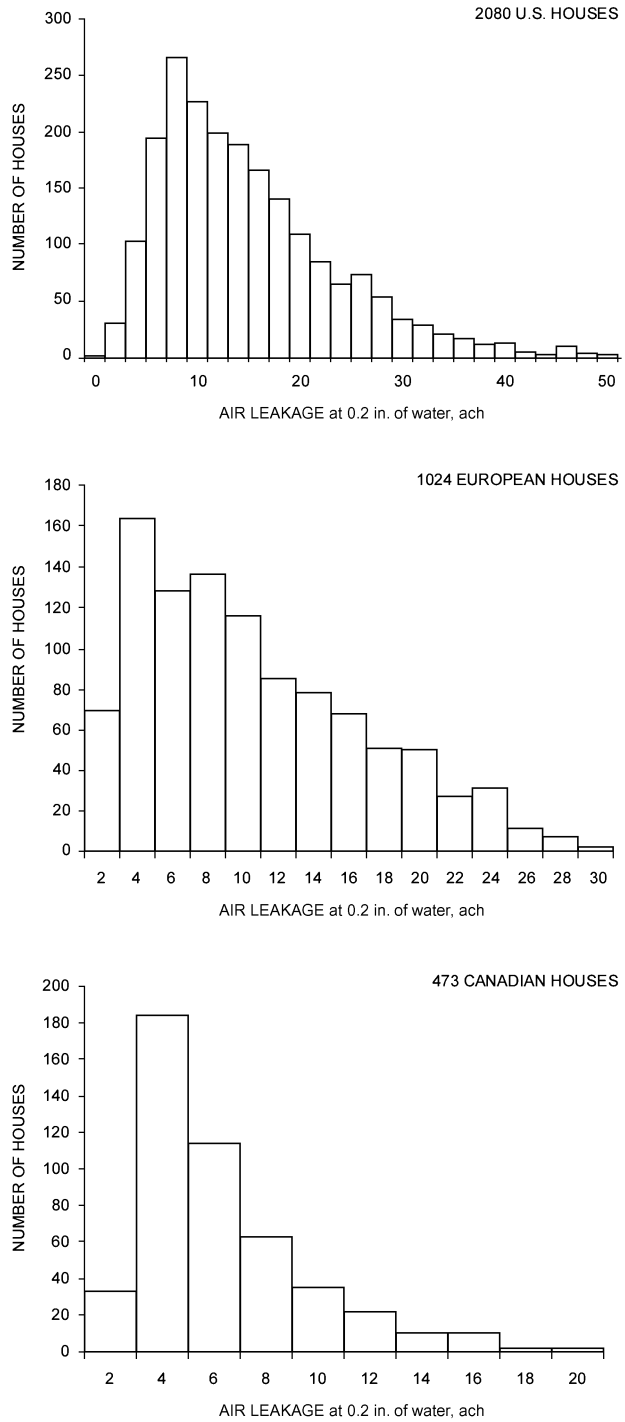


Fig. 10 Envelope Leakage Measurements

and temperature differences during the measurement period are sufficiently mild, pressure differences they induce during the test interfere with test pressures and cause measurement errors. Modera and Wilson (1990) and Persily (1982) studied the effects of wind speed on pressurization test results. Several experimental studies also showed variations on the order of 20 to 40% over a year in the measured airtightness in homes (Kim and Shaw 1986; Persily 1982; Warren and Webb 1986).

Figure 10 summarizes envelope leakage measured North American housing (Sherman and Dickerhoff 1998) and from several European and Canadian sources (AIVC 1994). This figure shows the large range of measured envelope tightness, but can still be used to illustrate typical and extreme values in the housing stock.

ASHRAE *Standard* 119 establishes air leakage performance levels for residential buildings. These levels are in terms of the normalized leakage area A_n:

$$A_n = 6.944(A_L/A_f)(H/H_o)^{0.3} \tag{46}$$

where

A_n = normalized leakage area, dimensionless
A_L = effective leakage area at 0.016 in. of water (C_D = 1.0), in^2
A_f = gross floor area (within exterior walls), ft^2
H = building height, ft
H_o = reference height of one-story building = 8 ft

Air Leakage of Building Components

The fan pressurization procedure discussed in the section on Envelope Leakage Measurement allows whole-building air leakage to be measured. The location and size of individual openings in building envelopes are extremely important because they influence the air infiltration rate of a building as well as the envelope's heat and moisture transfer characteristics. Additional test procedures for pressure-testing individual building components such as windows, walls, and doors are discussed in ASTM *Standards* E283 and E783 for laboratory and field tests, respectively.

Leakage Distribution

Dickerhoff et al. (1982) and Harrje and Born (1982) studied air leakage of individual building components and systems. The following points summarize the percentages of whole-building air leakage area associated with various components and systems. Values in parentheses include the range determined for each component and the mean of the range.

Walls (18 to 50%; 35%). Both interior and exterior walls contribute to the leakage of the structure. Leakage can occur between the sill plate and foundation; through cracks below the bottom of the gypsum wallboard, electrical outlets, and plumbing penetrations; and into the attic at the top plates of walls.

Ceiling details (3 to 30%; 18%). Leakage across the top ceiling of the heated space is particularly insidious because it reduces the effectiveness of insulation on the attic floor and contributes to infiltration heat loss. Ceiling leakage also reduces the effectiveness of ceiling insulation in buildings without attics. Recessed lighting, plumbing, and electrical penetrations leading to the attic are some particular areas of concern.

Forced-air heating and/or cooling systems (3 to 28%; 18%). The location of the heating or cooling equipment, air handler, or ductwork in conditioned or unconditioned spaces; the venting arrangement of a fuel-burning device; and the existence and location of a combustion air supply all affect air leakage. Modera et al. (1991) and Robison and Lambert (1989), among others, found that the variability of leakage in ducts passing through unconditioned spaces is high, the coefficient of variation being on the order of 50%. Field studies have also shown that in-situ repairs can eliminate one-quarter to two-thirds of the observed leakage (Cummings and Tooley 1989; Cummings et al. 1990; Jump et al. 1996; Robison and Lambert 1989). The 18% contribution of ducts to total leakage significantly underestimates their effect because, during system operation, pressure differentials across duct leaks are approximately ten times higher than typical pressure differences across envelope leaks (Modera 1989; Modera et al. 1991) and result in large (factors of two to three) changes in ventilation rate (Cummings et al. 1990; Walker 1999; Walker et al. 1999).

Windows and doors (6 to 22%; 15%). More variation in window leakage is seen among window types (e.g., casement versus double-hung) than among new windows of the same type from different manufacturers (Weidt et al. 1979). Windows that seal by compressing the weather strip (casements, awnings) show significantly lower leakage than windows with sliding seals.

Fireplaces (0 to 30%; 12%). When a fireplace is not in use, poorly fitting dampers allow air to escape. Glass doors reduce excess air while a fire is burning, but rarely seal the fireplace structure more tightly than a closed damper does. Chimney caps or fireplace plugs (with signs that warn they are in place) effectively reduce leakage through a cold fireplace.

Vents in conditioned spaces (2 to 12%; 5%). Exhaust vents in conditioned spaces frequently have either no dampers or dampers that do not close properly.

Diffusion through walls (<1%). Compared to infiltration through holes and other openings in the structure, diffusion is not an important flow mechanism. At 0.02 in. of water, the permeability of building materials produces an air exchange rate of less than 0.01 ach by wall diffusion in a typical house.

Component leakage areas. Individual building component leakage areas vary widely from house to house. Typical variability for an individual component is about a factor of 10, depending on the component's construction and installation. Testing should be used to establish the installed leakage of a component in applications where leakage is critical to building performance.

Multifamily Building Leakage

Leakage distribution is particularly important in multifamily apartment buildings. These buildings often cannot be treated as single zones because of the internal resistance between apartments. Moreover, leakage between apartments varies widely, from very small for well-constructed buildings with air/moisture retarders between units, to as high as 60% of the total apartment leakage in turn-of-the-century brick walk-up apartment buildings (Diamond et al. 1986; Modera et al. 1991).

Controlling Air Leakage

New Buildings. It is much easier to build a tight building than to tighten an existing building. Elmroth and Levin (1983), Eyre and Jennings (1983), Marbek Resource Consultants (1984), and Nelson et al. (1985) provide information and construction details on airtight building design for houses.

A continuous air infiltration retarder is one of the most effective means of reducing air leakage through walls, around window and door frames, and at joints between major building elements. Particular care must be taken to ensure its continuity at all wall, floor, and ceiling joints; at window and door frames; and at all penetrations of the retarder, such as electrical outlets and switches, plumbing connections, and utility service penetrations. Joints in the **air/vapor retarder** must be lapped and sealed. Plastic vapor retarders installed in the ceiling should be tightly sealed with the vapor retarder in the outside walls and should be continuous over the partition walls. A seal at the top of the partition walls prevents leakage into the attic; a plate on top of the studs generally gives a poor seal. The air infiltration retarder can be installed either on the inside of the wall framing, in which case it usually functions as a vapor retarder as well, or on the outside of the wall framing, in which case it should have a permeance rating high enough to allow diffusion of water vapor from the wall. For a discussion of moisture transfer in building envelopes, see Chapters 25 and 26.

Interior air/vapor retarders must be lapped and sealed at electrical outlets and switches, at joints between walls and floors and between walls and ceilings, and at plumbing connections penetrating the wall's interior finish. A continuous exterior air infiltration retarder installed on the outside of wall framing can cover these problem areas. Joints in the air infiltration retarder should be lapped and sealed or taped. Exterior air infiltration retarders are generally made of a material stronger than plastic film and are more likely to withstand damage during construction. Sealing the wall against air leakage at the exterior of the insulation also reduces convection currents within the wall cavity, allowing insulation to retain more of its effectiveness.

Existing Buildings. Air leakage sites must first be located to tighten the envelope of an existing building. As discussed earlier, air leakage in buildings is caused by not only windows and doors, but also a wide range of unexpected and unobvious construction defects. Many important leakage sites can be very difficult to find. A variety of techniques developed to locate leakage sites are described in ASTM *Standard* E1186 and Charlesworth (1988).

Once leakage sites are located, they can be repaired with materials and techniques appropriate to the size and location of the leak. Diamond et al. (1982), Energy Resource Center (1982), and Harrje et al. (1979) include information on airtightening or "weatherization" in existing residential buildings with caulking, sealing, weatherstripping, and use of door sweeps, for example. With these procedures, air leakage of residential buildings can be reduced dramatically: anywhere from 5% to more than 50%, depending on the extent of the tightening effort and the experience of those doing the work (Blomsterberg and Harrje 1979; Giesbrecht and Proskiw 1986; Harrje and Mills 1980; Jacobson et al. 1986; Verschoor and Collins 1986). Much less information is available for airtightening large, commercial buildings, but the same general principles apply (Parekh et al. 1991; Persily 1991).

RESIDENTIAL VENTILATION

Typical infiltration values in housing in North America vary by a factor of about ten, from tightly constructed housing with seasonal average air exchange rates as low as 0.1 air changes per hour (ach) to loosely constructed housing with air exchange rates as great as 2.0 ach. Figures 11 and 12 show histograms of infiltration rates measured in two different samples of North American housing (Grimsrud et al. 1982; Grot and Clark 1979). Figure 11 shows the average seasonal infiltration of 312 houses located in different areas in North America. The median infiltration value of this sample is 0.5 ach. Figure 12 represents measurements in 266 houses located in 16 U.S. cities. The median value of this sample is 0.9 ach. The group of houses in the Figure 11 sample is biased toward then-new, energy-efficient houses, whereas the group in Figure 12 represents older, low-income housing.

