CHAPTER 32

SPACE AIR DIFFUSION

TERMINOLOGY

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

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

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

Cold air. General term used for supply air at 1.5 to 4.5°C.

Core area. Total plane area of that portion of a grille, included within lines tangent to the outer edges of the outer openings, through which air can pass.

Damper. Device used to vary the volume 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. Distribution of air within a space by an outlet discharging supply air in various directions and planes.

Draft. Undesired local cooling of a body caused by low temperature and movement of air.

Drop. Vertical distance that the lower edge of a horizontally projected airstream drops 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 room air into the jet caused by the airstream discharged from the outlet (secondary air motion).

Entrainment ratio. Total air divided by the air discharged 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 the openings in the air outlet or inlet through which air can pass.

Grille. A covering for any area through which air passes.

Induction. See Entrainment.

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

Lower zone. Room volume below the stratification level created by displacement ventilation.

Nonisothermal jet. Air jet with an initial temperature different from the surrounding air.

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

Primary air. Air delivered to the outlet by the 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., 0.25, 0.5, 0.75, or 1.0 m/s).

Register. Grille equipped with a damper or control valve.

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

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

Supply opening or outlet. Any opening 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 17 of the *ASHRAE Handbook—Systems and Equipment*.)

Temperature differential. Temperature difference between primary and room air.

Terminal velocity. Maximum airstream velocity at the end of the throw.

Throw. Horizontal or vertical axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal velocity (e.g., 0.25, 0.5, 0.75, or 1.0 m/s), defined by ASHRAE *Standard* 70.

Total air. Mixture of discharged air and entrained air.

Upper zone. Room volume above the stratification level created by displacement ventilation.

Vane. Thin plate in the opening of a grille.

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

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

PRINCIPLES OF JET BEHAVIOR

AIR JET CLASSIFICATION

As a rule, air supplied into rooms through the various types of outlets (e.g., grilles, ceiling diffusers, perforated panels) is distributed by turbulent air jets. These air jets are 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 the occupied zone. If the air jet is not obstructed by walls, ceiling, or other obstructions, it is considered a **free jet**. If the air jet is attached to a surface, it is an **attached air jet**.

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

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

Depending on diffuser type, air jets can be classified as follows:

- **Compact** air jets are formed by cylindrical tubes, nozzles, square or rectangular openings with a small aspect ratio (unshaded or shaded by perforated plates), grilles, etc. Compact air jets are three-dimensional and axisymmetric at least at some distance from the diffuser opening. The maximum velocity in the cross section of the compact jet is on the axis.
- **Linear** air jets are formed by slots or rectangular openings with a large aspect ratio. The jet flows are approximately two-dimensional. Air velocity is symmetric in the plane at which air velocities in the cross section are maximum. At some distance from the diffuser, linear air jets tend to transform into compact jets.
- **Radial** air jets are formed by ceiling cylindrical air diffusers with flat disks or multidiffusers that direct the air horizontally in all directions.
- **Conical** air jets are formed by cone-type or regulated multidiffuser ceiling-mounted air distribution devices. They have an axis of symmetry. The air flows parallel to the conical surface (the angle at the top of the cone is 120°) with the maximum velocities in the cross section perpendicular to the axis.
- **Incomplete radial** air jets are formed by outlets with grilles having diverging vanes and a forced angle of expansion. At a distance, this jet tends to transform into a compact one.
- **Swirling** air jets are formed by diffusers with vortex-forming devices. These devices create rotation, which has, in addition to the axial component of velocity vectors, tangential and radial ones. Depending on the type of air diffuser, swirling jets can be compact, conical, or radial.

Isothermal Jets

The shape of jets at a short distance from the outlet face is very similar whether the outlet is round, rectangular, grille-like, or a perforated panel. The jet discharged from a round opening forms an expanding cone; jets from rectangular outlets rapidly pass from a rectangular to an elliptical cross-sectional shape and then to a circular shape, at a rate depending primarily on the aspect ratio and jet width. Even for wide-angle grilles and annular outlets, the similarities permit the same performance analysis for both.

For many conditions of jet discharge, it is possible to analyze jet performance and determine (1) the angle of divergence of the jet boundary, (2) the velocity patterns along the jet axis, (3) the velocity profile at any cross section in the zone of maximum engineering importance, and (4) the entrainment ratios in the same zone (Tuve 1953).

Using the data in this section, the following must be considered:

1. Because the method of finding the jet velocities is based on several approximations, the two recommended equations [Equations (3) and (4)] must be used cautiously for extreme axial and radial distances.
2. The characteristics of the low-velocity regions of ventilating jets are not well understood, and the effects at various Reynolds numbers are not fully known for axial or radial jets.
3. The quantitative treatment of the forces governing room air diffusion problems is limited, and nonisothermal conditions involving buoyant forces are more difficult to predict.
4. Most investigations have addressed free jets, whereas airstreams in practical room air diffusion are not free streams but are influenced by walls, ceilings, floors, and other obstructions.

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. The internal forces governing this air motion are extremely delicate (Nottage et al. 1952b).

