CHAPTER 20

SPACE AIR DIFFUSION

ROOM air distribution systems are intended to provide thermal comfort and ventilation for space occupants and processes. Although air terminals (inlets and outlets), terminal units, local ducts, and rooms themselves may affect room air diffusion, this chapter addresses only air terminals and their direct effect on occupant comfort. This chapter is intended to present HVAC designers the fundamental characteristics of air distribution devices. For information on naturally ventilated spaces, see Chapter 16. For a discussion of various air distribution strategies, tools, and guidelines for design and application, see Chapter 56 in the 2007 *ASHRAE Handbook—HVAC Applications*. Chapter 19 in the 2008 *ASHRAE Handbook—HVAC Systems and Equipment* provides descriptions of the characteristics of various air terminals (inlets and outlets) and terminal units, as well as selection tools and guidelines. Other fundamental references include Bauman and Daly (2003), Chen and Glicksman (2003), Kirkpatrick and Elleson (1996), Rock and Zhu (2002), and Skistad et al. (2002).

Room air diffusion methods can be classified as one of the following:

- **Mixed systems** produce little or no thermal stratification of air within the space. Overhead air distribution is an example of this type of system.
- **Fully (thermally) stratified systems** produce little or no mixing of air within the occupied space. Thermal displacement ventilation is an example of this type of system.
- **Partially mixed systems** provide some mixing within the occupied and/or process space while creating stratified conditions in the volume above. Most underfloor air distribution designs are examples of this type of system.
- **Task/ambient conditioning systems** focus on conditioning only a certain portion of the space for thermal comfort and/or process control. Examples of task/ambient systems are personally controlled desk outlets (sometimes referred to as personal ventilation systems) and spot-conditioning systems.

Air distribution systems, such as displacement ventilation (DV) and underfloor air distribution (UFAD), that deliver air in cooling mode at or near floor level and return air at or near ceiling level produce varying amounts of room air stratification. Figure 1 presents a series of simplified vertical profiles of temperature and pollutant concentration representing the spectrum of stratified conditions that may exist under cooling operation, from fully stratified (e.g., DV systems) to fully mixed (e.g., conventional overhead systems). For floor-level supply, thermal plumes that develop over heat sources in the room play a major role in driving overall floor-to-ceiling air motion. The amount of stratification in the room is primarily determined by the balance between total room airflow and heat load. In practice, the actual temperature (or concentration) profile depends on the combined effects of various factors, but is largely driven by the characteristics of the room supply airflow and heat load configuration.

For room supply airflow, the major factors are

- Total room supply airflow quantity
- Room supply air temperature
- Diffuser type
- Diffuser throw height (or outlet velocity); this is associated with the amount of mixing provided by a floor diffuser (or room conditions near a low-sidewall DV diffuser)

For room heat loads, the major factors are

- Magnitude and number of loads in space
- Load type (point or distributed source)
- Elevation of load (e.g., overhead lighting, person standing on floor, floor-to-ceiling glazing)
- Radiative/convective split
- For pollutant concentration profiles, whether pollutants are associated with heat sources

INDOOR AIR QUALITY AND SUSTAINABILITY

Air diffusion methods affect not only indoor air quality (IAQ) and thermal comfort, but also energy consumption over the building's life. Choices made early in the design process are important. The U.S. Green Building Council's (USGBC 2005) Leadership in Energy and Environmental Design (LEED®) rating system, which was originally created in response to indoor air quality concerns, now includes prerequisites and credits for increasing ventilation effectiveness and improving thermal comfort. These requirements and optional points are relatively easy to achieve if good room air diffusion design principles, methods, and standards are followed.

Environmental tobacco smoke (ETS) control is a LEED prerequisite. Banning indoor smoking is a common approach, but if indoor smoking is to be allowed, ANSI/ASHRAE *Standard* 62.1 requires that more than the base non-ETS ventilation air be provided where ETS is present in all or part of a building. Rock (2006) provides additional guidance on dealing with ETS.

The **air change effectiveness** is affected directly by the room air distribution system's design, construction, and operation, but is very difficult to predict. Many attempts have been made to quantify air change effectiveness, including ASHRAE *Standard* 129. However, this standard is only for experimental tests in well-controlled laboratories, and should not be applied directly to real buildings.

ANSI/ASHRAE *Standard* 62.1-2007 provides a table of typical values to help predict ventilation effectiveness. For example, well-designed ceiling-based air distribution systems produce near-perfect air mixing in cooling mode, and yield an air change effectiveness of almost 1.0.

Displacement and underfloor air distribution (UFAD) systems have the potential for values greater than 1.0. More information on ceiling- and wall-mounted air inlets and outlets can be found in Rock and Zhu (2002). Displacement system performance is described in Chen and Glicksman (2003). Bauman and Daly (2003)

The preparation of this chapter is assigned to TC 5.3, Room Air Distribution.

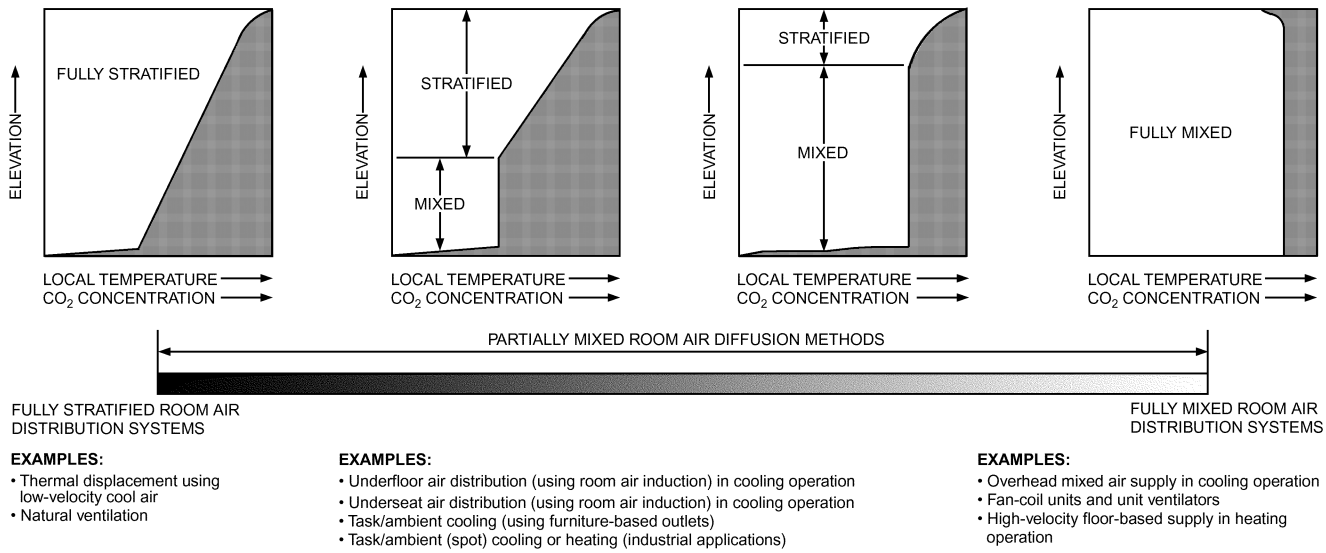


Fig. 1 Classification of Air Diffusion Methods

discuss UFAD in detail. (These three ASHRAE books were produced by research projects for Technical Committee 5.3.) More information on ANSI/ASHRAE *Standard* 62.1-2007 is available in its user's manual (ASHRAE 2007).

APPLICABLE STANDARDS AND CODES

The following standards and codes should be reviewed when applying various room air diffusion methods:

- ASHRAE *Standard* 55 specifies the combination of indoor thermal environmental factors and personal factors that will produce thermal acceptability to a majority of space occupants.
- ASHRAE *Standard* 62.1 establishes the ventilation requirements for acceptable indoor environmental quality. This standard is adopted as part of many building codes.
- ASHRAE/IESNA *Standard* 90.1 provides energy efficiency requirements that affect supply air characteristics.
- ASHRAE *Standard* 113 describes a method for evaluating the effectiveness of various room air distribution systems in achieving thermal comfort.
- ASHRAE *Standard* 129 specifies a method for measuring air-change effectiveness in mechanically ventilated spaces.

Local codes should also be checked to see how they apply to each of these subjects.

TERMINOLOGY

Adjacent zone. Area adjacent to an outlet in which long term occupancy is not recommended because of potential discomfort. Also called **clear** or **near zone**.

Aspect ratio. Ratio of length to width of opening or core of a grille.

Axial flow jet. Stream of air with motion approximately symmetrical along a line, although some spreading and drop or rise can occur from diffusion and buoyancy effects.

CAV. Constant air volume.

Coanda effect. Effect of a moving jet attaching to a parallel surface because of negative pressure developed between jet and surface.

Coefficient of discharge. Ratio of area at vena contracta to area of opening.

Cold air. General term for supply air, typically between 35 to 45°F.

Core area. Area of a register, grille, or linear slot pertaining to the frame or border, whichever is less.

Damper. Device used to vary the volume flow rate of air passing through a confined cross section by varying the cross-sectional area.

Diffuser. Outlet discharging supply air in various directions and planes.

Diffusion. Dispersion of air within a space.

Distribution. Moving air to or in a space by an outlet discharging supply air.

Draft. Undesired or excessive local cooling of a person caused by low temperature and air movement.

Drop. Vertical distance that the lower edge of a horizontally projected airstream descends between the outlet and the end of its throw.

Effective area. Net area of an outlet or inlet device through which air can pass; equal to the free area times the coefficient of discharge.

Entrainment. Movement of space air into the jet caused by the airstream discharged from the outlet (also known as secondary air motion).

Entrainment (or induction) ratio. Volume flow rate of total air (primary plus entrained air) divided by the volume flow rate of primary air at a given distance from the outlet.

Envelope. Outer boundary of an airstream moving at a perceptible velocity.

Exhaust opening or inlet. Any opening through which air is removed from a space.

Free area. Total minimum area of openings in an air outlet or inlet through which air can pass.

Grille. Functional or decorative device covering any area through which air passes.

Induction. See Entrainment.

Isothermal jet. Air jet with same temperature as surrounding air.

Lower (mixed) zone. In partially mixed systems, zone directly adjacent to floor, in which air is relatively well mixed.

Neck area. Nominal area of duct connection to air outlet or inlet.

Nonisothermal jet. Air jet with a discharge temperature different from surrounding air.

Occupied zone. Room volume where occupants are located (typically 6 ft above floor level and 1 ft from walls).

Outlet velocity. Average velocity of air emerging from outlet, measured in plane of opening.

Primary air. Air delivered to an outlet by a supply duct.

Radius of diffusion. Horizontal axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level (e.g., 50, 100, 150, or 200 fpm).

Register. Grille equipped with a flow control damper.

Spread. Divergence of airstream in horizontal and/or vertical plane after it leaves an outlet.

Stagnant zone. Area characterized by stratification and little air motion. This term does not necessarily imply poor air quality.

Stratification height. Vertical distance from floor to horizontal plane that defines lower boundary of upper mixed zone (in a fully stratified or partially mixed system).

Stratified zone. Zone in which air movement is entirely driven by buoyancy caused by convective heat sources. Typically found in fully stratified or partially mixed systems

Supply opening or outlet. Any opening or device through which supply air is delivered into a ventilated space being heated, cooled, humidified, or dehumidified. Supply outlets are classified according to their location in a room as sidewall, ceiling, baseboard, or floor outlets. However, because numerous designs exist, they are more accurately described by their construction features. (See Chapter 19 of the 2008 *ASHRAE Handbook—HVAC Systems and Equipment*.)

Terminal velocity. Maximum airstream velocity at end of throw.

Throw. Horizontal or vertical axial distance an airstream travels after leaving an air outlet before maximum stream velocity is reduced to a specified terminal velocity (e.g., 50, 100, 150, or 200 fpm), defined by ASHRAE *Standard* 70.

Total air. Mixture of discharged and entrained air.

Upper (mixed) zone. Zone in which air is relatively well mixed, with generally low average air velocities caused by the momentum of thermal plumes penetrating its lower boundary. Typically found in fully stratified or partially mixed systems.

Vane. Component of supply air outlet that imparts direction to the discharge jet.

Vane ratio. Ratio of depth of a vane to the space between two adjacent vanes.

VAV. Variable air volume.

Vena contracta. Smallest cross-sectional area of a fluid stream leaving an orifice.

PRINCIPLES OF JET BEHAVIOR

Air Jet Fundamentals

Air supplied to rooms through various types of outlets (e.g., grilles, ceiling diffusers, perforated panels) can be distributed by turbulent air jets (mixed and partially mixed systems) or in a low-velocity, unidirectional manner (stratified systems). The air jet discharged from an outlet is the primary factor affecting room air motion. Baturin (1972), Christianson (1989), and Murakami (1992) have further information on the relationship between the air jet and occupied zone.

If an air jet is not obstructed or affected by walls, ceiling, or other surfaces, it is considered a **free jet**.

Characteristics of the air jet in a room might be influenced by reverse flows created by the same jet entraining ambient air. If the supply air temperature is equal to the ambient room air temperature, the air jet is called an **isothermal jet**. A jet with an initial temperature different from the ambient air temperature is called a **nonisothermal jet**. The air temperature differential between supplied and ambient room air generates thermal forces (buoyancy) in jets, affecting the jet's (1) trajectory, (2) location at which it attaches to and separates from the ceiling/floor, and (3) throw. The significance of these effects depends on the ratio between the thermal buoyancy of the air and inertial forces.

Angle of Divergence. The angle of divergence is well defined near the outlet face, but the boundary contours are billowy and easily affected by external influences. Near the outlet, as in the room, air movement has local eddies, vortices, and surges. Internal forces governing this air motion are extremely delicate (Nottage et al. 1952a).

Measured angles of divergence (spread) for discharge into large open spaces usually range from 20 to 24°, with an average of 22°. Coalescing jets for closely spaced multiple outlets expand at smaller angles, averaging 18°, and jets discharging into relatively small spaces show even smaller angles of expansion (McElroy 1943). When outlet area is small compared to the dimensions of the space normal to the jet, the jet may be considered free as long as

$$X \le 1.5\sqrt{A_R} \tag{1}$$

where

X = distance from face of outlet, ft
A_R = cross-sectional area of confined space normal to jet, ft^2

Jet Expansion Zones. The full length of an air jet, in terms of the maximum or centerline velocity and temperature differential at the cross section, can be divided into four zones:

- **Zone 1**, a short core zone extending about four diameters or widths from the outlet face, in which the maximum velocity (temperature) of the airstream remains practically unchanged.
- **Zone 2**, a transition zone, with its length determined by the type of outlet, aspect ratio of the outlet, initial airflow turbulence, etc.
- **Zone 3**, a zone of fully established turbulent flow that may be 25 to 100 equivalent air outlet diameters (widths for slot air diffusers) long.
- **Zone 4**, a zone of diffuser jet degradation, where maximum air velocity and temperature decrease rapidly. Distance to this zone and its length depend on the velocities and turbulence characteristics of ambient air. In a few diameters or widths, air velocity becomes less than 50 fpm. Characteristics of this zone are still not well understood.