Additional studies have found average values for houses in regional areas. Palmiter and Brown (1989) and Parker et al. (1990) found a heating season average of 0.40 ach (range: 0.13 to 1.11 ach) for 134 houses in Pacific Northwest climates. In a comparison of 292 houses incorporating energy-efficient features (including measures to reduce air infiltration and provide ventilation heat recovery) with

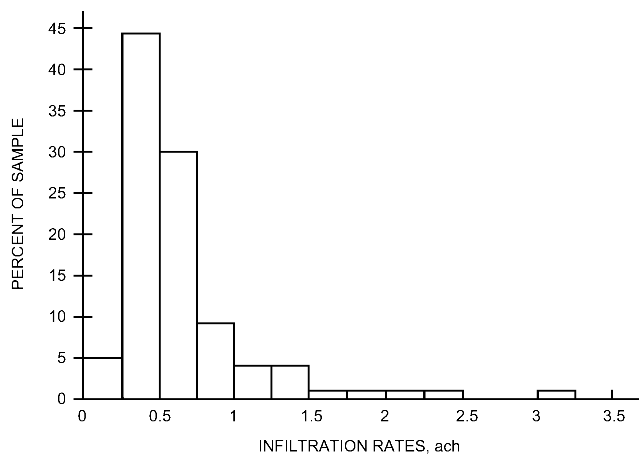


Fig. 11 Histogram of Infiltration Values for Then-New Construction

331 control houses, Parker et al. (1990) found an average of about 0.25 ach (range: 0.02 to 1.63 ach) for the energy-efficient houses versus 0.49 (range: 0.05 to 1.63 ach) for the control. Ek et al. (1990) found an average of 0.5 ach (range: 0.26 to 1.09) for 93 double-wide manufactured homes in the Pacific Northwest. Canadian housing stock has been characterized by Riley (1990) and Yuill and Comeau (1989). Although these studies do not represent random samples of North American housing, they indicate the distribution of infiltration rates expected in a group of buildings.

Occupancy influences have not been measured directly and vary widely. Desrochers and Scott (1985) estimated that they add an average of 0.10 to 0.15 ach to unoccupied values. Kvisgaard and Collet (1990) found that, in 16 Danish dwellings, occupants on average provided 63% of the total air exchange rate.

Ventilation air requirements for houses in the United States have traditionally been met on the assumption that the building envelope is leaky enough that infiltration will suffice. Possible difficulties with this approach include low infiltration when natural forces (temperature difference and wind) are weak; unnecessary energy consumption when these forces are strong; drafts in cold climates; lack of control of ventilation rates to meet changing needs; poor humidity control; potential for interstitial condensation from exfiltration in cold climates or infiltration in hot humid climates; and lack of opportunity to recover energy used to condition ventilation air. The solution to these concerns is to have a tight building envelope and a properly designed and operated mechanical ventilation system.

ASHRAE *Standard* 119 and the National Building Code of Canada (NRCC 1995) encourage the transition to tighter envelope construction. Hamlin (1991) found a 30% increase in airtightness of tract-built Canadian houses between 1982 and 1989. Also, 82% of newer houses had natural air exchange rates below 0.3 ach in March. Yuill (1991) derived a procedure to show the extent to which infiltration contributes toward meeting ventilation air requirements. As a result, the National Building Code of Canada has requirements for mechanical ventilation capability in all new dwelling units.

Canadian Standards Association (CSA) *Standard* F326 expands the requirements for residential mechanical ventilation systems to cover air distribution within the house, thermal comfort, minimum temperatures for equipment and ductwork, system controls, pressurization and depressurization of the dwelling, installation requirements, and verification of compliance. Verification can be by design or by test, but the total rate of outside air delivery must be measured.

Mechanical ventilation is required by ASHRAE *Standard* 62.2 and by code in some U.S. states; some details of these requirement are described in this chapter. The net benefit of using mechanical ventilation has been demonstrated and studied in various energy-efficient and advanced housing programs (Barley 2001; Palmiter et al. 1991; Riley 1990). Systems can be characterized as local or central; exhaust, supply, or balanced; with forced-air or radiant/ hydronic heating/cooling systems; with or without heat recovery; and with continuous operation or controlled by occupants, demand (i.e., by pollutant sensing), timers, or humidity. Note that not all combinations are viable. Various options are described by Fisk et al. (1984), Hekmat et al. (1986), Holton et al. (1997), Lubliner et al. (1997), Palmiter et al. (1991), Reardon and Shaw (1997), Sherman and Matson (1997), Sibbitt and Hamlin (1991), and Yuill et al. (1991).

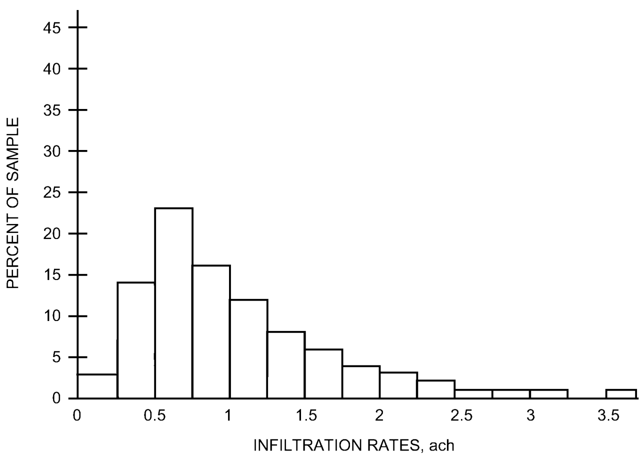


Fig. 12 Histogram of Infiltration Values for Low-Income Housing

The simplest systems use bathroom and kitchen fans to exhaust moisture and pollutants and to augment infiltration. Noise, installed capacity, durability under continuous operation, distribution to all rooms (especially bedrooms), envelope moisture, combustion safety, and energy efficiency issues need to be addressed. Many present bath and kitchen fans are ineffective ventilators because of poor installation and design, and many fail to exhaust outdoors. However, properly specified and installed exhaust fans can form part of good whole-house ventilation systems and are so specified in some Canadian building codes.

Some central supply systems use a central air-handling unit blower to induce air from the outdoors and distribute it. However, the blower operates intermittently if thermostatically controlled and provides little ventilation in mild weather. Continuous blower operation increases energy consumption. If the blower operates continuously when the heat source is off, the combination of lower mixed air temperature and high air speed can cause cold air drafts. To offset these problems, some systems use electronically commutated blower motors, which allow efficient continuous operation at lower speeds. Some others use a timer to cycle the blower when thermostatic demands are inadequate to cause the blower to operate when needed for ventilation (Rudd 1998).

Central exhaust systems use leakage sites and, in some cases, intentional and controllable openings in the building envelope as the supply. Such systems are suitable for retrofit in existing houses. Energy can be recovered from the exhaust airstream with a heat pump to supplement domestic hot-water and/or space heating.

For new houses with tightly constructed envelopes, balanced ventilation with passive heat recovery (air-to-air heat exchangers or heat recovery ventilators) can be appropriate in some climates. Fan-induced supply and exhaust air flows at nearly equal rates over a heat exchanger, where heat and sometimes moisture is transferred between the airstreams. This typically reduces the energy required to condition ventilation air by 60 to 80% (Cutter 1987). It also reduces the thermal discomfort that occurs when untempered air is introduced directly into the house. Airflow balance, leakage between streams, biological contamination of wet surfaces, frosting, and first cost are concerns associated with these systems.

Air-side economizers, which allow outside air to be up to 100% of the supply air at appropriate times, are not typically used in small buildings with low internal heat gains relative to the building envelope. Because of heat transfer through building envelopes, these small buildings quickly require heating or cooling as the outside air temperature falls or rises. Consequently, from an energy conservation point of view, small envelope-load-dominated buildings do not benefit as much as internal-load-dominated buildings from daytime use of air-side economizers; night ventilation during the cooling season may be very attractive. Also, ventilation rates increase dramatically when air-side economizers are in operation, so the extra moisture introduced or removed must be considered.

The type of ventilation system can be selected based on house leakage class as defined in ASHRAE *Standard* 119. Balanced air-to-air systems with heat recovery are optimal for tight houses (leakage classes A–C). The leakier the house is, the larger is the contribution from infiltration and the less effective is heat recovery ventilation. Tightening the envelope beyond the level of ASHRAE *Standard* 119 may be warranted in extreme climates to better use the heat recovery effect (Sherman and Matson 1997). In mild climates, these systems can also effectively be used in leakage classes D–F. Central exhaust systems should not be used for leakage classes A–C unless special provisions are made for air inlets; otherwise their operation may depressurize the house enough to cause backdrafting through fossil-fueled appliances. Unbalanced systems (either supply or exhaust) are optimal for leakage classes D–F. Ventilation systems are normally not needed for leakage classes G–J, but when they are needed, an unbalanced system is usually the best choice. More discussion of mechanical systems for residences is available in Russell et al. (2005); some information on practices outside North America can be found in McWilliams and Sherman (2005).

Residential Ventilation Zones

For guidance in the selection of residential ventilation systems, Sherman (1995) developed four climatic zones for the United States. These zones are shown in Figure 13 for the continental United States. Alaska is in zone 1, and Hawaii is in zone 4.

Zone 1 includes the severe climates of the northern tier of states. A zone 1 residence that meets airtightness and energy conservation standards probably cannot meet its ventilation needs through infiltration, and will require mechanical ventilation. Zone 2 includes moderate climates where careful design and construction may allow buildings to simultaneously meet energy standards and ventilation needs through infiltration and mechanical exhaust. The mild climates in zone 3 allow residences to meet both ASHRAE *Standards* 119 and 62.2 over a substantial range of airtightness. Zone 4 residences have relatively small energy penalties associated with infiltration or ventilation. In this zone, natural ventilation is usually preferred to mechanical ventilation as a technique to supplement infiltration.

Shelter in Place

The most fundamental function of a house is to provide shelter from outdoor conditions. The building is intended to be the first line of defense at separating the relatively uncontrolled outdoor environment from the desired indoor environment.

A first response to poor outdoor air quality is to go inside, close the windows, and turn off central heating, air-conditioning, and ventilating systems, as well as any other fans. Closing windows and other air intakes reduces air exchange with the outdoors, decreasing the immediate intrusion of outdoor air into the home. However, because no home is perfectly airtight, closing doors and windows does not eliminate intrusion. Because all indoor air ultimately comes from outdoors, all else being equal, indoor conditions eventually come to dynamic equilibrium with outdoor conditions. The tighter the building, the longer the time needed to come to equilibrium.

The delay time (the time it takes to completely change the air in a building) is determined by the ventilation rate. The effectiveness of sheltering within the home thus depends on envelope tightness. For a home with 0.35 air changes per hour, the delay time is roughly 3 h. For a tight house without mechanical ventilation, the delay time can easily be twice as long. Most houses in the United States are leaky (i.e., typically one air change per hour) and thus could have a delay time on the order of one hour (Sherman and Matson 1997).

Reactive gases in outdoor air, such as ozone, can be decreased to some degree by the building envelope. For other outdoor contaminants, the building envelope serves to delay, not reduce, their introduction into the indoor environment. Such a delay is not very helpful at reducing exposures to outdoor contaminants that persist over days, but can be an effective strategy for short-duration (less than a few hours) sources. In houses without indoor ozone sources, ozone levels tend to be higher in houses that do not have air conditioners than in those with air conditioners; ozone levels also are higher when windows are open than when they are closed (Weschler 2000). For outdoor exposure times shorter than the delay time, the house serves as a reservoir of cleaner air. After the outdoor contaminant is gone, windows can be opened to flush out pollutants that entered during the exposure period.

Safe Havens

Simply going inside may not be sufficient for highly unusual but potentially lethal events. Chemical spills or fires, explosions, bioterrorism, or similar toxic air pollutant releases can temporarily create dangerous outdoor conditions that render other air quality issues insignificant. With sufficient warning, occupants should leave the vicinity, but the unexpected nature of these events means that the only viable alternative may be to shelter in place.

This strategy may work for short-term releases. Homes are often too leaky to provide the protection needed for longer-duration events, but individual rooms can be temporarily sealed to become safe havens. A safe haven should be chosen to have as little contact as possible with outside walls, and preferably be on the side of the house furthest downwind from the source. Duct tape can be used to seal leaks, cracks, seams, register grilles, and doors, with thick plastic sheeting used to span larger gaps (Sorensen and Vogt 2001). If such a shelter has an air change rate of 0.15 ach with the house, it will take 4 to 6 h for contaminated outdoor air to reach the safe haven.

There may, however, be a very small population of at-risk individuals or locations for which emergencies are somewhat more likely. In such cases, a safe haven can be designed in advance with a highly efficient particle/gas-phase filtration system capable of providing several hours, days, or weeks of protection (Ormerod 1983). A short-term safe haven might be effectively combined with other emergency shelters (e.g., tornado, hurricane, civil defense) to reduce cost.