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). In cases where the 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, m
A_R = cross-sectional area of confined space normal to jet, m^2

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

Zone 1. A core zone; a short 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, the length of which depends on the type of outlet, aspect ratio of the outlet, initial airflow turbulence, and so forth.

Zone 3. A zone of fully established turbulent flow that may be 25 to 100 equivalent air outlet diameters (widths for slot-type air diffusers) long.

Zone 4. A zone of diffuser jet degradation, where the maximum air velocity and temperature decreases rapidly. The distance to this zone and its length depend on the velocities and turbulence characteristics of the ambient air. In a few diameters or widths, the air velocity becomes less than 0.25 m/s. The 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,

$$\frac{V_x}{V_o} = \sqrt{\frac{1.13KH_o}{X}} \tag{2}$$

where

V_x = centerline velocity at distance X from outlet, m/s
V_o = V_c/C_dR_{fa} = average initial velocity at discharge from open-ended duct or across contracted stream at vena contracta of orifice or multiple-opening outlet, m/s
V_c = nominal velocity of discharge based on core area, m/s
C_d = discharge coefficient (usually between 0.65 and 0.90)
R_{fa} = ratio of free area to gross (core) area

Table 1 Recommended Values for Centerline Velocity Constant K for Commerical Supply Outlets

Outlet Type	Discharge Pattern	Area A	K[a]
High sidewall grilles (Figure 1A)	0° deflection [b]	Core	5.0
	Wide deflection	Core	3.7
High sidewall linear (Figure 1B)	Core less than 100 mm high [c]	Core	3.9
	Core more than 100 mm high	Core	4.4
Low sidewall (Figure 1C)	Up and on wall, no spread	Core	4.4
	Wide spread [c]	Core	2.6
Baseboard (Figure 1C)	Up and on wall, no spread	Duct	3.9
	Wide spread [e]	Duct	1.8
Floor (Figure 1C)	No spread [c]	Core	4.1
	Wide spread	Core	1.4
Ceiling circular directional (Fig. 1D)	360° horizontal [d]	Duct	1.0
	Four-way—little spread	Duct	3.3
Ceiling linear (Figure 1E)	One-way—horizontal along ceiling [c]	Core	4.8

[a] These values are representative for commercial outlets with discharge patterns as shown in Figure 1.
[b] Free area is about 80% of core area.
[c] Free area is about 50% of core area.
[d] Cone free area is greater than duct area.
[e] Face free area is greater than duct area.

H_o = width of jet at outlet or at vena contracta, m
K = centerline velocity constant depending on outlet type and discharge pattern (see Table 1)
X = distance from outlet to measurement of centerline velocity V_x, m

The aspect ratio (Tuve 1953) and turbulence (Nottage et al. 1952b) primarily affect the 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 100 mm. This is particularly true of nonisothermal jets. When H_o is very small, the induced air can penetrate the core of the jet, thus reducing the centerline velocities. The difference in performance between the radial outlet with a small H_o and the axial outlet with a large H_o shows the importance of the thickness of the jets.

When air is discharged from relatively large perforated panels, the constant velocity core formed by the coalescence of the individual jets extends a considerable distance from the panel face. In Zone 1, when the ratio $X/\sqrt{A_c}$ (Distance from Panel/$\sqrt{\text{Panel Area}}$) is less than 5, the following equation should be used 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 with accuracy from the following equations:

$$\frac{V_x}{V_o} = \frac{K D_o}{X} = \frac{1.13 K \sqrt{A_o}}{X} \tag{4}$$

$$V_x = \frac{1.13 K V_o \sqrt{A_o}}{X} = \frac{1.13 KQ}{X\sqrt{A_o}} \tag{5}$$

$$V_x = \frac{1.13 KQ}{X\sqrt{A_c C_d R_{fa}}} \tag{6}$$

where

K = proportionality constant
D_o = effective or equivalent diameter of stream at discharge from open-end duct or at contracted section, m

Table 2 Recommended Values of Centerline Velocity Constant for Standard Openings

Type of Outlet	K: V_o = 2.5 to 5 m/s	K: V_o = 10 to 50 m/s
Free openings		
Round or square	5.0	6.2
Rectangular, large aspect ratio (<40)	4.3	5.3
Annular slots, axial or radial[a]	—	—
Grilles and grids		
Free area 40% or more	4.1	5.0
Perforated panels		
Free area 3 to 5%	2.7	3.3
Free area 10 to 20%	3.5	4.3

[a]For radial slots, use X/H instead of $X/\sqrt{A}$. H is height or width of slot.
Note: K is an index of loss in axial kinetic energy. Interpolate as required. Departures from maximum value indicate losses in Zones 1 and 2 when compared with the jet from a rounded-entrance, circular nozzle.

A_o = core area A_c or duct area, m^2
A_c = measured gross (core) area of outlet, m^2
Q = discharge from outlet, m^3/s

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.

Equation (4) is nondimensional and requires only that consistent units be used. Values of K are listed in Table 2 (Tuve 1953, Koestel et al. 1950).

In multiple-opening outlets and annular ring outlets, the streams coalesce into a solid jet before actual jet expansion takes place. This coalescence affects the proportionality constant K and accounts for some divergence in reported values for similar outlets.