Zone 3 is of major engineering importance because, in most cases, the diffuser jet enters the occupied area within this zone.

Centerline Velocities in Zones 1 and 2. In zone 1, the ratio V_x/V_o is constant and equal to the ratio of the center velocity of the jet at the start of expansion to the average velocity. The ratio V_x/V_o varies from approximately 1.0 for rounded entrance nozzles to about 1.2 for straight pipe discharges; it has much higher values for diverging discharge outlets.

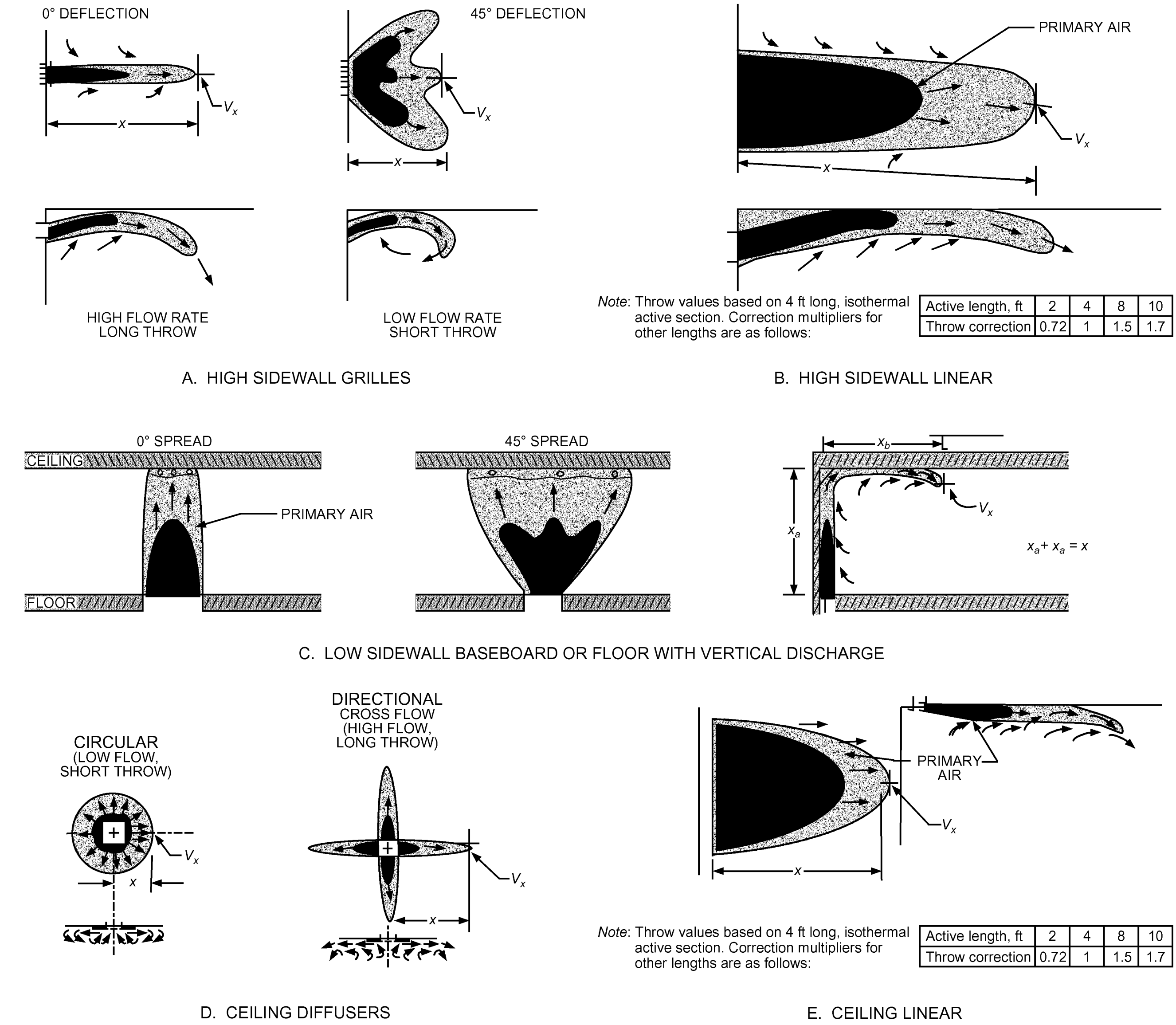
Experimental evidence indicates that, in zone 2,

$$\frac{V_x}{V_o} = \sqrt{\frac{K_c H_o}{X}} \tag{2}$$

where

V_x = centerline velocity at distance X from outlet, fpm
V_o = $V_c/C_d R_{fa}$ = average initial velocity at discharge from open-ended duct or across contracted stream at vena contracta of orifice or multiple-opening outlet, fpm
V_c = nominal velocity of discharge based on core area, fpm
C_d = discharge coefficient (usually between 0.65 and 0.90)
R_{fa} = ratio of free area to gross (core) area
H_o = width of jet at outlet or at vena contracta, ft
K_c = centerline velocity constant, depending on outlet type and discharge pattern (see Table 1)
X = distance from outlet to measurement of centerline velocity V_x, ft

The aspect ratio (Tuve 1953) and turbulence (Nottage et al. 1952a) primarily affect centerline velocities in zones 1 and 2. Aspect ratio has little effect on the terminal zone of the jet when H_o is greater than 4 in. This is particularly true of nonisothermal jets. When H_o is very small, induced air can penetrate the core of the jet, thus reducing centerline velocities. The difference in performance between a radial outlet with small H_o and an axial outlet with large H_o shows the importance of jet thickness.



Note: Airflow patterns shown with darker shading indicate primary air patterns for terminal velocities above 150 fpm.

Fig. 2 Airflow Patterns of Different Diffusers

Table 1 Recommended Values for Centerline Velocity Constant K_c for Commercial Supply Outlets

Outlet Type	Discharge Pattern	A_o	K_c
High sidewall grilles	0° deflection[a]	Free	5.7
(Figure 2A)	Wide deflection	Free	4.2
High sidewall linear	Core less than 4 in. high[b]	Free	4.4
(Figure 2B)	Core more than 4 in. high	Free	5.0
Low sidewall	Up and on wall, no spread	Free	4.5
(Figure 2C)	Wide spread[b]	Free	3.0
Baseboard	Up and on wall, no spread	Core	4.0
(Figure 2C)	Wide spread	Core	2.0
Floor grille	No spread[b]	Free	4.7
(Figure 2C)	Wide spread	Free	1.6
Ceiling (Figure 2D)	360° horizontal[c]	Neck	1.1
	Four-way; little spread	Neck	3.8
Ceiling linear slot (Figure 2E)	One-way; horizontal along ceiling[b]	Free	5.5

[a]Free area is about 80% of core area.
[b]Free area is about 50% of core area.
[c]Cone free area is greater than duct area.

When air is discharged from relatively large perforated panels, the constant-velocity core formed by coalescence of individual jets extends a considerable distance from the panel face. In zone 1, when the ratio is less than 5, use the following equation for estimating centerline velocities (Koestel et al. 1949):

$$V_x = 1.2 V_o \sqrt{C_d R_{fa}} \tag{3}$$

Centerline Velocity in Zone 3. In zone 3, maximum or centerline velocities of straight-flow isothermal jets can be determined accurately from the following equations:

$$\frac{V_x}{V_o} = \frac{K_c H_o}{X} = \frac{K_c \sqrt{A_o}}{X} \tag{4}$$

$$V_x = \frac{K_c V_o \sqrt{A_o}}{X} = \frac{K_c Q_o}{X \sqrt{A_o}} \tag{5}$$

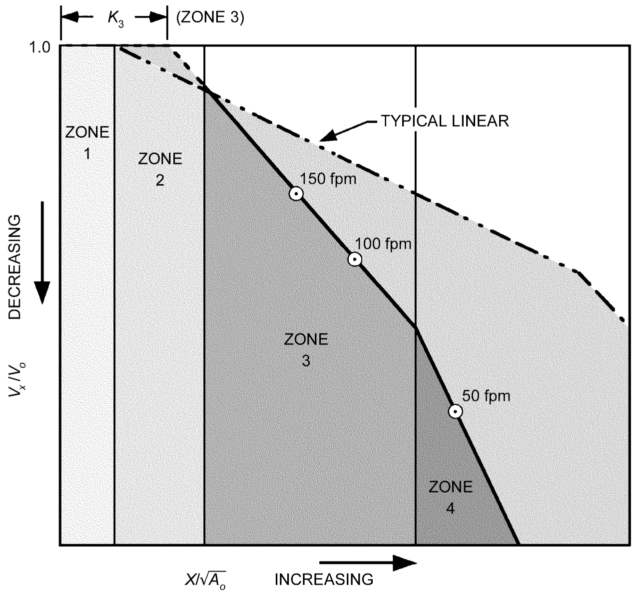


Fig. 3 Chart for Determining Centerline Velocities of Axial and Radial Jets

where

K_c = centerline velocity constant
H_o = effective or equivalent diameter of stream at discharge from open-ended duct or at contracted section, ft
A_o = core area or neck area as shown in Table 1, ft²
A_c = measured gross (core) area of outlet, ft²
Q_o = discharge from outlet, cfm

Because A_o equals the effective area of the stream, the flow area for commercial registers and diffusers, according to ASHRAE *Standard* 70, can be used in Equation (4) with the appropriate value of K.

Determining Centerline Velocities. To correlate data from all four zones, centerline velocity ratios are plotted against distance from the outlet in Figure 3.

Airflow patterns of diffusers are related to the throw K-factors and throw distance. In general, diffusers with a circular airflow pattern have a shorter throw than those with a directional or cross-flow pattern. During cooling, the circular pattern tends to curl back from the end of the throw toward the diffuser, reducing the drop and ensuring that the cool air remains near the ceiling.

Cross-flow airflow patterns have a longer throw, and the individual side jets react similarly to jets from sidewall grilles. Jets with this pattern have a longer throw, and airflow does not roll back to the diffuser at the end of the throw, but continues to move away from the diffuser at low velocities.

Throw. Equation (5) can be transposed to determine the throw X of an outlet if the discharge volume and the centerline velocity are known:

$$X = \frac{K_c Q_o}{V_x \sqrt{A_o}} \tag{6}$$

The following example illustrates the use of Table 1 and Figure 3.

Example 1. A 12 by 18 in. high sidewall grille with an 11.25 by 17.25 in. core area is selected. From Table 1, K_c = 5 for zone 3. If the airflow is 600 cfm, what is the throw to 50, 100, and 150 fpm?

Solution:

From Equation (6),

$$X = \frac{K_c Q_o}{V_x \sqrt{A_o}} = \frac{5 \times 600}{V_x \sqrt{11.25 \times 17.25 / 144}} = \frac{2920}{V_x}$$

Solving for 50 fpm throw,

$$X = 2920/50 = 58.4 \text{ ft}$$

But, according to Figure 3, 50 fpm is in zone 4, which is typically 20% less than calculated in Equation (4), or

$$X = 58.4 \times 0.80 = 47 \text{ ft}$$

Solving for 100 fpm throw,

$$X = 2920/100 = 29 \text{ ft}$$

Solving for 150 fpm throw,

$$X = 2920/150 = 19 \text{ ft}$$

Velocity Profiles of Jets. In zone 3 of both axial and radial jets, the velocity distribution may be expressed by a single curve (Figure 3) in terms of dimensionless coordinates; this same curve can be used as a good approximation for adjacent portions of zones 2 and 4. Temperature and density differences have little effect on cross-sectional velocity profiles.

Velocity distribution in zone 3 can be expressed by the Gauss error function or probability curve, which is approximated by the following equation:

$$\left(\frac{r}{r_{0.5V}}\right)^2 = 3.3 \log \frac{V_x}{V} \tag{7}$$

where

r = radial distance of point under consideration from centerline of jet
$r_{0.5V}$ = radial distance in same cross-sectional plane from axis to point where velocity is one-half centerline velocity (i.e., $V = 0.5V_x$)
V_x = centerline velocity in same cross-sectional plane
V = actual velocity at point being considered

Experiments show that the conical angle for $r_{0.5V}$ is approximately one-half the total angle of divergence of a jet. The velocity profile curve for one-half of a straight-flow turbulent jet (the other half being a symmetrical duplicate) is shown in Figure 4. For multiple-opening outlets, such as grilles or perforated panels, the velocity profiles are similar, but the angles of divergence are smaller.

Entrainment Ratios. The following equations are for entrainment of circular jets and of jets from long slots. For third-zone expansion of circular jets,

$$\frac{Q_x}{Q_o} = \frac{2X}{K_c \sqrt{A_o}} \tag{8}$$

By substituting from Equation (4),

$$\frac{Q_x}{Q_o} = 2\frac{V_o}{V_x} \tag{9}$$

For a continuous slot with active sections up to 10 ft and separated by 2 ft,

$$\frac{Q_x}{Q_o} = \sqrt{\frac{2}{K_c}}\sqrt{\frac{X}{H_s}} \tag{10}$$

or, substituting from Equation (2),

$$\frac{Q_x}{Q_o} = \sqrt{2}\frac{V_o}{V_x} \tag{11}$$

where

Q_x = total volumetric flow rate at distance X from face of outlet, cfm
Q_o = discharge from outlet, cfm
X = distance from face of outlet, ft
K_c = centerline velocity constant

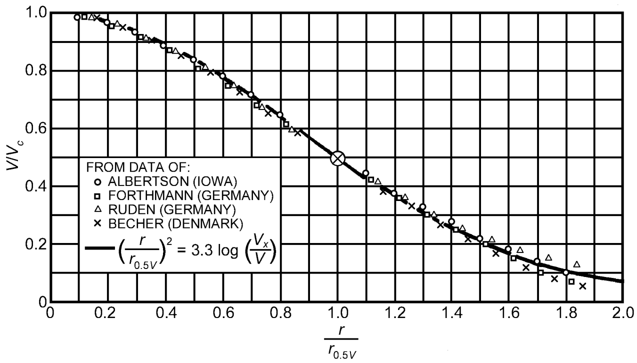


Fig. 4 Cross-Sectional Velocity Profiles for Straight-Flow Turbulent Jets

A_o = core area or neck area free (see Table 1), ft
H_s = width of slot, ft

The entrainment ratio Q_x/Q_o is important in determining total air movement at a given distance from an outlet. For a given outlet, the entrainment ratio is proportional to the distance X [Equation (8)] or to the square root of the distance X [Equation (10)] from the outlet. Equations (9) and (11) show that, for a fixed centerline velocity V_x, the entrainment ratio is proportional to outlet velocity. Equations (9) and (11) also show that, at a given centerline and outlet velocity, a circular jet has greater entrainment and total air movement than a long slot. Comparing Equations (8) and (10), the long slot should have a greater rate of entrainment. The entrainment ratio at a given distance is less with a large K than with a small K.

Isothermal Radial Flow Jets

In a radial jet, as with an axial jet, the cross-sectional area at any distance from the outlet varies as the square of this distance. Centerline velocity gradients and cross-sectional velocity profiles are similar to those of zone 3 of axial jets, and the angles of divergence are about the same.