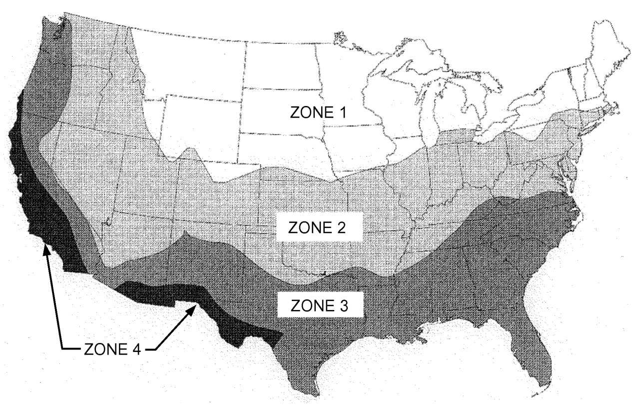


Fig. 13 Airtightness Zones for Residences in the United States
(Sherman 1995)

RESIDENTIAL VENTILATION AND IAQ CONTROL REQUIREMENTS

ASHRAE *Standard* 62.2 presents minimum requirements for residential ventilation air and acceptable indoor air quality, and its user's manual (ASHRAE 2006) has detailed information for designing and constructing residential buildings in compliance with the standard. Best or good practice may require going beyond the standard's minima. This section describes good practice; however, this presumes that the minimum requirements of 62.2 are met as well.

Traditionally, ventilation air for residences has been provided by natural ventilation and infiltration. Sherman and Matson (1997) showed that most of the older building stock is sufficiently leaky that infiltration alone can meet the minimum requirements of ASHRAE *Standard* 62.2. Houses built or retrofitted to new standards have substantially tighter envelopes and insufficient infiltration to meet ventilation standards. Studies have shown that concerns over safety, noise, comfort, air quality, and energy minimize occupant use of operable windows (Johnson and Long 2005; Price and Sherman 2006). As a result, these houses require supplemental mechanical ventilation to satisfy current standards.

Simply meeting minimum residential ventilation rates is not always sufficient to adequately dilute all contaminants. For some buildings, such ventilation may not meet the requirements of individuals with allergies or chemical sensitivities or when there are unusual sources such as radon or mold. In these cases, source control or extra ventilation is required to manage the contaminant levels. Therefore, especially in single-family dwellings, occupants must be responsible for introducing, monitoring, and controlling the sources in the indoor environment, as well as for operating the dwelling unit to meet their individual needs. Increasingly, residences are also used for business or hobby purposes, which may introduce air contaminants not addressed in *Standard* 62.2; portions of these residences may require ventilation air as required by *Standard* 62.1 or industrial guidelines.

Source Control

When considering how much whole-house ventilation should be supplied, typical and unusual significant sources of indoor pollution need to be controlled. This can be done either by mitigating the source itself or by using local exhaust to extract contaminants before they can mix into the indoor environment. Typical sources that should be considered include the following:

Clothes Dryers and Central Vacuum Systems. Clothes dryer exhaust is heavily laden with moisture and laundry by-products such as flammable lint and various gaseous contaminants. Many moisture problems have been traced to clothes dryers vented indoors. Exhaust from clothes dryers, which is typically about 150 cfm, should be vented directly to the outdoors. Similarly, central vacuum systems should be vented directly outdoors to exhaust the finer particles that pass through their filters.

Combustion. Water and carbon dioxide are always emitted during combustion of hydrocarbons in air. Other dangerous compounds are created, as well. All these by-products should be vented directly outdoors, preferably using sealed combustion or direct-vent equipment. Venting should meet all applicable codes. For buildings with naturally aspirated combustion appliances, excessive depressurization by exhaust systems must be avoided, which can be done by keeping combustion equipment outside the pressure boundary. In addition, a depressurization safety test should be considered, such as described in ASTM *Standard* E1998 or CGSB *Standard* 51.71. Fireplace combustion products should be isolated from the occupied space using tight-fitting doors and outdoor air intakes, when necessary. Flues and chimneys must be designed and installed to disperse combustion products well away from air intakes and operable windows, for example. Chapter 34 of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* has more information on venting systems.

Carbon monoxide is one of the most pervasive indoor contaminants. It can come from virtually any source of combustion, including automobiles. Because even combustion appliances that meet manufacturers' specifications can interact with the building and emit carbon monoxide, at least one carbon monoxide alarm meeting safety standards such as CSA *Standard* 6.19 should be installed near sleeping areas in each dwelling, including each unit of multifamily residential buildings, that has combustion appliances (e.g., fireplaces, stoves, furnaces, water heaters) within the pressure boundary, or has attached garages or storage sheds. Carbon monoxide alarms also should be considered for nonresidential buildings: poisonings have occurred in many building types, including hotels, motels, stores, restaurants, nursing homes, dormitories, laundromats, and schools.

Garages. Garages and storage spaces contain many sources of contaminants. Doors between them and occupied space should be well sealed with gaskets or weatherstripping and possibly be self-closing. Depressurized sections of HVAC systems, such as air handlers or return or intake ducts, should not be located in garages. If such sections must pass through garages, they must be well sealed. Care should be taken to ensure that there is a good pressure barrier between the garage and the occupied space, typically using an air/moisture retarder such as heavy polyethylene, and other measures. Carbon monoxide sources may be present in garages, so pressure barriers, fire-rated compartmentation, and ventilation of attached residences are life-safety measures. Separate ventilation systems that slightly depressurize attached garages and storage spaces and exhaust directly outdoors should be considered, especially when these support spaces are tightly constructed or are in cold climates. Several studies (Batterman et al. 2006; Emmerich et al. 2003; Fugler 2004) of contaminant sources and transport in garages found that, in some cases, significant fractions of infiltration air enter houses from attached garages, and that modern residential garages are tighter than older garages, which were commonly assumed to be leaky enough to avoid many IAQ problems.

Particulates. The ventilation system should be designed such that return and outdoor air is filtered before passing through the thermal-conditioning components. Pressure drops associated with this filtration should be considered in the design of the air-handling system. Particulate filters or air cleaners should have a minimum efficiency of 60% for 0.125 in. particles, which is equivalent to a MERV 6 designated filter according to ASHRAE *Standard* 52.2.

Microbiologicals. Because ventilation can increase the source as well as removal rates of various air pollutants, it is, at best, moderately effective at reducing exposures to many airborne microbiologicals. Ventilation can, however, be part of the moisture balance that is critical to retarding fungal growth on surfaces and spores released into the air, depending on indoor/outdoor conditions.

Radon and Soil Gas. Buildings are exposed to gases that migrate from the soil through cracks or leaks. Soil gases vary with time and conditions, and can contain toxins from pesticides, landfill, fuel, or sewer gas, but the highest-profile pollutant in this category is radon and its radioactive-decay-produced "daughters." Source control measures, such as differential pressure control and airtightening, are far more effective than ventilation mechanisms at controlling exposure to soil gas. See Chapter 11 for more information.

Volatile Organic Compounds (VOCs). VOCs are ubiquitous in modern life. Products that emit VOCs include manufactured wood products, paints, stains, varnishes, solvents, pesticides, adhesives, wood preservatives, waxes, polishes, cleansers, lubricants, sealants, dyes, air fresheners, fuels, plastics, copy machines, printers, tobacco products, perfumes, cooking by-products, and dry-cleaned clothes. Whenever possible, VOCs and other toxic compounds should be stored outside the occupied space in loosely constructed or ventilated enclosures such as garden sheds, and away from occupied buildings' ventilation intakes. When unusual amounts of such compounds are present, additional ventilation should be considered.

Outdoor Air. Outdoor air may at times contain unacceptably high levels of pollutants, including ozone, pollen, carbon monoxide, particulate matter, odors, toxic agents, etc. At such times, it may be impossible to provide acceptable indoor air quality using solely outdoor air, and increased ventilation rates can actually decrease indoor air quality. In areas in which this problem may be anticipated, automatic or manual controls should be provided to allow reducing the ventilation rate. Cleaning recirculated air or using effective portable air cleaners should be considered for sensitive individuals.

Local Exhaust

After source elimination, the single most important source control mechanism in dwellings is local exhaust. All wet rooms and other spaces (e.g., kitchens, utility rooms, bathrooms, lavatories, toilets) designed to allow specific contaminant release should be provided with local exhaust. Workshops, recreation rooms, smoking areas, art studios, greenhouses, and hobby rooms may also require local ventilation and/or air cleaning to remove contaminants generated by the activities involved. Contaminants of concern should be evaluated to determine how much additional ventilation is required. Many of these rooms can be adequately ventilated by following the requirements for kitchens or bathrooms. If unvented combustion appliances must be used, rooms with these appliances should also meet general ventilation requirements for kitchens, because such appliances generate significant amounts of moisture and, often, ultrafine particles, even when burning properly.

Mechanical exhaust is the preferred method of providing local ventilation. Normally, it is designed to operate intermittently under manual control to exhaust contaminated air outside when the contaminant is being produced and occupants recognize the need for ventilation. However, in many circumstances, a continuous, lower-flow-rate exhaust can work as well.

Continuous Local Mechanical Exhaust. A continuously operating mechanical exhaust is intended to operate without occupant intervention. This exhaust may be part of a balanced mechanical ventilation system. The system should be designed to operate during all hours in which the dwelling is occupied. Override control should be provided if needed. The minimum delivered ventilation should be at least that given in Table 1.

Intermittent Local Mechanical Exhaust. An intermittently operating local mechanical exhaust is intended to be operated as needed by the occupant and should be designed with this intent. Shutoff timers, occupancy controls, multiple-speed fans, and switching integral with room lighting are helpful, provided they do not impede occupant control. The minimum airflow rate should be at least that given in Table 2.

Alternatives. Cleaning recirculated air can sometimes be substituted for local exhaust, if it can be shown to be effective in removing contaminants of concern. Natural ventilation is not generally a suitable method for local exhaust and ventilation air needs in most climates and spaces. Using natural ventilation can cause reentrainment problems when air flows into rather than out of the space, and contaminated exhaust or exfiltrating air reenters the building. In milder climates, natural ventilation may be acceptable when the contaminant of concern is related to odor rather than health or safety. Purpose-designed passive exhaust systems have shown acceptable ventilation in some European settings, and may be considered in lieu of mechanical systems. Axley (2001b) discusses evaluation and design of passive residential ventilation systems further.

Table 1 Continuous Exhaust Airflow Rates

Application	Airflow Rate	Notes
Kitchen	5 ach	Based on kitchen volume
Utility room, bathroom, toilet, lavatory	20 cfm	Not less than 2 ach

Table 2 Intermittent Exhaust Airflow Rates

Application	Airflow Rate	Notes
Kitchen	100 cfm	Vented range hood required if less than 5 ach
Utility room, bathroom, toilet, lavatory	50 cfm	Not less than 2 ach

Table 3 Total Ventilation Air Requirements

Area Based	Occupancy Based
1 cfm/100 ft^2 of floor space	7.5 cfm per person, based on normal occupancy

Whole-House Ventilation

Although control of significant sources of pollution in a dwelling is important, whole-house ventilation through centrally introduced, conditioned, and distributed outside air may still be needed. Each dwelling should be provided with outdoor air according to Table 3. The rate is the sum of the Area-Based and Occupancy-Based columns. Design occupancy can be based on the number of bedrooms as follows: first bedroom, two persons; each additional bedroom, one person. Additional ventilation should be considered when occupant densities exceed 1/250 ft^2.

Natural whole-house ventilation that relies on occupant operation should not be used to make up any part of the minimum total whole-house ventilation air requirement. However, because occupancy and sources vary significantly, the capacity to ventilate above minimum rates can be provided by operable exterior openings such as doors and windows.

Air Distribution

Ventilation air should be provided to each habitable room through mechanical and natural air distribution. If a room does not have a balance between air supply and return or exhaust, pathways for transfer air should be provided. These pathways may be door undercuts, transfer ducts with grilles, or simply grilles where ducts are not necessary or required by code.

In houses without central air handlers, special provisions to distribute outdoor air may be required. Rooms in which occupants spend many continuous hours, such as bedrooms, may require special consideration. Local and whole-house ventilation equipment should be chosen to be energy efficient, easy to maintain, reliable, durable, and quiet. Heat recovery should be considered, especially in cold climates.

Selection Principles for Residential Ventilation Systems

Occupant comfort, energy efficiency, ease of use, service life, first and life-cycle cost, value-added features, and indoor environmental quality should be considered when selecting a strategy and system. HVAC and related systems can be a potential cause of poor indoor air quality. For example, occupants may not use the ventilation systems as intended if operation results in discomfort (e.g., drafts) or excessive energy use. The resulting lack of ventilation might produce poor indoor air quality. Therefore, careful design, construction, commissioning, operation, and maintenance is necessary to provide optimum effectiveness.