For perforated panels of relatively large size, the values of K given in Table 2 apply only when the ratio $X/\sqrt{A_c}$ is larger than 5 (see the section on Centerline Velocities in Zones 1 and 2).

Low-velocity test results, in the range $V_x < 0.75$ m/s, indicate that normal values of K should be reduced about 20% for $V_x = 0.25$ m/s, as used later in Equation (9) for throw.

Determining Centerline Velocities. To correlate data from all four zones, centerline velocity ratios are plotted against distance from the outlet in Figure 2.

The airflow patterns of diffusers are related to the throw K-factors and to the throw distance. In general, diffusers that have a circular airflow pattern have a shorter throw than those with a directional or crossflow pattern. During cooling, the circular pattern has a tendency to curl back from the end of the throw toward the diffuser. This action reduces the drop and ensures that the cool air remains near the ceiling.

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

The following example illustrates the use of Table 1 and Figure 2.

Example 1. A 300 mm by 450 mm high sidewall grille with a 280 mm by 430 mm core area is selected. From Table 1, K = 5 for Zone 3. If the airflow is 0.3 m^3/s, what is the throw to 0.25 m/s, 0.5 m/s, and 0.75 m/s? The grille has 80% free area.

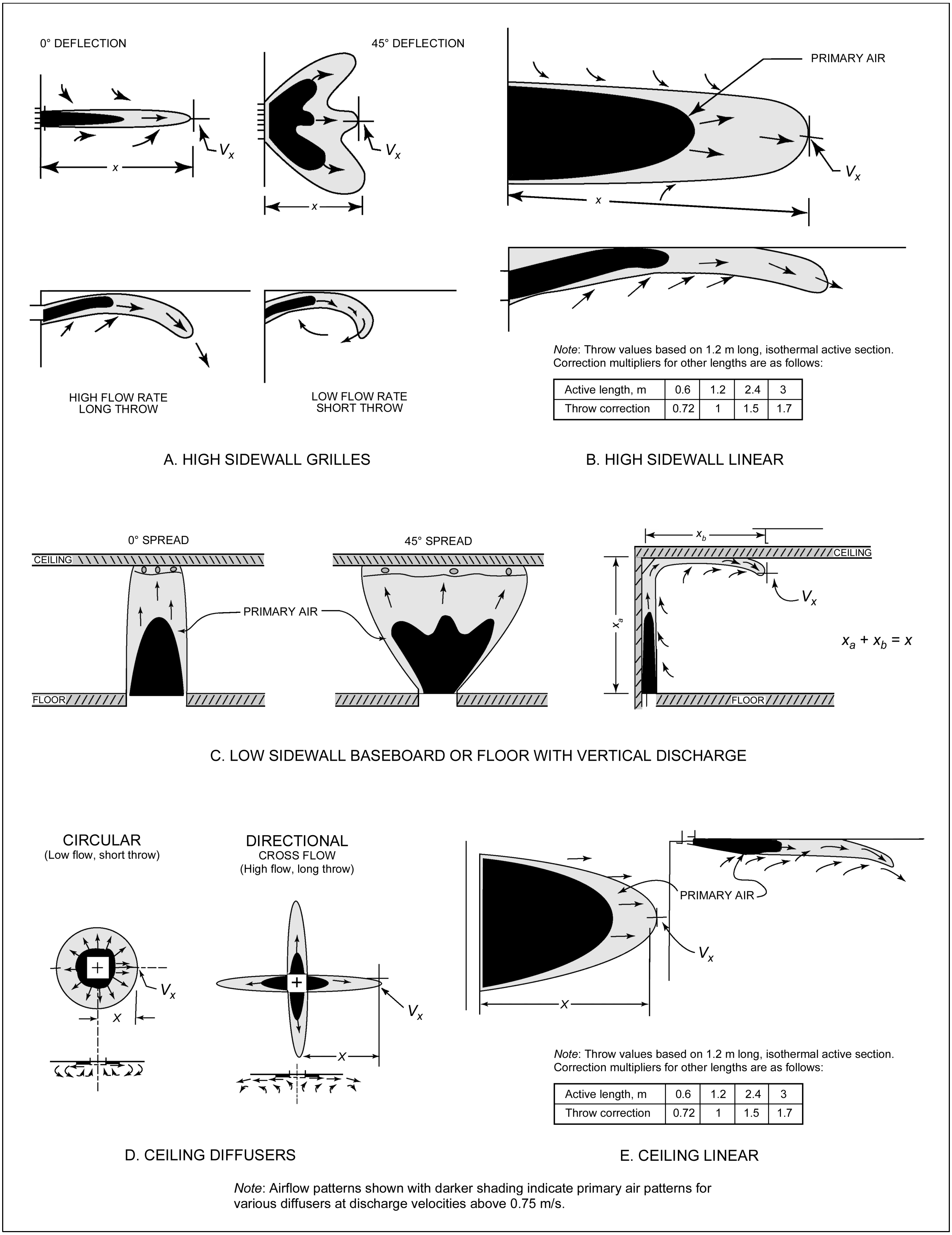

Fig. 1 Airflow Patterns of Different Diffusers

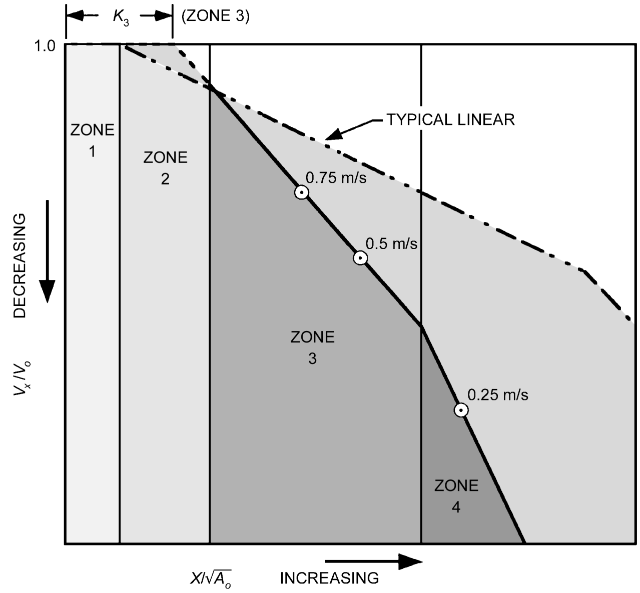

Fig. 2 Chart for Determining Centerline Velocities of Axial and Radial Jets

Solution:

From Equation (5):

$$X = \frac{1.13KQ}{V_x\sqrt{A_o}} = \frac{1.13 \times 5 \times 0.3}{V_x\sqrt{280 \times 430/10^6}} = \frac{4.885}{V_x}$$

Solving for 0.25 m/s throw,

$$X = \frac{4.885}{0.25} = 19.5 \text{ m}$$

But according to Figure 2, 0.25 m/s is in Zone 4, which is typically 20% less than calculated in Equation (4), or

