

VENTILATION AND INFILTRATION

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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 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. Choosing appropriate ventilation and infiltration rates to solve thermal comfort problems and to reduce energy consumption can affect indoor air quality and may be against code, so such procedures 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*. This standard is reviewed regularly and contains ventilation design and evaluation requirements for commercial and residential buildings. In the design of a new building or the 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 it may be grandfathered under 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* has a list of current standards.

This chapter focuses on commercial and institutional buildings, where ventilation concerns usually dominate, and on single- and multifamily residences, where infiltration is important. The 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 ASHRAE Handbook—Applications](#). However, many of the fundamental ideas and terminology covered in this chapter can also be applied to industrial 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 the 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 8](#) of this volume also addresses thermal comfort. Additionally, the flow of air into buildings and between zones will affect fires and the movement of smoke. Smoke management is addressed in [Chapter 52 of the ASHRAE Handbook—Applications](#).

Ventilation and Infiltration

Air exchange of outdoor air with the air already in a building can be divided into two broad classifications: ventilation and infiltration. **Ventilation air** is air used to provide acceptable indoor air quality. It may be composed of forced or natural ventilation, infiltration, suitably treated recirculated air, transfer air, or an appropriate combination (ASHRAE *Standard 62*). **Ventilation** includes the intentional introduction of air from the outside into a building; it is further subdivided into natural ventilation and forced 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. **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; it is also called **mechanical ventilation**.

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 the 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.

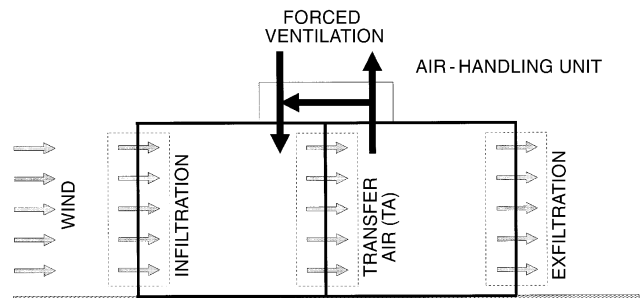


Fig. 1 Two-Space Building with Forced Ventilation, Infiltration, and Exfiltration

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

These modes of air exchange 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 for the proper design and operation of an HVAC system.

Modern commercial and institutional buildings are normally required to have forced ventilation and are usually pressurized somewhat to reduce or eliminate infiltration. Forced 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 62* and *Standard 55* requirements are followed. Forced ventilation equipment and systems are described in [Chapters 1, 2, and 9 of the ASHRAE Handbook—Systems and Equipment](#).

In commercial and institutional buildings, natural ventilation, such as 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 forced ventilation, an air- or water-side economizer cycle 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 typically rely on infiltration and natural ventilation to meet their ventilation needs. Neither is reliable for ventilation 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 may 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 forced ventilation in residences, and heat recovery heat exchangers are popular for reducing the energy impact. Residential buildings with low rates of infiltration and natural ventilation require forced ventilation at rates given in ASHRAE *Standard 62*.

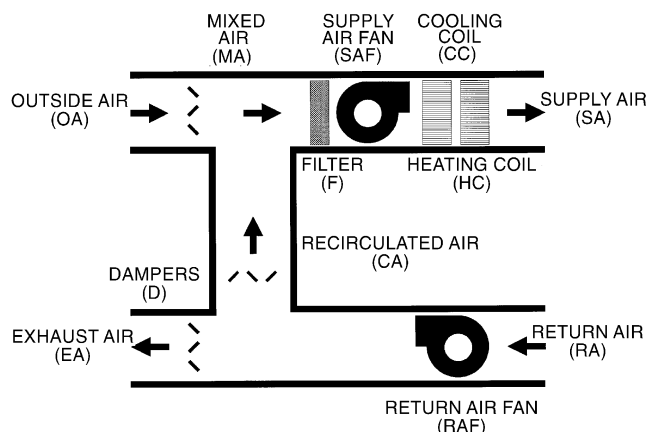


Fig. 2 Simple All-Air Air-Handling Unit with Associated Airflows

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 portion of the return air that is discharged to the environment is **exhaust air** (ea), and the part of the return air that is reused is **recirculated air** (ca). Air brought in intentionally from the environment is **outdoor** or **outside air** (oa). Because outside air may need treatment to be acceptable for use in a building, it should not be called “fresh air.” The outside air and the 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). Due to 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 being 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:

$$X_{oa} = \frac{Q_{oa}}{Q_{sa}} = \frac{Q_{oa}}{Q_{ma}} = \frac{Q_{oa}}{Q_{oa} + Q_{ca}} \quad (1)$$

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 through evaluating the requirements of ASHRAE *Standard 62*. 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), or exhausted by separate 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).

When outside air via forced ventilation is used to provide ventilation air (as is common in commercial and institutional buildings), this outside air is 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 the flow rate of the supply air is reduced to meet a particular thermal load.

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. No mixing of the room air occurs in ideal displacement flow, 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 within the room has been called short-circuiting flow

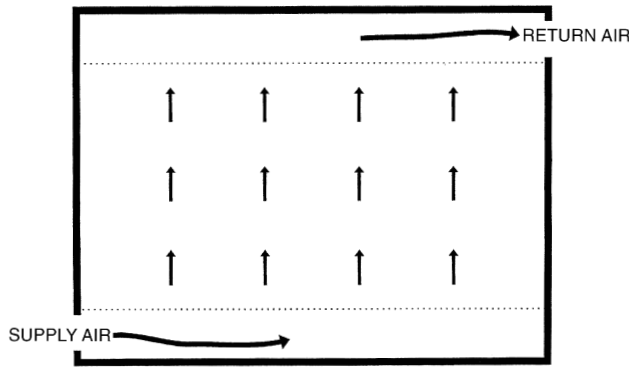


Fig. 3 Displacement Flow Within a Space

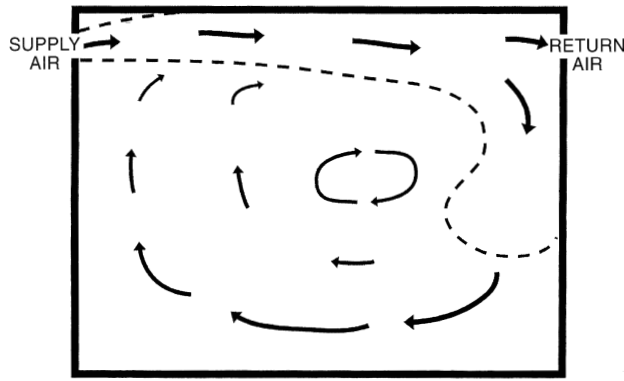


Fig. 4 Entrainment Flow Within a Space

because much of the supply air leaves the room without mixing with the room air. There is little evidence that properly designed, installed, and operated air distribution systems exhibit short-circuiting behavior. There is some evidence that poorly designed, installed, or operated systems can exhibit short-circuiting behavior, especially ceiling-based systems in the 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 **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 2 of ASHRAE *Standard* 62 assume delivery of ventilation air with perfect mixing within spaces. For more detailed information on space air diffusion, see [Chapter 32](#).

The 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 air and secondary air at a specific point in a jet. The term primary air is also used to describe the supply air provided to fan-powered mixing boxes by a central air-handling unit.

Air Exchange Rate

The **air exchange rate** I compares airflow to volume and is

$$I = \frac{Q}{V} \quad (2)$$

where

Q = volumetric flow rate of air into space, m^3/s
 V = interior volume of space, m^3

The air exchange rate has units of 1/time. 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 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

$$I_N = \frac{Q_{oa}}{V} \quad (3)$$

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:

$$I_S = \frac{Q_{sa}}{V} \quad (4)$$

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:

$$\tau = \frac{1}{I} = \frac{V}{Q} \quad (5)$$

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

$$\tau_N = \frac{V}{Q_{oa}} \quad (6)$$

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:

$$\tau_S = \frac{V}{Q_{sa}} \quad (7)$$

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, and Yuill 1991) describes the

proper ventilation rate averaging process. In this concept, the average (effective) rate is the steady-state rate that would yield 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 due to the nonlinear response of indoor concentrations to the 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

$$I_m = \frac{\bar{Q}}{V} = \frac{F}{V\bar{C}} = \frac{1}{\bar{\tau}_e} \quad (8)$$

The time-averaged effective turnover time $\bar{\tau}_e$ 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):

$$\bar{\tau}_e = \frac{1}{N} \sum_{i=1}^N \tau_{e,i} \quad (9)$$

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

$$\text{for } I_i = 0, \tau_{e,i} = \Delta t + \tau_{e,i-1} \quad (11)$$

where

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

ASHRAE *Standard* 136 provides a set of factors to assist in calculating the annual effective air exchange rate for houses. These factors were determined using Equations (9) through (11).

Age of Air

The **age of air** θ_{age} (Sandberg 1981) is the length of time 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 forced 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, a longer age of air suggests poorer outside air delivery compared to a shortage of air for the same location. The age of

air has units 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 the 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. Once 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} = 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 does not have knowledge or control of actual pollutant sources within buildings, so Table 2 of ASHRAE *Standard* 62 defines outdoor air requirements for typical, expected building uses. For most projects, therefore, the air change effectiveness is of more relevance to HVAC system design than the ventilation effectiveness. Various definitions for air change effectiveness have been proposed. The specific measure that meets the 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:

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

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

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

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:

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

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).

Currently, the HVAC design engineer must assume that a properly designed, installed, operated, and maintained air distribution

system provides an air change effectiveness of about 1. Therefore, the Table 2 values of ASHRAE *Standard* 62 are appropriate for the design of commercial, institutional, and residential buildings when the Ventilation Rate Procedure is used. If the Indoor Air Quality Procedure of *Standard* 62 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.

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 due to temperature differences between indoor and outdoor air (buoyancy, or the stack effect). Mechanical air-moving systems also induce pressure differences across the envelope due to the 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 by [Chapter 6](#). As a result, standard conditions should not be used to calculate the density. For example, a building site at 1500 m has an air density that is about 20% less than if the building were at sea level. An air temperature increase from -30 to 20°C causes a similar air density difference. Combined, these elevation and temperature effects reduce the air density about 45%. Moisture effects on density are generally negligible, so the dry air density can be used instead, except in hot, humid climates when the air is hot and close to saturation. For example, saturated air at 40°C has a density about 5% greater 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

$$p_s = p_r - \rho g H \quad (16)$$

where

- p_s = stack pressure, Pa
- p_r = stack pressure at reference height, Pa
- g = gravitational constant, 9.81 m/s^2
- ρ = indoor or outdoor air density, kg/m^3
- H = height above reference plane, m

For tall buildings or when significant temperature stratification occurs indoors, Equation (16) 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. 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. The building is then characterized by an effective stack height and neutral pressure level (NPL) or leakage distribution (see 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

$$\begin{aligned} \Delta p_s &= (\rho_o - \rho_i)g(H_{\text{NPL}} - H) \\ &= \rho_o \left(\frac{T_o - T_i}{T_i} \right) g(H_{\text{NPL}} - H) \end{aligned} \quad (17)$$

where

- T_o = outdoor temperature, K
- T_i = indoor temperature, K
- ρ_o = outdoor air density, kg/m^3
- ρ_i = indoor air density, kg/m^3
- H_{NPL} = height of neutral pressure level above reference plane without any other driving forces, m

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 (17) 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, in the absence of other driving forces and assuming no stack effect is within the flow elements themselves, when the indoor air is warmer than outdoors, the base of the building is depressurized and the top is pressurized relative to outdoors; when the indoor air is cooler than outdoors, the reverse is true.

In the absence of 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 nor is it necessarily unique. 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 a vertical separation between its inlet and outlet and (2) the air inside the flow element is not at the indoor or outdoor temperature, such as in a chimney, more complex analyses than Equation (17) 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 pressure of the undisturbed airstream. The wind pressure or velocity pressure is given by the Bernoulli equation, assuming no height change or pressure losses:

$$p_w = C_p \rho \frac{U^2}{2} \quad (18)$$

where

- p_w = wind surface pressure relative to outdoor static pressure in undisturbed flow, Pa
- ρ = outside air density, kg/m^3 (about 1.2)
- U = wind speed, m/s
- C_p = wind surface pressure coefficient, dimensionless

C_p is a function of location on the building envelope and wind direction. [Chapter 16](#) provides additional information on the 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. A harmonic trigonometric function was developed by Walker and Wilson (1994) 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

$$C_p(\phi) = \frac{1}{2} \{ [C_p(1) + C_p(2)](\cos^2\phi)^{1/4} + [C_p(1) - C_p(2)](\cos\phi)^{3/4} + [C_p(3) + C_p(4)](\sin^2\phi)^2 + [C_p(3) - C_p(4)]\sin\phi \} \quad (19)$$

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

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 the geometry effects on flow around a building, the application of this interpolation function is limited to low-rise buildings that are 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 16](#) 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 the wind speed measured at the local weather station, typically the nearest airport. This wind speed needs to be corrected for reductions due to the difference between the height where the wind speed is measured and the height of the building and reductions due to 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 10 m) and at a different location. The difference in terrain between the measurement station and the building under study must also be accounted for. [Chapter 16](#) 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 due to the reduction in 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 (20) to find the wind shelter for intermediate wind angles:

$$s = \frac{1}{2} \left\{ \begin{aligned} &[s(1) + s(2)]\cos^2\phi + [s(1) - s(2)]\cos\phi \\ &+ [s(3) + s(4)]\sin^2\phi + [s(3) - s(4)]\sin\phi \end{aligned} \right\} \quad (20)$$

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 the above method gives a realistic variation of wind shelter effects with wind direction, estimates for the numerical values of wind shelter factor s for each of the four cardinal directions must be provided. [Table 11](#) in the section on Residential Calculation Examples lists typical shelter factors. The wind speed used in Equation (18) is then given by

$$U = sU_H \quad (21)$$

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

Mechanical Systems

The operation of mechanical equipment, such as supply or exhaust systems and vented combustion devices, affects pressure differences across the building envelope. The interior static pressure adjusts such that the sum of all airflows through the 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 (Tamura and Wilson 1966, 1967b; Persily and Grot 1985a).