A jet from a ceiling plaque has the same form as half of a free radial jet. The jet is wider and longer than a free jet, with maximum velocity close to the surface. Koestel (1957) provides an equation for radial flow outlets.

Nonisothermal Jets

When the temperature of introduced air is different from the room air temperature, the diffuser air jet is affected by thermal buoyancy caused by air density difference. The trajectory of a nonisothermal jet introduced horizontally is determined by the Archimedes number (Baturin 1972):

$$\mathrm{Ar} = \frac{gL_o(T_o - T_A)}{V_o^2 T_A} \tag{12}$$

where

g = gravitational acceleration rate, ft/min^2
L_o = length scale of diffuser outlet equal to hydraulic diameter of outlet, ft
$(T_o - T_A)$= initial temperature of jet – temperature of ambient air, °F
V_o = initial air velocity of jet, fpm
T_A = room air temperature, °R

The influence of buoyant forces on horizontally projected heated and chilled jets is significant in heating and cooling with wall outlets. Koestel's (1955) equation describes the behavior of these jets.

Helander and Jakowatz (1948), Helander et al. (1953, 1954, 1957), Knaak (1957), and Yen et al. (1956) developed equations for outlet characteristics that affect the downthrow of heated air. Koestel (1954, 1955) developed equations for temperatures and velocities in heated and chilled jets. Kirkpatrick and Elleson (1996) and Li et al. (1993, 1995) provide additional information on nonisothermal jets.

Nonisothermal Horizontal Free Jet

A horizontal free jet rises or falls according to the temperature difference between it and the ambient environment. The horizontal jet throw to a given distance follows an arc, rising for heated air and falling for cooled air. The distance from the diffuser to a given terminal velocity along the discharge jet remains essentially the same.

Comparison of Free Jet to Attached Jet

Most manufacturers' throw data obtained in accordance with ASHRAE *Standard* 70 assume the discharge is attached to a surface. An attached jet induces air along the exposed side of the jet, whereas a free jet can induce air on all its surfaces. Because a free jet's induction rate is larger compared to that of an attached jet, a free jet's throw distance will be shorter. To calculate the throw distance X for a noncircular free jet from catalog data for an attached jet, the following estimate can be used.

$$X_{free} = X_{attached} \times 0.707 \tag{13}$$

Circular free jets generally have longer throws compared to noncircular jets.

Jets from ceiling diffusers initially tend to attach to the ceiling surface, because of the force exerted by the Coanda effect. However, cold air jets will detach from the ceiling if the airstream's buoyancy forces are greater than the inertia of the moving air stream.

With separation, a cold draft may enter the occupied space, resulting in thermal discomfort. The thermal discomfort is caused by two factors: the cold draft of the separated jet in the occupied space, and the lack of adequate mixing in areas of the room not reached by the separated jet. The separation distance parameter x_s is the distance from the diffuser at which a jet separates from the ceiling.

Separation distance correlates with outlet jet conditions. Separation distance depends on the velocity constant K, outlet temperature, flow rate, and static pressure drop. For slot and round diffusers,

$$x_s = (11.91)(1.2)K^{1/2}(\Delta T/T)^{-1/2}Q_o^{\,1/4}\Delta P^{\,3/8} \tag{14}$$

where

x_s = jet separation distance, ft
K_c = centerline velocity constant
ΔT = room-jet temperature difference, °F
T = average absolute room temperature, °R
Q_o = outlet flow rate, cfm
ΔP = diffuser static pressure drop, in. of water

A representative value of C_s that has been found to best match the results of analyses and experiments of a wide variety of diffusers is 1.2.

Surface Jets (Wall and Ceiling)

Attached jets travel at a higher velocity and entrain less air than a free jet. Values of centerline velocity constant K are approximately those for a free jet multiplied by $\sqrt{2}$; that is, the normal maximum of 6.2 for K for free jets becomes 8.8 for a similar jet discharged parallel to an adjacent surface.

When a jet is discharged parallel to but at some distance from a solid surface (wall, ceiling, or floor), its expansion in the direction of the surface is reduced, and entrained air must be obtained by recirculation from the jet instead of from ambient air (McElroy 1943; Nottage et al. 1952b; Zhang et al. 1990). The restriction to entrainment caused by the solid surface induces the **Coanda effect**, which makes the jet attach to a surface a short distance after it leaves the diffuser outlet. The jet then remains attached to the surface for some distance before separating again.

In nonisothermal cases, the jet's trajectory is determined by the balance between thermal buoyancy and the Coanda effect, which depends on jet momentum and distance between the jet exit and solid surface. The behavior of such nonisothermal surface jets has been studied by Kirkpatrick et al. (1991), Oakes (1987), Wilson et al. (1970), and Zhang et al. (1990), each addressing different factors. More systematic study of these jets in room ventilation flows is needed to provide reliable guidelines for designing air distribution systems.

Multiple Jets

Twin parallel air jets act independently until they interfere. The point of interference and its distance from outlets vary with the distance between outlets. From outlets to the point of interference, maximum velocity, as for a single jet, is on the centerline of each jet. After interference, velocity on a line midway between and parallel to the two jet centerlines increases until it equals jet centerline velocity. From this point, maximum velocity of the combined jet stream is on the midway line, and the profile seems to emanate from a single outlet of twice the area of one of the two outlets.

Airflow in Occupied Zone

Mixing Systems. Laboratory experiments on jets usually involve recirculated air with negligible resistance to flow on the return path. Experiments in small-cross-sectional mine tunnels, where return flow meets considerable resistance, show that jet expansion terminates abruptly at a distance that is independent of discharge velocity and is only slightly affected by outlet size. These distances are determined primarily by the return path's size and length. In a long tunnel with a cross section of 5 by 6 ft, a jet may not travel more than 25 ft; in a tunnel with a relatively large section (25 by 60 ft), the jet may travel more than 250 ft. McElroy (1943) provides data on this phase of jet expansion.

Zhang et al. (1990) found that, for a given heat load and room air supply rate, air velocity in the occupied zone increases when outlet discharge velocity increases. Therefore, the design supply air velocity should be high enough to maintain the jet traveling in the desired direction, to ensure good mixing before it reaches the occupied zone. Excessively high outlet air velocity induces high air velocity in the occupied zone and results in thermal discomfort.

Turbulence Production and Transport. Air turbulence in a room is mainly produced at the diffuser jet region by interaction of supply air with room air and with solid surfaces (walls or ceiling) in the vicinity. It is then transported to other parts of the room, including the occupied zone (Zhang et al. 1992). Turbulence is also damped by viscous effect. Air in the occupied zone usually contains very small amounts of turbulent kinetic energy compared to the jet region. Because turbulence may cause thermal discomfort (Fanger et al. 1989), air distribution systems should be designed so that stationary occupants are not subjected to the region where primary mixing between supply and room air occurs (except in specialized applications such as task ambient or spot-conditioning systems).

SYSTEM DESIGN

MIXED-AIR SYSTEMS

In mixed-air systems, high-velocity supply jets from air outlets maintain comfort by mixing room air with supply air. This air mixing, heat transfer, and resultant velocity reduction should occur outside the occupied zone. Occupant comfort is maintained not directly by motion of air from the outlets, but from secondary air motion that results from mixing in the unoccupied zone. Comfort is maximized when uniform temperature distribution and room air velocities of less than 50 fpm are maintained in the occupied zone.

Outlet Types

Straub and Chen (1957) and Straub et al. (1956) classified outlets into five groups:

Group A. Outlets mounted in or near the ceiling that discharge air horizontally.

Group B. Outlets mounted in or near the floor that discharge air vertically in a nonspreading jet.

Group C. Outlets mounted in or near the floor that discharge air vertically in a spreading jet.

Group D. Outlets mounted in or near the floor that discharge air horizontally.

Group E. Outlets mounted in or near the ceiling that project primary air vertically.

Analysis of outlet performance was based on primary air pattern, total air pattern, stagnant air layer, natural convection currents, return air pattern, and room air motion. Figures 5 to 9 show room air motion characteristics of the five outlet groups; exterior walls are depicted by heavy lines. The principles of air diffusion emphasized by these figures are as follows:

- Primary air (shown by dark envelopes in Figures 5 to 9) from the outlet down to a velocity of about 150 fpm can be treated analytically. Heating or cooling load has a strong effect on the characteristics of primary air.
- Total air, shown by light gray envelopes in Figures 5 to 9, is influenced by primary air and is of relatively high velocity (but less than 150 fpm). Total air is also influenced by the environment and drops during cooling or rises during heating; it is not subject to precise analytical treatment.
- Natural convection currents form a stagnant zone from the ceiling down during cooling, and from the floor up during heating. This zone forms below the terminal point of the total air during heating and above the terminal point during cooling. Because this zone results from natural convection currents, its air velocities are usually low (approximately 20 fpm), and the air stratifies in layers of increasing temperatures. The concept of a stagnant zone is important in properly applying and selecting outlets because it considers the natural convection currents from warm and cold surfaces and internal loads.
- A return inlet affects room air motion only in its immediate vicinity. The intake should be located in the stagnant zone to return the warmest room air during cooling or the coolest room air during heating. The importance of the location depends on the relative size of the stagnant zone, which depends on the type of outlet.
- The general room air motion (shown by arrows in white areas in Figures 5 to 9) is a gentle drift toward the total air. Room conditions are maintained by entraining room air into the total airstream. The room air motion between the stagnant zone and the total air is relatively slow and uniform. The highest air motion occurs in and near the total airstreams.

Group A Outlets. This group includes high sidewall grilles, sidewall diffusers, ceiling diffusers, linear ceiling diffusers, and similar outlets. High sidewall grilles and ceiling diffusers are illustrated in Figure 5.

Primary air envelopes (**isovels**) show a horizontal, two-jet pattern for the high sidewall and a 360° diffusion pattern for the ceiling outlet. Although variation of vane settings might cause a discharge in one, two, or three jets in the case of the sidewall outlet, or have a smaller diffusion angle for the ceiling outlet, the general effect in each is the same.

During cooling, the total air drops into the occupied zone at a distance from the outlet that depends on air quantity, supply velocity, temperature differential between supply and room air, deflection setting, ceiling effect, and type of loading within the space. Analytical

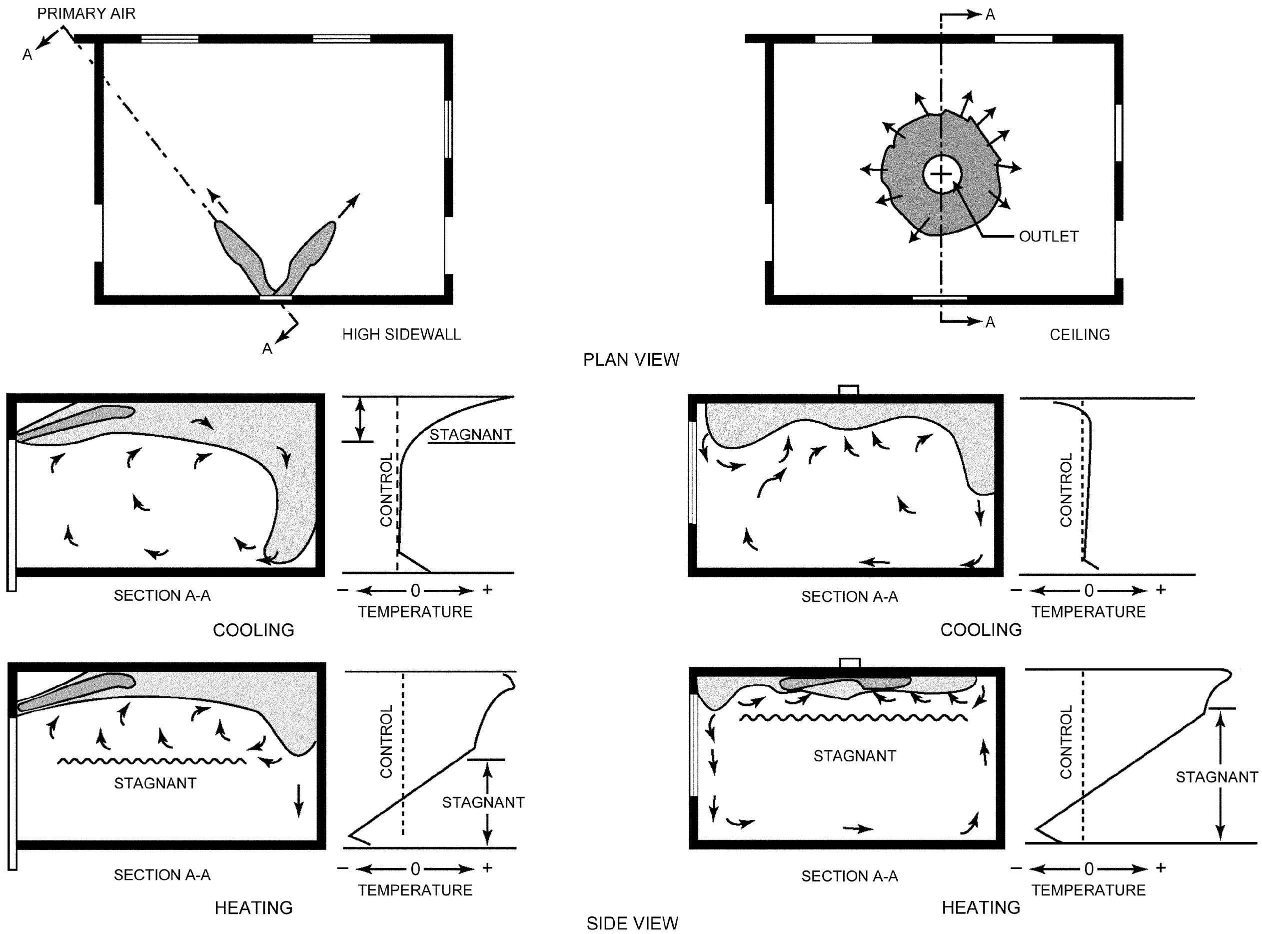


Fig. 5 Air Motion Characteristics of Group A Outlets
(Straub et al. 1956)

methods of relating some of these factors are presented in the section on Principles of Jet Behavior.

The cooling diagram for the high sidewall outlet shows an overthrow condition, which causes the total air to drop along the opposite wall and flow slowly for some distance across the floor. Velocities of about 100 to 150 fpm may be found near the wall but dissipate within about 4 in. of the wall.

The cooling diagram for the ceiling outlet shows that total air movement is counteracted by rising natural convection currents on the heated wall, and, therefore, drops before reaching the wall. On the other hand, the total air reaches the inside wall and descends for some distance along it. With this type of outlet, temperature variations in the room are minimized, with minimal stagnant volume. The maximum velocity and maximum temperature variation occur in and near the total air envelope; therefore, the drop region becomes important because it is an area with high effective draft temperature T_{ed} [see Equation (18)]. Consequently, how far the air drops before velocities and temperatures reach acceptable limits must be known.

Because these outlets discharge horizontally near the ceiling, the warmest air in the room is mixed immediately with cool primary air far above the occupied zone. Therefore, the outlets are capable of handling relatively large quantities of air at large temperature differentials.