$$X = 19.5 \times 0.80 = 15.6 \text{ m}$$

Solving for 0.50 m/s throw,

$$X = \frac{4.885}{0.50} = 9.8 \text{ m}$$

Solving for 0.75 m/s throw,

$$X = \frac{4.885}{0.75} = 6.5 \text{ m}$$

Throw. Equation (6) can be transposed to determine the throw X of an outlet if the discharge volume and the centerline velocity are known:

$$X = \frac{1.13K}{V_x}\frac{Q}{\sqrt{A_c C_d R_{fa}}} \tag{7}$$

Or, if $Z = \sqrt{C_d R_{fa}}$,

$$X = \frac{1.13K}{V_x}\frac{Q}{Z\sqrt{A_c}} \tag{8}$$

The maximum throw T_V is usually defined as the distance from the outlet face to where the centerline velocity is 0.25 m/s. Therefore, for V_T = 0.25 m/s,

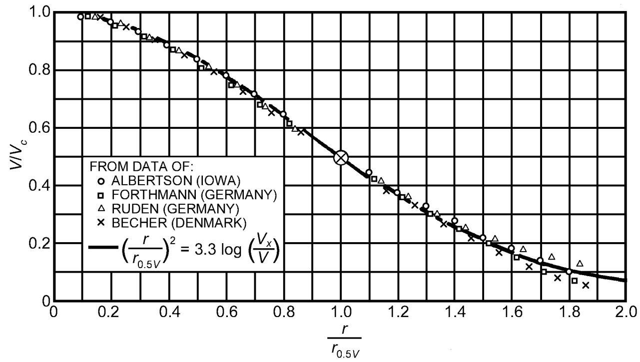

Fig. 3 Cross-Sectional Velocity Profiles for Straight-Flow Turbulent Jets

$$T_V = X = \frac{1.13K}{0.25}\frac{Q}{Z\sqrt{A_c}} \tag{9}$$

Any other terminal centerline velocity could be inserted in Equation (9) for V_T.

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:

$$\left(\frac{r}{r_{0.5V}}\right)^2 = 3.3 \log\frac{V_x}{V} \tag{10}$$

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 3. 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 are equations for the entrainment of circular jets and of jets from long slots.

For third-zone expansion of circular jets,

$$\frac{Q_x}{Q_o} = \frac{2X}{1.13K\sqrt{A_o}} \tag{11}$$

By substituting from Equation (4),

$$\frac{Q_x}{Q_o} = 2\frac{V_o}{V_x} \tag{12}$$

For a continuous slot with active sections up to 3 m and separated by 0.6 m,

$$\frac{Q_x}{Q_o} = \sqrt{\frac{2}{1.13K}}\sqrt{\frac{X}{H_o}} \tag{13}$$

or, substituting from Equation (2),

$$\frac{Q_x}{Q_o} = \sqrt{2}\,\frac{V_o}{V_x} \tag{14}$$

where

Q_x = total volumetric flow rate at distance X from face of outlet, m³/s
Q_o = discharge from outlet, m³/s
X = distance from face of outlet, m
K = proportionality constant
A_o = core area A_c or duct area, m²
H_o = width of slot, m