Air exhausted from a building by a whole-building exhaust system must be balanced by increasing the airflow into the building through other openings. As a result, the airflow at some locations changes from outflow to inflow. For supply fans, the situation is reversed and envelope inflows become outflows. Thus, the effects a mechanical system has 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 the 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. A similar phenomenon, but in reverse, can occur during the heating season in cold climates if the building is depressurized.

The interaction between mechanical systems and the building envelope also pertains to systems serving zones of buildings. The performance of zone-specific exhaust or pressurization systems is affected by the 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 register. 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

with the return duct tends to depressurize relative to outside. This is caused by the resistance of internal door undercuts 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 3 to 6 Pa (Modera et al. 1991).

Building envelope airtightness and interzonal airflow resistance can also affect the 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 34](#) of this volume and [Chapter 18 of the ASHRAE Handbook—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 will tend to depressurize the building; return leaks to the outside will tend to pressurize it.

Combining Driving Forces

The pressure differences due to wind pressure, stack pressure, and mechanical systems are considered in combination by adding them together and then determining the airflow rate through each opening due to this total pressure difference. The air flows 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:

$$P_U = \rho_o \frac{U_H^2}{2} \quad (22)$$

$$P_T = g \rho_o \left(\frac{T_o - T_i}{T_i} \right) \quad (23)$$

where T = air temperature, K.

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

$$\Delta p = s^2 C_p P_U + H P_T + \Delta p_I \quad (24)$$

where Δp_I = pressure that acts to balance inflows and outflows (including mechanical system flows). Equation (24) can then be applied to every leak for the building with the 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 pressure difference. Internal partitions, stairwells, elevator shafts, utility ducts, chimneys, vents, operable windows, and mechanical supply and exhaust systems complicate the analysis of NPL location. An opening with a large area relative to the total building leakage causes the NPL to shift toward the location of 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 5](#) 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 5A](#), 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 5B](#) presents qualitative uniform pressure differences caused by wind alone, with opposing effects on the windward and leeward sides. When the temperature difference and wind effects both exist, the pressures due to each are added together to determine the total pressure difference across the building envelope. In [Figure 5B](#), there is no NPL because no locations on the building envelope have zero pressure difference. [Figure 5C](#) shows the combination, where the wind force of [Figure 5B](#) has just balanced the thermal force of [Figure 5A](#), causing no pressure difference at the top windward or bottom leeward side.

The relative importance of the wind and stack pressures in a building depends on building height, internal resistance to vertical airflow, location and flow resistance characteristics of envelope

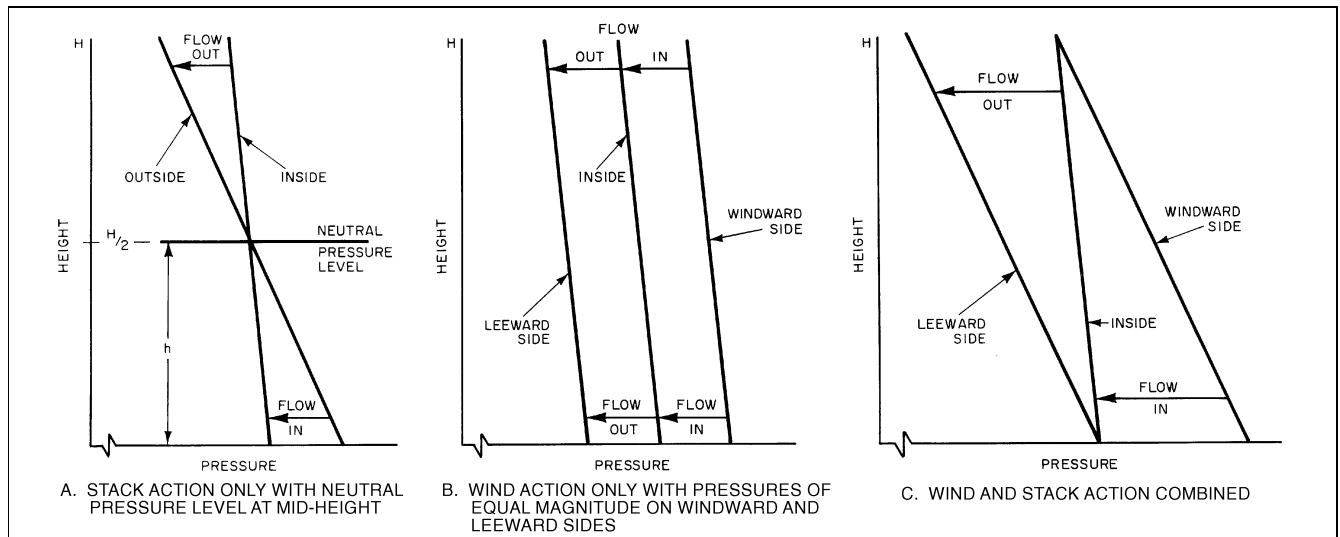


Fig. 5 Distribution of Inside and Outside Pressures over Height of Building

openings, local terrain, and the immediate shielding of the building. The taller the building is and the smaller its internal resistance to airflow, the stronger the stack effect will be. The more exposed a building is, the more susceptible it will be to wind. For any building, there will be ranges of wind speed and temperature difference for which the building's infiltration is dominated by the stack effect, the wind, or the driving pressures of both (Sinden 1978a). 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 the 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 the internal airflow resistance. Pressurizing all levels uniformly has little effect on the pressure differences across floors and vertical shaft enclosures, but pressurizing individual stories increases the pressure drop across these internal separations. Pressurization of the ground level is often used in tall buildings to reduce the stack 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, 1967a). 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 (17) provides a maximum stack pressure difference, given no internal airflow resistance. The sum of the pressure differences across the exterior wall at the bottom and at the top of the building, as calculated by these equations, equals the total theoretical draft for the building. The sum of the 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 the 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 6A), and the thermal draft coefficient equals 1. In a building with airtight separations at each floor, each story acts independently, its own stack effect being unaffected by that of any other floor (Figure 6B). The theoretical draft is minimized in this case, and each story has an NPL.

Real multistory buildings are neither open inside (Figure 6A), nor airtight between stories (Figure 6B). Vertical air passages, stairwells, elevators, and other service shafts allow airflow between floors. Figure 6C 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 6A, and the discontinuity at each floor (Figure 6B) 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 means of controlling indoor-outdoor pressure differences due to the stack effect and therefore infiltration (Lovaett and Wilson 1994). Good separation is also conducive to the proper operation of mechanical ventilation and smoke management systems. However, care is needed to avoid pressure differences that could prevent door opening in an emergency. Tamura and Wilson (1967b) showed that when vertical shaft

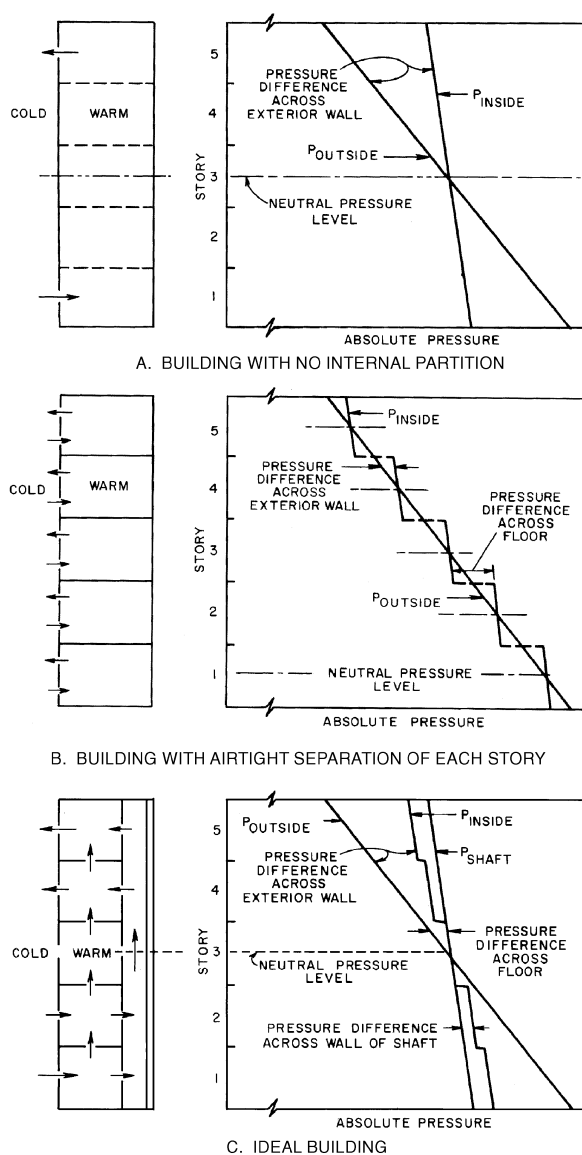


Fig. 6 Compartmentation Effect in Buildings

leakage is at least two times the 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 (1967a) indicated that the thermal draft coefficient ranged from 0.8 to 0.9 with the ventilation systems off.

ASHRAE Standard 62

ASHRAE Standard 62 provides guidance on ventilation and indoor air quality and includes two alternative procedures for determining design ventilation rates. In the **Ventilation Rate Procedure**, indoor air quality is assumed to be acceptable if (1) the concentrations of six pollutants in the incoming outdoor air meet the U.S. EPA national ambient air quality standards (EPA 1986), and (2) the outdoor air ventilation rates meet or exceed values (which depend on the space type) provided in a table.

The minimum outside air ventilation per person for any type of space is 8 L/s. This minimum rate will maintain the indoor CO_2 concentration within 0.07% (700 parts per million) of the outdoor concentration, assuming a typical CO_2 generation rate per occupant

(Janssen 1989). This minimum outside air ventilation rate was based, in part, on research by Berg-Munch et al. (1986), Yaglou et al. (1936), Iwashita et al. (1989), and Cain et al. (1993) that indicated that 8 L/s was required to satisfy the odor perceptions of 80% or more of visitors.

The other design approach in ASHRAE *Standard 62* is the **Indoor Air Quality Procedure**. In this procedure, acceptable indoor air quality is achieved through the control of indoor contaminant concentrations. Such control can be realized through source control, air cleaning, and ventilation. It allows for either or both improved indoor air quality and reduced energy consumption. [Chapter 24 of the ASHRAE Handbook—Systems and Equipment](#) has information on air cleaning.

A combination of source control and local exhaust, as opposed to dilution with ventilation air, is the method of choice in industrial environments. Industrial ventilation is discussed in [Chapters 29 and 30 of the ASHRAE Handbook—Applications](#) and in *Industrial Ventilation: A Manual of Recommended Practice* (ACGIH 1998).

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 (CO₂), 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, the maintenance of acceptable indoor concentrations of a variety of additional pollutants that are not generated primarily by occupants has been a major concern. Information on contaminants can be found in [Chapter 12](#), and odors are covered in [Chapter 13](#).

Indoor pollutant concentrations depend on the strength of pollutant sources and the total rate of pollutant removal. Pollutant sources include the outdoor air; indoor sources such as occupants, furnishings, and appliances; and the soil adjacent to the building. 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) the indoor air is thoroughly mixed, and (3) the pollutant source strength and ventilation rate have been stable for a sufficient period; then the steady-state indoor pollutant concentration is given by

$$C_i = C_o + S/Q_{oa} \quad (25)$$

where

- C_i = steady-state indoor concentration, $\mu\text{g}/\text{m}^3$
- C_o = outdoor concentration, $\mu\text{g}/\text{m}^3$
- S = total pollutant source strength, $\mu\text{g}/\text{s}$
- Q_{oa} = ventilation rate, m^3/s

Variation in pollutant source strengths (rather than variation in ventilation rate) is considered the largest cause of building-to-building variation in the 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, maintenance of 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 due to the lack of health data, makes the specification of minimum ventilation rates even more difficult.

In cases of high contaminant source strengths, impractically high rates of ventilation are required to control contaminant levels, and other methods of control are more effective. Removal or reduction of contaminant sources is a very effective means of control. Controlling a localized source by means of local exhaust, such as range hoods or bathroom exhaust fans, can also be effective.