All exhaust, supply, or air-handler fans have the potential to change the pressure of the living space relative to the outside. High-volume fans, such as the air handler and some cooking exhaust fans, can cause high levels of depressurization, particularly in tightly constructed homes. Considering these effects is essential in design. Excessive depressurization of the living space relative to outside may cause backdrafting of combustion appliances and the migration of contaminants such as radon or other soil gases, car exhaust, or insulation particles into the living space. Depressurization can also result in moisture intrusion into building cavities in warm, moist climates, which may cause structural damage and fungal growth. Pressurization of the living space can cause condensation in building cavities in cold climates, also resulting in structural damage. Excess pressure can best be prevented by balanced ventilation systems and tightly sealed duct systems. In addition, adequate pathways must be available for all return air to the air-handling devices.

Occupant activities, operation of fans that exhaust air from the home, and leaky ducts on air conditioners, furnaces, or heat pumps may depressurize the structure. Options to address backdrafting concerns include

- Using combustion appliances with isolated (or sealed) combustion systems

- Locating combustion appliances in a ventilated room isolated from depressurized zones by well-sealed partitions
- Installing supply fans to balance or partially balance exhaust from the zone
- Testing to ensure that depressurization is not excessive

The system must be designed, built, operated, and maintained in a way that discourages growth of biological contaminants. Typical precautions include sloping condensate drain pans toward the drain, keeping condensate drains free of obstructions, keeping cooling coils free of dirt and other obstructions, maintaining humidifiers, and checking and eliminating any cause of moisture inside ducts.

Outside and exhaust airstreams of ventilation systems can be coupled using a heat pump or other device to recover thermal energy, when appropriate. Such heat pump or other equipment may reverse mode with the seasons or sensed temperature differences, for example. Heat can also be recovered from air to preheat potable water, for example.

SIMPLIFIED MODELS OF RESIDENTIAL VENTILATION AND INFILTRATION

This section describes several calculation procedures, ranging from simple estimation techniques to more physical models. Orme (1999) provides a more thorough review of simplified models. A building's air exchange rate cannot be reliably deduced from the building's construction or age, or from a simple visual inspection. Some measurement is necessary, such as a pressurization test of envelope airtightness or a detailed quantification of the leakage sites and their magnitude. The air exchange rate of a building may be calculated given (1) the location and leakage function for every opening in the building envelope and between major building zones, (2) the wind pressure coefficients over the building envelope, and (3) any mechanical ventilation airflow rates. These inputs are generally unavailable for all except very simple structures or extremely well studied buildings. Therefore, their values must be assumed. The appropriateness of these assumptions influences the accuracy of predictions of air exchange rates.

Empirical Models

These models of residential infiltration are based on statistical fits of infiltration rate data for specific houses. They use pressurization test results to account for house airtightness and take the form of simple relations between infiltration rate, an airtightness rating, and, in most cases, weather conditions. Empirical models account for envelope infiltration only and do not deal with intentional ventilation. In one approach, the calculated air exchange rate at 0.20 in. of water based on a pressurization test is simply divided by a constant approximately equal to 20 (Sherman 1987). This technique does not account for the effect of infiltration-driving mechanisms on air exchange. Empirical models that do account for weather effects have been developed by Kronvall (1980), Reeves et al. (1979), and Shaw (1981).

The latter two models account for building air leakage using the values of c and n from Equation (40). The only other inputs required are wind speed and temperature difference. These empirical models predict long-term (one-week) infiltration rates very well in the houses from which they were developed; they do not, however, work as well in other houses because of the building-specific nature of leakage distribution, wind pressure, and internal partitioning. Persily (1986) and Persily and Linteris (1983) compared measured and predicted house infiltration rates for these and other models. The average long-term differences between measurements and predictions are generally on the order of 40%, although individual predictions can be off by 100% or more (Persily 1986; Walker and Wilson 1998).

Multizone Models

Multicell models of air exchange treat buildings as a series of interconnected zones and assume that air within each zone is well mixed. Several such models have been developed by Allard and Herrlin (1989), Etheridge and Alexander (1980), Feustel and Raynor-Hoosen (1990), Herrlin (1985), Liddament and Allen (1983), Walton (1984, 1989), and Walton and Dols (2003). They are all based on a mass balance for each zone of the building. These mass balances are used to solve for interior static pressures in the building by requiring that inflows and outflows for each zone balance to zero. The user must input information describing building envelope leakage, values to account for wind pressure on the building envelope, temperatures for each zone, and any mechanical ventilation airflow rates. Wind pressure coefficient data in the literature, air leakage measurement results from the building or its components, and air leakage data from the literature can be used as estimates. These models not only solve for whole-building and individual zone air exchange rates, but also determine airflow rates and pressure differences between zones. These interzone airflow rates are useful for predicting pollutant transport within buildings with well mixed zones. Chapter 13 has more details on multizone airflow and IAQ modeling.

Single-Zone Models

Several procedures have been developed to calculate building air exchange rates that are based on physical models of the building interior as a single zone. These single-zone models are only appropriate for buildings with no internal resistance to airflow, and are therefore inappropriate for large, multizone buildings. Some models of this type have been developed by Cole et al. (1980), Sherman and Grimsrud (1980), Walker and Wilson (1998), and Warren and Webb (1980). The section on Residential Calculation Examples uses both basic and enhanced models (Bradley 1993; CHBA 1994; Hamlin and Pushka 1994; Palmiter and Bond 1994; Walker and Wilson 1998).

The **basic model** uses effective air leakage area A_L at 0.016 in. of water, which can be obtained from a whole-building pressurization test. The **enhanced model** uses pressurization test results to characterize house air leakage through leakage coefficient c and pressure exponent n. The enhanced model improves on the basic model by using a power law to represent envelope leakage, including a flue as a separate leakage site, and having separate wind effects for houses with crawlspaces or slab/basement foundations.

For both models, the user must input wind speed, temperature difference, information on distribution of leakage over the building envelope, a wind shelter (or local shielding for the basic model) parameter, and a terrain coefficient. The predictive accuracy of the enhanced model can be very good, typically ±10% when parameters are well known for the building in question (Palmiter and Bond 1994; Sherman and Modera 1986; Walker and Wilson 1998). All these single-zone models are sensitive to values of inputs, which are quite difficult to determine.

Superposition of Wind and Stack Effects

Simplified physical models of infiltration solve the problem of two natural driving forces, wind and stack, separately and then combine them in a process called **superposition**. Superposition is necessary because each physical process can affect internal and external pressures on the structure, which can cause interactions between physical processes that are otherwise independent. An exact solution is impossible because detailed properties of all the building leaks are unknown and because leakage is a nonlinear process. For this reason, most modelers have developed a simplified superposition process to combine stack and wind effects. Sherman (1992b) compared various superposition procedures and derived a generalized superposition equation involving simple leakage distribution parameters, and showed that the result is always subadditive.

Typically, only 35% of infiltration from the smaller effect can be added to the larger effect. Depending on details, that percentage could go as high as 85% or as low as zero. Walker and Wilson (1993) compared several superposition techniques to measured data. Sherman, as well as Walker and Wilson, found quadrature, shown in Equation (47), to be a robust superposition technique:

$$Q = \sqrt{Q_s^2 + Q_w^2} \tag{47}$$

The following sections discuss how superposition is combined with calculation of wind and stack flows to determine total flow.

Residential Calculation Examples

Basic Model. The following calculations are based on the Sherman and Grimsrud (1980) model, which uses the effective air leakage area at 0.016 in. of water. This leakage area can be obtained from a whole-building pressurization test. Using effective air leakage area, the airflow rate from infiltration is calculated according to

$$Q = A_L \sqrt{C_s \Delta t + C_w U^2} \tag{48}$$

where

Q = airflow rate, cfm
A_L = effective air leakage area, in^2
C_s = stack coefficient, cfm^2/(in^4·°F)
Δt = average indoor-outdoor temperature difference for time interval of calculation, °F
C_w = wind coefficient, cfm^2/(in^4·mph^2)
U = average wind speed measured at local weather station for time interval of calculation, mph

Table 4 presents values of C_s for one-, two-, and three-story houses. The value of wind coefficient C_w depends on the local shelter class of the building (described in Table 5) and the building height. Table 6 presents values of C_w for one-, two-, and three-story houses in shelter classes 1 through 5. In calculating values in Tables 4 and 6, the following assumptions were made regarding input to the basic model:

Table 4 Basic Model Stack Coefficient C_s

	House Height (Stories)		
	One	Two	Three
Stack coefficient	0.0150	0.0299	0.0449

Table 5 Local Shelter Classes

Shelter Class	Description
1	No obstructions or local shielding
2	Typical shelter for an isolated rural house
3	Typical shelter caused by other buildings across street from building under study
4	Typical shelter for urban buildings on larger lots where sheltering obstacles are more than one building height away
5	Typical shelter produced by buildings or other structures immediately adjacent (closer than one house height): e.g., neighboring houses on same side of street, trees, bushes, etc.

Table 6 Basic Model Wind Coefficient C_w

	House Height (Stories)		
Shelter Class	One	Two	Three
1	0.0119	0.0157	0.0184
2	0.0092	0.0121	0.0143
3	0.0065	0.0086	0.0101
4	0.0039	0.0051	0.0060
5	0.0012	0.0016	0.0018

- Terrain used for converting meteorological to local wind speeds is that of a rural area with scattered obstacles
- R = 0.5 (half the building leakage in the walls)
- X = 0 (equal amounts of leakage in the floor and ceiling)
- Heights of one-, two-, and three-story buildings = 8, 16, and 24 ft, respectively

Example 2. Estimate the infiltration at design conditions for a two-story house in Lincoln, Nebraska. The house has effective air leakage area of 77 in^2 and volume of 12,000 ft^3, and the predominant wind is perpendicular to the street (shelter class 3). The indoor air temperature is 68°F.

Solution: The 99% design temperature for Lincoln is –2°F. Assume a design wind speed of 15 mph. From Equation (48), with C_s = 0.0299 from Table 4 and C_w = 0.0086 from Table 6, the airflow rate caused by infiltration is

$$Q = 77\sqrt{(0.0299)(70) + (0.0086)(15^2)} = 155 \text{ cfm} = 9300 \text{ ft}^3/\text{h}$$

From Equation (2), air exchange rate I is equal to Q divided by the building volume:

$$I = (9300 \text{ ft}^3/\text{h})/12{,}000 \text{ ft}^3 = 0.78 \text{ h}^{-1} = 0.78 \text{ ach}$$

Example 3. Predict the average infiltration during a one-week period in January for a one-story house in Portland, Oregon. During this period, the average indoor/outdoor temperature difference is 30°F, and average wind speed is 6 mph. The house has volume of 9000 ft^3 and effective air leakage area of 107 in^2, and it is located in an area with buildings and trees within 30 ft in most directions (shelter class 4).

Solution: From Equation (48), the airflow rate caused by infiltration is

$$Q = 107\sqrt{(0.0150)(30) + (0.0039)(6^2)} = 82.2 \text{ cfm} = 4930 \text{ ft}^3/\text{h}$$

The air exchange rate is therefore

$$I = 4930/9000 – 0.55 \text{ h}^{-1} = 0.55 \text{ ach}$$

Example 4. Estimate the average infiltration over the heating season in a two-story house with volume of 11,000 ft^3 and leakage area of 131 in^2. The house is located on a lot with several large trees but no other close buildings (shelter class 3). Average wind speed during the heating season is 7 mph, and the average indoor/outdoor temperature difference is 36°F.

Solution: From Equation (48), the airflow rate from infiltration is

$$Q = 131\sqrt{(0.0299)(36) + (0.0086)(7^2)} = 160 \text{ cfm} = 9620 \text{ ft}^3/\text{h}$$

The average air exchange rate is therefore

$$I = 9620/11{,}000 = 0.87 \text{ h}^{-1} = 0.87 \text{ ach}$$

Enhanced Model. This section presents a simple, single-zone approach to calculating air infiltration rates in houses based on the Walker and Wilson (1998) model. The airflow rate from infiltration is calculated using

$$Q_s = c\,C_s \Delta t^n \tag{49}$$

$$Q_w = cC_w(sU)^{2n} \tag{50}$$

where

Q_s = stack airflow rate, cfm
Q_w = wind airflow rate, cfm
c = flow coefficient, cfm/(in. of water)^n
C_s = stack coefficient, (in. of water)/°F)^n
C_w = wind coefficient, (in. of water/mph^2)^n
s = shelter factor

In calculating tabulated values of C_s, C_w, and s, the following assumptions were made:

- Each story is 8 ft high.
- The flue is 6 in. in diameter and reaches 6 ft above the upper ceiling.