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 (11)] or to the square root of the distance X [Equation (13)] from the outlet. Equations (12) and (14) show that, for a fixed centerline velocity V_x, the entrainment ratio is proportional to the outlet velocity. Equations (12) and (14) 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 (11) and (13), 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 the 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 behavior of the diffuser air jet is affected by the thermal buoyancy due to air density difference. The trajectory of a nonisothermal jet introduced horizontally is determined by the **Archimedes number** (Baturin 1972):

$$\mathrm{Ar} = \frac{gL_o(t_o - t_s)}{V_o^2 T_s} \tag{15}$$

where

g = gravitational acceleration rate, m/s²
L_o = length scale of diffuser outlet equal to hydraulic diameter of outlet, m
t_o = initial temperature of jet, °C
t_s = temperature of surrounding air, °C
V_o = initial air velocity of jet, m/s
T_s = room air temperature, K

The paths assumed by horizontally projected heated and chilled jets influenced by buoyant forces are significant in heating and cooling with wall outlets. Koestel's equation (1955) describes the behavior of these jets.

Helander and Jakowatz (1948), Helander et al. (1953, 1954, 1957), Yen et al. (1956), and Knaak (1957) 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. Li et al. (1993, 1995) and Kirkpatrick and Elleson (1996) provide additional information on nonisothermal jets.

Surface Jets (Wall and Ceiling)

Jets discharging parallel to a surface with one edge of the outlet coinciding with the surface take the form of one-half of an axial jet discharging from an outlet twice as large, similar to radial jets from ceiling plaques. Entrainment takes place almost exclusively along the surface of a half cone, and the maximum velocity remains close to the surface (Tuve 1953).

Values of 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 and adjacent to a 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. 1952a, 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 from the surface again.

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

MULTIPLE JETS

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

Koestel and Austin (1956) determined the spacing between outlets for noninterference between the jets. For a K value of 6.5, the outlets should be placed three to eight diameters apart, with V_o values from 2.5 to 7.5 m/s.

AIRFLOW IN OCCUPIED ZONE

Mixing Systems. Laboratory experiments on jets usually involve recirculated air with negligible resistance to flow on the return path of the jet air. Experiments in mine tunnels of small cross-sectional areas, where the return flow of jet air to outlets meets considerable resistance, show that expansion of the jet terminates abruptly at a distance that is independent of discharge velocity and is only slightly affected by the size of the outlet. These distances are determined primarily by the size and length of the return path. In a long tunnel with a cross section of 1.5 m by 1.8 m, a jet may not travel more than 7.5 m; in a tunnel with a relatively large section (7.5 m by 18 m), the jet may travel more than 75 m. 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 the outlet discharge velocity increases. Therefore, the design supply air velocity should be high enough to maintain the jet traveling in the desired direction in order to ensure good mixing before it reaches the occupied zone. Excessively high outlet air velocity would induce high air velocity in the occupied zone and result in thermal discomfort.

Turbulence Production and Transport. The air turbulence within a room is mainly produced at the diffuser jet region by interaction of the supply air with the room air and with the 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). Meanwhile, the turbulence is also damped by viscous effect. Air in the occupied zone usually contains very small amounts of turbulent kinetic energy compared to that in the jet region. Because turbulence may cause thermal discomfort (Fanger et al. 1989), air diffusion systems should be designed so that the primary mixing between the introduced air and the room air occurs away from occupied regions.

ROOM AIR DIFFUSION METHODS

Room air diffusion systems can be classified as mixing, displacement, unidirectional, underfloor, and task ambient conditioning.

MIXING SYSTEMS

In mixing systems, conditioned air is normally discharged from air outlets at velocities much greater than those acceptable in the occupied zone. Conditioned air temperature may be above, below, or equal to the air temperature in the occupied zone, depending on the heating/cooling load. The diffuser jets mix with the ambient room air by entrainment, which reduces the air velocity and equalizes the air temperature. The occupied zone is ventilated either by the decayed air jet directly or by the reverse flow created by the jets. Mixing air distribution creates relatively uniform air velocity, temperature, humidity, and air quality conditions in the occupied zone.

Outlet Classification and Performance

Straub et al. (1956) and Straub and Chen (1957) 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 4 through 8 show the 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:

1. The primary air (shown by dark envelopes in Figures 4 through 8) from the outlet down to a velocity of about 0.75 m/s can be treated analytically. The heating or cooling load has a strong effect on the characteristics of the primary air.
2. The total air, shown by gray envelopes in Figures 4 through 8, is influenced by the primary air and is of relatively high velocity (but less than 0.75 m/s). The total air is also influenced by the environment and drops during cooling or rises during heating; it is not subject to precise analytical treatment.
3. 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, the air velocities within it are usually low (approximately 0.1 m/s), 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.
4. A return inlet affects the room air motion only within 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.
5. The general room air motion (shown by white areas in Figures 4 through 8) is a gentle drift toward the total air. Room conditions are maintained by the entrainment of the 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 4.

The 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 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 0.5 to 0.75 m/s may be found near the wall but will dissipate within about 100 mm of the wall.

The cooling diagram for the ceiling outlet shows that the total air movement is counteracted by the 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 the 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 θ [see Equation (16)]. 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 the 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.