Particles can be removed with various types of air filters. Gaseous contaminants with higher relative molecular mass can be controlled with activated carbon or alumina pellets impregnated with a substance such as potassium permanganate. [Chapter 24 of the ASHRAE Handbook—Systems and Equipment](#) has information on air cleaning.

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. [Chapter 28](#) and [Chapter 29](#) cover thermal loads in more detail.

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

$$q_s = Q\rho c_p \Delta t = 1200Q\Delta t \quad (26)$$

where

- q_s = sensible heat load, W
- Q = airflow rate, m^3/s
- ρ = air density, kg/m^3 (about 1.2)
- c_p = specific heat of air, $\text{J}/(\text{kg}\cdot\text{K})$ (about 1000)
- Δt = temperature difference between indoors and outdoors, K

Second, air exchange 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

$$q_l = Q\Delta W[4775 + 1.998\Delta t] \quad (27)$$

where

- q_l = latent heat load, W
- ΔW = humidity ratio difference between indoors and outdoors, mass water/unit mass dry air (kg/kg)

Finally, air exchange can change a building's thermal load by altering the performance of the envelope insulation system. Airflow through the insulation can decrease the thermal load due to heat exchange between infiltrating or exfiltrating air and the thermal 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, a number of researchers (Wolf 1966; Berlad et al. 1978; Bankvall 1987; Lecompte 1987) have 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 insu-

lation system can also decrease the system's performance due to moisture condensation in and on the insulation.

Even if air flows only through cracks instead of through the insulation, the actual heating/cooling load due to the combined effect of conduction and airflow heat transfer can be lower than the heating/cooling load calculated by Equation (26). 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)**. This effect appears to be significant in building envelopes, based on preliminary laboratory and numerical work under controllable conditions by several investigators.

Using a computer simulation, Kohonen et al. (1987) found that the conduction/infiltration thermal interaction reduced total heating load by 15%. Several experimental studies by Claridge et al. (1988), Claridge and Bhattacharyya (1990), Liu and Claridge (1992a, 1992b, 1992c, 1995), Timusk et al. (1992), and others 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 (27). Judkoff et al. (1997) measured heat recovery using a mobile home under steady-state conditions. They found that up to 40% heat recovery occurs during exfiltration through the envelope of the mobile home. Buchanan and Sherman (2000) carried out two- and three-dimensional computational fluid dynamics 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.

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 due to 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 due to 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).

Natural Ventilation Openings

Natural ventilation openings include (1) windows, doors, dormer (monitor) openings, and skylights; (2) roof ventilators; (3) stacks connecting to registers; 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 ventilators can have manual dampers or dampers controlled by thermostat or wind velocity.

A natural-draft roof ventilator should be positioned so that it receives the full, unrestricted wind. Turbulence created by surrounding obstructions, including higher adjacent buildings, impairs a ventilator's ejector action. The ventilator inlet should be conical or bell mounted to give a high flow coefficient. The opening area at the 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.

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 (26) and (27) 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

$$Q = C_D A \sqrt{2\Delta p / \rho} \quad (28)$$

where

Q = airflow rate, m³/s

C_D = discharge coefficient for opening, dimensionless

A = cross-sectional area of opening, m²

ρ = air density, kg/m³

Δp = pressure difference across opening, Pa

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

Factors due to wind forces 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. The following equation shows the rate of air forced

through ventilation inlet openings by wind or determines the proper size of openings to produce given airflow rates:

$$Q = C_v A U \quad (29)$$

where

- Q = airflow rate, m^3/s
 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, m^2
 U = wind speed, m/s

Inlets should face directly into the prevailing wind. If they are not advantageously placed, flow will be less than that predicted by Equation (29); if the 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 16 gives a general description of the wind pressure distribution on a building. The inlets should be placed in the exterior high-pressure regions; the outlets should be placed in the exterior low-pressure regions.

Flow Caused by Thermal Forces Only

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

$$Q = C_D A \sqrt{2g \Delta H_{\text{NPL}} (T_i - T_o) / T_i} \quad (30)$$

where

- Q = airflow rate, m^3/s
 C_D = discharge coefficient for opening
 ΔH_{NPL} = height from midpoint of lower opening to NPL, m
 T_i = indoor temperature, K
 T_o = outdoor temperature, K

Equation (30) 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.

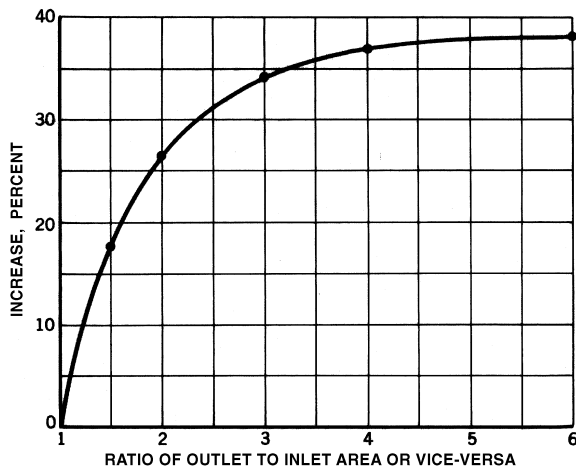


Fig. 7 Increase in Flow Caused by Excess Area of One Opening over the Other

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):

$$C_D = 0.40 + 0.0045 |T_i - T_o| \quad (31)$$

If enough other openings are available, the 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 (30) and (31) 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 (30) and add the increase as determined from Figure 7.

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) or other design considerations.

1. In hot, humid climates, use mechanical cooling. If mechanical cooling is not available, air velocities should be maximized in the occupied zones. In hot, arid climates, consider evaporative cooling. Airflow throughout the building should be maximized for structural cooling, particularly at night when the temperature is low.
2. 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.
3. The building should be shaped to expose maximum shell openings to breezes.
4. Architectural elements such as wing walls, parapets, and overhangs should be used to promote airflow into the building interior.
5. The long facade of the building and the majority of the door and window openings should be oriented with respect to the prevailing summer breezes. If there is no prevailing direction, openings should be sufficient to provide ventilation regardless of wind direction.
6. Windows should be located in opposing pressure zones. Two openings on opposite sides of a space increase the 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.
7. If a room has only one external wall, better airflow is achieved with two widely spaced windows.
8. If the 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.
9. Vertical distance between openings is required to take advantage of the stack effect; the greater the vertical distance, the greater the ventilation.

10. 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 the pressure across the opening.
11. 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.
12. Openings with areas much larger than calculated are sometimes desirable when anticipating increased occupancy or very hot weather.
13. 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.
14. Window openings should be accessible to and operable by occupants.
15. 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.
16. 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.

RESIDENTIAL AIR LEAKAGE

Most infiltration in residential buildings in the U.S. is dominated by envelope leakage. However, trends in new construction are towards tighter envelopes such that envelope leakage is reduced in newer housing.

Envelope Leakage Measurement

Envelope leakage of a building 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.

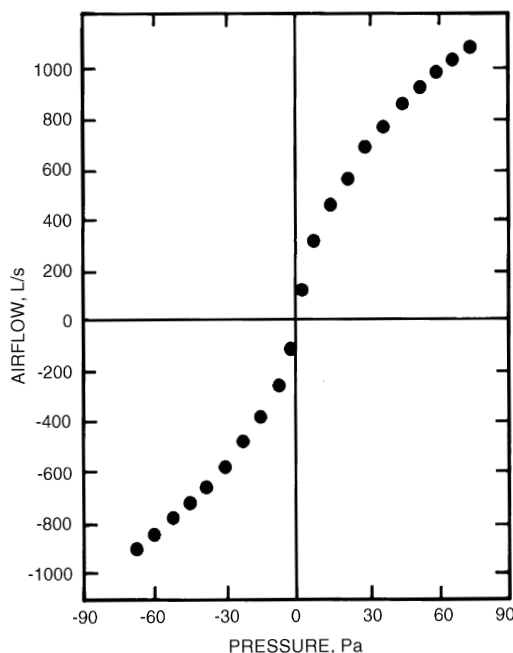


Fig. 8 Airflow Rate Versus Pressure Difference Data from Whole-House Pressurization Test

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 *Standard* E 779, ASTM *Standard* E 1827, ISO *Standard* 9972, and CGSB *Standard* 149.10). 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 10 Pa to 75 Pa.

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 8. 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 Hopkins and Hansford (1974), Etheridge (1977), Kronvall (1980), Chastain et al. (1987), and Walker et al. (1997).

The 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 (32), commonly called the **power law equation**:

$$Q = c(\Delta p)^n \quad (32)$$

where

c = flow coefficient, $\text{m}^3/(\text{s} \cdot \text{Pa}^n)$

n = pressure exponent, dimensionless

Sherman (1992b) showed how the power law can be developed analytically by looking at developing laminar flow in short pipes. Equation (32) only approximates the relationship between Q and Δp . Measurements of single cracks (Honma 1975, Krieth et al. 1957) have shown 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) further 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 using fan pressurization techniques.

Airtightness Ratings

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

$$A_L = 10\,000 Q_r \frac{\sqrt{\rho/2\Delta p_r}}{C_D} \quad (33)$$

where

A_L = equivalent or effective air leakage area, cm^2

Q_r = predicted airflow rate at Δp_r (from curve fit to pressurization test data), m^3/s

ρ = air density, kg/m^3

Δp_r = reference pressure difference, Pa

C_D = discharge coefficient

All the 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 being used. Reference pressure differences include 4, 10, 25, 50, and 75 Pa. Reference pressure differences of 4 and 10 Pa are advocated because they are closer to the pressure differences that actually induce air exchange and, therefore, better model the flow characteristics of the openings. While this may be true, they are outside the range of measured values in the test; therefore, the predicted airflow rates at 4 and 10 Pa are subject to significant uncertainty. The uncertainty in these predicted airflow rates and the implications for quantifying airtightness are discussed in Persily and Grot (1985b), Chastain (1987), and Modera and Wilson (1990). 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 4 Pa, with errors as great as $\pm 40\%$, mainly due to the fan calibration errors at low flow.

Some common airtightness ratings include the effective air leakage area at 4 Pa assuming $C_D = 1.0$ (Sherman and Grimsrud 1980); the equivalent air leakage area at 10 Pa assuming $C_D = 0.611$ (CGSB *Standard* 149.10); and the airflow rate at 50 Pa, divided by the building volume to give units of air changes per hour (Blomsterberg and Harrie 1979).

Conversion Between Ratings

Air leakage areas at one reference pressure difference can be converted to air leakage areas at an other reference pressure difference according to:

$$A_{r,2} = A_{r,1} \left(\frac{C_{D,1}}{C_{D,2}} \right) \left(\frac{\Delta p_{r,2}}{\Delta p_{r,1}} \right)^{n-0.5} \quad (34)$$

where

- $A_{r,1}$ = air leakage area at reference pressure difference $\Delta p_{r,1}$, cm^2
- $A_{r,2}$ = air leakage area at reference pressure difference $\Delta p_{r,2}$, cm^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 (32)

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

$$Q_{r,2} = \frac{C_{D,1} A_{r,1}}{10\,000} \sqrt{\frac{2}{\rho}} (\Delta p_{r,1})^{0.5-n} (\Delta p_{r,2})^n \quad (35)$$

where $Q_{r,2}$ = airflow rate at reference difference $\Delta p_{r,2}$, m^3/s .

The flow coefficient c in Equation (32) may be converted to an air leakage area according to

$$A_L = \frac{10\,000 c}{C_D} \sqrt{\frac{2}{\rho}} \Delta p_r^{(n-0.5)} \quad (36)$$

Finally, an air leakage area may be converted to the flow coefficient c in Equation (32) according to

$$c = \frac{C_D A_L}{10\,000} \sqrt{\frac{2}{\rho}} (\Delta p_r)^{0.5-n} \quad (37)$$

Equations (34) through (37) require the assumption of a value of n , unless it is reported with the measurement results. When whole-building pressurization test data are fitted to Equation (32), 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 the wind and temperature differences during the measurement period are sufficiently mild, the pressure differences they induce during the test will interfere with the test pressures and cause measurement errors. Persily (1982) and Modera and Wilson (1990) studied the effects of wind speed on pressurization test results. Several experimental studies have also shown variations on the order of 20 to 40% over a year in the measured airtightness in homes (Persily 1982, Kim and Shaw 1986, Warren and Webb 1986).

Figure 9 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 :

$$A_n = 0.1 \left(\frac{A_L}{A_f} \right) \left(\frac{H}{H_o} \right)^{0.3} \quad (38)$$

where

- A_n = normalized leakage area, dimensionless
- A_L = effective leakage area at 4 Pa ($C_D = 1.0$), cm^2
- A_f = gross floor area (within exterior walls), m^2
- H = building height, m
- H_o = reference height of one-story building = 2.5 m

Air Leakage of Building Components

The fan pressurization procedure discussed in the section on Envelope Leakage Measurement enables the measurement of whole-building air leakage. 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 heat and moisture transfer characteristics of the envelope. Additional test procedures exist for pressure-testing individual building components such as windows, walls, and doors; they are discussed in ASTM *Standards* E 283 and E 783 for laboratory and field tests, respectively.