During heating, warm supply air introduced at the ceiling can cause stratification in the space if there is insufficient induction of room air at the outlet. Selecting diffusers properly, limiting the room supply temperature differential, and maintaining air supply rates at a level high enough to ensure air mixing by induction provide adequate air diffusion and minimize stratification.

Several building codes and ASHRAE *Standard* 90.1 require sufficient insulation in exterior walls, so most perimeter spaces can be heated effectively by ceiling air distribution systems. Interior spaces, which generally have only cooling demand conditions, seldom require long-term heating and are seldom a design problem.

Flow rate and velocity for both heating and cooling are the same for the outlets shown in Figure 5. The heating diagram for the sidewall unit shows that, under these conditions, total air does not descend along the wall. Consequently, higher velocities might be beneficial in eliminating the stagnant zone, because high velocity causes some warm air to reach floor level and counteract stratification of the stagnant region.

The heating diagram for the ceiling outlet shows the effect of natural convection currents that produce a larger throw toward the cold exposed wall. The velocity of total air toward the exposed wall complements natural convection currents. However, the warm total air loses its downward momentum at its terminal point, and buoyancy forces cause it to rise toward the primary air. Although these forces are complementary, the heating effect of total air replaces cool natural convection currents with warm total air.

Group B Outlets. This group includes floor registers, baseboard units, low sidewall units, linear-type grilles in the floor or windowsill, and similar outlets. Figure 6 illustrates a floor outlet adjacent to an inside wall.

Because these outlets have no deflecting vanes, primary air is discharged in a single, vertical jet. When total air strikes the ceiling,

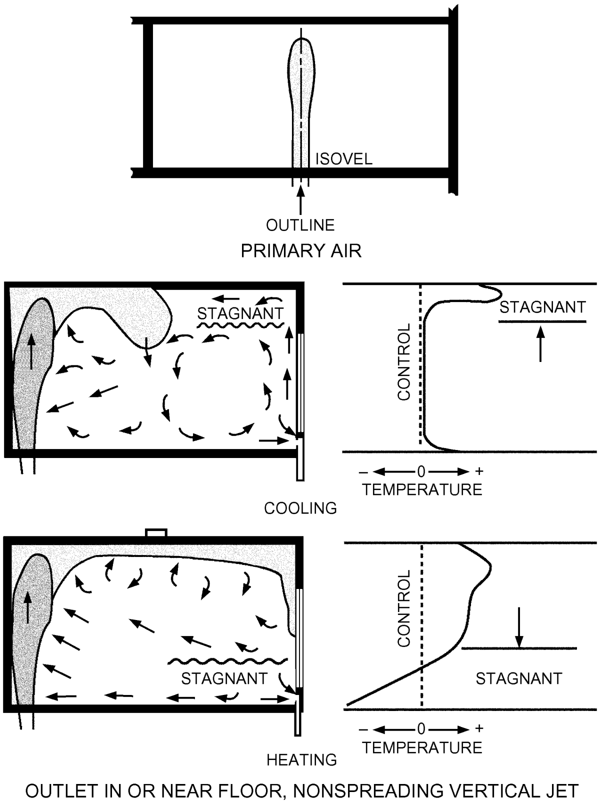


Fig. 6 Air Motion Characteristics of Group B Outlets
(Straub et al. 1956)

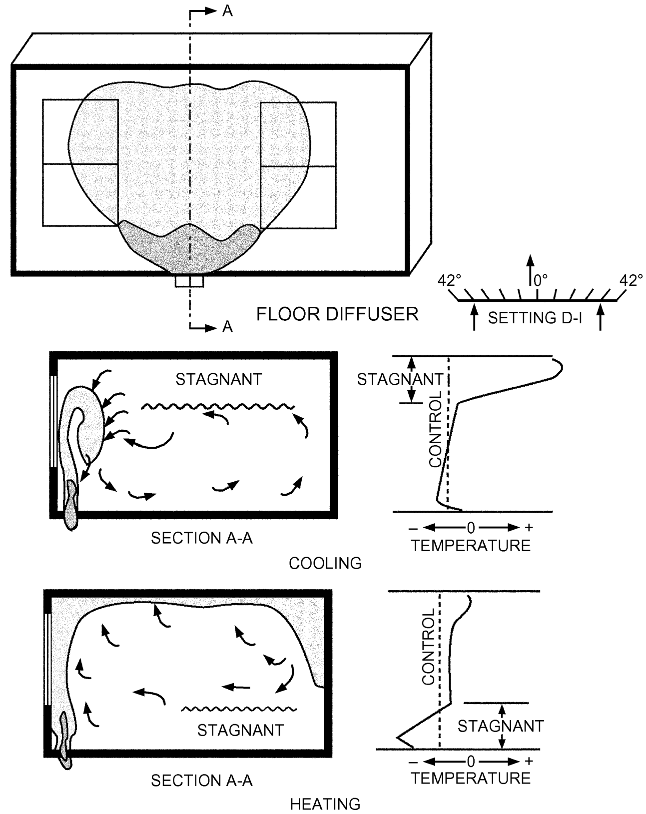


Fig. 7 Air Motion Characteristics of Group C Outlets
(Straub et al. 1956)

it fans out in all directions and, during cooling, follows the ceiling for some distance before dropping toward the occupied zone. During heating, the total airflow follows the ceiling across the room, then descends partway down the exterior wall.

The cooling diagram shows that a stagnant zone forms outside the total air region above its terminal point. Below the stagnant zone, air temperature is uniform, effecting complete cooling. Also, the space below the terminal point of total air is cooled satisfactorily. For example, if total airflow is projected upward for 8 ft, the region from this level down to the floor will be cooled satisfactorily. However, this does not apply to an extremely large space. Judgment is needed to determine the acceptable size of the space outside the total air. A distance of 15 to 20 ft between the drop region and the exposed wall is a conservative design value.

Comparison of Figures 5 and 6 for heating shows that the stagnant region is smaller for group B outlets than for group A outlets because air entrained close to the outlet is taken mainly from the stagnant region, which is the coolest air in the room. This results in greater temperature equalization and less buoyancy in the total air than occurs with group A outlets.

Although temperature gradients for both outlet groups are about the same, the stagnant layer is lower for group B than for group A.

Group C Outlets. This group includes floor diffusers, sidewall diffusers, linear-type diffusers, and other outlets installed in the floor or windowsill (Figure 7).

Although group C outlets are related to group B outlets, they are characterized by wide-spreading jets and diffusing action. Total air and room air characteristics are similar to those of group B, although the stagnant zone is larger during cooling and smaller during heating. Primary air diffusion usually causes the total air to fold back on the primary and total air during cooling, instead of following the ceiling. This makes it more difficult to project cool air, but it also provides a greater area for induction of room air. This is beneficial during heating because induced air comes from the lower regions of the room.

Group D Outlets. This group includes baseboard and low sidewall registers and similar outlets (Figure 8) that discharge primary air in single or multiple jets. During cooling, because air is discharged horizontally across the floor, the total air remains near the floor, and a large stagnant zone forms in the entire upper region of the room.

During heating, the total air rises toward the ceiling because of the buoyant effect. Temperature variations are uniform, except in the total air region.

Group E Outlets. This group includes ceiling diffusers, linear grilles, sidewall diffusers and grilles, and similar outlets mounted or designed for vertical downward air projection. Figure 9 shows the heating and cooling diagrams for such a ceiling diffuser.

During cooling, the total air projects to and follows the floor, producing a stagnant region near the ceiling. During heating, the total airflow reaches the floor and folds back toward the ceiling. If projected air does not reach the floor, a stagnant zone results.

Outlet Selection and Location

The design of a mixed-air distribution system is influenced by the same factors that affect design of an air-conditioning plant: building use, size, and construction type. Location and selection of supply outlets is further influenced by the interior design of the building, local sources of heat gain or loss, and outlet performance and design.

Local sources of heat gain or loss promote convection currents or cause stratification; they may, therefore, determine both the type and location of supply outlets. Outlets should be located to neutralize any undesirable convection currents set up by a concentrated load. If a concentrated heat source is located in the occupied zone, the heating effect can be counteracted by (1) directing cool air toward the source

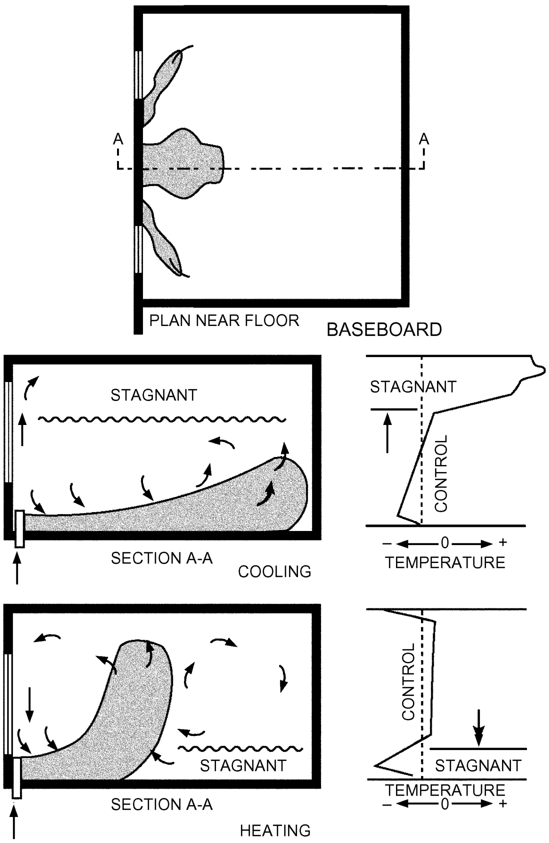


Fig. 8 Air Motion Characteristics of Group D Outlets
(Straub et al. 1956)

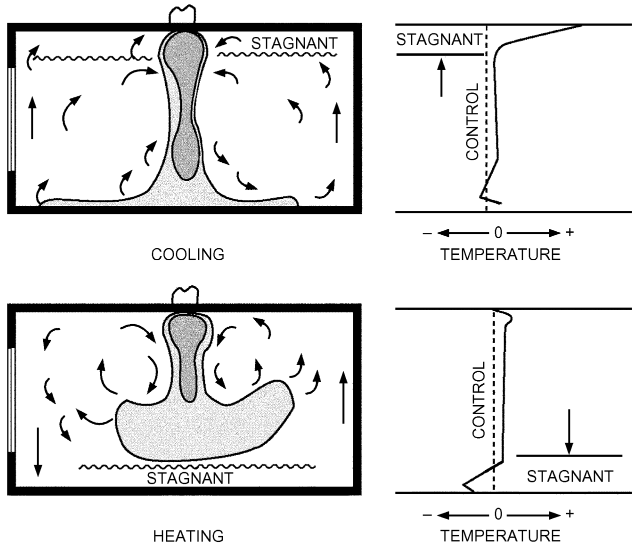


Fig. 9 Air Motion Characteristics of Group E Outlets
(Straub et al. 1956)

or (2) locating an exhaust or return grille adjacent to the source (more economical for cooling applications, because heat is withdrawn at its source rather than dissipated into the space). Where lighting loads are heavy (5 W/ft^2) and ceilings relatively high (above 15 ft), outlets should be located below the lighting load, and the stratified warm air should be removed by an exhaust or return fan. An exhaust fan is recommended if the wet-bulb air temperature is above that of the outdoors; a return fan is recommended if the wet-bulb temperature is below this temperature. These methods reduce the requirements for supply air.

The following selection considerations for outlets in groups A through E are based on analysis of outlet performance tests conducted by Straub and Chen (1957) and Straub et al. (1956).

Group A Outlets. Outlets mounted in or near the ceiling with horizontal air discharge should not be used with temperature differentials exceeding 25°F during heating. Hart and Int-Hout (1980) and Lorch and Straub (1983) recommended that temperature differentials not exceed 15°F during heating. Consequently, these outlets should be used for heating buildings in regions where winter heating is only a minor problem and, in northern latitudes, solely for interior spaces. However, these outlets are particularly suited for cooling and can be used with high airflow rates and large temperature differentials. They are usually selected for their cooling characteristics.

Performance is affected by various factors. Vane deflection settings reduce throw and drop by changing air from a single straight jet to a wide-spreading or fanned-out jet. Accordingly, a sidewall outlet with 0° deflection has a longer throw and a greater drop than a ceiling diffuser with a single 360° angle of deflection. Sidewall grilles and similar outlets with other deflection settings may have performance characteristics between these two extremes.

Wide deflection settings also cause a ceiling effect, which increases throw and decreases drop. To prevent smudging, total air should be directed away from the ceiling, but this is rarely practicable, except for very high ceilings. For optimum air diffusion in areas with normal ceilings, total air should scrub the ceiling surface.

Drop increases and throw decreases with larger temperature differentials. For constant temperature differential, airflow rate affects drop more than velocity. Therefore, to avoid drop, several small outlets in a room may be better than one large outlet.

With the data in the section on Principles of Jet Behavior, throw may be selected for part of the distance between outlet and wall or, preferably, for the entire distance. For outlets in opposite walls, throw should be one-half the distance between the walls. Following these recommendations, the air drops before striking the opposite wall or the opposing airstream. To counteract specific sources of heat gain or to provide higher air motion in rooms with high ceilings, a longer throw may be necessary. In no case should the drop exceed the distance from the outlet to the 6 ft level.

To maintain maximum ventilation effectiveness with ceiling diffusers, throws should be kept as long as possible. With VAV designs, some overthrow at maximum design volumes is desirable; the highest induction can be maintained at reduced flows. Adequate induction by a ceiling-mounted diffuser prevents short-circuiting unmixed supply air between supply outlet and ceiling-mounted returns.

Group B Outlets. In selecting these outlets, it is important to provide enough throw to project air high enough for proper cooling in the occupied zone. Increased supply air velocity improves air diffusion during both heating and cooling. Also, a terminal velocity of about 150 fpm is found at the same distance from the floor during both heating and cooling. Therefore, outlets should be selected from data given in the section on Principles of Jet Behavior, with throw based on a terminal velocity of 150 fpm.

With outlets installed near the exposed wall, primary air is drawn toward the wall, resulting in a wall effect similar to the ceiling effect for ceiling outlets. This scrubbing of the wall increases heat gain or loss. To reduce scrubbing, outlets should be installed some distance from the wall, or supply air should be deflected away from the wall. However, to prevent air from dropping into the occupied zone before it reaches maximum projection, the distance should not be too large nor the angle too wide. A distance of 6 in. and an angle of 15° is satisfactory.

These outlets do not counteract natural convection currents unless they are installed in sufficient numbers around the space perimeter, preferably in locations of greatest heat gain or loss (under windows). The effect of drapes and blinds must be considered with outlets installed near windows. Correctly installed, these outlets handle large airflow rates with uniform air motion and temperatures.

Group C Outlets. These outlets can be used for heating, even with severe heat load conditions. Higher supply velocities produce better room air diffusion than lower velocities, but velocity is not critical in selecting these units for heating.

To achieve required projection for cooling, use temperature differentials of less than 15°F. With higher temperature differentials, supply air velocity is not sufficient to project the total air up to the desired level.