- The flue is unsheltered.
- Half of envelope leakage (not including the flue) is in the walls and one-quarter each is at the floor and ceiling, respectively.
- $n = 0.67$

Using typical values for terrain factors, house height, and wind speed measurement height, wind speed multiplier G (given in Table 7) uses a relationship based on equations found in Chapter 24 and used in the following examples.

Example 5. Estimate the infiltration at design conditions for a two-story slab-on-grade house with a flue in Lincoln, Nebraska. The house has a flow coefficient of $c = 2.6$ cfm/(in. of water)^n and a pressure exponent of $n = 0.67$ (this corresponds to effective leakage area of 77 in^2 at 0.016 in. of water). The building volume is 12,000 ft^3. The 97.5% design temperature is –2°F, and design wind speed is 15 mph.

Solution: For a slab-on-grade two-story house with a flue, Table 8 gives $C_s = 2.41$ [(in. of water)/°F]^n and $C_w = 2.14$ [(in. of water)/ mph^2]^n. The house is maintained at 68°F indoors. The building wind speed is determined by taking design wind speed U_{met} and multiplying by the wind speed multiplier G from Table 7:

$$U = GU_{met} = 0.59(15) = 8.9 \text{ mph}$$

From Table 5, the shelter class for a typical urban house is 4. Table 9 gives the shelter factor for a two-story house with a flue and shelter class 4 as $s = 0.64$. The stack flow is calculated using Equation (49):

$$Q_s = (2.6)(2.41)[68 - (-2)]^{0.67} = 108 \text{ cfm}$$

The wind flow is calculated using Equation (50):

$$Q_w = (2.6)(2.14)[0.64 \times 8.9)]^{1.34} = 57 \text{ cfm}$$

Substituting Q_s and Q_w into Equation (47) gives Q = 122 cfm = 7300 ft^3/h. From Equation (2), air exchange rate I is equal to Q divided by building volume:

$$I = (7300 \text{ ft}^3/\text{h})/12{,}000 \text{ ft}^3 = 0.61 \text{ h}^{-1} = 0.61 \text{ ach}$$

Example 6. Estimate the average infiltration over a one-week period for a single-story crawlspace house in Redmond, Washington. The house has a flow coefficient of $c = 4.1$ cfm/(in. of water)^n and a pressure exponent of $n = 0.6$ (this corresponds to effective leakage area of 107 in^2 at 0.016 in. of water). The building volume is 9000 ft^3. During this period, the average indoor/outdoor temperature difference is 29°F, and wind speed is 6 mph. The house is electrically heated and has no flue.

Solution: For a single-story house with no flue, $C_s = 1.46$ [(in. of water)/°F]^n. For a crawlspace, $C_w = 1.75$ [(in. of water)/mph^2]^n. From Table 7, for a one-story house, $G = 0.48$.

$$U = GU_{met} = 0.48(6) = 2.9 \text{ mph}$$

Table 9 gives shelter factor $s = 0.50$ for a house with no flue and shelter class 4. Stack flow is calculated using Equation (49):

$$Q_s = (4.1)(1.46)(29)^{0.6} = 45 \text{ cfm}$$

Wind flow is calculated using Equation (50):

$$Q_w = (4.1)(1.75)[0.50 \times 8.9)]^{1.2} = 11.2 \text{ cfm}$$

Substituting Q_s and Q_w into Equation (47) gives Q = 46 cfm = 2800 ft^3/h. From Equation (2), air exchange rate I is equal to Q divided by building volume:

$$I = (2800 \text{ ft}^3/\text{h})/9000 \text{ ft}^3 = 0.31 \text{ h}^{-1} = 0.31 \text{ ach}$$

Example 7. Estimate the infiltration for a three-story house in San Francisco, California. The house has a flow coefficient of $c = 5.2$ cfm/(in. of water)^n and a pressure exponent of $n = 0.67$ (this corresponds to effective leakage area of 155 in^2 at 0.016 in. of water). The building volume is 14,200 ft^3. The indoor/outdoor temperature difference is 9°F and wind speed is 10 mph. The house has a flue and a crawlspace.

Solution: For a three-story house with a flue, $C_s = 2.92$ [(in. of water)/ °F]^n. For a crawlspace, $C_w = 2.11$ [(in. of water)/mph^2]^n. From Table 7, for a three-story house, $G = 0.67$.

$$U = GU_{met} = 0.67(10) = 6.7 \text{ mph}$$

The prevailing wind blows along the row of houses parallel to the street, so the house has a shelter class of 5. Table 9 gives the shelter factor for a three-story house with a flue and shelter class 5 as $s = 0.43$.

$$Q_s = (5.2)(2.92)(9)^{0.67} = 66 \text{ cfm}$$

$$Q_w = (5.2)(2.11)[0.43 \times 6.7)]^{1.34} = 45 \text{ cfm}$$

Substituting Q_s and Q_w in Equation (47) gives Q = 80 cfm = 4800 ft^3/h.

$$I = (4800 \text{ ft}^3/\text{h})/(14{,}200) \text{ ft}^3 = 0.34 \text{ h}^{-1} = 0.34 \text{ ach}$$

Combining Residential Infiltration and Mechanical Ventilation

Significant infiltration and mechanical ventilation often occur simultaneously in residences. The pressure difference from Equation (31) can be used for each building leak, and the flow network (including mechanical ventilation) for the building can be solved to find the flow through all the leaks while accounting for the effect of the mechanical ventilation. However, for simplified models, natural infiltration and mechanical ventilation are usually determined separately and require a superposition method to combine the flow rates.

Sherman (1992b) compared various superposition procedures and derived a generalized superposition equation that involves simple leakage distribution parameters. The result is always subadditive. For small unbalanced fans, typically only half the flow contributes to the total, but this fraction can be anywhere between 0 and 100%, depending on leakage distribution. When fan flow is large, infiltration may be ignored.

In special cases when the leakage distribution is known and highly skewed, it may be necessary to work through the superposition method in more detail. For example, in a wind-dominated situation, a supply fan has a much bigger effect than an exhaust fan on changing the total ventilation rate; the same is true for houses with high neutral levels in cold climates. For the general case, when

Table 7 Enhanced Model Wind Speed Multiplier G

	House Height (Stories)		
	One	Two	Three
Wind speed multiplier G	0.48	0.59	0.67

Table 8 Enhanced Model Stack and Wind Coefficients

	One Story		Two Story		Three Story	
	No Flue	With Flue	No Flue	With Flue	No Flue	With Flue
C_s	1.46	1.87	2.13	2.41	2.68	2.92
C_w for basement slab	2.14	1.95	2.34	2.14	2.34	2.29
C_w for crawlspace	1.75	1.75	1.95	1.95	2.07	2.11

Table 9 Enhanced Model Shelter Factor s

Shelter Class	No Flue	One Story with Flue	Two Story with Flue	Three Story with Flue
1	1.00	1.10	1.07	1.06
2	0.90	1.02	0.98	0.97
3	0.70	0.86	0.81	0.79
4	0.50	0.70	0.64	0.61
5	0.30	0.54	0.47	0.43

details are not known or can be assumed to be broad and typical, the following superposition gives good results:

$$Q_{comb} = Q_{bal} + \sqrt{Q_{unbal}^2 + Q_{infiltration}^2} \tag{51}$$

Typical Practice

The preceding sections on estimating infiltration in low-rise residences represent current analytical techniques typically used for research and remediation purposes, but most small residential buildings are designed and constructed without direct involvement of ventilation engineers. Contractors, who typically prepare these buildings' designs, are required to follow mandates in various codes and standards, and they apply experience-based rules of thumb when determining, for example, exhaust needs. Often, leaky buildings or air quality problems result. Research and experience has shown that tightening building envelopes, and potentially using mechanical ventilation with heat recovery, can yield improved indoor air quality and reduced energy consumption. Retaining the services of a ventilation engineer before construction begins is advisable in some situations.

COMMERCIAL AND INSTITUTIONAL AIR LEAKAGE

Commercial Building Envelope Leakage

ASTM *Standard* E779 and CGSB *Standard* 149.10 include methods to measure the airtightness of building envelopes of single-zone buildings. Although many multizone buildings can be treated as single-zone buildings by opening interior doors or by inducing equal pressures in adjacent zones, these standards provide no guidelines for dealing with problems arising in tall buildings, such as stack and wind effects. Tall buildings require refinement and extensions of established procedures because they have obstacles to accurate measurement not present in small buildings, including large envelope leakage area, interfloor leakage, vertical shafts, and large wind and stack pressures. In conducting a fan pressurization test in a large building, the building's own air-handling equipment sometimes can be used to induce test pressures, as described in CGSB *Standard* 149.15. In other cases, a large fan is brought to the building to perform the test, as described by CIBSE *Standard* TM-23. Bahnfleth et al. (1999) also discuss how to address some of these issues.

Building envelopes of large commercial buildings are often assumed to be quite airtight. Tamura and Shaw (1976a) found that, assuming a flow exponent n of 0.65 in Equation (40), air leakage measurements in eight Canadian office buildings with sealed windows ranged from 0.120 to 0.480 cfm/ft^2. Persily and Grot (1986) ran whole-building pressurization tests in large office buildings that showed that pressurization airflow rate divided by building volume is relatively low compared to that of houses. However, if these airflow rates are normalized by building envelope area instead of by volume, the results indicate envelope airtightness levels similar to those in typical American houses. The same study also looked at eight U.S. office buildings and found air leakage ranging from 0.213 to 1.028 cfm/ft^2 at 0.30 in. of water. This means that office building envelopes are leakier than expected. Typical air leakage values per unit wall area at 0.30 in. of water are 0.10, 0.30, and 0.60 cfm/ft^2 for tight, average, and leaky walls, respectively (Tamura and Shaw 1976a).

Emmerich and Persily (2005) summarize available measured airtightness data for 203 U.S. commercial and institutional buildings. Sources of data included 9 buildings tested by the National Institute of Standards and Technology (Musser and Persily 2002; Persily and Grot 1986; Persily et al. 1991), 90 tested by the Florida Solar Energy Center (Cummings et al. 1996, 2000), 2 tested by Pennsylvania State University (Bahnfleth et al. 1999), 23 tested by Camroden Associates (Brennan et al. 1992 and previously unpublished data), and 79 buildings tested by the U.S. Army Corps of Engineers (previously unpublished data). Tested buildings were of a wide range of types and ages but were primarily low-rise buildings. The overall average airtightness of 1.55 cfm/ft^2 of above-grade envelope surface area at 0.0109 psi is in the same range as that reported for typical U.S. houses, and is similar to averages reported by Potter (2001) for U.K. commercial buildings built before recent airtightness regulations. The data show that taller buildings tend to be tighter, and a lack of correlation between year of construction and observed building air leakage. This study also found a trend, with considerable scatter, toward tighter buildings in colder climates. The authors caution that conclusions from this analysis are limited by the small sample size and a lack of random sampling. None of the buildings are known to have been constructed to meet a specified air leakage criterion, which has been identified as a key to achieving tight building envelopes in practice.

Grot and Persily (1986) also found that eight recently constructed office buildings had infiltration rates ranging from 0.1 to 0.6 ach with no outdoor air intake. The infiltration rates of these buildings exhibited varying degrees of weather dependence, generally much lower than that measured in houses. Infiltration in commercial buildings can have many negative consequences, including reduced thermal comfort, interference with proper operation of mechanical ventilation systems, degraded indoor air quality, moisture damage of building envelope components, and increased energy consumption. These results suggest strongly that commercial buildings' envelopes require tighter construction, and that continuous air barrier systems should be used in all conditioned buildings. Since 1997, the Building Environment and Thermal Envelope Council of the National Institute of Building Sciences has sponsored several symposia on air barriers for buildings in North American climates. Others have also published articles on the importance of limiting air leakage in commercial buildings (Anis 2001; Ask 2003; Fennell and Haehnel 2005).