Leakage Distribution

Dickerhoff et al. (1982) and Harrie and Born (1982) studied the 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. The 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 between the sill plate and the foundation, cracks below the bottom of the gypsum wallboard, electrical outlets, plumbing penetrations, and leaks into the attic at the top plates of walls all occur.

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 infil-

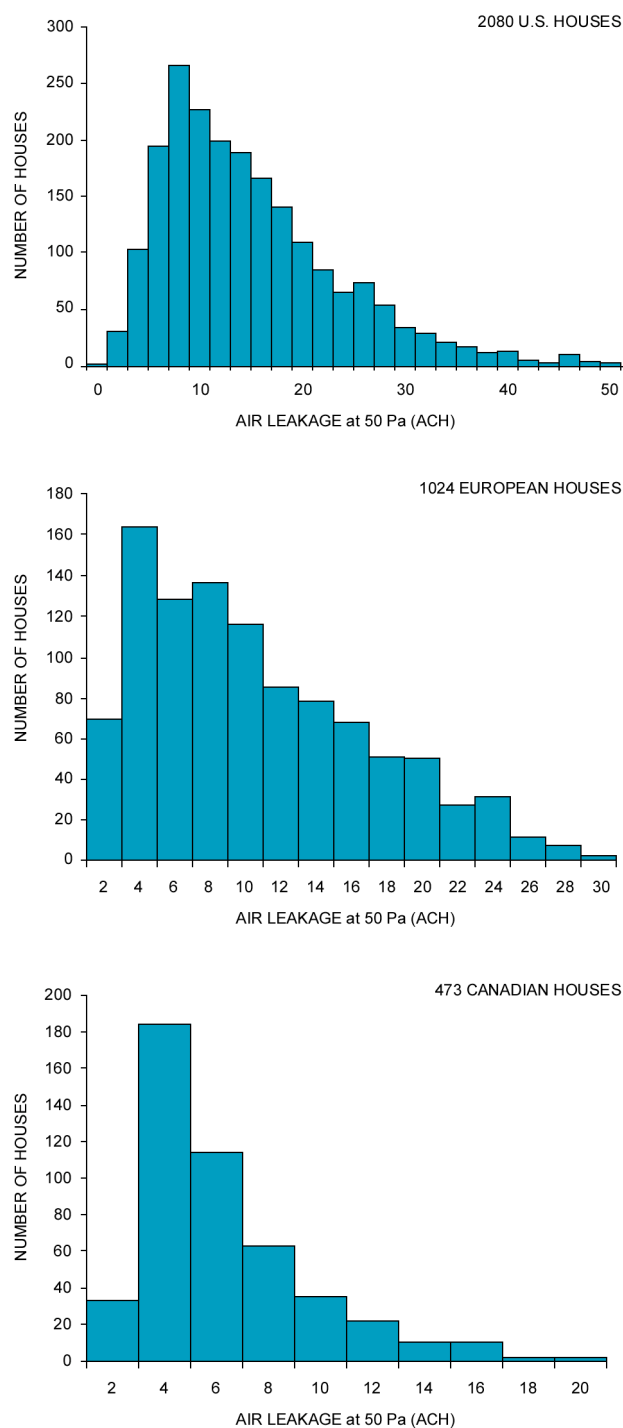


Fig. 9 Envelope Leakage Measurements

tration 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 leakage. Modera et al. (1991) and Robison and Lambert (1989), among others, have shown

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, Robison and Lambert 1989, Jump et al. 1996). The 18% contribution of ducts to total leakage significantly underestimates their impact because during system operation, the pressure differentials across the duct leaks are approximately ten times higher than typical pressure differences across the envelope leaks (Modera 1989, Modera et al. 1991) and result in large (factors of two to three) changes in ventilation rate (Walker 1999, Walker et al. 1999, Cummings et al. 1990).

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%). Diffusion, in comparison to infiltration through holes and other openings in the structure, is not an important flow mechanism. At 5 Pa, 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. Table 1 shows effective air leakage areas for a variety of residential building components at 4 Pa with C_D assumed equal to 1 (Colliver et al. 1992). The values in the table present results in terms of air leakage area per unit component. Per unit component means per component, per unit surface area, or per unit length of crack or sash, whichever is appropriate. These air leakage areas may be converted to air leakage areas at other reference pressures, airflow rates, or flow coefficients using Equations (34) through (37). Table 1 can be used to estimate the air leakage area of the building if test data are not available. To obtain the building's total air leakage area, multiply the overall dimensions or number of occurrences of each building component by the appropriate table entry. The sum of the resulting products is the total building air leakage area. Table 2 gives the result of an example calculation of the effective air leakage area of a residence. Each leakage component is identified in the first column and described in the second. The length, area, or number of the component is in the third column. The fourth column contains the air leakage area per unit component, from Table 1, and the fifth contains the total air leakage area associated with that component. The sum of the terms in the last column is the total air leakage area of the building, in this case 848 cm². Klote and Milke (1992) describe a method for estimating airflows through gaps such as those found around doors.

Multifamily Building Leakage

Leakage distribution is particularly important in multifamily apartment buildings. These buildings often cannot be treated as single zones due to the internal resistance between apartments. Moreover, the leakage between apartments varies widely, tending to be small in modern construction, and ranging as high as 60% of the total apartment leakage in turn-of-the-century brick walk-up apartment buildings (Modera et al. 1991, Diamond et al. 1986). Little information on interzonal leakage has been reported because of the difficulty and expense of these measurements.

Table 1 Effective Air Leakage Areas (Low-Rise Residential Applications Only)

	Units (see note)	Best Estimate	Mini- mum	Maxi- mum		Units (see note)	Best Estimate	Mini- mum	Maxi- mum
Ceiling					Piping/Plumbing/Wiring penetrations				
General	cm ² /m ²	1.8	0.79	2.8	Uncaulked	cm ² ea	6	2	24
Drop	cm ² /m ²	0.19	0.046	0.19	Caulked	cm ² ea	2	1	2
Ceiling penetrations					Vents				
Whole-house fans	cm ² ea	20	1.6	21	Bathroom with damper closed	cm ² ea	10	2.5	20
Recessed lights	cm ² ea	10	1.5	21	Bathroom with damper open	cm ² ea	20	6.1	22
Ceiling/Flue vent	cm ² ea	31	28	31	Dryer with damper	cm ² ea	3	2.9	7
Surface-mounted lights	cm ² ea	0.82			Dryer without damper	cm ² ea	15	12	34
Chimney	cm ² ea	29	21	36	Kitchen with damper open	cm ² ea	40	14	72
Crawl space					Kitchen with damper closed	cm ² ea	5	1	7
General (area for exposed wall)	cm ² /m ²	10	8	17	Kitchen with tight gasket	cm ² ea	1		
200 mm by 400 mm vents	cm ² ea	129			Walls (exterior)				
Door frame					Cast-in-place concrete	cm ² /m ²	0.5	0.049	1.8
General	cm ² ea	12	2.4	25	Clay brick cavity wall, finished	cm ² /m ²	0.68	0.05	2.3
Masonry, not caulked	cm ² /m ²	5	1.7	5	Precast concrete panel	cm ² /m ²	1.2	0.28	1.65
Masonry, caulked	cm ² /m ²	1	0.3	1	Low-density concrete block, unfinished	cm ² /m ²	3.5	1.3	4
Wood, not caulked	cm ² /m ²	1.7	0.6	1.7	Low-density concrete block, painted or stucco	cm ² /m ²	1.1	0.52	1.1
Wood, caulked	cm ² /m ²	0.3	0.1	0.3	High-density concrete block, unfinished	cm ² /m ²	0.25		
Trim	cm ² /lmc	1			Continuous air infiltration barrier	cm ² /m ²	0.15	0.055	0.21
Jamb	cm ² /lmc	8	7	10	Rigid sheathing	cm ² /m ²	0.35	0.29	0.41
Threshold	cm ² /lmc	2	1.2	24	Window framing				
Doors					Masonry, uncaulked	cm ² /m ²	6.5	5.7	10.3
Attic/crawl space, not weatherstripped	cm ² ea	30	10	37	Masonry, caulked	cm ² /m ²	1.3	1.1	2.1
Attic/crawl space, weatherstripped	cm ² ea	18	8	18.5	Wood, uncaulked	cm ² /m ²	1.7	1.5	2.7
Attic fold down, not weatherstripped	cm ² ea	44	23	86	Wood, caulked	cm ² /m ²	0.3	0.3	0.5
Attic fold down, weatherstripped	cm ² ea	22	14	43	Windows				
Attic fold down, with insulated box	cm ² ea	4			Awning, not weatherstripped	cm ² /m ²	1.6	0.8	2.4
Attic from unconditioned garage	cm ² ea	0	0	0	Awning, weatherstripped	cm ² /m ²	0.8	0.4	1.2
Double, not weatherstripped	cm ² /m ²	11	7	22	Casement, weatherstripped	cm ² /lmc	0.24	0.1	3
Double, weatherstripped	cm ² /m ²	8	3	23	Casement, not weatherstripped	cm ² /lmc	0.28		
Elevator (passenger)	cm ² ea	0.26	0.14	0.35	Double horizontal slider, not weatherstripped	cm ² /lmc	1.1	0.019	3.4
General, average	cm ² /lmc	0.31	0.23	0.45	Double horizontal slider, wood, weatherstripped	cm ² /lmc	0.55	0.15	1.72
Interior (pocket, on top floor)	cm ² ea	14			Double horizontal slider, aluminum, weatherstripped	cm ² /lmc	0.72	0.58	0.8
Interior (stairs)	cm ² /lmc	0.9	0.25	1.5	Double-hung, not weatherstripped	cm ² /lmc	2.5	0.86	6.1
Mail slot	cm ² /lmc	4			Double-hung, weatherstripped	cm ² /lmc	0.65	0.2	1.9
Sliding exterior glass patio	cm ² ea	22	3	60	Double-hung with storm, not weatherstripped	cm ² /lmc	0.97	0.48	1.7
Sliding exterior glass patio	cm ² /m ²	5.5	0.6	15	Double-hung with storm, weatherstripped	cm ² /lmc	0.79	0.44	1
Storm (difference between with and without)	cm ² ea	6	3	6.2	Double-hung with pressurized track, weatherstripped	cm ² /lmc	0.48	0.39	0.56
Single, not weatherstripped	cm ² ea	21	12	53	Jalousie	cm ² /louver	3.38		
Single, weatherstripped	cm ² ea	12	4	27	Lumped	cm ² /lms	0.471	0.009	2.06
Vestibule (subtract per each location)	cm ² ea	10			Single horizontal slider, weatherstripped	cm ² /lms	0.67	0.2	2.06
Electrical outlets/Switches					Single horizontal slider, aluminum	cm ² /lms	0.8	0.27	2.06
No gaskets	cm ² ea	2.5	0.5	6.2	Single horizontal slider, wood	cm ² /lms	0.44	0.27	0.99
With gaskets	cm ² ea	0.15	0.08	3.5	Single horizontal slider, wood clad	cm ² /lms	0.64	0.54	0.81
Furnace					Single-hung, weatherstripped	cm ² /lms	0.87	0.62	1.24
Sealed (or no) combustion	cm ² ea	0	0	0	Sill	cm ² /lmc	0.21	0.139	0.212
Retention head or stack damper	cm ² ea	30	20	30	Storm inside, heat shrink	cm ² /lms	0.018	0.009	0.018
Retention head and stack damper	cm ² ea	24	18	30	Storm inside, rigid sheet with magnetic seal	cm ² /lms	0.12	0.018	0.24
Floors over crawl spaces					Storm inside, flexible sheet with mechanical seal	cm ² /lms	0.154	0.018	0.833
General	cm ² /m ²	2.2	0.4	4.9	Storm inside, rigid sheet with mechanical seal	cm ² /lms	0.4	0.045	0.833
Without ductwork in crawl space	cm ² /m ²	1.98			Storm outside, pressurized track	cm ² /lmc	0.528		
With ductwork in crawl space	cm ² /m ²	2.25			Storm outside, 2-track	cm ² /lmc	1.23		
Fireplace					Storm outside, 3-track	cm ² /lmc	2.46		
With damper closed	cm ² /m ²	43	10	92					
With damper open	cm ² /m ²	350	145	380					
With glass doors	cm ² /m ²	40	4	40					
With insert and damper closed	cm ² /m ²	36	26	46					
With insert and damper open	cm ² /m ²	65	40	90					
Gas water heater	cm ² ea	20	15	25					
Joints									
Ceiling-wall	cm ² /lmc	1.5	0.16	2.5					
Sole plate, floor/wall, uncaulked	cm ² /lmc	4	0.38	5.6					
Sole plate, floor/wall, caulked	cm ² /lmc	0.8	0.075	1.2					
Top plate, band joist	cm ² /lmc	0.1	0.075	0.38					

Note: Air leakage areas are based on values found in the literature. The effective air leakage area (in square centimetres) is based on a pressure difference of 4 Pa and $C_D = 1$.