These outlets have been used successfully for residential heating, but they may also offer a solution for applications where heating requirements are severe and cooling requirements are moderate. For throw, refer to the section on Principles of Jet Behavior.

Group D Outlets. These outlets direct high-velocity total air into the occupied zone, and, therefore, are not recommended for comfort, particularly for summer cooling. For heating, outlet velocities should not be higher than 300 fpm, so that air velocities in the occupied zone will not be excessive. These outlets have been applied successfully to process installations where controlled air velocities are desired.

Group E Outlets. The different throws shown in the heating and cooling diagrams for these outlets become critical in selecting and applying the outlets. Because the total air enters the occupied zone for both cooling and heating, outlets are used for either cooling or heating, but seldom for both.

During cooling, temperature differential, supply air velocity, and airflow rate strongly influence projection. Therefore, low values of each should be selected.

During heating, it is important to select the correct supply air velocity to project warm air into the occupied zone. Temperature differential is also critical because a small temperature differential reduces variation of throw during cyclic fluctuation of the supply air temperature. Vane setting for deflection is as important here as it is for group B and C outlets.

Investigations by Miller and Nevins (1969) and Nevins and Ward (1968) in full-scale interior test rooms indicate that air temperatures and velocities throughout a room cooled by a ventilating ceiling are a linear function of room load (heat load per unit area), and are not affected significantly by variations in ceiling type, total air temperature differential, or air volumetric flow rate. Higher room loading produces wider room air temperature variations and higher velocities, which decrease performance.

These studies also found no appreciable difference in the performance of air-diffusing ceilings and circular ceiling diffusers for lower room loads (20 $Btu/h \cdot ft^2$). For higher room loads (80 $Btu/h \cdot ft^2$), an air-diffusing ceiling system has only slightly larger vertical temperature variations and slightly lower room air velocities than a ceiling diffuser system.

When the ventilating ceiling is used at exterior exposures, the additional load at the perimeter must be considered. During heating, the designer must provide for the cold-wall effect (radiation, convection, and conduction loads), as with any ceiling supply distribution system. Sound generated by the air supply device must also be considered in total system analysis to ensure that room sound levels do not exceed the design criteria.

Noise. Noise generated by diffusers transmits to the occupied space directly and cannot be attenuated. Therefore, the distribution system design should meet the sound level criteria specified in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Inlet Selection and Location

Selection. Selection of return and exhaust inlets depends on (1) velocity in the occupied zone near the inlets, (2) permissible pressure drop through the inlets, and (3) noise.

Velocity. Airflow patterns and room air movement are not influenced by the location of return and exhaust inlets beyond a distance of one characteristic length of the return or exhaust inlet (e.g., square root of the inlet area). Air handled by the inlet approaches from all directions, and its velocity decreases rapidly as distance from the inlet increases. Therefore, drafty conditions rarely occur near return inlets.

Permissible pressure drop. Permissible pressure drop depends on the designer's choice. Proper pressure drop allowances should be made for control or directive devices.

Noise. Noise generation and transmission through return inlets should also be taken into account in space acoustical space calculations.

Location. Inlets should be located to minimize short-circuiting of supply air, although tests conducted under ASHRAE *Standard* 129 show little short circuiting with cold ceiling supply and return air. If air is supplied by jets attached to the ceiling, exhaust inlets should be located between the jets or at the side of the room, away from supply air jets. In rooms with vertical temperature stratification, such as foundries, computer rooms, theaters, bars, kitchens, dining rooms, and club rooms, exhaust inlets should be located near the ceiling to collect warm air, odors, and fumes.

For industrial rooms with gas release, selection of exhaust inlet locations depends on the density of released gases and their temperature; locations should be specified for each application.

Exhaust inlets located in walls and doors, depending on their elevation, have the characteristics of either floor or ceiling returns. In large buildings with many small rooms, return air may be brought through door grilles or door undercuts into the corridors and then to a common return or exhaust. If pressure drop through door returns is excessive, air diffusion to the room may be seriously unbalanced by opening or closing doors. Outward leakage through doors or windows cannot be counted on for dependable results.

Ceiling-Based Air Diffusion

For the best thermal comfort conditions and highest ventilation effectiveness in an occupied space (e.g., office or retail store), the entire system performance of air diffusers should be considered. This is particularly true for open spaces, where airstreams from diffusers may interact with each other, and for perimeter spaces, where airstreams from diffusers interact with hot or cold perimeter walls. Although throw data for individual diffusers are used in system design, a mixed-air distribution system should maintain a high quality of air diffusion in the occupied space with low temperature variation, good air mixing, and no objectionable drafts in the occupied space (typically 6 in. to 6 ft above the floor).

Adequate ventilation requires that the selected diffusers effectively mix (by entrainment) the total air in the room with the conditioned supply air, which is assumed to contain adequate ventilation air.

Interior Spaces. An interior space is conditioned exclusively for cooling loads, except after unoccupied periods when the space may have cooled to below a comfortable temperature. Tests by Hart and Int-Hout (1981), Miller (1979), Miller and Nash (1971), and Miller and Nevins (1970) suggest that the air diffusion performance index (ADPI) (see the section on ADPI under System Performance Evaluation) can be improved by moving diffusers closer together (i.e., specifying more diffusers for a given space and air quantity) and by limiting the supply air/room air temperature difference. In a given system of diffusers, these studies found an optimum operating range of air volumetric flow rates at a given thermal load. Operating load varies with diffuser design, ceiling height, thermal load, and diffuser orientation. This information can be obtained by constructing a mock-up representing the proposed building space, with several alternatives tested for ADPI values, in accordance with ASHRAE *Standard* 113. Usually, the diffuser manufacturer performs these tests and can provide the best choice of design options for a particular building. For a VAV system, diffuser spacing selection should not be based on maximum or design air volumes, but rather on the air volume range in which the system is expected to operate most of the time. For VAV applications, Miller (1979) recommends that the designer consider the expected variation in outlet air volume to ensure that ADPI values remain above a specified minimum. An ADPI of 80% or greater ensures that the space complies with the ASHRAE *Standard* 55 limit of 5.4°F in the occupied zone.

For an office environment in cooling mode, the design goal should be an ADPI greater than 80. The ADPI should not be used as a measure of performance for heating conditions. In both cases, ASHRAE *Standard* 55 recommends that the maximum temperature gradient (the difference in temperature between any two points) should not exceed 5.4°F.

Perimeter Spaces. Modern office buildings commonly use all-air mechanical systems to handle both heating and cooling thermal loads, instead of baseboards for heating and forced air for cooling. State energy codes (most based on the ASHRAE *Standard* 90 series) require that commercial buildings have exterior walls that meet minimum thermal performance criteria for a particular location. Typically, walls of new buildings have design heat losses as low as 100 to 300 Btu/h per linear foot of wall.

A successful all-air heating/cooling mechanical system requires the designer to consider several design variables (Hart and Int-Hout 1980; Lorch and Straub 1983; Rousseau 1983). The most important design variables include

- Supply air/room air temperature difference
- Diffuser type and design
- Design heating and cooling loads
- Supply air volumetric flow rates
- Distance between diffusers and perimeter wall
- Direction of air throw (toward wall, away from wall, or both)
- Ceiling height
- Desired air diffusion performance criteria

Linear diffusers placed parallel to the perimeter wall perform well. For year-round operation, linear diffusers with two-way throw (i.e., both toward and away from the perimeter wall) work best. Lorch and Straub (1983) reported optimum performance with a diffuser that throws warm air toward the perimeter wall during heating and chilled air in both directions during cooling. Performance was less than optimum with high discharge temperatures (greater than 15°F above ambient), both with one-way throw of air away from a cold wall and with one-way throw of chilled air toward the perimeter wall. During heating, the supply air temperature must be limited to avoid excessive thermal stratification. Diffusers should be located such that the published 150 fpm isothermal throw (which is typically unaffected by Δt) extends to within 4.5 ft of the floor. According to ASHRAE *Standard* 62.1, if throw does not meet this requirement, and the discharge-to-room temperature differential exceeds 15°F, the ventilation rate must be increased by 25%. Furthermore, when the room-to-discharge differential exceeds 15°F, it is unlikely that the vertical temperature limitation of ASHRAE *Standard* 55 will be met. Figure 10 can be used to predict approximate vertical projection on heated and cooled jets.

To resolve any uncertainty about performance, construct a mock-up with provisions for a cold wall; several variations of the design should be tested so that the best diffuser wall spacing and supply air volumes can be selected. The ADPI, room temperature gradients, or both, measured in accordance with ASHRAE *Standard* 113, can help gage system performance.

The following principles provide the best air diffusion quality and minimum energy use:

- For cooling, return air should exhaust from a location that takes advantage of any thermal stratification design. Often, this should be a high point, to take advantage of rising warm air. Cooling supply air should be introduced as close to the heat sources as possible. Alternatively, stratification designs may condition only part of the total space. In these cases, conditioned air is supplied and exhausted as close to the occupants as possible. In either case, comfort zone temperature gradients should be maintained within 5°F.
- For heating, thermal stratification should be discouraged. Heat should be introduced at points low in the large space. Ceiling-mounted fans may reduce stratification.

System Performance Evaluation

The object of air diffusion in warm-air heating, ventilating, and air-conditioning is to create the proper combination of temperature, humidity, and air motion in the occupied zone of the conditioned room (from the floor to 6 ft above floor level) (Miller 1989). The effective draft temperature considers the physiological effects on a human body of air temperature, air motion, and relative humidity. Variation from accepted standard limits (see ASHRAE *Standard* 55) may cause occupant discomfort. Lack of uniform conditions in the space or excessive fluctuation of conditions in the same part of the space also produces discomfort. Discomfort can be caused by any of the following conditions:

- Excessive air motion (draft)
- Excessive room air temperature variations (horizontal, vertical, or both)
- Failure to deliver or distribute air according to load requirements at different locations
- Overly rapid fluctuation of room temperature

Draft. Koestel and Tuve (1955) and Reinmann et al. (1959) defined draft as any localized feeling of coolness or warmth of any portion of the body caused by both air movement and air temperature, with humidity and radiation considered constant. The warmth or coolness of a draft was measured above or below a controlled room condition of 76°F db at the center of the room, 30 in. above the floor, with air moving at about 30 fpm.

To define the **effective draft temperature** T_{ed} (difference in temperature between any point in the occupied zone and the control condition), the investigators used the following equation proposed

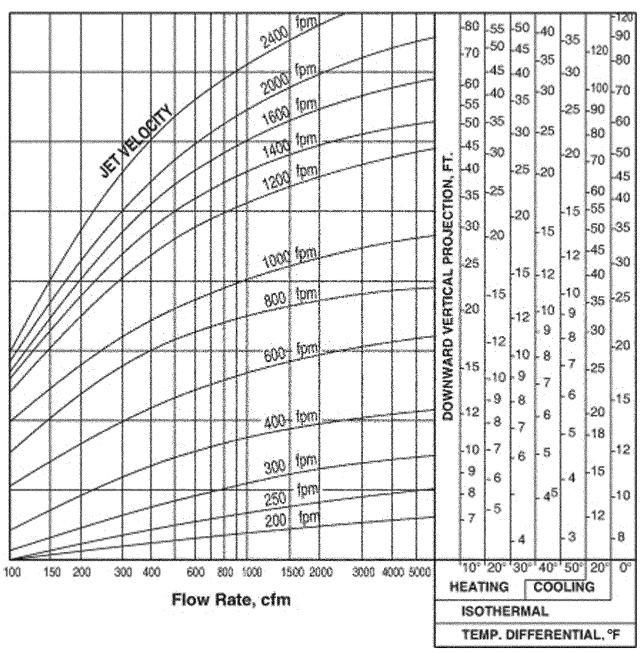


Fig. 10 Approximate Downward Vertical Projection of Heated and Cooled Jets Along Perimeter Spaces

by Rydberg and Norback (1949) and modified by Straub (Straub and Chen 1957; Straub et al. 1956) in discussion of a paper by Koestel and Tuve (1955):

$$\theta_{ed} = (T_x - T_c) - 0.07(V_x - 30) \tag{15}$$

where

θ_{ed} = effective draft temperature, °F
T_x = local airstream dry-bulb temperature, °F
T_c = average (control) room dry-bulb temperature, °F
V_x = local airstream centerline velocity, fpm

Equation (15) accounts for the feeling of coolness produced by air motion and is used to establish the neutral line in Figure 11. In summer, the local airstream temperature T_x is below the control temperature T_c. Hence, both temperature and velocity terms are negative when velocity V_x is greater than 30 fpm, and they both add to the feeling of coolness. In winter, if T_x is above T_c, any air velocity above 30 fpm subtracts from the feeling of warmth produced by T_x. Therefore, it is usually possible to have zero difference in effective temperature between location x and the control point in winter, but not in summer.

Houghten et al. (1938) presented data to statistically interpret the percentage of room occupants that will object to a given draft condition. Figure 11 presents the data in the form used by Koestel and Tuve (1955), showing that a person tolerates higher velocities and lower temperatures at ankle level than at neck level. Because of this, conditions in the zone approximately 30 to 60 in. above the floor are more critical than conditions nearer the floor.

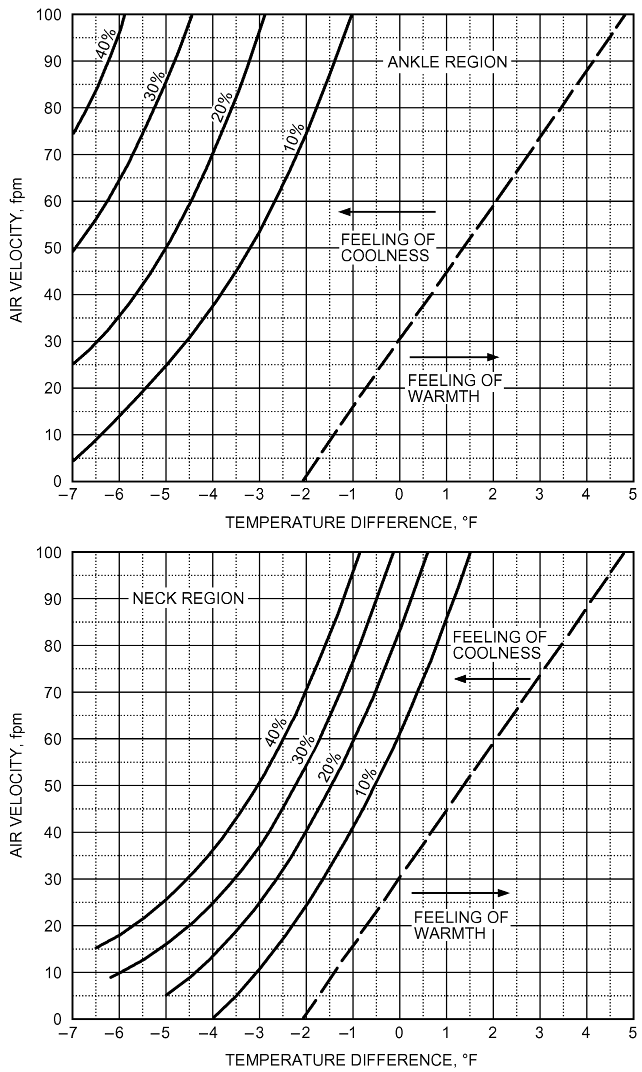


Fig. 11 Percentage of Occupants Objecting to Drafts in Air-Conditioned Room

Air Velocity. Room air velocities less than 50 fpm are generally preferred, but even higher velocities may be acceptable to some occupants (Figure 11). ASHRAE *Standard* 55 recommends elevated air speeds at elevated air temperatures. No minimum air speeds are recommended for comfort, although air speeds below 20 fpm are usually imperceptible.