Envelope leakage in commercial buildings also depends on HVAC system operation. Often, commercial buildings, and their HVAC systems, are in operation during normal daytime business hours but switch into "unoccupied" operation at nights and on weekends. If pressurized while their HVAC systems operate, infiltration is often very low or even eliminated in buildings with tight envelopes. However, in unoccupied mode, this pressurization is often lost, so infiltration and potentially moisture intrusion may be significant at times.

Air Leakage Through Internal Partitions

In large buildings, air leakage associated with internal partitions becomes very important. Elevator, stair, and service shaft walls; floors; and other interior partitions are the major separations of concern in these buildings. Their leakage characteristics are needed to determine infiltration through exterior walls and airflow patterns in a building. These internal resistances are also important in the

Table 10 Air Leakage Areas for Internal Partitions in Commercial Buildings (at 0.30 in. of water and C_D = 0.65)

Construction Element	Wall Tightness	Area Ratio
		A_L/A_w
Stairwell walls	Tight	0.14×10^{-4}
	Average	0.11×10^{-3}
	Loose	0.35×10^{-3}
Elevator shaft walls	Tight	0.18×10^{-3}
	Average	0.84×10^{-3}
	Loose	0.18×10^{-2}
		A_L/A_f
Floors	Average	0.52×10^{-4}

A_L = air leakage area A_w = wall area A_f = floor area

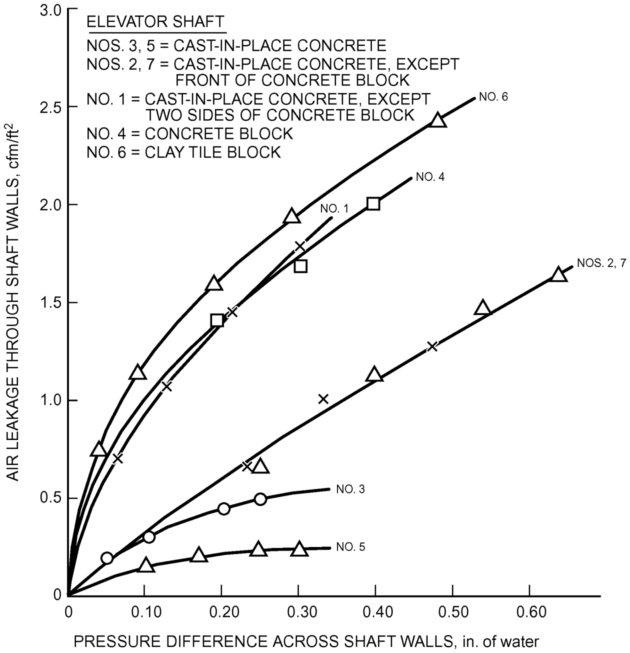


Fig. 14 Air Leakage Rates of Elevator Shaft Walls

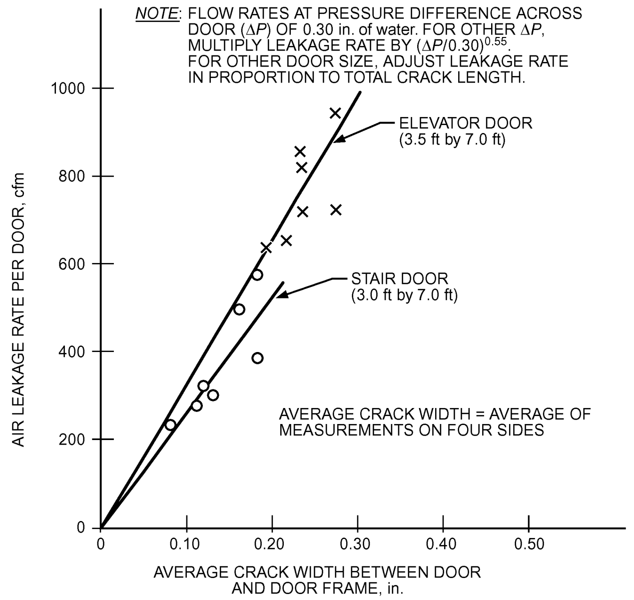


Fig. 15 Air Leakage Rate of Door Versus Average Crack Width

event of a fire to predict smoke movement patterns and evaluate smoke management systems.

Table 10 gives air leakage areas calculated at 0.30 in. of water with C_D = 0.65 for different internal partitions of commercial buildings (Klote and Milke 2002). Figure 14 presents examples of measured air leakage rates of elevator shaft walls (Tamura and Shaw 1976b), the type of data used to derive the values in Table 10. Consult Chapter 52 of the 2007 *ASHRAE Handbook—HVAC Applications* for performance models and applications of smoke management systems.

Leakage openings at the top of elevator shafts are equivalent to orifice areas of 620 to 1550 in^2. Air leakage rates through stair shaft and elevator doors are shown in Figure 15 as a function of average crack width around the door. Air leakage areas associated with other openings in commercial buildings are also important for air movement calculations. These include interior doors and partitions, suspended ceilings in buildings where space above the ceiling is used in air distribution, and other components of the air distribution system.

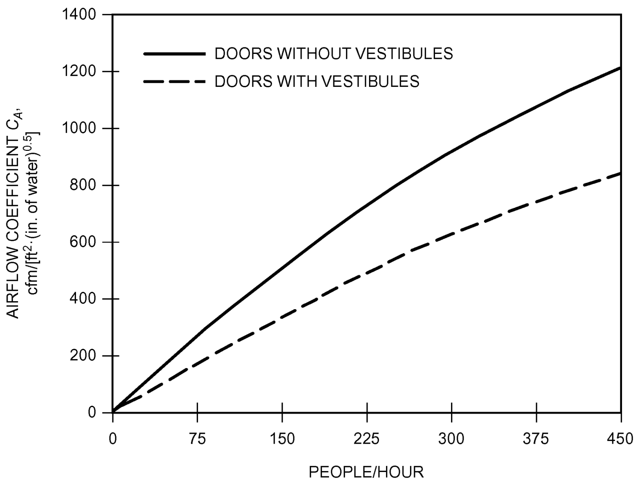


Fig. 16 Airflow Coefficient for Automatic Doors

Air Leakage Through Exterior Doors

Door infiltration depends on the type and use of door, room, and building, and on air speed and pressure differentials. In residences and small buildings where doors are used infrequently, air exchange associated with a door can be estimated based on air leakage through cracks between door and frame. Airflow increases significantly as door-opening frequency increases. Vestibules or revolving doors should be considered for high-frequency applications.

Air Leakage Through Automatic Doors

Automatic swinging, sliding, rotating, or overhead doors are a major source of air leakage in buildings. They are normally installed where large numbers of people use the doors or bulk goods are transported through the doorways. These doors stay open longer with each use than manual doors. Air leakage through automatic doors can be reduced by installing a vestibule. However, pairs of automatic doors on the inside and outside of a vestibule normally have overlapping open periods, even when used by only one person at a time. Therefore, it is important that designers take into account airflow through automatic doors when calculating heating and cooling loads in adjacent spaces.

To calculate the average airflow rate through an automatic door, the designer must take into account the area of the door, the pressure difference across it, the discharge coefficient of the door when it is open, and the fraction of time that it is open. Obtaining the discharge coefficient is complicated by the fact that it changes as the door opens and closes.

To simplify this calculation, ASHRAE research project RP-763 (Yuill 1996) developed Figure 16 to combine the discharge coefficients of doors as they open and close with the fraction of time that doors are open at a particular level of use. This figure presents an overall airflow coefficient as a function of the number of people using a door per hour. To obtain the average infiltration rate through an automatic door, multiply this coefficient by the door's opening area and by the square root of the pressure difference between the outdoor and indoor air at the door's location. The pressure difference across a door in a building depends on wind pressure on the building, stack effect caused by the indoor/outdoor temperature difference, and effects of air-handling system operation. It also depends on leakage characteristics of the building's exterior walls and of internal partitions.

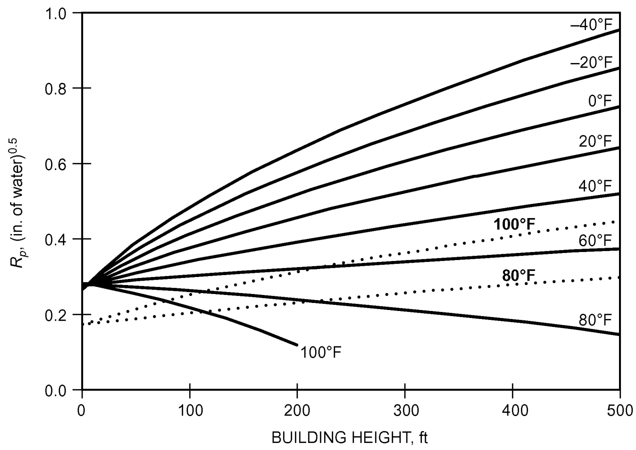


Fig. 17 Pressure Factor for Automatic Doors

Two simple methods are presented here. The first method uses simplifying assumptions to determine design values for R_p, the square root of the pressure difference across the automatic door, given in Figure 17. The second method requires explicit calculation of envelope pressures.

In Figure 17, airflows shown for outside air temperatures of 80 and 100°F, represented by dotted lines, are outward flows. They intercept the vertical axis at a lower point than the other lines because wind pressure coefficients on the building's downwind face, where the greatest outward flows occur, are lower than on the upward face. In many buildings, air pressure in the building is controlled by varying the flow rate through return fan(s) or by controlling the relief air dampers. These systems are usually set to maintain a pressure slightly above ambient in the lobby, but in a large building, multiple sensors may be used to regulate air pressure on each floor, for example. Subtracting the interior pressure maintained from the wind pressure gives the net pressure for estimating airflow through an exterior door.

Method 1. For the first method, the infiltration rate through the automatic door is given by

$$Q = C_A A R_p \tag{52}$$

where

Q = airflow rate, cfm
C_A = airflow coefficient from Figure 16, cfm/[ft^2·(in. of water)$^{0.5}$]
A = area of the door opening, ft^2
R_p = pressure factor from Figure 17, in. of water$^{0.5}$

Method 2. Airflow Q is given by

$$Q = C_A A \sqrt{\Delta p} \tag{53}$$

where

Q = airflow rate, cfm
C_A = airflow coefficient from Figure 16, cfm/[ft^2·(in. of water)$^{0.5}$]
A = area of the door opening, ft^2
Δp = pressure difference across door, in. of water

To find Δp, it is necessary to find the pressure differential from wind and that from stack effect. To give the largest possible pressure difference across the door, there are no interactions between the two natural pressures:

$$\Delta p = p_w - \Delta p_s \tag{54}$$

where

p_w = wind-induced surface pressure relative to static pressure, in. of water
Δp_s = pressure difference due to stack effect, in. of water

Example Calculations

Find the maximum possible infiltration through an automatic door located on the ground floor of a 20-story building. The area of the door is 36 × 84 in. = 3024 in^2 = 21 ft^2. Each floor is 13 ft high. Approximately 300 people per hour pass through the door. The design wind conditions are 15 mph, indoor temperature is 70°F, and outdoor temperature is 20°F. The airflow coefficient from Figure 16, using the line for doors without vestibules, is approximately 920 cfm/[ft^2·(in. of water)$^{0.5}$].

Method 1:

The pressure factor from Figure 17 is 0.5 (in. of water)$^{0.5}$. Equation (52) gives the door flow as

$$Q = 920(21)0.5 = 9660 \text{ cfm}$$

Method 2:

The worst possible case for wind surface pressure coefficient C_p at any point and in any position on the ground floor of the building is inferred from figures in Chapter 24 to be about 0.75. Using this in Equation (25), together with the specified wind speed, results in p_w = 0.082 in. of water. Assume that H is one-half the door height (42 in.). To have maximum pressure across the door, assume the neutral pressure plane is located halfway up the building such that

$$H_{\text{NPL}} = \frac{1}{2}(20 \text{ stories})\frac{13 \text{ ft}}{\text{story}} = 130 \text{ ft}$$

Substituting these values into Equation (24) gives Δp_s = –0.19 in. of water. This is the maximum stack pressure difference given no internal resistance to airflow. To find the actual stack pressure difference, it is necessary to multiply this by a draft coefficient. For this example, the coefficient is assumed to be 0.9, which is the highest value that has been found for tall buildings. Therefore, Δp_s = 0.9(–0.19 in. of water) = –0.17 in. of water. The total pressure is then Δp = 0.082 – (–0.17) = 0.252 in. of water. Substituting into Equation (53),

$$Q = 920(21)\sqrt{0.252} = 9700 \text{ cfm}$$

If the building has a vestibule, the airflow coefficient is read from Figure 16 using the line for doors with vestibules, and it is approximately 626 cfm/[ft^2·(in. of water)$^{0.5}$], reducing airflow to 6600 cfm into the building.