Abbreviations: m² = gross area in square metres
ea = each
lmc = linear metre of crack
lms = linear metre of sash

Table 2 Example of Calculation of Building Effective Air Leakage Area Based on Component Leakage Areas

Component	Description	Size or Number	$\times A_L$ per unit	$= A_{L,t}$ cm^2
Sills	Uncaulked	43.2 m	4.0 cm^2/m	173
Electrical outlets		20	0.5 cm^2 ea	10
Windows	Sliding	13.1 m^2	4.0 cm^2/m^2	75
Framing		13.1 m^2	1.7 cm^2/m^2	
Exterior doors	Single	5.7 m^2	7.7 cm^2/m^2	54
Framing		5.7 m^2	1.7 cm^2/m^2	
Fireplace	Without damper	1	350 cm^2 ea	350
Penetrations	Pipes	7	6.0 cm^2 ea	42
Heating ducts	Ducts untaped, in basement	1	144 cm^2 ea	144
Calculated total building air leakage area $A_c =$				848 cm^2

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 permit diffusion of water vapor from the wall. For a discussion of moisture transfer in building envelopes, see [Chapter 23](#) and [Chapter 24](#).

A continuous air infiltration retarder installed on the outside of wall framing can cover many difficult construction details associated with the installation of continuous air-vapor retarders. 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. The exterior air infiltration retarder can cover these problem areas continuously. 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 cuts down on convection currents within the wall cavity, allowing insulation to retain more of its effectiveness.

Existing Buildings. The air leakage sites must first be located in order to tighten the envelope of an existing building. As discussed earlier, air leakage in buildings is due not only to windows and doors, but to 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 E 1186* 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. Harje et al. (1979), Diamond et al. (1982), and Energy Resource Center (1982) include information on airtightening in

existing residential buildings. With these procedures, the air leakage of residential buildings can be reduced dramatically. Depending on the extent of the tightening effort and the experience of those doing the work, residential buildings can be tightened anywhere from 5% to more than 50% (Blomsterberg and Harje 1979, Harje and Mills 1980, Jacobson et al. 1986, Verschoor and Collins 1986, Giesbrecht and Proskiw 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 of about 0.2 air changes per hour (ACH) to loosely constructed housing with air exchange rates as great as 2.0 ACH. [Figure 10](#) and [Figure 11](#) show histograms of infiltration rates measured in two different samples of North American housing (Grimsrud et al. 1982, Grot and Clark 1979). [Figure 10](#) 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 11](#) represents measurements in 266 houses located in 16 cities in the United States. The median value of this sample is 0.9 ACH. The group of houses contained in the [Figure 10](#) sample is biased toward new, energy-efficient houses, while the group in [Figure 11](#) represents older, low-income housing in the United States.

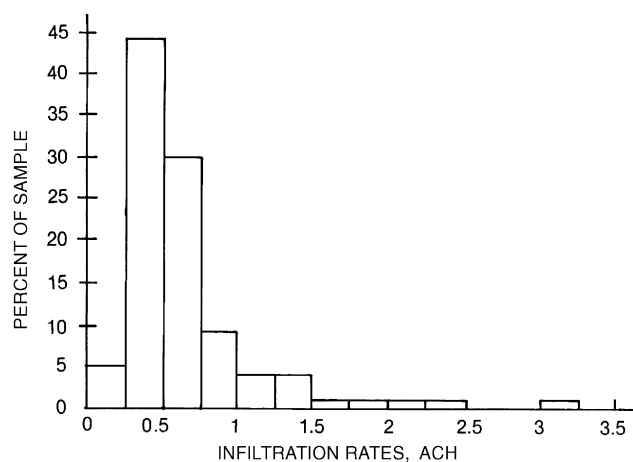


Fig. 10 Histogram of Infiltration Values—New Construction

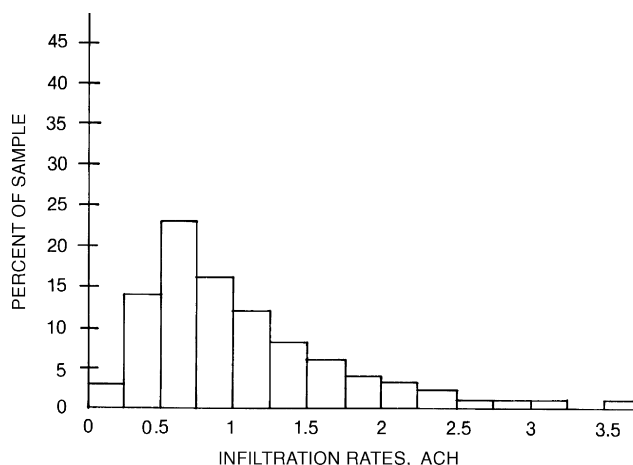


Fig. 11 Histogram of Infiltration Values—Low-Income Housing

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 the Pacific Northwest. In a comparison of 292 houses incorporating energy-efficient features (including measures to reduce air infiltration and provide ventilation heat recovery) with 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 also in the Pacific Northwest. Canadian housing stock has been characterized by Yuill and Comeau (1989) and Riley (1990). While 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, the users on average provided 63% of the total air exchange rate.

Ventilation air requirements for houses in the U.S. 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 such 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 the energy used to condition the ventilation air. The solution to these concerns is to have a reasonably 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) shows a 30% increase in airtightness of tract-built Canadian houses between 1982 and 1989. Also, 82% of the 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.

ASHRAE *Standard* 62 gives ventilation air requirements for houses, essentially 0.35 ACH with at least 8 L/s per occupant. 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 being used in houses, especially in energy-efficient housing demonstration programs (Riley 1990, Palmiter et al. 1991). Possible 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), Sibbitt and Hamlin (1991), Palmiter et al. (1991), Yuill et al. (1991), Holton et al. (1997), Sherman and Matson (1997), Reardon and Shaw (1997), and Lubliner et al. (1997).

The simplest systems use bathroom and kitchen fans 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. 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 thermostat 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 reduces the energy required to condition the ventilation air by typically 60 to 80% (Cutter 1987). It also reduces the thermal comfort problem 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.

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 for those cases in which they are, an unbalanced system is usually the best choice.

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 12](#) 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 forced (mechanical) ventilation. Zone 2 includes the 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 over a substantial range of airtightness. Zone 4 residences have relatively small energy penalties associated

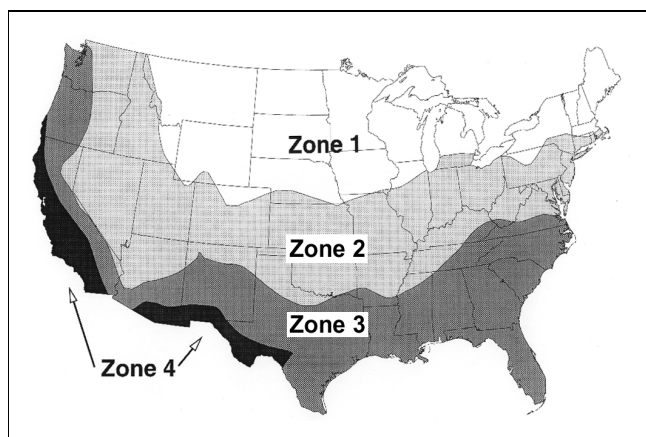


Fig. 12 Airtightness Zones for Residences in the United States
(Sherman 1995)

with infiltration or ventilation. In this zone, natural ventilation is usually preferred to forced ventilation as a technique to supplement infiltration.

RESIDENTIAL VENTILATION REQUIREMENTS

Traditionally, residential ventilation 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. Houses built or retrofitted to new standards have substantially tighter envelopes and insufficient infiltration to meet ventilation standards. In most circumstances, concerns over safety, noise, comfort, air quality, and energy minimize occupant use of operable windows. As a result, these houses require supplemental mechanical ventilation to satisfy these standards.

Simply meeting the 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. 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.

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 the contaminants before they can mix into the indoor environment. Typical sources that should be considered include the following:

Clothes Dryers. Clothes dryer exhaust is heavily laden with moisture and laundry by-products. Many moisture problems have been traced to clothes dryers vented indoors. Exhaust from clothes dryers, which is typically about 70 L/s, should be vented directly to the outdoors.

Combustion. Water and carbon dioxide are always emitted during combustion. Other more dangerous compounds can be emitted as well. All these by products should be vented directly outdoors. Venting of combustion appliances should meet all applicable codes, but for buildings with naturally aspirated combustion appliances within the pressure boundary, excessive depressurization due to

exhaust systems should be avoided. In addition, a depressurization safety test should be considered, such as described in ASTM *Standard* E 1998 or CGSB *Standard* 51.71.

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 UL *Standard* 2034 should be installed in each dwelling that has combustion appliances (including fireplaces) within the pressure boundary or attached garages.

Garages. Garages contain many sources of contaminants. Doors between garages 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 ducts, should not be located in garages. If such sections must pass through garages, they should be well sealed.

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 3 μ m particles, which is equivalent to a MERV 6 designated filter according to ASHRAE *Standard* 52.2.

Outdoor Air. Outdoor air may contain unacceptably high levels of pollutants, including ozone, pollen, carbon monoxide, particulate matter, odors, etc. In such cases, it may be impossible to provide acceptable indoor air quality; increased ventilation rates can actually decrease indoor air quality. In areas in which this problem may be anticipated, controls should be provided to allow the occupants to temporarily reduce the ventilation rate. Air cleaning should be considered for sensitive individuals.

Local Exhaust Ventilation

The single most important source control mechanism in dwellings apart from source elimination is local exhaust. All wet rooms and other spaces designed to allow specific contaminant release should be provided with local exhaust. These spaces include kitchens, utility rooms, bathrooms, and toilets. Workshops, recreation rooms, smoking areas, art studios, and hobby rooms may also require local ventilation and/or air cleaning to remove contaminants generated by the activities involved. If unvented combustion appliances must be used, then rooms with these appliances should meet the same general ventilation requirements for kitchens because such appliances generate significant amounts of moisture even when burning properly.

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

Continuous Local Mechanical Ventilation. 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 3](#).

Table 3 Continuous Exhaust Airflow Rates

Application	Continuous Flow	Notes
Kitchen	5 ACH	Based on kitchen volume
Utility room	10 L/s	Not less than 2 room air changes per hour
Bathroom	10 L/s	
Toilet	10 L/s	

Table 4 Intermittent Exhaust Airflow Rates

Application	Continuous Flow	Notes
Kitchen	50 L/s	Vented range hood required if less than 5 kitchen air changes per hour
Utility room	25 L/s	Not less than 2 room air changes per hour
Bathroom	25 L/s	
Toilet	25 L/s	

Table 5 Total Ventilation Air Requirements

Area Based	Occupancy Based
0.10 L/s per square metre of floor space	8 L/s per person, based on normal occupancy

Intermittent Local Mechanical Ventilation. An intermittently operating local mechanical exhaust is intended to be operated as needed by the occupant and should be designed with this intent. Shut-off timers, occupancy controls, multiple speed fans, and switching integral with room lighting are helpful, provided they do not impede occupant control. The minimum airflow rating should be at least that given in [Table 4](#), or as mandated by local codes.

Alternative Local Ventilation. Cleaning of recirculated air can sometimes be substituted for local ventilation, if it can be shown to be effective in removing contaminants of concern. Natural ventilation is not generally a suitable method for local ventilation in most climates and spaces. Use of natural ventilation can cause re-entrainment problems when air flows into rather than out of the space, and contaminated exfiltrating air reenters the building. In milder climates, it may be acceptable to use natural ventilation, when the contaminant of concern is related to odor rather than health or safety. Purpose-designed passive exhaust systems have shown acceptable ventilation in European settings and may be considered in lieu of mechanical systems.

Whole-House Ventilation

While control of significant sources of pollution in a dwelling is important, whole-house ventilation may still be needed. Each dwelling should be provided with outdoor air according to [Table 5](#). 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.

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. However, because occupancy and sources vary significantly, the capacity to ventilate above minimum rates can be provided by operable openings such as doors and windows.

Continuous whole-house mechanical ventilation systems are intended to operate without occupant intervention. The ventilation effects of multiple local systems may be combined to meet whole-house requirements.

Intermittent whole-house mechanical ventilation systems are intended to be operated automatically and regularly, but not necessarily continuously. The system may consist of supply, exhaust, or balanced mechanical systems. It should be designed so that it can operate automatically based on a timer. A supplementary control mechanism such as a humidistat or indoor air quality sensor may be used.

The effective ventilation rate of an intermittent system is the product of its delivered capacity, its fractional on-time, and the temporal ventilation effectiveness from [Table 6](#). Systems cycling at least once every 3 h may be assumed to have perfect ventilation effectiveness. Multiple systems may be combined, but the combination should only be supply or exhaust values and not both.

Table 6 Ventilation Effectiveness for Intermittent Fans

Fractional On-Time, f	Temporal Ventilation Effectiveness
$f \leq 35\%$	0.33
$35\% < f \leq 60\%$	0.50
$60\% < f \leq 80\%$	0.75
$80\% < f$	1.0

Air Distribution

Ventilation air should be provided to each habitable room through mechanical and natural air distribution. If a room does not have a balanced air supply (or inlet) 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.

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.