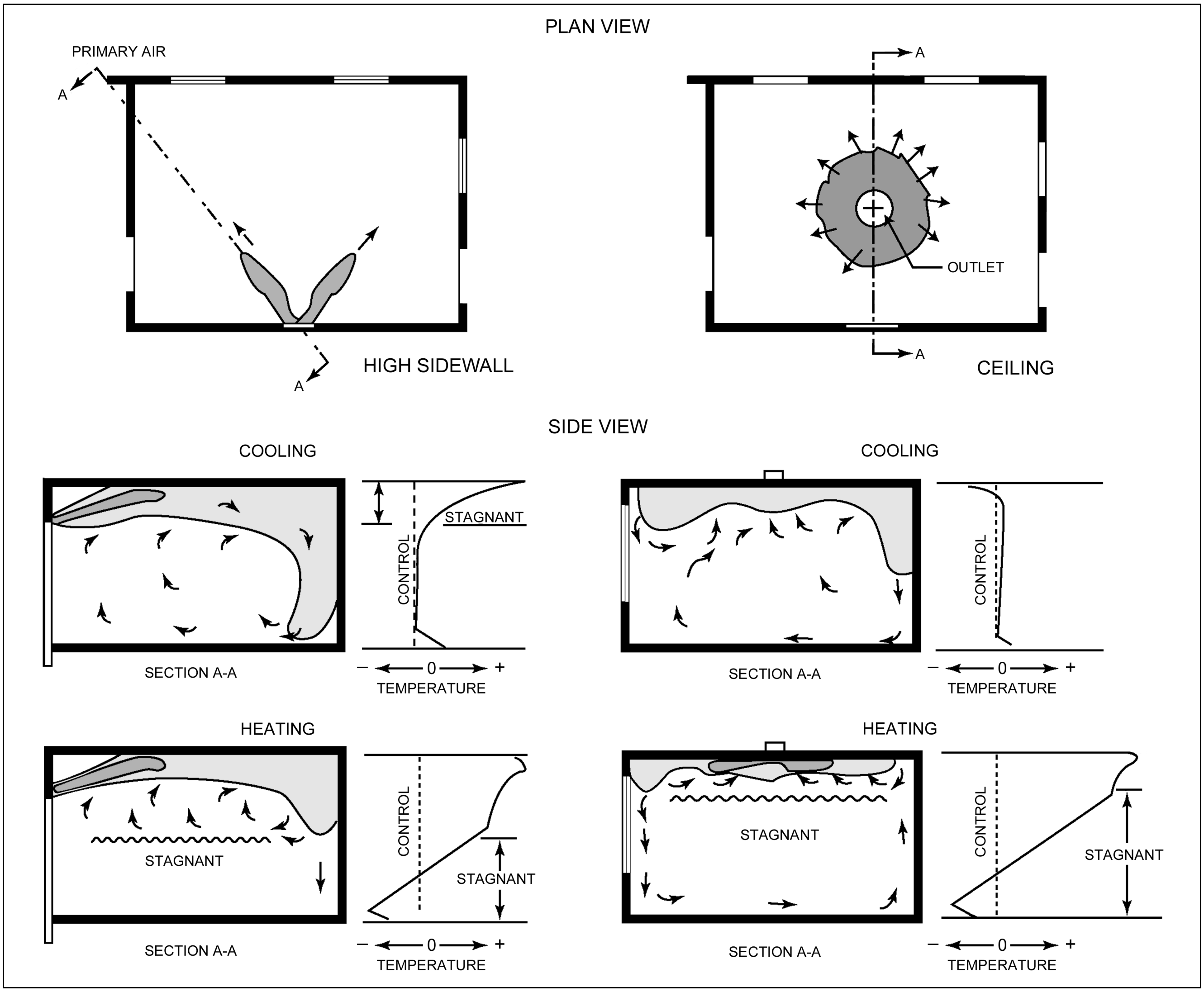

Fig. 4 Air Motion Characteristics of Group A Outlets
(Straub et al. 1956)

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 diffusion 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 4. The heating diagram for the sidewall unit shows that, under these conditions, the total air does not descend along the wall. Consequently, higher velocities might be beneficial in eliminating the stagnant zone, since 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 the natural convection currents that produce a larger throw toward the cold exposed wall. The velocity of the total air toward the exposed wall complements the 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 the total air replaces the 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 5 illustrates a floor outlet adjacent to an inside wall.

Because these outlets have no deflecting vanes, the primary air is discharged in a single, vertical jet. When the total air strikes the ceiling, 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 the total air is cooled satisfactorily. For example, if total airflow is projected upward for 2.4 m, the region from this level down to the floor will be cooled satisfactorily. This, however, does not apply to an extremely large space. Judgment to determine the acceptable size of the space outside the total air is needed. A distance of 4.5 to 6 m between the drop region and the exposed wall is a conservative design value.