Air Diffusion Performance Index (ADPI). A high percentage of people are comfortable in sedentary (office) occupations when the effective draft temperature θ_{ed}, as defined in Equation (18), is between –3 and +2°F and the air velocity is less than 70 fpm. If several measurements of air velocity and air temperature are made throughout the occupied zone of an office, the ADPI is the percentage of measurement locations where these specifications for effective draft temperature and air velocity were met. An ADPI approaching 100% indicates the most desirable conditions (Miller 1971; Miller and Nash 1971; Miller and Nevins 1969, 1970, 1972; Nevins and Miller 1972; Nevins and Ward 1968).

The ADPI is based only on air velocity and effective draft temperature (a combination of local temperature variations from the room average) and is not directly related to dry-bulb temperature or relative humidity. These and similar effects, such as mean radiant temperature, must be accounted for separately according to ASHRAE *Standard* 55.

ADPI is applicable only for cooling-mode conditions; a measurement technique is specified in ASHRAE *Standard* 113. Heating conditions can be evaluated using ASHRAE *Standard* 55 guidelines or ISO *Standard* 7730, and can also be measured using ASHRAE *Standard* 113. The ADPI can be predicted from isothermal throw data determined under ASHRAE *Standard* 70 (see Table 3) to predict what will happen under cooling conditions, within the maximum range of room loads presented. These data were obtained, and are therefore most usable, in spaces with ceiling heights between 8 and 10 ft. In a room with a single diffuser, ADPI may be overly sensitive to high airflow rates, because diffuser throws wash the room's walls.

Jet Throw. The throw of a jet is the distance from the outlet to a point where the maximum velocity in the stream cross section has been reduced to a selected terminal velocity. To estimate ADPI, terminal velocity V_T was selected for all diffusers as 50 fpm, except for ceiling slot diffusers, for which it was selected as 100 fpm. Manufacturers give data for jet throw from various diffusers for isothermal conditions and without a boundary wall interfering with the jet.

The throw distance of a jet is denoted by X_{VT}, where subscript VT indicates the terminal velocity for which the throw is given. Characteristic room length L is the distance from the diffuser to the nearest boundary wall in the principle horizontal direction of the airflow. However, where air injected into the room does not impinge on a wall surface but collides with air from a neighboring diffuser, L is one-half the distance between diffusers plus the distance the mixed jet travels downward to reach the occupied zone. Table 2 defines characteristic length for various diffusers.

The midplane between diffusers also can be considered the module line when diffusers serve equal modules throughout a space, and a characteristic length consideration can be based on module dimension.

Load Considerations. Recommendations in Table 3 cover cooling loads of up to 80 Btu/h per square foot of floor surface. The loading is distributed uniformly over the floor up to about 7 Btu/h·ft^2, lighting contributes about 10 Btu/h·ft^2, and the remainder is supplied by a concentrated load against one wall that simulates a business machine or a large sun-loaded window. Over this range of data, the maximum ADPI condition is lower for the

Table 2 Characteristic Room Length for Several Diffusers

Diffuser Type	Characteristic Length L
High sidewall grille	Distance to wall perpendicular to jet
Circular ceiling pattern diffuser	Distance to closest wall or intersecting air jet
Sill grille	Length of room in direction of jet flow
Ceiling slot diffuser	Distance to wall or midplane between outlets
Light troffer diffusers	Distance to midplane between outlets plus distance from ceiling to top of occupied zone
Cross-flow pattern ceiling diffusers	Distance to wall or midplane between outlets

Table 3 Air Diffusion Performance Index (ADPI) Selection Guide

Terminal Device	Room Load, Btu/h·ft²	X_{50}/L for Maximum ADPI	Maximum ADPI	For ADPI Greater than	Range of X_{50}/L
High sidewall grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5 to 2.2
	40	1.6	78	70	1.2 to 2.3
	20	1.5	85	80	1.0 to 1.9
	<10	1.4	90	80	0.7 to 2.1
Circular ceiling diffusers	80	0.8	76	70	0.7 to 1.3
	60	0.8	83	80	0.7 to 1.2
	40	0.8	88	80	0.5 to 1.5
	20	0.8	93	80	0.4 to 1.7
	<10	0.8	99	80	0.4 to 1.7
Sill grille, straight vanes	80	1.7	61	60	1.5 to 1.7
	60	1.7	72	70	1.4 to 1.7
	40	1.3	86	80	1.2 to 1.8
	20	0.9	95	90	0.8 to 1.3
Sill grille, spread vanes	80	0.7	94	90	0.6 to 1.5
	60	0.7	94	80	0.6 to 1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling slot diffusers (for T_{100}/L)	80	0.3	85	80	0.3 to 0.7
	60	0.3	88	80	0.3 to 0.8
	40	0.3	91	80	0.3 to 1.1
	20	0.3	92	80	0.3 to 1.5
Light troffer diffusers	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Cross-flow pattern diffusers	11 to 50	2.0	96	90	1.4 to 2.7
	11 to 50	2.0	96	80	1.0 to 3.4

highest loads; however, the optimum design condition changes only slightly with load.

Design Conditions. The quantity of air must be known from other design specifications. If it is not known, the solution must be obtained by trial and error.

The devices for which data were obtained are (1) high sidewall grilles; (2) circular pattern ceiling diffusers; (3) sill grilles; (4) two- and four-slot ceiling diffusers; (5) light troffer diffusers; and (6) square-faced one-, two-, three-, and four-jet pattern (cross-flow) ceiling diffusers. Table 3 summarizes recommendations on X_{VT}/L by giving the value of X_{50}/L at which ADPI is maximized for various loads, as well as a range of values of X_{50}/L for which ADPI is above a minimum specified value.

FULLY STRATIFIED SYSTEMS

Fully stratified air distribution systems have been used in industrial applications for many years. In the 1980s, they became a popular alternative for office and classroom HVAC in Europe, and their popularity has recently spread to North America because of their high contaminant removal efficiencies and their possible energy savings, especially in relatively mild climates. **Thermal displacement ventilation (TDV) systems** are the most widely used variant of these systems.

The main objective of a mixed-air system is to create a homogenous mixture of supply and room air throughout the space. Contaminants and heat are diluted and then extracted through the return inlet. TDV systems (Figure 12) do not attempt to mix heat and contaminants; instead, they allow them to escape into the upper uninhabited zone, from which they are extracted. With a TDV system, supply air is introduced directly into the occupied zone at low velocity and a temperature lower than that of room air. Contaminants and heat in the space are carried by convective flows (created by space heat sources) into the upper part of the room. Warm air in the upper zone does not recirculate into the occupied zone, so the temperature and concentration of most impurities at the exhaust inlet exceed those in the occupied zone and at the breathing level.

TDV systems offer increased ventilation effectiveness and may reduce HVAC energy consumption. Applications include classrooms, conference rooms, theaters, restaurants, supermarkets, and spaces with high ceilings (10 ft and above) (Skistad et al. 2002).

Sandberg and Blomqvist (1989) suggest that the maximum convective cooling load in office buildings with TDV not exceed about 8 Btu/h·ft², so that the maximum vertical temperature gradient in the occupied zone is not larger than 5°F. Kegel and Schulz (1989) and Svensson (1989) suggested higher cooling load limits of 10 to 13 Btu/h·ft². However, Chen and Glicksman (1999) demonstrated that cooling loads up to 40 Btu/h·ft² can be handled in the office environment if the ventilation rate is increased. Howe et al. (2003) reported successful application of TDV in a telecommunication equipment room with cooling loads up to 108 Btu/h·ft², although thermal comfort was not the primary objective of this application.

Convective Flows Associated with Space Heat Sources

Convective heat flows in the space are the driving forces behind TDV systems. When the surface temperature of a heat source exceeds that of the air surrounding it, heat is transferred to ambient air by convection. This transfer warms the air and causes it to rise because of buoyancy. These rising plumes grow as they entrain room air. Radiant heat transfer does not directly affect heat plume formation, but may indirectly influence development of other heat source plumes by raising the surface temperature of the source.

Each space heat source forms its own thermal plume. Formation of the plume and its vertical travel are determined by several factors:

- Shape and surface area of heat source
- Intensity of heat source
- Air turbulence around heat source (turbulence discourages plume formation)
- Temperature gradient in the space (affects plume volume)

The heat plume rises until it encounters ambient air of similar temperature.

The Archimedes number [Equation (15)] relates the ratio between buoyancy forces and velocity forces of the air surrounding the heat source. Larger Archimedes numbers indicate that buoyancy dominates the air behavior, whereas smaller numbers indicate that inertia (velocity) dominates. Lower Archimedes numbers in mixed-air systems usually inhibit plume formation.

Characteristics of Thermal Plumes

As a thermal plume rises because of natural convection above a heat source, it entrains surrounding air and therefore increases in size and volume, and decreases in velocity (Figure 13). The maximum height to which a plume rises depends primarily on the heat source's strength, and secondarily on stratification in the room (which decreases the rising plume's buoyancy). The **stratified zone**

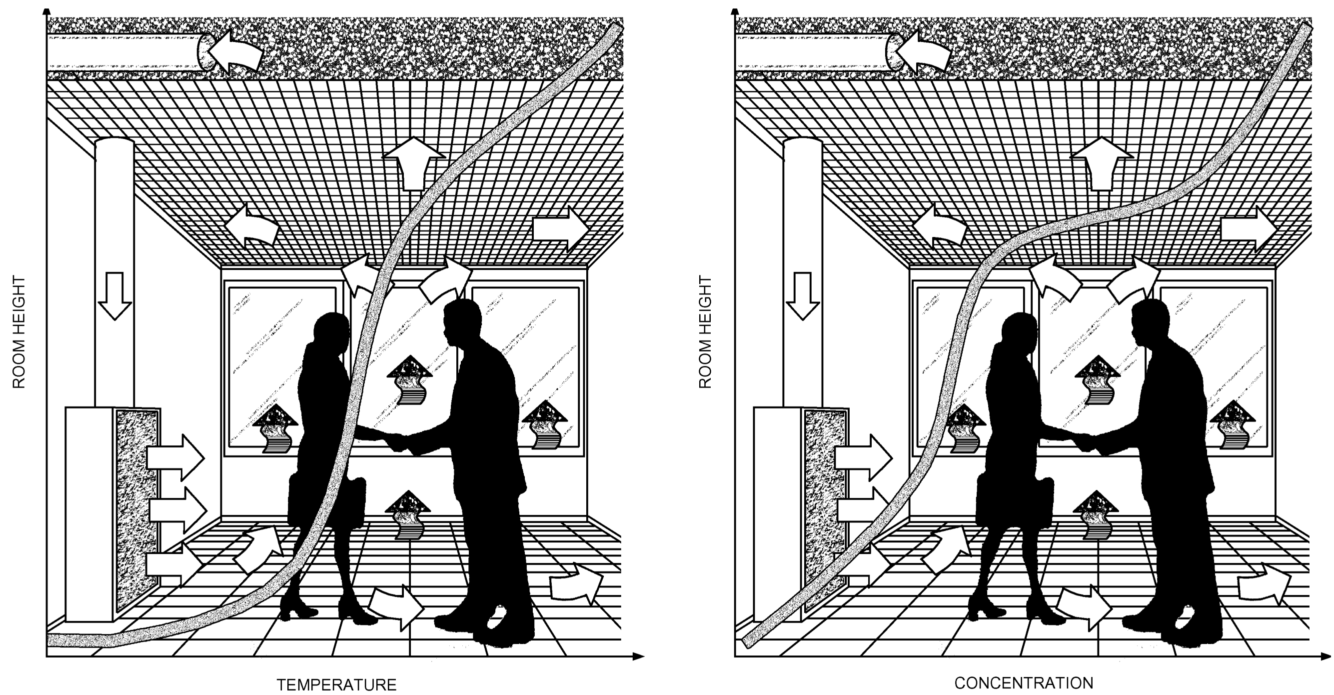


Fig. 12 Displacement Ventilation

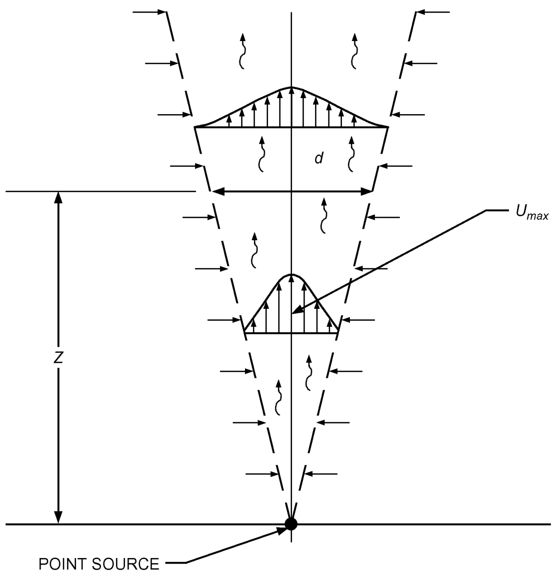


Fig. 13 Thermal Plume from Point Source

has little or no recirculation. In this region, cool supply air gradually flows across the room in a thin layer, typically 4 to 6 in. thick. It is drawn horizontally toward the heat sources, where it joins rising air in the plumes and is entrained upward. These plumes expand and rise until they encounter equally warm air in the upper regions of the space. The **upper zone** above the stratification height is characterized by low-velocity recirculation, which produces a fairly well-mixed layer of warm air with greater contaminant concentration than that in the lower levels of the space.

Typically, warmer, more polluted air will not reenter the stratified zone. This principle is the basis for the improved ventilation effectiveness and heat removal efficiency of TDV systems. In some situations (e.g., morning start-up, winter), there are also sources of cooling in the space, such as cold perimeter windows. The resulting cold downdraft may transport some air from the upper zone back down to the stratified zone. Figure 14 shows these basic elements in a simplified schematic of a TDV system. In the figure, q_0 represents the supply airflow into the room from a low sidewall diffuser, q_1 is the upward-moving airflow in thermal plumes that form above heat sources, and q_2 is the downward-moving airflow resulting from cool surfaces. In this simplified configuration, the stratification height occurs at a height SH, where the net upward moving flow $q_1 - q_2$ equals q_0. An important objective in designing and operating a TDV system is to maintain stratification above the occupied zone.