Selection Principles for Ventilation Systems

Determining the right ventilation approach is part of optimizing the entire building, and it can rarely be done without considering the building envelope, the climate, the needs of the users, and costs. For example, mechanical exhaust systems should not be used in hot, humid climates (because of possible condensation problems) when the building has or is likely to have air conditioning installed. Similarly, mechanical supply ventilation systems should not be used in very cold climates.

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, 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. 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, insulation particles, etc.) 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, resulting in damage to the structural integrity of the home. Excess pressure can best be prevented by balanced ventilation systems and nonleaky duct systems. In addition, adequate pathways must be available for all return air to the air-handling devices.

Occupant activities and operation of fans that exhaust air from the home (including leaky ducts on air conditioners, furnaces, or heat pumps) may produce depressurization of the structure. Several 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 the exhaust from the zone
- Testing to ensure that depressurization will not be excessive

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

Selecting a Whole-House Ventilation System

Whole-house ventilation can be provided through mechanical or natural means or by a combination thereof. Regardless of the approach chosen, it is necessary to consider where the outdoor air comes from, how it enters the house, how it is distributed, and how it leaves the house. Systems that are uncomfortable, expensive to operate, unsafe, noisy, or in other ways unacceptable to the occupants are not likely to be used.

Natural Ventilation. The use of operable windows and other natural ventilation openings is rarely an acceptable means of providing base levels of whole-house ventilation (Wilson and Walker 1992), but when it is, it can be a cost-effective alternative. Operable windows allow the user to control the ventilation by the size of the opening. Windows are present for light and egress and can serve as a ventilation system; most detached dwellings have sufficient operable windows for natural ventilation. Environmental factors may inhibit effective occupant use of the windows. During cold periods, open windows can cause thermal discomfort. Noise or security issues may also reduce the desirability of opening windows. Finally, the uncontrolled nature of the natural ventilation may increase energy use in many climates, but it can be energy-efficient in some mild climates.

Mechanical Ventilation. Whole-house systems may run continuously or intermittently. Intermittent systems must supply more ventilation air and, thus, may cost more to temper the outside air and to run the fans. The system can consist of supply, exhaust, or a balanced combination of the two. Fans that are noisy are likely to be unacceptable to many occupants and will be disabled; noise should be reduced by using quiet fans or by remote mounting. Continuous ventilation requires less power if the system is designed with low-resistance ducting. The electrical energy required for operation should be calculated together with the energy required to temper the outside air introduced through ventilation.

Supply Ventilation. A supply ventilation system usually has a single intake for the ventilation air. This air is distributed through the house either by a dedicated duct system or by the thermal distribution system. The outdoor air may be filtered in a supply ventilation system to remove pollen and dust.

Envelope leakage, exhaust stacks, and flues provide pathways for exhaust air. Supply ventilation can result in indoor pressurization. This is generally unacceptable in very cold climates because the exfiltrating air can cause condensation in the building envelope. However, continuous supply ventilation systems can be designed in conjunction with specific building envelope features to account for this condensation.

Supply ventilation can mitigate radon entry or backdrafting problems and may reduce interstitial moisture problems in hot, humid climates. In temperate or severe climates, the supply air, if delivered directly to rooms without tempering, can cause thermal discomfort or draftiness. A dedicated duct system increases the cost of a supply ventilation system.

Exhaust Ventilation. An exhaust ventilation system usually has a central mechanical exhaust. Air enters the house through envelope leakage, open windows, or designed inlets. Because the air intake is dispersed, thermal discomfort is less than that with supply ventilation. An exhaust air heat pump can recover energy, but it may not be economical in many climates. The exhaust fan depressurizes the house, which can aggravate radon or backdrafting problems. Because of potential interstitial space moisture problems, exhaust

ventilation may be unacceptable in hot, moist climates when the indoor air is mechanically cooled.

Although air intake through building leaks reduces the particulate concentration somewhat, filtration of outdoor air is not generally possible with exhaust ventilation. In unusually tight houses, envelope leakage may be insufficient to provide enough supply air, so designed intakes may be needed.

Balanced Ventilation. In a balanced mechanical ventilation system, there is usually a mechanical exhaust either centrally located or ducted from locations likely to have high contaminant levels. There is a single outside air intake for the ventilation air, which is then dispersed through the house. The systems are designed to produce equal supply and exhaust flows. With equal flows, the system creates neither pressurization nor depressurization, so problems associated with house pressures are reduced. It is also possible to use slight positive pressurization to reduce infiltration. Filtration and tempering of the incoming air can be accomplished at the central unit.

Most balanced systems feature either sensible heat recovery or total heat recovery. Sensible heat recovery systems provide tempering of the incoming air using the exiting air as a source for heat (or cooling). Total heat recovery systems provide both sensible and latent tempering of the incoming air.

Because of its energy recovery properties, a balanced system becomes more attractive as total space-conditioning costs increase, such as in severe climates. The balanced system is initially the most costly of the three systems, but total operating costs may be less, particularly if high-efficiency fan/motor assemblies are used.

Selecting a Local Ventilation System

The two most common methods for providing local ventilation to damp rooms are natural ventilation through an operable window and mechanical exhaust fans. These methods have the same design issues as whole-house systems. Balanced or supply-only systems should not be used for local ventilation to reduce spreading of moisture and contaminants through the house.

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. The air exchange rate of a building 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, assumptions as to their values must be made. The appropriateness of these assumptions determines 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 50 Pa 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 Reeves et al. (1979), Kronvall (1980), and Shaw (1981).

The latter two models account for building air leakage using the values of c and n from Equation (32). The only other inputs required are wind speed and temperature difference. Such 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 due to the building-specific nature of leakage distribution, wind pressure, and internal partitioning. Persily and Linteris (1983) and Persily (1986) show comparisons between 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.

Multizone Models

Multicell models of air exchange treat buildings as a series of interconnected zones and assume that the air within each zone is well mixed. Several such models have been developed by Allard and Herrlin (1989), Etheridge and Alexander (1980), Liddament and Allen (1983), Walton (1984, 1989), Herrlin (1985), and Feustel and Raynor-Hoosen (1990). 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 within the building by requiring that the inflows and outflows for each zone balance to zero. The models require the user to input a location and leakage function for every opening in the building envelope and in relevant interior partitions, a value for the wind pressure coefficient C_p at the location of each building envelope leakage site, temperatures for each zone, and any mechanical ventilation airflow rates. Such detailed information is difficult to obtain for a building. 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. 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. Caution should be used when applying them to the prediction of smoke movement patterns in the event of a fire. Multizone models have the advantage of being able to model very complex representations of buildings by using personal computers; however, determining the correct inputs to these models is difficult. As a result of these uncertainties, multizone models are best used to bound a solution rather than determining an absolute solution. Monte Carlo simulation coupled with the multizone modeling is a useful technique to determine these bounds, if the probability distribution for the uncertain parameters can be defined.

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 to buildings with no internal resistance to airflow and are therefore inappropriate to large, multizone buildings. Models of this type have been developed by the Institute of Gas Technology (IGT) (Cole et al. 1980), the Building Research Establishment (Warren and Webb 1980), the Lawrence Berkeley National Laboratory (LBNL) (Sherman and Grimsrud 1980), and the University of Alberta (AIM-2) (Walker and Wilson 1998). These models have exhibited average errors on the order of 40% for many measurements on groups of houses and can be off by 100% in individual cases (Persily 1986, Walker and Wilson 1998). The following section on Residential Calculation Examples uses a basic model and an enhanced model (Walker and Wilson 1998, Palmiter and Bond 1994, Hamlin and Pushka 1994, CHBA 1994, Bradley 1993).

The **basic model** uses the effective air leakage area A_L at 4 Pa, which can be obtained from a whole-building pressurization test. If a test value is not available, the data in Table 1 can be used to estimate the air leakage area of the building. To obtain the building's total air leakage area, multiply the overall dimensions or number of occurrences of each building component by the appropriate table entry. The sum of the resulting products is the total building air leakage area. See Table 2 for an example.

The **enhanced model** uses pressurization test results to characterize house air leakage through the leakage coefficient c and the pressure exponent n . The enhanced model improves on the basic model by using a power law to represent the envelope leakage, including a flue as a separate leakage site, and having separate wind effects for houses with crawl spaces 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 $\pm 10\%$ when the parameters are well known for the building in question (Walker and Wilson 1998, Palmiter and Bond 1994, Sherman and Modera 1986). All these single-zone models are sensitive to the values of the 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 the 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 (1992a) compares various superposition procedures and derives a generalized superposition equation that involves some simple leakage distribution parameters. He shows that the result is always sub-additive. Typically only 35% of the infiltration due to the smaller effect can be added to the larger effect. Depending on the 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 (39)] to be a robust superposition technique:

$$Q = \sqrt{Q_s^2 + Q_w^2} \quad (39)$$

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

Residential Calculation Examples

Basic Model. The following calculations are based on the LBNL model (Sherman and Grimsrud 1980), which uses the effective air leakage area at 4 Pa. This leakage area can be obtained from a whole-building pressurization test. Using the effective air leakage area, the airflow rate due to infiltration is calculated according to

$$Q = \frac{A_L}{1000} \sqrt{C_s \Delta T + C_w U^2} \quad (40)$$

where

$$\begin{aligned} Q &= \text{airflow rate, m}^3/\text{s} \\ A_L &= \text{effective air leakage area, cm}^2 \\ C_s &= \text{stack coefficient, (L/s)}^2/(\text{cm}^4 \cdot \text{K}) \end{aligned}$$

Δt = average indoor-outdoor temperature difference for time interval of calculation, K
 C_w = wind coefficient, (L/s)²/[cm⁴ · (m/s)²]
 U = average wind speed measured at local weather station for time interval of calculation, m/s

Table 7 presents values of C_s for one-, two-, and three-story houses. The value of the wind coefficient C_w depends on the local shelter class of the building (described in Table 8) and the building height. Table 9 presents values of C_w for one-, two-, and three-story houses in shelter classes 1 through 5. In calculating the values in Table 7 and Table 9, the following assumptions were made regarding input to the basic model:

- Terrain used for converting meteorological to local wind speeds is that of a rural area with scattered obstacles
- $R = 0.5$ (half of 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 = 2.5, 5.0, and 7.5 m, respectively

Example 1. Estimate the infiltration at design conditions for a two-story house in Lincoln, Nebraska. The house has an effective air leakage area of 500 cm² and a volume of 340 m³, and the predominant wind is perpendicular to the street (shelter class 3). The indoor air temperature is 20°C.

Solution: The 99% design temperature for Lincoln is -19°C. Assume a design wind speed of 6.7 m/s. From Equation (40), with $C_s = 0.000\ 290$ from Table 7 and $C_w = 0.000\ 231$ from Table 9, the airflow rate due to infiltration is

$$Q = \frac{500}{1000} \sqrt{(0.000\ 290 \times 39) + (0.000\ 231 \times 6.7^2)}$$

$$= 0.0736\ \text{m}^3/\text{s} = 265\ \text{m}^3/\text{h}$$

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

Table 7 Basic Model Stack Coefficient C_s

	House Height (Stories)		
	One	Two	Three
Stack coefficient	0.000 145	0.000 290	0.000 435

Table 8 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 the street from the 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 that are immediately adjacent (closer than one house height): e.g., neighboring houses on the same side of the street, trees, bushes, etc.

Table 9 Basic Model Wind Coefficient C_w

Shelter Class	House Height (Stories)		
	One	Two	Three
1	0.000 319	0.000 420	0.000 494
2	0.000 246	0.000 325	0.000 382
3	0.000 174	0.000 231	0.000 271
4	0.000 104	0.000 137	0.000 161
5	0.000 032	0.000 042	0.000 049

$$I = (265\ \text{m}^3/\text{h})/340\ \text{m}^3$$

$$= 0.78\ \text{h}^{-1} = 0.78\ \text{ACH}$$

Example 2. Calculate 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 17 K, and the average wind speed is 2.7 m/s. The house has a volume of 255 m³ and an effective air leakage area of 690 cm², and it is located in an area with buildings and trees within 10 m in most directions (shelter class 4).

Solution: From Equation (40), the airflow rate due to infiltration is

$$Q = \frac{690}{1000} \sqrt{(0.000\ 145 \times 17) + (0.000\ 104 \times 2.7^2)}$$

$$= 0.0392\ \text{m}^3/\text{s} = 141\ \text{m}^3/\text{h}$$

The air exchange rate is therefore

$$I = 141/255 = 0.55\ \text{h}^{-1} = 0.55\ \text{ACH}$$

Example 3. Estimate the average infiltration over the heating season in a two-story house with a volume of 310 m³ and the air leakage area calculated in Table 2 (848 cm²). The house is located on a lot with several large trees but no other close buildings (shelter class 3). The average wind speed during the heating season is 3.2 m/s, while the average indoor-outdoor temperature difference is 20 K.