A comparison of Figures 4 and 5 for heating shows that the stagnant region is smaller for Group B outlets than for Group A outlets

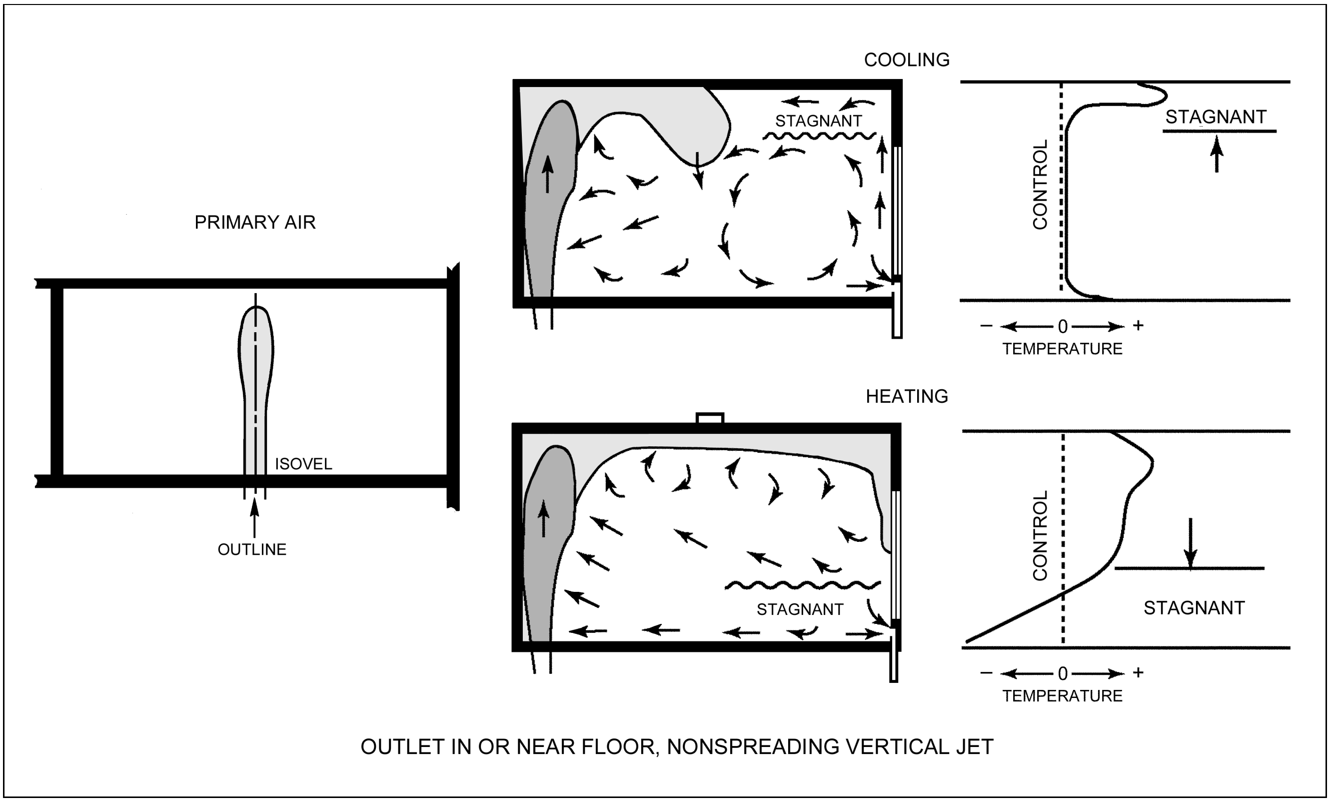

Fig. 5 Air Motion Characteristics of Group B Outlets
(Straub et al. 1956)