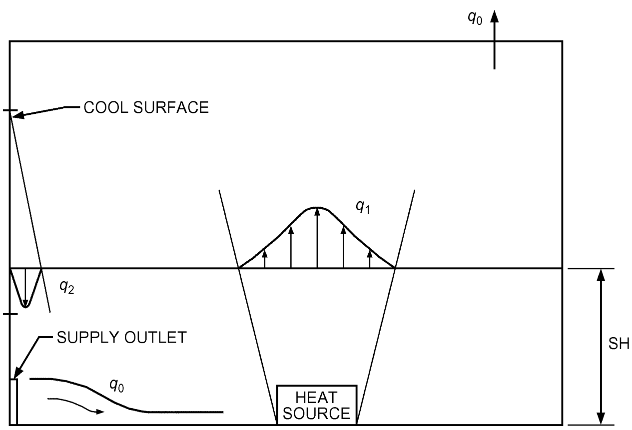


Fig. 14 Schematic Diagram of Major Flow Elements in Room with Displacement Ventilation

Vertical Temperature Distribution

Thermal displacement ventilation (TDV) outlets discharge conditioned (typically 60 to 65°F) air at very low outlet velocities (less than 70 fpm). Cool air drops almost immediately to the floor, because of its negative buoyancy. The buoyancy of the supply causes it to remain near the floor until it comes into close contact with a convective heat source. The ascending plume associated with the source creates a stack effect that entrains supply air from the

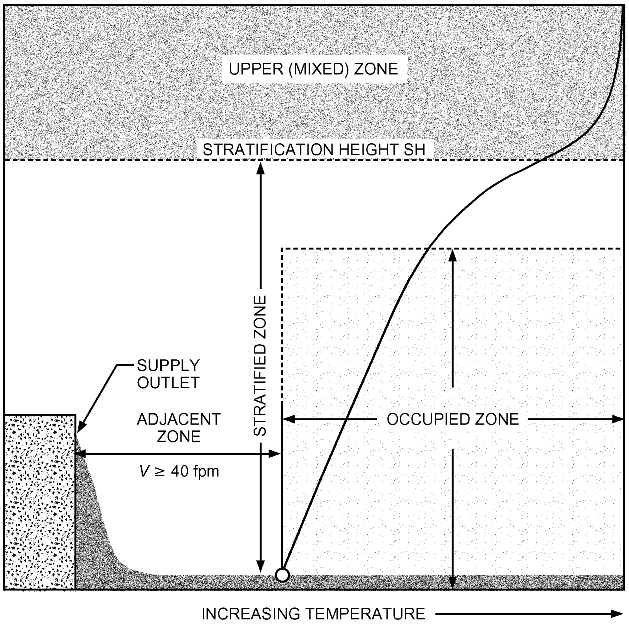


Fig. 15 Vertical Temperature Gradients in Thermal Displacement System

floor. As it passes vertically over the source, the displaced supply air is warmed and becomes part of the convective plume.

Spaces conditioned by TDV systems can be categorized into two basic zones. Figure 15 illustrates a vertical temperature gradient for a TDV system applied in a space with conventional-height ceilings. In addition, the **adjacent zone** (sometimes referred to as the **near** or **clear zone**) is the portion of the space near the air outlet where horizontal velocities in excess of 40 fpm may be found. This zone is defined as the room volume bordered horizontally by the outlet and the vertical plane corresponding to the furthest extent of an isovel of 40 fpm. Stationary occupants should not normally be located in this zone, because local velocities and supply air temperatures are likely to cause draft sensations. Once the supply air is reduced to a terminal velocity of 40 fpm or less when measured 4 in. above the floor, its temperature has usually increased to a level not likely to produce draft.

Once the conditioned air mixture has passed through the adjacent zone, it pools out across the floor and only moves vertically when entrained by a convective plume associated with a space heat source. The plume rises through the naturally stratified environment (the **stratification zone**) to a level where it encounters equally warm air. The plume then dissipates horizontally across the space. The level at which this occurs is called the **stratification height** (**SH**; sometimes also called the **shift height**).

Displaced heat and contaminants (whose buoyancy exceeds that of room air) pool in the space above the stratification height. This is referred to as the **upper zone**.

Contaminant Distribution

One of the key benefits of fully stratified systems is increased removal efficiency of contaminants associated with space heat sources. These contaminants are directly conveyed to the upper zone by thermal plumes associated with the heat sources.

Figure 16 illustrates how stratification height influences indoor air quality in the occupied zone for the idealized case of a TDV system serving a space with a single heat source (person) and its associated contaminant (person's breathing) (Skistad 1994). The figure shows two typical vertical profiles of pollutants from a person's breathing. Normalized pollutant concentrations (c/c_R) are plotted

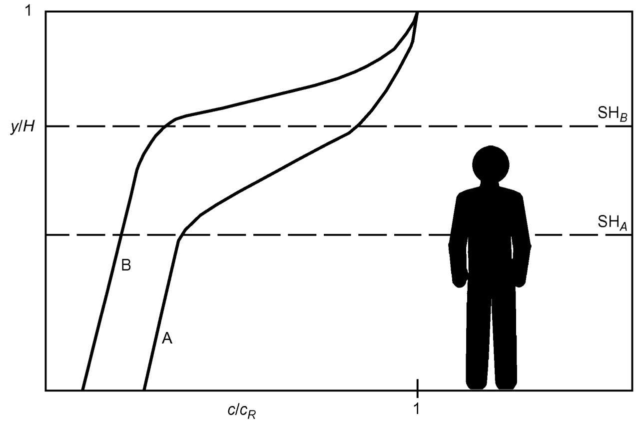


Fig. 16 Vertical Profiles of Pollutant Concentrations in Room with Displacement Ventilation

against normalized room height (y/H), where c_R is the concentration at the return grille near ceiling level and H is room height. Both profiles demonstrate how a large increase in pollutant concentration occurs at the stratification height, with cleaner, less polluted air in the stratified zone and higher pollutant concentrations in the upper zone. Profile A is produced by a lower airflow rate that results in a stratification height (SH_A) somewhat below head height of a standing occupant. By increasing the airflow rate (loads remain constant), stratification height (SH_B) is raised above head height in profile B, producing improved indoor air quality at the breathing height. Stratification height also can be locally displaced about 0.7 ft upward around a person (Nielsen 1996). This represents the entrainment of cleaner air from lower levels in the room by the thermal plume rising around a person up to their breathing height.

Design Methods

Depending on space requirements, there are two options for design. **Temperature-based design** is used when heat removal is the main objective (e.g., in schools, offices, auditoriums, sport facilities). The **shift-zone method** is used when contaminants should be considered, such as in smoking rooms and other facilities with gaseous contaminants (lighter than air or of the same density as air) associated with convective plumes from heat sources. Whereas the objective of the temperature-based method is simply to satisfy temperature conditions in the occupied zone, shift-zone design also seeks to stratify contaminants by maintaining the stratification height above the occupants' breathing level.

Design methods for TDV systems are addressed in Chapter 56 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Ventilation and Heating

Skistad et al. (2002) reported that displacement ventilation can be combined successfully with radiators and convectors at exterior walls to offset space heat losses. Radiant heating panels and heated floors also can be used with displacement ventilation. When the secondary heating system is used, displacement diffusers supply air at 4°F lower than room air temperature. In that case, displacement ventilation performs the same way as in cooling mode and all the benefits associated with better indoor air quality are preserved.

When warm air is supplied through displacement outlets, the system's performance is similar to that of a mixed-air system in heating mode.

Outlet Types

Outlets in TDV systems are designed to limit outlet velocities to about 70 fpm or less. This results in supply airflow capacities of typically 50 to 70 cfm per square foot of discharge area. Therefore, outlets used in TDV systems tend to be large compared to those used for mixed-air systems. Properly designed TDV outlets also incorporate provisions that create an equal distribution of supply air across their face to minimize discharge velocity variations. Equally distributing flow across the discharge area reduces the length of the outlet's adjacent zone.

TDV system outlets are available in various geometries and capacities. They may be mounted flush in a partition wall, either contained within the wall or extending into a mechanical space behind it. They are also often mounted adjacent to the partition wall (with either a 90 or 180° discharge pattern) or in a corner (with a quarter round discharge pattern). Others are free-standing and column-shaped, with a full radial discharge pattern.

Because TDV outlets discharge the air at such low velocities, they impart very little directional guidance to the supply airstream. This allows them to be mounted behind architectural elements such as louvers or screens and remain visually unobtrusive.

Floor diffusers that discharge supply air horizontally at low airflow rates can be used to create a fully stratified room environment. Floor diffusers with vertical discharge of turbulent air jets do not create such conditions, and are covered in the section on Partially Mixed Systems.

Much smaller versions of TDV outlets are used for underseat supply in public assembly (theaters, lecture halls, sports arenas, etc.) HVAC applications. These outlets are generally designed for much lower individual supply airflow capacities, and typically use warmer supply air. They may have internal mixing devices to help reduce their adjacent zone length.

Outlet Selection and Location

TDV outlets should be selected and located so that stationary space occupants are not located within their adjacent zone where draft risks are high. The low discharge velocities of TDV terminals create very little noise and thus are only of concern in very sensitive acoustical applications. System noise (noise from fans, dampers, ductwork, etc.) should be considered, however, because it is transmitted with the supply airflow.

Return Inlet Selection and Location

TDV return air inlets used should conform to the same requirements as those for mixed-air systems. They should always be located above the occupied zone, and should be provided within any confined space served by a TDV supply air terminal.

System Performance Evaluation

The primary comfort criterion of TDV systems is maintaining the design room air temperature (usually specified at the head level of the predominant space occupants) while limiting the vertical temperature difference between occupants' ankle and head levels to no more than 5.4°F. Because velocities in the occupied zone are very low, they are of minimal concern. Additional information may be found in ASHRAE *Standard* 113.

ADPI should not be used to evaluate fully stratified systems, because it essentially measures the degree of mixing achieved by the room air distribution system. A fully mixed environment would have the highest ADPI rating.

PARTIALLY MIXED SYSTEMS

Partially mixed room air distribution systems used for space cooling generally discharge conditioned air from a low sidewall or floor location, and the diffuser discharge turbulence is considerably greater than in fully stratified (TDV) systems. This creates a zone of high entrainment near the plane of discharge. A common example of partially mixed systems is **underfloor air distribution (UFAD) systems**.

UFAD systems differ from TDV systems primarily in the way air is delivered to the space: (1) air is supplied at higher velocities through smaller supply outlets, and (2) local air supply conditions are generally under the control of occupants, allowing comfort conditions to be optimized. By introducing supply air with greater momentum, UFAD systems alter conditions in the lower region of the space by increasing the amount of mixing and reducing the temperature gradient. At higher elevations in the room, above the influence of supply outlets, overall airflow performance is similar to that of TDV systems. Based on recent experimental results (Webster et al. 2002a, 2002b; Yamanaka et al. 2002) and an extension of displacement theory, three distinct zones in the room can be used to describe the room air diffusion for UFAD systems.

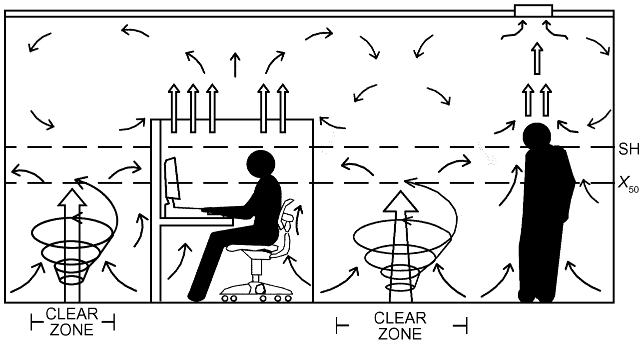


Fig. 17 Underfloor Air Distribution System with Diffuser Throw below Stratification Height

Figure 17 shows a schematic of typical airflow patterns in an UFAD system in an office environment. The diagram identifies two characteristic heights in the room that define the three zones in the room: (1) the throw height (X_{50}) of the floor diffusers, and (2) the stratification height (SH), similar to that found in TDV systems. As shown, UFAD diffusers typically create adjacent zones that have excessive draft and cool temperatures, making long-term occupancy not recommended. When under direct individual control by the occupant, however, these local thermal conditions may be acceptable, and even desirable. Increased mixing in the occupied zone diminishes ventilation effectiveness, compared to TDV systems. In any case, control and optimization of stratification is crucial to system design and sizing, energy-efficient operation, and comfort performance of UFAD systems.

Figure 18 compares typical vertical temperature profiles for UFAD, TDV, and conventional overhead mixing systems. The profiles shown are representative of normal operating conditions and are intended to demonstrate key differences and similarities between the three air distribution systems. The UFAD profile is based on temperatures in a space outside the direct influence of supply outlets (outside adjacent zones), and can vary significantly depending on several control factors (see the section on Controlling Stratification) (Webster et al. 2002a). In Figure 18, the nondimensional temperature (temperature ratio) is plotted versus room height, where T_H is room air temperature as a function of height, T_S is supply temperature, and T_E is temperature at the ceiling. The linear profile for TDV systems is based on the 50% rule of thumb that applies to rooms of conventional height and normal heating loads (Skistad 1994); the temperature near the floor is assumed to be halfway between the supply and exhaust temperatures. The TDV profile is assumed to join the UFAD profile at the stratification height. As long as the throw heights of the UFAD diffusers are below the stratification height, the upper zone is assumed to perform in a similar manner for both systems (for the same room-load-to-supply-volume ratio). The fully mixed system profile represents a uniformly mixed room with the temperature equal to the exhaust temperature.

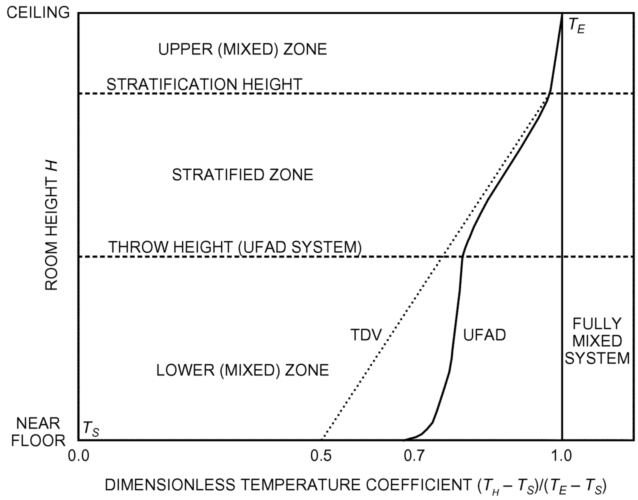


Fig. 18 Comparison of Typical Vertical Temperature Profiles for Underfloor Air Distribution, Displacement Ventilation, and Mixing Systems

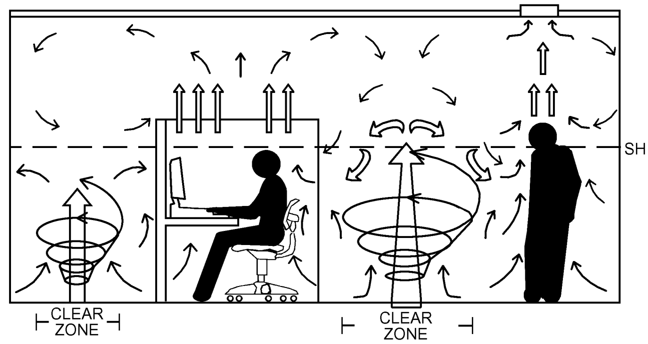


Fig. 19 Underfloor Air Distribution System with Diffuser Throw above Stratification Height

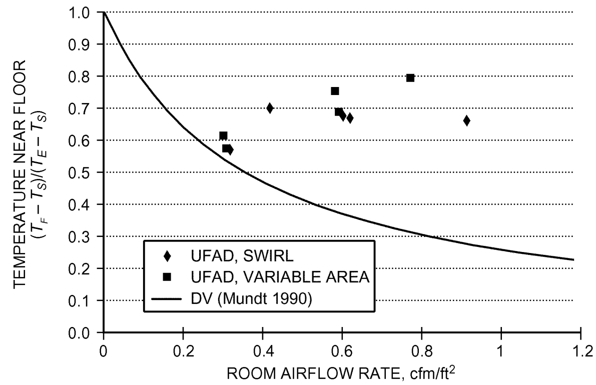


Fig. 20 Nondimensional Temperature near Floor Versus Room Airflow Rate

Experimental UFAD data taken from Webster et al. (2002a); TDV results from Mundt (1990).