Solution: From Equation (40), the airflow rate due to infiltration is

$$Q = \frac{848}{1000} \sqrt{(0.000\ 290 \times 20) + (0.000\ 231 \times 3.2^2)}$$

$$= 0.077\ \text{m}^3/\text{s} = 276\ \text{m}^3/\text{h}$$

The average air exchange rate is therefore

$$I = 276/310 = 0.89\ \text{h}^{-1} = 0.89\ \text{ACH}$$

Enhanced Model. This section presents a simple, single-zone approach to calculating air infiltration rates in houses based on the AIM-2 model (Walker and Wilson 1998). The airflow rate due to infiltration is calculated using

$$Q_s = c C_s \Delta t^n \quad (41)$$

$$Q_w = c C_w (sU)^{2n} \quad (42)$$

where

Q_s = stack airflow rate, m³/s
 Q_w = wind airflow rate, m³/s
 c = flow coefficient, m³/(s·Pa ^{n})
 C_s = stack coefficient, (Pa/K) ^{n}
 C_w = wind coefficient, (Pa·s²/m²) ^{n}
 s = shelter factor

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

- Each story is 2.5 m high.
- The flue is 15 cm in diameter and reaches 2 m above the upper ceiling.
- The flue is unsheltered.
- Half of the 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, the windspeed multiplier G (given in Table 10) uses a relationship based on equations found in Chapter 16.

Table 10 Enhanced Model Wind Speed Multiplier G

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

Table 11 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

Table 12 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	0.054	0.069	0.078	0.089	0.098	0.107
C_w for basement/slab	0.156	0.142	0.170	0.156	0.170	0.167
C_w for crawl space	0.128	0.128	0.142	0.142	0.151	0.154

Example 4. 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 $c = 0.051 \text{ m}^3/(\text{s} \cdot \text{Pa}^n)$ and a pressure exponent of $n = 0.67$ (this corresponds to effective leakage area of 500 cm^2 at 4 Pa). The building volume is 340 m^3 . The 97.5% design temperature is -19°C , and the design wind speed is 6.7 m/s .

Solution: For a slab-on-grade two-story house with a flue, Table 12 gives $C_s = 0.089 \text{ (Pa/K)}^n$ and $C_w = 0.156 \text{ (Pa} \cdot \text{s}^2/\text{m}^2)^n$. The house is maintained at 20°C indoors. The building wind speed is determined by taking the design wind speed and multiplying by the wind speed multiplier G from Table 10:

$$U = GU_{met} = 0.59(6.7) = 3.95 \text{ m/s}$$

From Table 8, the shelter class for a typical urban house is 4. Table 11 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 (41):

$$Q_s = (0.051)(0.089)[20 - (-19)]^{0.67} = 0.053 \text{ m}^3/\text{s}$$

The wind flow is calculated using Equation (42):

$$Q_w = (0.051)(0.156)(0.64 \times 3.95)^{1.34} = 0.027 \text{ m}^3/\text{s}$$

Substituting Q_s and Q_w into Equation (39) gives $Q = 0.059 \text{ m}^3/\text{s} = 214 \text{ m}^3/\text{h}$. From Equation (2), the air exchange rate I is equal to Q divided by the building volume:

$$I = (214 \text{ m}^3/\text{h})/340 \text{ m}^3 \\ = 0.63 \text{ h}^{-1} = 0.63 \text{ ACH}$$

Example 5. Estimate the average infiltration over a one-week period for a single-story crawl space house in Redmond, Washington. The house has a flow coefficient of $c = 0.078 \text{ m}^3/(\text{s} \cdot \text{Pa}^n)$ and a pressure exponent of $n = 0.6$ (this corresponds to effective leakage area of 690 cm^2 at 4 Pa). The building volume is 255 m^3 . During this period, the average indoor-outdoor temperature difference is 16 K , and the wind speed is 2.7 m/s . The house is electrically heated and has no flue.

Solution: For a single-story house with no flue, $C_s = 0.054 \text{ (Pa/K)}^n$. For a crawl space, $C_w = 0.128 \text{ (Pa} \cdot \text{s}^2/\text{m}^2)^n$. From Table 10, for a one-story house, $G = 0.48$.

$$U = GU_{met} = 0.48(2.7) = 1.3 \text{ m/s}$$

Table 11 gives shelter factor $s = 0.50$ for a house with no flue and shelter class 4. The stack flow is calculated using Equation (41):

$$Q_s = (0.078)(0.054)(16)^{0.6} = 0.022 \text{ m}^3/\text{s}$$

The wind flow is calculated using Equation (42):

$$Q_w = (0.078)(0.128)(0.50 \times 1.3)^{1.2} = 0.006 \text{ m}^3/\text{s}$$

Substituting Q_s and Q_w into Equation (39) gives $Q = 0.023 \text{ m}^3/\text{s} = 83 \text{ m}^3/\text{h}$. From Equation (2), the air exchange rate I is equal to Q divided by the building volume:

$$I = (83 \text{ m}^3/\text{h})/(255 \text{ m}^3) \\ = 0.32 \text{ h}^{-1} = 0.32 \text{ ACH}$$

Example 6. Estimate the infiltration for a three-story house in San Francisco, California. The house has a flow coefficient of $c = 0.102 \text{ m}^3/(\text{s} \cdot \text{Pa}^n)$ and a pressure exponent of $n = 0.67$ (this corresponds to effective leakage area of 1000 cm^2 at 4 Pa). The building volume is 395 m^3 . The indoor-outdoor temperature difference is 5 K , and the wind speed is 4.47 m/s . The house has a flue and a crawl space foundation.

Solution: For a three-story house with a flue, $C_s = 0.107 \text{ (Pa/K)}^n$. For a crawl space, $C_w = 0.154 \text{ (Pa} \cdot \text{s}^2/\text{m}^2)^n$. From Table 10, for a three-story house, $G = 0.67$.

$$U = GU_{met} = 0.67(4.47) = 3.0 \text{ m/s}$$

The prevailing wind blows along the row of houses parallel to the street, so the house has a shelter class of 5. Table 11 gives the shelter factor for a three-story house with a flue and shelter class 5 as $s = 0.43$.

$$Q_s = (0.102)(0.107)(5)^{0.67} = 0.032 \text{ m}^3/\text{s}$$

$$Q_w = (0.102)(0.154)(0.43 \times 3.0)^{1.34} = 0.022 \text{ m}^3/\text{s}$$

Substituting Q_s and Q_w in Equation (39) gives $Q = 0.039 \text{ m}^3/\text{s} = 140 \text{ m}^3/\text{h}$.

$$I = (140 \text{ m}^3/\text{h})/395 \text{ m}^3 \\ = 0.35 \text{ h}^{-1} = 0.35 \text{ ACH}$$

Combining Residential Infiltration and Mechanical Ventilation

Significant infiltration and mechanical ventilation often occur simultaneously in residences. The pressure difference from Equation (24) 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, the natural infiltration and mechanical ventilation are usually determined separately and require a superposition method to combine the flow rates.

Sherman (1992) compares various superposition procedures and derives a generalized superposition equation that involves some simple leakage distribution parameters. He shows that 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 the leakage distribution. When the fan flow becomes 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 will have a much bigger effect than an exhaust fan on changing the total ventilation rate; the same situation holds for houses having high neutral levels in cold climates. For the general case, when the details are not known or can be assumed to be broad and typical, the following superposition gives good results:

$$Q_{comb} = Q_{bal} + \sqrt{Q_{unbal}^2 + Q_{infiltration}^2} \quad (43)$$

NONRESIDENTIAL AIR LEAKAGE

Commercial Building Envelope Leakage

There is currently only one industry standard for the measurement of envelope leakage in tall buildings: CGSB *Standard* 149.15. It utilizes the building's own air handlers. ASTM *Standard* E779 and CGSB *Standard* 149.10 are intended for small detached buildings such as houses and 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 of obstacles to accurate measurement not present in small buildings, including large envelope leakage area, interfloor leakage, vertical shafts, and large wind and stack pressures. Recent work by Bahnfleth et al. (1999) also shows how to deal with some of these issues.

The building envelopes of large commercial buildings are often thought to be quite airtight. The National Association of Architectural Metal Manufacturers specifies a maximum leakage per unit of exterior wall area of $300 \text{ cm}^3/(\text{s} \cdot \text{m}^2)$ at a pressure difference of 75 Pa exclusive of leakage through operable windows. Tamura and Shaw (1976a) found that, assuming a flow exponent n of 0.65 in Equation (32), air leakage measurements in eight Canadian office buildings with sealed windows ranged from 610 to $2440 \text{ cm}^3/(\text{s} \cdot \text{m}^2)$. 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. In a study of eight U.S. office buildings, Persily and Grot (1986) found air leakage ranging from 1080 to $5220 \text{ cm}^3/(\text{s} \cdot \text{m}^2)$ at 75 Pa. This means that office building envelopes are leakier than expected. Typical air leakage values per unit wall area at 75 Pa are 500, 1500, and $3000 \text{ cm}^3/(\text{s} \cdot \text{m}^2)$ for tight, average, and leaky walls, respectively (Tamura and Shaw 1976a).

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.

Air Leakage Through Internal Partitions

In large buildings, the 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 within a building. These internal resistances are also important in the event of a fire to predict smoke movement patterns and evaluate smoke management systems.

Table 13 gives air leakage areas (calculated at 75 Pa with $C_D = 0.65$) for different internal partitions of commercial buildings (Klote and Milke (1992)). Figure 13 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 13. Chapter 52 of the

Table 13 Air Leakage Areas for Internal Partitions in Commercial Buildings (at 75 Pa and $C_D = 0.65$)

Construction Element	Wall Tightness	Area Ratio
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

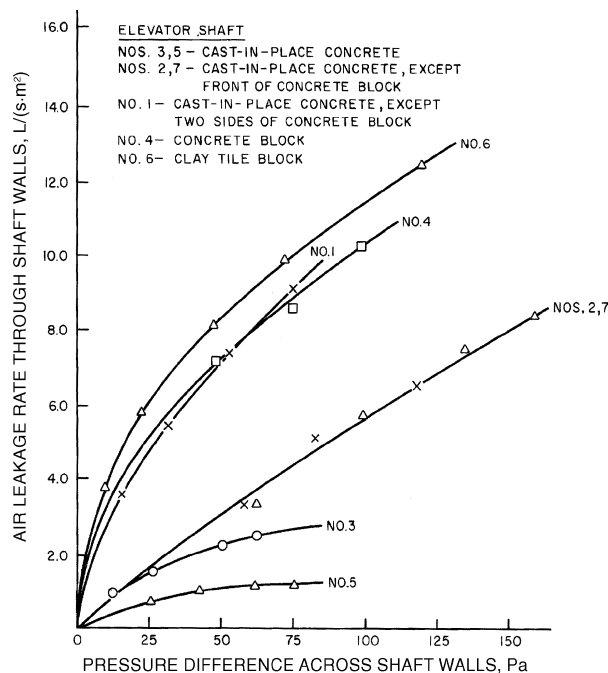


Fig. 13 Air Leakage Rates of Elevator Shaft Walls

ASHRAE Handbook—Applications should be consulted for performance models and applications of smoke management systems.

Leakage openings at the top of elevator shafts are equivalent to orifice areas of 0.4 to 1.0 m^2 . Air leakage rates through stair shaft and elevator doors are shown in Figure 14 as a function of average crack width around the door. The air leakage areas associated with other openings within commercial buildings are also important for air movement calculations. These include interior doors and partitions, suspended ceilings in buildings where the space above the ceiling is used in the air distribution system, and other components of the air distribution system.

Air Leakage Through Exterior Doors

Door infiltration depends on the type of door, room, and building. In residences and small buildings where doors are used infrequently, the air exchange associated with a door can be estimated based on air leakage through cracks between the door and the frame. A frequently opened single door, as in a small retail store, has a much larger amount of airflow than a closed door.