Air Exchange Through Air Curtains

Air curtains are jets of air projected across envelope openings with the intention of reducing air exchange and the entrance of dust and insects, for example. They are commonly applied to loading dock doorways and high-use building entrances. Performance of air curtains is highly dependent on factors such as jet characteristics, wind, and building pressurization. More discussion on air curtain performance is available in Chapter 17 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*.

COMMERCIAL AND INSTITUTIONAL VENTILATION

ASHRAE *Standard* 62.1 contains requirements on ventilation and indoor air quality for commercial, institutional, and high-rise residential buildings. These requirements address system and equipment issues, design ventilation rates, commissioning and systems start-up, and operation and maintenance. The user's manual for *Standard* 62.1-2007 (ASHRAE 2007) provides details to help the user design, install, and operate buildings to meet requirements. The design requirements include two alternative procedures:

- The prescriptive **ventilation rate procedure (VRP)** contains a table of outdoor air ventilation requirements for a variety of space types, with adjustments for air distribution in rooms and systems serving multiple spaces. These requirements consist of both a per-person rate and a per-floor-area rate. Minimum outside air ventilation rates are based, in part, on research by Berg-Munch et al. (1986), Cain et al. (1983), Iwashita et al. (1989), and Yaglou et al.

(1936), as well as years of experience of designers and building operators.

- The **indoor air quality procedure (IAQP)**, which achieves acceptable indoor air quality by controlling indoor contaminant concentrations through source control, air cleaning, and ventilation. It allows for either or both improved indoor air quality and reduced energy consumption. Chapter 28 of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* has information on air cleaning.

The ventilation rate procedure is by far the more commonly used.

Combining source control and local exhaust, as opposed to dilution with ventilation air, is the method of choice in many industrial environments. Industrial ventilation is discussed in Chapters 29 and 30 of the 2007 *ASHRAE Handbook—HVAC Applications* and in *Industrial Ventilation: A Manual of Recommended Practice* (ACGIH 2001). Ventilation of medical facilities, where high indoor air quality is expected, is discussed in ASHRAE *Standard* 62.1, Chapter 7 of the 2007 *ASHRAE Handbook—HVAC Applications*, and other publications [e.g., AIA et al. (2006)].

Commercial and institutional building ventilation systems are typically designed to provide slight pressurization to minimize infiltration. This pressurization is achieved by having the outside or makeup airflow rate higher than the exhaust or relief airflow rate. In these buildings, infiltration is usually neglected except in areas such as lobbies and loading docks, where infiltration can be important because of doors. However, as discussed previously, this may only be achieved in practice with tight envelope construction such as by including a continuous air barrier. As discussed in the section on Driving Mechanisms for Ventilation and Infiltration, wind and the stack effect can also cause significant infiltration and exfiltration. Ventilation airflow rates for commercial and institutional buildings are typically determined using procedures in ASHRAE *Standard* 62.1. In these procedures for designing mechanical ventilation systems, no credit is given for infiltration. However, weather-driven pressure differentials may be significant and need to be considered when designing the ventilation system.

Ventilation Rate Procedure

Per ASHRAE *Standard* 62.1, the design ventilation rate is determined based on a table of minimum ventilation requirements for different space types. These requirements are expressed as an outdoor airflow rate per occupant or per unit floor area, or often both, depending on space type. These ventilation rates are based on air pollutants generated by people, activities, and building materials and furnishings. The rates are then adjusted for various parameters (e.g., multiple zones, type of room air distribution).

The HVAC designer faces several challenges in designing an air distribution system to deliver outdoor air to building occupants. The first is to determine whether the outdoor air is acceptable for use, and to design a system for cleaning the air if it is not acceptable. A second is to design an air intake and distribution system that will *deliver* the required level of outdoor air to the occupied portions of the building, and not just *admit* it to an air handler. This outdoor air must be delivered not only at design conditions, but throughout the year. The task is complicated by weather-related variations in indoor/outdoor pressure difference. Other complications include pressure variations caused by building components such as exhaust fans or dirty filters, and probably most significantly by supply flow variations associated with variable-air-volume (VAV) systems (Janu et al. 1995; Mumma and Wong 1990). This delivery issue is related to the discussion in the section on Air Change Effectiveness.

Survey of Ventilation Rates in Office Buildings

Relatively few measurements of as-built office building ventilation performance have been conducted, and those data generally have not used consistent measurement methods or involved representative collections of buildings. The U.S. Environmental Protection Agency (EPA) Building Assessment Survey and Evaluation (BASE) study involved indoor environmental measurements, including ventilation, in 100 randomly selected office buildings using a standardized protocol (EPA 2003). Persily et al. (2005) analyzed the BASE data and found that outdoor ventilation rates measured using duct traverses at air handler intakes were higher than might be expected, with a mean of about 117 cfm per person. However, these elevated values are partially explained by low occupant density (mean of about four persons per 1000 ft^2) and high outdoor air fractions (mean of about 35%). Considering only values that correspond to minimum outdoor air intake, the mean ventilation rate was 23 cfm per workstation. About one-half the ventilation rates under minimum outdoor air intake were below 20 cfm per person. Another key outcome of this study is documentation of measured airflow rates that are quite different from their design values. This finding highlights the need for good system commissioning and maintenance to achieve design intent. Designing and configuring systems to encourage regular maintenance by providing easy access to key system components is also important.

OFFICE BUILDING EXAMPLE

Ventilation and infiltration principles from this chapter, *Standard* 62.1-2007, and elsewhere are applied to a conventional office building in Atlanta, Georgia. The building's floor plans, elevations, and wall sections are available in Chapter 18. The infiltration, local exhaust, or ventilation airflow rates in this example can be used later in the design process (1) as input for the heating and cooling load calculations; (2) for sizing fans, ducts, and dampers; and (3) for inclusion in the construction documents' air-handling units (AHUs) schedules and specifications.

This example relies on the 2007 edition of ASHRAE *Standard* 62.1; because this and other standards are updated frequently, users should check for the latest edition.

Location

The example building is about 8 mi northeast of downtown Atlanta, and is close to a major highway and its access roads. Atlanta's climate is hot and humid in the summer, and has relatively mild winters. The average annual outdoor air temperature is about 60.6°F and the heating degree-days per year, base 65°F (HDD_{65}), are about 3265 (Rock 2005). From Chapter 14, the winter 99% design outdoor air (OA) temperature is 23°F, whereas the 1% cooling dry-bulb temperature is 91°F, with a mean coincident wet-bulb temperature of 74°F. The 99.6 and 0.4% design wind speeds are 12 mph in the winter and 9 mph in the summer, both out of the northwest. Warm and humid winds also travel north from the Gulf of Mexico, and occasionally the wind is from the Atlantic Ocean from the southeast.

Building

The approximately 30,500 ft^2 building is a two-story, flat-roofed, slab-on-grade commercial office building with a substantial roof overhang in each direction. Materials and construction quality are average commercial grade. The double-paned windows, and similar spandrel glass, are fixed in their metal curtain wall frames; all windows are nonoperable. The remaining portions of the exterior walls are brick. There are relatively few doors to the outside, as described later in this example. The building is surrounded by black asphalt driveways, a parking lot, and some vegetation. The nearby highway is across a parallel two-lane access road, to the northwest.

Occupancy

The building is occupied during normal weekday business hours, and occasionally for special weekend events. Night and weekend thermostat setbacks are used. On the perimeter of the building are mostly single-person offices and conference rooms. The core of the building is mainly open-plan with cubicle workspaces, as well as various support rooms, restrooms, two stair towers, and an elevator. There is a large mailroom on the first floor and a lunchroom on the second. Occupant density is high during workdays. The overhead fluorescent lighting is typical of such office buildings, and there are significant computing, printing, and copying equipment loads. Smoking is not allowed in the building.

The building is owner-occupied. Owners generally have long-term interests in minimizing costs, and in maximizing indoor air quality and thermal comfort so that productivity is high.

Infiltration

For this example, assume a conventional all-air overhead HVAC system, and that the building is well sealed. Consequently, a slightly positive overall building pressurization is assumed during occupied hours, because many existing commercial buildings are too leaky to be pressurized effectively. Because water condensation in the exterior envelope of the building is possible, air pressurization should be as low as is practical, and continuous vapor retarders should be installed. As a more expensive alternative to slight pressurization, the automatic control system could actively manage the dampers' positions and fans' operation to maintain an average neutral pressurization, relative to the outdoors. In either case, a good assumption is that infiltration is minimized, the windows and spandrel glass are fixed and well sealed, and the exterior doors are normally kept closed. During high-wind conditions beyond design, windward perimeter spaces may have some infiltration loads, but under non-peak outdoor temperatures, a well-zoned HVAC system should have enough capacity to handle these extra loads. If both the OA temperature and wind are extreme, then these upwind perimeter spaces may become slightly uncomfortable. These extreme conditions are expected to occur only a few hours in a typical year.

Spaces with exterior doors can experience significant infiltration loads when people enter and leave. The first-floor vestibules on the north and south sides of the building help limit this infiltration through the two main entrances. The double doors from stair tower #2 have infrequent use, and a high level of thermal comfort in stair towers is not typically expected. Thus, brief infiltration surges in stair tower #2 are deemed acceptable. However, the doors from the parking lot to the mailroom are frequently used by staff for shipping, receiving, entrance, and egress, and infiltration loads on the vestibules and mailroom are of concern. Many designers choose to ignore these extra loads in pressurized buildings, because they are transient and not easily characterized; the systems' capacities are likely sufficient to minimize uncomfortable conditions in these spaces. In this example, however, the HVAC designer is concerned about summertime airborne moisture, especially in the mailroom where books and other publications are stored, because strong, humid, southerly winds easily overcome a slight indoor pressurization when the large doors to the parking lot are open.

This chapter and many of its supporting references describe detailed methods for estimating infiltration or air leakage. Typically, pressure differences, openings' coefficients, and hour-by-hour weather data are required to perform these transient calculations, usually using a computer program separate from that used for thermal load calculations. For HVAC design purposes for a building similar to the example, an air exchange rate of unconditioned outside air through infiltration, per space, expressed in air changes per hour (ach) or airflow rate (cfm) is of more immediate use. Either value is then entered into the load calculation program. Unfortunately, accurate air changes per hour are difficult, if not impossible, to predict, so design estimates must be made. For example,

North Vestibule, Room 101

- Gross floor area ≈ 11 ft × 13 ft = 143 ft^2
- Room volume ≈ 143 ft^2 × 9 ft = 1287 ft^3
- $ach_{inf} \approx 1.0$, so
- $Q_{inf} \approx (1287 \text{ ft}^3 \times 1.0)/60 \text{ min/h} \approx 22 \text{ cfm}_{oa}$

Either 1.0 ach or 22 cfm of infiltration is then used as input for the load calculation program for this space. The 1.0 ach assumption was made by the designer during on-site observation that these particular manually operated exterior doors have low usage. If passage rates were known, Yuill's (1996) flow rate estimation method would have been used instead.

South Vestibule, Room 115

- Gross floor area ≈ 8 ft × 10 ft = 80 ft^2
- Room volume ≈ 80 ft^2 × 9 ft = 720 ft^3
- $ach_{inf} \approx 2.0$, so
- $Q_{inf} \approx (720 \text{ ft}^3 \times 2.0)/60 \text{ min/h} \approx 24 \text{ cfm}_{oa}$

In practice, this back entrance from the parking lot on the southeast side of the building is the primary means of entrance and egress, and as such, the estimated infiltration for it is increased to 2.0 ach, compared to the north vestibule's 1.0 ach.

In colder U.S. climates, it is common practice for low-cost commercial buildings to have only space heating, and not cooling, in stair towers and vestibules. However, for this building in the Southeast, the designer decided to provide cooling for these vestibules. Thus, the estimated infiltration rates are applied to both the heating and cooling load calculations for these spaces. The building's mailroom, which also has exterior doors, is to be heated and cooled, too.