Lower (Mixed) Zone

The lower zone is directly adjacent to the floor and varies in depth according to the vertical projection of the floor-based supply outlets used. The air in this layer is relatively well mixed because of the influence of high-velocity jets near the supply air outlets. The upper boundary of the lower zone coincides with the elevation at which supply air reaches a terminal velocity of around 50 fpm. The greater mixing in this zone increases the temperature ratio near floor to about 0.7, and reduces the gradient in comparison to TDV systems. The lower zone always exists, although its height may vary greatly depending on the vertical projection of supply outlets and the ratio of space heat load to supply airflow.

Stratified Zone

The stratified zone is a transition region between the lower and upper zones. Air movement in this zone is entirely buoyant, driven by rising thermal plumes around convective space heat sources. Formation of these plumes is uninhibited in this region, because air movement is not affected by supply air jets. Therefore, the vertical temperature gradient in this zone tends to be greatest, approaching that for TDV systems. The stratified zone only exists when the throw height of supply outlets is below the stratification height.

Upper (Mixed) Zone

The upper zone comprises warm (contaminated) air deposited by rising heat plumes within the space. Although its average air velocities are generally quite low, air in this zone is relatively well mixed as a result of the momentum of thermal plumes penetrating its lower boundary. This zone is analogous to the upper zone found in spaces served by TDV systems. Its bottom boundary, coincident with the stratification height, is primarily a function of the ratio of space heat load to supply airflow rate. If jets from supply outlets penetrate this zone, its depth (or even existence) may be affected, though, if properly controlled, this may be a secondary effect (Figure 19).

Temperature Near Floor

As shown in Figure 18, the greater mixing provided by turbulent supply outlets used in UFAD systems increases the temperature near the floor compared to TDV systems (for the same supply air temperature and volume). This effect is shown more clearly in Figure 20, which plots the nondimensional temperature near the floor as a function of overall room airflow rate, where T_F is the temperature near the floor, T_s is the supply temperature at the floor, and T_e is the temperature at the ceiling. Measurement heights for T_F are in the range of 3 to 4 in. Experimental data for both swirl and variable-area floor diffusers are taken from Webster et al. (2002a). The curve for TDV systems is based on numerous measurements in different rooms (Mundt 1990).

Stratification Height

If vertical throw is equal to or less than the stratification height (see Figure 17), the only airflow crossing it will be from buoyancy effects, similar to TDV systems. As throw and mixing are reduced, UFAD systems tend to approach the operation of TDV systems. If throw height is close to or greater than the stratification height, cooler supply air penetrates the warmer upper layer before dropping back down into the lower region, bringing warm air with it (see Figure 19). Although a subject of ongoing research, recent results indicate that, as long as diffuser throw does not penetrate too far into the upper zone (up to 7 ft in a 10 ft high room), relatively similar comfort conditions are produced in the occupied zone, compared to diffusers with lower throws (Webster et al. 2002a).

The amount of air brought down influences temperatures in the lower region, and can also increase stratification height, but this is a secondary effect. Higher throws that penetrate the stratification height result in slightly warmer temperatures and a smaller gradient in the lower region.

When a very strong supply air jet penetrates far into the upper zone, it is possible to disrupt the stratified airflow pattern. For example, laboratory experiments (Bauman et al. 1991; Fisk et al. 1991) demonstrated that, when a fan-driven floor supply module was operated at higher air supply volumes, cool supply jets were

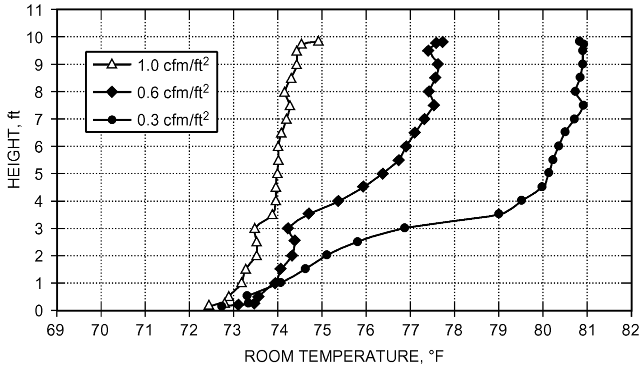


Fig. 21 Effect of Room Airflow Variation at Constant Heat Input, Swirl Diffusers, Interior Zone

able to reach the ceiling, thereby minimizing stratification and producing close to uniform ventilation conditions. This operating strategy of providing a well-mixed space reduces or eliminates the potential improvements in energy and ventilation performance described. To avoid eliminating a stratified space with UFAD systems, maximum vertical throws of diffusers should be limited to no closer than 2 to 3 ft from the ceiling.

Controlling Stratification

Laboratory experiments have investigated the thermal stratification performance of UFAD systems using floor diffusers (Webster et al. 2002a, 2002b). Figure 21 shows the effect of variations in total room airflow on stratification for swirl diffusers operating in a simulated interior space with total heat input of 18 Btu/h·ft^2 and a supply air temperature of 64°F. For constant heat input, stratification increases when room airflow is reduced. Figure 21 also demonstrates how a control strategy might optimize stratification performance. At the highest flow rate of 1 cfm/ft^2, the temperature profile exhibits only a small amount of stratification, with a head/foot temperature difference of 1.3°F. This represents a case where the space is overventilated. On the other hand, at the lowest flow rate of 0.3 cfm/ft^2, the head/foot temperature difference increases to 6.8°F, exceeding the limit of 5.4°F specified in ASHRAE *Standard* 55. This temperature profile demonstrates the sensitivity to changes in airflow rate, although it is highly unlikely that a system with cooling loads of this magnitude would be operated at such a low airflow rate. To improve energy performance (reduce airflow) while maintaining thermal comfort (avoiding excessive stratification), the middle profile at a flow rate of 0.6 cfm/ft^2 may be a reasonable target, because it has a head/foot temperature difference of 3.2°F. The difference between the middle and first profiles also demonstrates that, despite a 40% reduction in airflow rate, the temperature in the space only increases by about 1°F up to a height of nearly 4 ft.

Heating Systems

In most applications, heating is primarily needed only near the building envelope, where heat loss to the outdoors can cool spaces and may cause discomfort. Heating may also be needed in some top-floor interior zones and during periods of low occupancy (e.g., nights and weekends).

In operation, delivering warm air from rapidly mixing diffusers near floor level is very effective at providing heat to the conditioned space. Because of buoyancy, the characteristic thermal stratification obtained in cooling operation is replaced with a well-mixed, uniform temperature distribution. Heating load calculation can therefore use the same methods as for conventional overhead air distribution systems.

Effective heating systems isolate the source of warm air from the thermal lag effect of the concrete slab (which is usually slightly cooler than room temperature). This can be done, for example, by ducting from an underfloor fan-coil unit, or by using baseboard radiation or convection units. Quick response on heating can be very important during morning start-up, particularly if night setback is used.

Outlets Types

Partially mixed systems use a wide variety of outlet types, because they are designed to promote mixing in a designated portion of the space. Most partially mixed systems are floor based, however, and those are the types discussed here.

Floor-based outlets used in partially mixed air distribution systems may be classified as passive or active. **Passive** diffusers are installed in the plenum under the raised access floor in UFAD systems. They are not directly ducted to either the conditioned air source or a fan-assisted terminal in the floor plenum. Instead, their supply airflow rate depends on the pressure in the raised-floor plenum that delivers the HVAC service. **Active** diffusers are connected to either a supply air duct or a fan-assisted terminal.

Both passive and active diffusers can be operated with constant or variable air volume. The supply airflow rate in variable-air-volume diffusers can be either automatically reset in response to a control signal, or manually adjusted by space occupants.

High-induction swirl diffusers are the most common type of UFAD supply air outlet. The swirling air pattern provides rapid mixing of supply air with room air up to the height of the diffuser's vertical throw. Although the discharge pattern for most swirl diffusers is not adjustable, occupants have limited control of the delivered air volume by rotating the face of the diffuser or opening the diffuser and adjusting a volume control damper. The maximum flow rate for most passive swirl diffusers operated at typical UFAD plenum pressures is about 100 cfm at 0.08 in. of water. Most are equipped with a catch basin for dirt and liquid spills.

Linear bar diffusers are also commonly used in UFAD systems. They are often used as active diffusers (supplied by fan-assisted terminals with reheat provisions) for heating and/or cooling perimeter zones adjacent to exterior windows. They may also be used as passive diffusers to supply cooling directly from the pressurized floor plenum. In the latter case, their cooling delivery is usually variable volume and is automatically modulated in accordance with the space thermostat demand.

Passive floor diffusers may also be configured as variable-air-volume diffusers, requiring control and power connections to automatically adjust an integral volume control damper. These terminals may either deliver supply airflow in proportion to a space thermostat signal or use pulse-width modulation to constantly reset an inlet damper from fully open to fully closed (two-position operation). In the latter case, air is supplied through a slotted square floor grill in a jet-type airflow pattern. Occupants can adjust supply jet direction by changing the grille's orientation.

Plenum boxes with integral airflow dampers can also be used to provide automatic variable volume through UFAD floor diffusers. These can be either passive or active in operation, depending on whether they are fan assisted or directly supplied by the pressurized floor plenum.

Outlet Selection and Location

Outlets used in partially mixed air distribution systems should be located such that stationary occupants are not within their prescribed adjacent zone. Outlets that are intended to be manually adjustable by space occupants should be located within a few feet of the occupants.

Return Inlet Selection and Location

As in fully stratified systems, return inlets in partially mixed systems should be located above the occupied zone. These inlets should be of sufficient size to result in inlet velocities no greater than

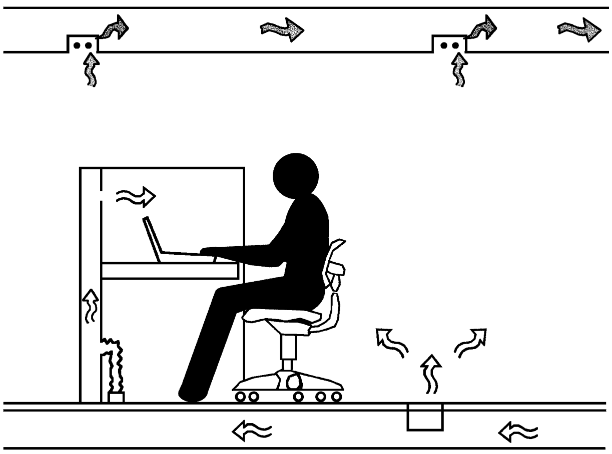


Fig. 22 Underfloor TAC and Personal HVAC System
(Matsunawa et al. 1995)

500 fpm, and should be located in every confined space with a supply air outlet.

System Performance Evaluation

The primary comfort criterion of UFAD systems is maintenance of the design room air temperature (usually specified at the head level of the predominant space occupants) while limiting the vertical temperature difference between occupants' ankle and head levels to no more than 5.4°F. Because velocities in the occupied zone are very low, they are of minimal concern. Additional information may be found in ASHRAE *Standard* 113.

ADPI should not be used to evaluate partially mixed systems, because it essentially measures the degree of mixing achieved by the room air distribution system. A fully mixed environment would have the highest ADPI rating.

TASK/AMBIENT CONDITIONING (TAC)

Task/ambient conditioning (TAC) is most commonly installed with underfloor air distribution (Arens et al. 1991; Bauman and Arens 1996; Bauman et al. 1991, 1993, 1995, 1998; Faulkner et al. 1993, 1999; Fisk et al. 1991; Matsunawa et al. 1995; Tsuzuki et al. 1999). TAC gives individuals some control over their local environment without adversely affecting that of nearby occupants. Typically, the occupant can control the speed, direction, and, in some cases, temperature of supply air. TAC systems are most frequently installed in open-plan offices to provide supply air and, in some cases, radiant heating directly into workstations. Figure 22 shows an underfloor TAC system with a local (personal HVAC) diffuser located in the partition in front of the office worker (Matsunawa et al. 1995).

Bauman et al. (1998) and de Dear and Brager (1999) found that building occupants who have no individual control capabilities are twice as sensitive to changes in temperature as occupants who do have individual thermal control.

SYMBOLS

A_c = measured gross (core) area of outlet, ft^2
A_o = core area or neck area, ft^2
A_R = cross-sectional area of confined space normal to jet, ft^2
Ar = Archimedes number [Equation (12)]
c = pollutant concentration
C_d = discharge coefficient (usually between 0.65 and 0.90)
c_R = concentration of pollutant at return grille near ceiling level
g = gravitational acceleration rate, ft/min^2
H = height or width of slot (Table 2), or of room
H_o = width of jet at outlet or at vena contracta, ft
H_s = width of slot, ft
K_c = centerline velocity constant
L = characteristic length, ft
L_o = length scale of diffuser outlet equal to hydraulic diameter of outlet, ft
ΔP = diffuser static pressure drop, in. of water
Q_o = discharge from outlet, cfm
Q_x = total volumetric flow rate at distance X from face of outlet, cfm
r = radial distance of point under consideration from centerline of jet
$r_{0.5V}$ = radial distance in same cross-sectional plane from axis to point where velocity is one-half centerline velocity (i.e., $V = 0.5V_x$)
R_{fa} = ratio of free area to gross (core) area
SH = stratification height
T = average absolute room temperature, °R
ΔT = room/jet temperature difference, °F
T_A = temperature of ambient air, °F
T_c = average (control) room dry-bulb temperature, °F
T_E = temperature at ceiling, °F
T_F = temperature near floor, °F
T_H = temperature at given height, °F
T_O = initial temperature of jet, °F
T_S = supply temperature, °F
T_x = local airstream dry-bulb temperature, °F
V = actual velocity at point being considered
V_c = nominal velocity of discharge based on core area, fpm
V_o = initial air velocity of jet, fpm
V_T = terminal velocity, fpm
V_x = centerline velocity, fpm
X = distance from face of outlet to location of centerline velocity V_X, ft
$X_{attached}$ = throw distance of attached jet, ft
X_{free} = throw distance of free jet, ft
X_H = throw height from floor outlet, ft
X_{VT} = distance to given terminal velocity, ft
θ_{ed} = effective draft temperature, °F

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