Air Leakage Through Automatic Doors

Automatic doors are a major source of air leakage in buildings. They are normally installed in locations where large numbers of

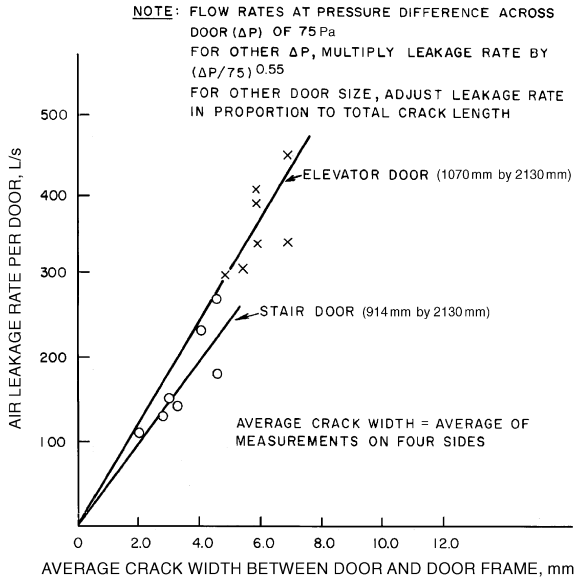


Fig. 14 Air Leakage Rate of Door Versus Average Crack Width

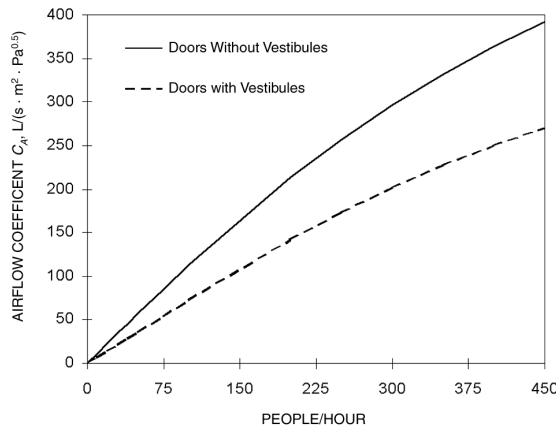


Fig. 15 Airflow Coefficient for Automatic Doors

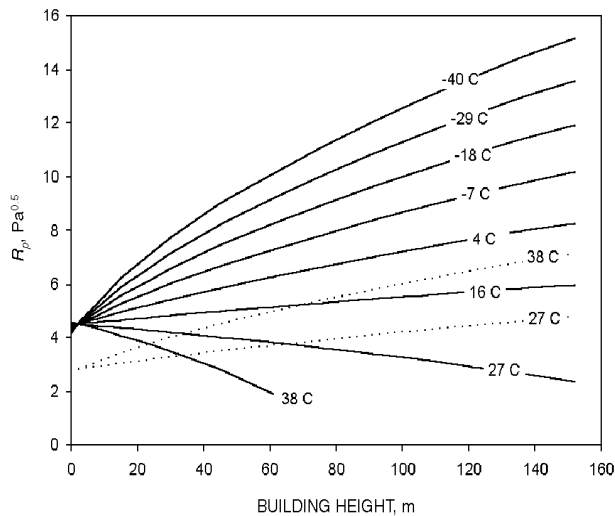


Fig. 16 Pressure Factor for Automatic Doors

people use the doors. They stay open longer with each use than manual doors. The air leakage through automatic doors can be reduced by the installation of 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 the airflow through automatic doors when calculating the heating and cooling loads in the spaces next to them.

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, Figure 15 has been developed (Yuill 1996) 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 people using a door per hour. To obtain the average infiltration rate through an automatic door, the user must multiply this coefficient by the opening area of the door and by the square root of the pressure difference between the outdoor air and the indoor air at the location of the door. The pressure difference across a door in a building depends on the wind pressure on the building, the stack effect due to the indoor-outdoor temperature difference, and the effect of the operation of the air-handling system. It also depends on the leakage characteristics of the exterior walls of the building and of internal partitions.

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 16. The second method requires explicit calculation of envelope pressures.

In Figure 16, the airflows shown for ambient temperatures of 27 and 38°C, represented by dotted lines, are outward flows. They intercept the vertical axis at a lower point than the other lines because the wind pressure coefficients on the downwind face of the building (where the greatest outward flows will occur) are lower than on the upward face. In many buildings, the pressure in the building is controlled by varying the flow rate through the return fan(s) or by controlling the relief air dampers. These systems are usually set to maintain a pressure above ambient in the lobby. Subtracting the interior pressure maintained in the lobby from the wind pressure gives the net pressure for estimating airflow through the door.

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

$$Q = C_A A R_p \quad (44)$$

where

Q = airflow rate, L/s

C_A = airflow coefficient from Figure 15, L/(s·m²·Pa^{0.5})

A = area of the door opening, m²

R_p = pressure factor from Figure 16, (Pa)^{0.5}

Method 2. The airflow Q is given by

$$Q = C_A A \sqrt{\Delta p} \quad (45)$$

where

Q = airflow rate, L/s

C_A = airflow coefficient from Figure 15, L/(s·m²·Pa^{0.5})

A = area of the door opening, m²

Δp = pressure difference across the door, Pa

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

$$\Delta p = p_U - \Delta p_s \quad (46)$$

where

p_U = wind-induced surface pressure relative to static pressure, Pa
 Δp_s = pressure difference due to stack effect, Pa

Example Calculations

It is desired to 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 $0.91 \text{ m} \times 2.1 \text{ m} = 1.9 \text{ m}^2$. Each floor is 4 m high. Approximately 300 people per hour pass through the door. The design wind conditions are 6.7 m/s, the indoor temperature is 21°C, and the outdoor temperature is -7°C. The airflow coefficient from [Figure 15](#) (using the line for doors without vestibules) is approximately $300 \text{ L/(s} \cdot \text{m}^2 \cdot \text{Pa}^{0.5})$.

Method 1:

The pressure factor from [Figure 16](#) is $7.9 \text{ Pa}^{0.5}$. Equation (44) gives the door flow as

$$Q = 300(1.9)7.9 = 4500 \text{ L/s}$$

Method 2:

The worst possible case for the 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 16](#) to be about 0.75. Using this in Equation (18), together with the specified wind speed, results in $p_w = 20 \text{ Pa}$. Assume that H is one-half the height of the door (1.1 m). In order to have the maximum pressure across the door, assume the neutral pressure plane is located halfway up the building such that

$$H_{\text{NPL}} = \frac{1}{2}(20 \text{ stories}) \frac{4 \text{ m}}{\text{story}} = 40 \text{ m}$$

Substituting these values into Equation (17) gives $\Delta p_s = -47 \text{ Pa}$. 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. We will assume that this coefficient is 0.9, which is the highest value that has been found for tall buildings. Therefore, $\Delta p_s = 0.9(-47 \text{ Pa}) = -42 \text{ Pa}$. The total pressure is then $\Delta p = 20 - (-42) = 62 \text{ Pa}$. Substituting into Equation (45),

$$Q = 306(1.9)\sqrt{62} = 4580 \text{ L/s}$$

If the building had a vestibule, the airflow coefficient would be read from [Figure 15](#) using the line for doors with vestibules, and it would be approximately $208 \text{ L/(s} \cdot \text{m}^2 \cdot \text{Pa}^{0.5})$, reducing the airflow to 3100 L/s into the building.

NONRESIDENTIAL VENTILATION

Commercial and institutional building ventilation systems are typically designed to provide a 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, where infiltration can be important due to doors. 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*, Ventilation for Acceptable Indoor Air Quality. In these procedures for designing forced 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.

ASHRAE Standard 62 includes two procedures for obtaining acceptable indoor air quality: the Ventilation Rate Procedure and the Indoor Air Quality Procedure. The Ventilation Rate Procedure is by far the more commonly used.

Ventilation Rate Procedure

In the Ventilation Rate Procedure, 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, depending on the space type. These ventilation rates are based on air pollutants generated by people, activities, and building materials and furnishings.

The HVAC designer faces several challenges in designing an air distribution system to deliver outdoor air to the occupants of a building. 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 goal 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. This outdoor air must be delivered not only at the design conditions, but throughout the year. The task is complicated by weather-related variations in indoor-outdoor pressure difference. Other complications include pressure variations due to building components such as exhaust fans or dirty filters, and probably most significantly by supply flow variations associated with the operation of variable air volume (VAV) systems (Mumma and Wong 1990, Janu et al. 1995). After the design level of outdoor air is brought into the building, it must then be delivered to the occupants. This issue is related to the discussion in the section on Air Change Effectiveness presented earlier in this chapter.

TRACER GAS MEASUREMENTS

The only reliable way to determine the air exchange rate of an existing building is to measure it. Several tracer gas measurement procedures exist (including a standard test method: *ASTM Standard E 741*), all involving an inert or nonreactive gas used to label the indoor air (Hunt 1980; Sherman et al. 1980; Harje et al. 1981; Lagus and Persily 1985; Dietz et al. 1986; Charlesworth 1988; Persily 1988; Fisk et al. 1989; Lagus 1989; Sherman 1989a, 1989b; Fortmann et al. 1990; Harje et al. 1990; Persily and Axley 1990; Sherman 1990). The tracer is released into the building in a specified manner, and the concentration of the tracer within the building is monitored and related to the building's air exchange rate. A variety of tracer gases and associated concentration detection devices have been used. Desirable qualities of a tracer gas are detectability, non-reactivity, 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 within the building. Assuming the outdoor concentration is zero and the indoor air is well mixed, this total balance takes the following form:

$$V \left(\frac{dC}{d\theta} \right) = F(\theta) - Q(\theta)C(\theta) \quad (47)$$

where

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

In Equation (47), density differences between indoor and outdoor air are generally ignored for moderate climates; therefore, Q also refers to the airflow rate into the building. While 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 (47) is based on the assumptions that (1) no unknown tracer gas sources exist; (2) the 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 within 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).

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 (47) is

$$C(\theta) = C_o e^{-I\theta} \quad (48)$$

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

Equation (48) 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 (48):

$$\ln C(\theta) = \ln C_o - I\theta \quad (49)$$

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 the calibration of the 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 and analyzed elsewhere.

The most serious problem with the decay technique is imperfect mixing of the tracer gas with the interior air, both at initial injection and during the decay. Equations (47) and (48) employ the assumption 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 will be subject to errors. It is difficult to estimate the magnitude of the errors due to poor mixing, and little analysis of this problem has been performed.

Growth. The growth or step-up method is similar to the decay method except that the initial tracer gas concentration is low and is increased 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 (47) reduces to

$$Q(\theta) = F(\theta)/C \quad (50)$$

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

Because the tracer gas injection is continuous, no initial mixing period is required. Another advantage is that the 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. This procedure has the disadvantage

of requiring the measurement of absolute tracer concentrations and injection rates. Also, imperfect mixing of the tracer and the 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 (47) becomes

$$C(\theta) = (F/Q)(1 - e^{-I\theta}) \quad (51)$$

After sufficient time, the transient term reduces to zero, the concentration attains equilibrium, and Equation (51) reduces to

$$Q = F/C \quad (52)$$

Equation (52) 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 the measurement of absolute concentrations and injection rates.

Dietz et al. (1986) used a special case of the constant injection technique. This technique uses permeation tubes as a tracer gas source. The tubes release the tracer at an ideally constant rate into the building being tested. 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 (47) for C and taking the time average gives

$$\langle C \rangle = \langle F/Q \rangle = F \langle 1/Q \rangle \quad (53)$$

where $\langle \dots \rangle$ denotes time average. (Note that the time average of $dC/d\theta$ is assumed to equal zero.)

Equation (53) shows that the average tracer concentration $\langle C \rangle$ and the 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 there were large changes in air exchange rate due to occupant airing of houses; errors from 5 to 30% were measured when the variation was due to 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 (47) 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 (Harrje et al. 1985, Sherman and Dickerhoff 1989, Fortmann et al. 1990, Harrje et al. 1990). These techniques are important when considering the transport of pollutants from one room of a building to another. For a theoretical development, see Sinden (1978b). 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).

SYMBOLS

A = area, m^2 or cm^2
 c = flow coefficient, $m^3/(s \cdot Pa^n)$
 c_p = specific heat, $J/(kg \cdot K)$ or $kJ/(kg \cdot K)$
 $\bar{C}$ = concentration
 $\bar{C}$ = time averaged concentration
 C_A = airflow coefficient for automatic doors, $L/(s \cdot m^2 \cdot Pa^{0.5})$
 C_D = discharge coefficient
 C_p = pressure coefficient
 C_s = stack flow coefficient, $(L/s)^2/(cm^4 \cdot K)$ or $(Pa/K)^n$
 C_v = effectiveness of openings
 C_w = wind flow coefficient, $(L/s)^2/[cm^4 \cdot (m/s)^2]$ or $[Pa/(m/s)^2]^n$
 F = tracer gas injection rate, m^3/s
 $\bar{F}$ = time-averaged contaminant source strength, m^3/s
 f = fractional on-time
 g = gravitational constant, m/s^2
 G = wind speed reduction factor
 h = specific enthalpy, kJ/kg
 H = height, m
 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, $K \cdot day$
 n = pressure exponent
 N = number of discrete time periods in period of interest
 p = pressure, Pa
 P = parameter
 q = heat rate, W
 $\dot{Q}$ = volumetric flow rate, m^3/s
 $\bar{Q}$ = effective volumetric flow rate, m^3/s
 s = shelter factor
 S = source strength, $\mu g/s$
 t = relative temperature, $^{\circ}C$
 T = absolute temperature, K
 U = wind speed, m/s
 V = volume, m^3
 W = humidity ratio, kg/kg
 ε_l = air change effectiveness
 ϕ = wind angle, degrees
 ρ = air density, kg/m^3
 θ = time
 θ_{age} = age of air
 τ = time constant

Subscripts

b = base
 ba = bypass air
 c = calculated
 ca = recirculated air
 e = effective
 ea = exhaust air
 f = floor
 i = indoor or time counter for summation (instantaneous)
 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 = outside air
 p = pressure
 r = reference
 s = sensible or stack
 sa = supply air
 S = space or source
 w = wind

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AIRBASE, a database of bibliographic references that contains abstracts in English of technical papers covering air infiltration in buildings, has been developed by the Air Infiltration and Ventilation Centre (AIVC 1992). Most of the articles are concerned with the prediction, measurement, and reduction of air infiltration and leakage rates. Abstracts are also included for selected papers relating to indoor air quality, occupant behavior, thermal comfort, ventilation efficiency, natural and mechanical ventilation, wind pressure and its influence on infiltration, energy-saving measures, and moisture and condensation.