Mailroom, Room 114

- Gross floor area ≈ (51 ft × 22 ft) + (33 ft × 10 ft) = 1452 ft^2
- Room volume ≈ 1452 ft^2 × 9 ft = 13,068 ft^3
- $ach_{inf} \approx 0.5$, so
- $Q_{inf} \approx (13{,}068 \text{ ft}^3 \times 0.5)/60 \text{ min/h} \approx 109 \text{ cfm}_{oa}$

Even though the mailroom has only a single layer of doors to the outside, and not a vestibule, the designer estimated the infiltration at a lower rate (0.5 ach) than those for the vestibules. This is because of the mailroom's large interior volume relative to its exterior doorway's area.

Note that *no* estimate of air changes will be accurate at all times; this portion of HVAC design is still largely an art because of the many unknowns and variability of weather and building use. For improved energy conservation, all exterior doors must be extremely well weatherstripped and have automatic closers, and a sign indicating doors should be kept closed when not in use should be placed on the mailroom's doors. High-quality gaskets and sealants for the windows and spandrel glass are also required.

Local Exhausts

(This section assumes that ANSI/ASHRAE *Standard* 62.1-2007 has been adopted into the local building code without modification.) At least 10 rooms require direct, powered air exhaust: the two restrooms per floor, the darkroom, three designated photocopy spaces, and the two janitors' closets. The restrooms have three flushable fixtures each, so from Table 6.4 of *Standard* 62.1, with intermittent use, each restroom requires

$$Q_{ea} = 3 \text{ units} \times 50 \text{ cfm/unit} = 150 \text{ cfm}$$

Also from Table 6.4, the darkroom (room 222) of the second floor needs

$$\text{Gross floor area} \approx 10 \text{ ft} \times 15 \text{ ft} = 150 \text{ ft}^2$$

$$Q_{ea} = 150 \text{ ft}^2 \times 1.00 \text{ cfm/ft}^2 = 150 \text{ cfm}$$

Similarly, the designated photocopy areas need 0.50 $\text{cfm}_{ea}/\text{ft}^2$, so

$$\text{First floor, plan east: } \approx 80 \text{ ft}^2 \times 0.50 \text{ cfm/ft}^2 = 40 \text{ cfm}_{ea}$$

$$\text{First floor, plan southwest: } \approx 160 \text{ ft}^2 \times 0.50 \text{ cfm/ft}^2 = 80 \text{ cfm}_{ea}$$

$$\text{Second floor, plan east: } \approx 112 \text{ ft}^2 \times 0.50 \text{ cfm/ft}^2 = 56 \text{ cfm}_{ea}$$

The two small janitors' closets, one on each floor, also require exhaust:

$$60 \text{ ft}^2 \times 1.00 \text{ cfm/ft}^2 = 60 \text{ cfm}_{ea}$$

These local exhaust airflow rates are then entered into the load calculation program. They are room loads, attached to each particular space, and are *not* combined and entered as systems-level loads. The load calculation program evaluates the room loads, appropriately combines them, and then finds the systems-level loads for various peak hours.

Some local code authorities amend the requirements of *Standard* 62.1, or have not yet adopted the most current version, so significant deviations from these examples are possible. For example, in much of the United States, janitorial closets and photocopy rooms have not been required to have local exhausts. *Standard* 62.1-2007 recognized that these spaces can be significant sources of airborne pollutants, and some direct exhaust from them can be very beneficial for improving indoor air quality.

Ventilation

(This section assumes that ANSI/ASHRAE *Standard* 62.1-2007 has been adopted into local code without changes.) Ventilation air is needed to maintain acceptable indoor air quality. The example building is well sealed, natural ventilation is not used, and no credit for any infiltration is taken toward ventilation air requirements, as is typical for conventional commercial buildings. Thus, minimum ventilation air required by *Standard* 62.1 is provided mechanically through the AHUs. Because smoking is not allowed in the building, no extra ventilation for environmental tobacco smoke (ETS) is needed. However, considering outdoor air pollution from the major highway nearby as well as metropolitan Atlanta's smog, some outdoor air pretreatment may be considered later in the design process.

Standard 62.1 has two methods for determining needed ventilation airflow rates: the performance IAQ procedure (IAQP), and the prescriptive ventilation rate procedure (VRP). Most HVAC designers of conventional buildings with normal occupancies and outdoor air conditions use the VRP, which is appropriate for this example building.

Required ventilation air (conditioned outside air) is admitted to this building through two air-handling units; each AHU serves one floor. Flow rates of outside air are input values for, and carried through to the results of, the load calculation simulation. Energy needed to condition the outside air ultimately is a systems-level load, because all of this ventilation air is conditioned by the AHUs before its introduction to the building.

Commercial load calculation programs often provide suggested values of ventilation airflow rates and occupancy schedules, but may not have been updated to reflect the latest VRP requirements and procedures of *Standard* 62.1. As such, it is difficult to present an example here; instead, a sample check using some assumed values for the first-floor executive director's office (room 132) is given. It is assumed that this room is a separate thermal zone because of its use and its location on the southwest corner of the building and its two solar exposures.

Executive Director's Office, Room 132

- Gross floor area $\approx$ 12 ft $\times$ 21 ft = 252 ft^2
- Room volume $\approx$ 252 ft^2 $\times$ 9 ft = 2268 ft^3
- Assumed supply air Q_{sa} = 412 cfm

The supply airflow rate was estimated at 300 ft^2/ton, a sensible heat factor of 0.9, a cooling supply (55°F) to room (75°F) air temperature difference of 20°F, 12,000 Btu/h per ton of cooling, and the sensible heat equation 1.1 $\times$ cfm$_{sa}$ $\times$ ΔT. From Table 6.1 of *Standard* 62.1, the office's population P can be estimated as

$$P = 252 \text{ ft}^2 \times 5 \text{ occupants/1000 ft}^2 = 1.26$$

In this case, however, there is only one regular occupant of the space. The needed ventilation airflow rate to the breathing zone V_{bz} is then found from the table as follows:

$$V_{bz} = R_p P_z + R_a A_z$$

$$V_{bz} = (5 \text{ cfm/person} \times 1 \text{ person}) + (0.06 \text{ cfm/ft}^2 \times 252 \text{ ft}^2) = 20.12 \text{ cfm}$$

where

R_p = outdoor airflow rate required per person, from *Standard* 62.1's Table 6-1, cfm
P_z = zone population (largest number of people expected to occupy the zone during typical use)
R_a = outdoor airflow rate required per unit area, from *Standard* 62.1's Table 6-1, cfm
A_z = occupiable floor area of zone, ft^2

Note that *Standard* 62.1's VRP includes a building component $R_a A_z$, as well as the traditional per-person people component.

Because this is a conventional office building, with ceiling plenums and no raised floors, overhead air supply and return is assumed. The cooling mode, not heating, is dominant in this and most other U.S. office buildings that have high internal heat gains as well as well-sealed envelopes. From the standard's Table 6.2, with ceiling supply of cool air, the zone air distribution effectiveness E_z is estimated as 1.0. From Equation (6-2) of the standard, the design zone outdoor airflow rate V_{oz} is then

$$V_{oz} = V_{bz}/E_z = 20.12 \text{ cfm}/1.0 = 20.12 \text{ cfm}$$

But this is still not the amount of outside air that must be conditioned by the air handler: the rate must be adjusted for inefficiencies and recirculation in the air distribution system.

Because single-duct VAV with terminal reheat air distribution systems were initially planned by the designer, *Standard* 62.1's multiple-zone recirculating systems adjustment is needed. For this thermal zone, the primary outdoor air fraction Z_p for its VAV terminal unit and downstream is

$$Z_p = V_{oz}/V_{pz} = 20.12 \text{ cfm}/412 \text{ cfm} = 0.05, \text{ or } 5\%$$

However, for VAV systems, the minimum expected primary airflow rate should be used. In this case, 412 cfm is the peak design airflow rate. Designers often assume about 30% of this peak flow as the minimum in VAV systems, so for this space, 412 $\times$ 0.3 = 124 cfm. The adjusted primary outdoor air fraction is then

$$Z_p = V_{oz}/V_{pz} = 20.12 \text{ cfm}/124 \text{ cfm} = 0.16, \text{ or } 16\%$$

The preceding calculations need to be performed for every thermal zone on each air handler. Then, for each system, the highest primary outside air fraction is used to estimate the air distribution systems' ventilation effectiveness; ASHRAE (2007) includes a spreadsheet for doing these calculations.

For the purposes of this example, 0.16 is assumed to be the maximum Z_p, so, from Table 6.3 of *Standard* 62.1, the system ventilation efficiency E_v is 0.9. If, instead, the standard's Appendix A method for determining E_v were used, a value closer to 1.0 (perfect mixing) would likely result for this example's conventional overhead all-air cooling system. Table 6.3's value of 0.9 is likely somewhat conservative, but is obtained quickly for design purposes.

Next, the uncorrected outdoor air intake flow rate V_{ou} is needed; *Standard* 62.1's Equation (6-6) includes diversity factor D to adjust the people component of the flow rate. All zones' flow rates are needed to perform this calculation. For this example, the uncorrected outdoor air intake flow rate for the first floor's AHU was estimated from floor area, an occupancy of 5 people per 1000 ft^2, and

20 cfm per person, and is assumed to be 1525 cfm. The adjusted outdoor air intake flow rate V_{ot} for this AHU is then

$$V_{ot} = V_{ou}/E_v = 1525 \text{ cfm}/0.9 = 1700 \text{ cfm}_{oa}$$

After load calculations are complete, these assumed airflow rates can be replaced with actual values for each zone, and the outside airflow rate can be updated. Repeating the load calculations may be necessary. The final value of the adjusted outdoor air intake flow rate is then reported on the AHU's schedule so that testing, adjusting, and balancing (TAB) personnel and others can use this information to ensure that the system admits the desired flow rate of ventilation air. The information is also used to select air cleaners, dampers, coils, ducts, and fans.

For more examples on determining ventilation air rates for commercial buildings, see the user's manual for *Standard* 62.1 (ASHRAE 2007). For low-rise residential buildings, consult ASHRAE *Standard* 62.2 and its user's manual (ASHRAE 2006).

SYMBOLS

A = area, ft^2 or in^2
c = flow coefficient, cfm/(in. of water)^n
c_p = specific heat, Btu/lb·°F
C = concentration, ppm
$\overline{C}$ = time averaged concentration
C_A = airflow coefficient for automatic doors, cfm/[ft^2·(in. of water)$^{0.5}$]
C_D = discharge coefficient
C_p = pressure coefficient
C_s = stack flow coefficient, cfm^2/(in^4·°F) or (in. of water/°F)^n
C_v = effectiveness of openings
C_w = wind flow coefficient, cfm^2/(in^4·mph^2) or (in. of water/mph^2)^n
E = system efficiency
F = tracer gas injection rate, cfm
$\overline{F}$ = time-averaged contaminant source strength, cfm
f = fractional on-time
g = gravitational acceleration, ft/s^2
G = wind speed multiplier, Table 7
h = specific enthalpy, Btu/lb$_m$
H = height, ft
i = hour of year
I = air exchange rate, 1/time
I_i = instantaneous air exchange rate, 1/time
I_m = effective air exchange rate, 1/time
IDD = infiltration degree-days, °F
n = pressure exponent
N = number of discrete time periods in period of interest
p = pressure, in. of water
P = parameter, or occupancy population
q = heat rate, Btu/h
Q = volumetric flow rate, cfm
$\overline{Q}$ = effective volumetric flow rate, cfm
R = outdoor airflow rate, cfm
s = shelter factor
S = source strength, cfm
t = time [Equations (9) to (11)]; temperature, °F or °R
U = wind speed, mph
V = volume, ft^3, or ventilation airflow rate, cfm
W = humidity ratio, lb$_m$ water/lb$_m$ dry air
ε_I = air change effectiveness
θ = time
θ_{age} = age of air
ρ = air density, lb$_m$/ft^3
τ = time constant
ϕ = wind angle, degrees

Subscripts

a = area
b = base
ba = bypass air
bz = breathing zone
c = calculated
ca = recirculated air
e = effective
ea = exhaust air
f = floor
i = indoor or time counter for summation (instantaneous)
inf = infiltration
H = building height, eaves or roof
ka = makeup air
l = latent
la = relief air
L = leakage or local
ma = mixed air
met = meteorological station location
n = normalized
N = nominal
NPL = neutral pressure level
o = outdoor, initial condition, or reference
oa = outdoor air
ot = adjusted outdoor air
ou = uncorrected outdoor air
oz = zone outdoor
p = pressure, or primary
r = reference
s = sensible or stack
sa = supply air
S = space or source
w = wind
v = ventilation
z = zone

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