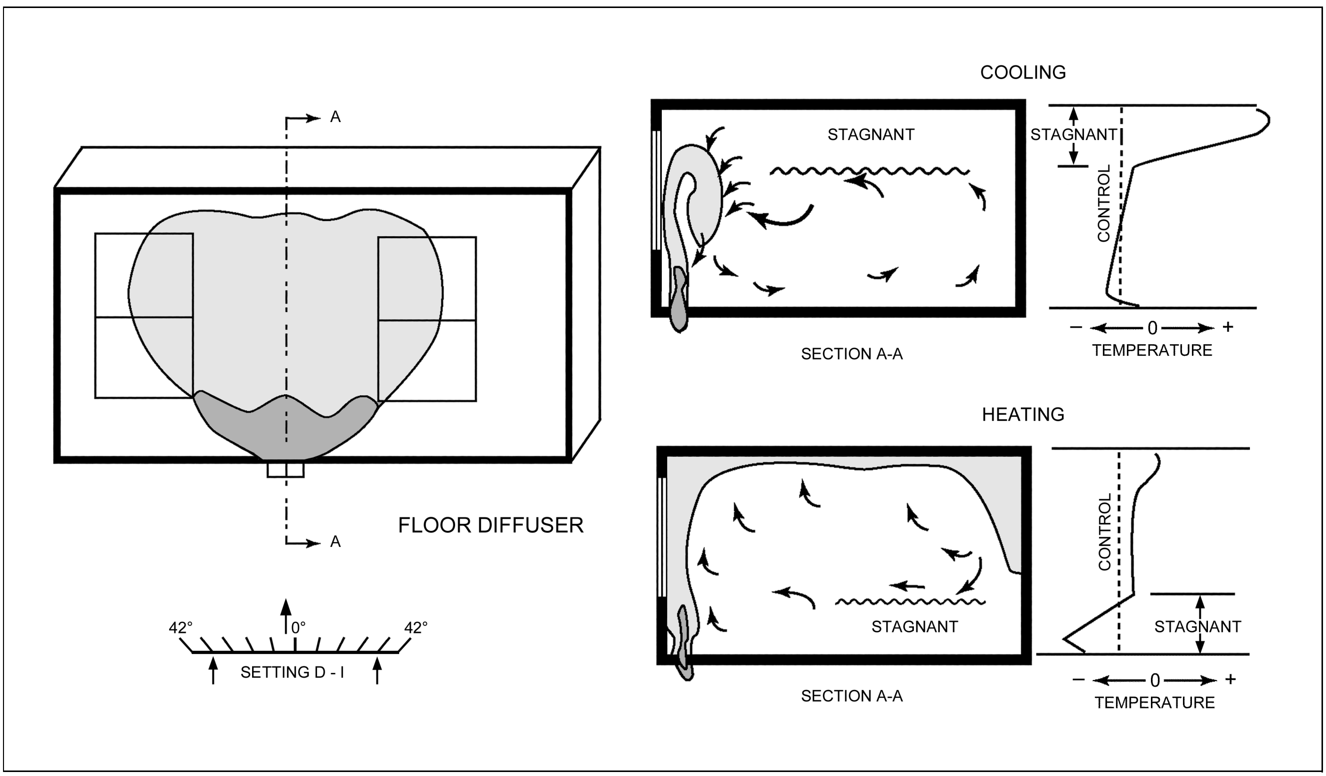

Fig. 6 Air Motion Characteristics of Group C Outlets
(Straub et al. 1956)

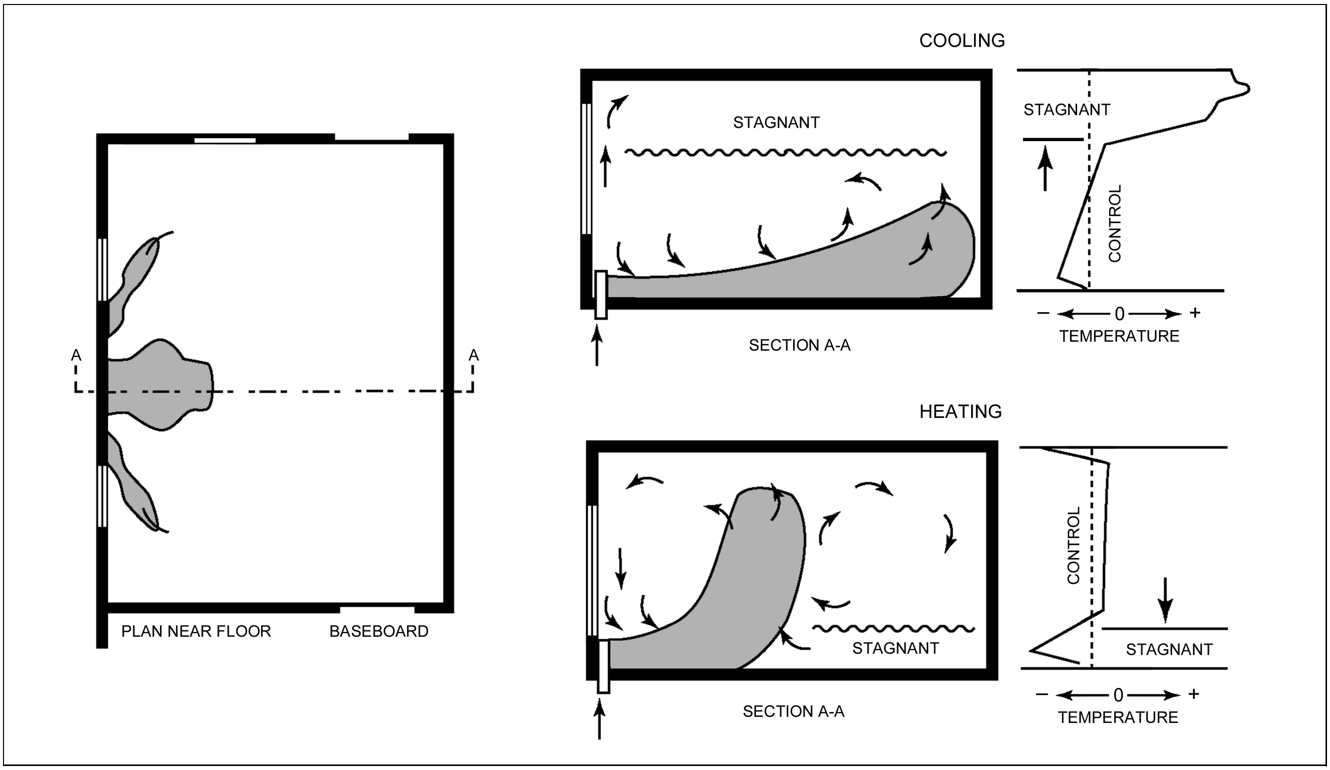

Fig. 7 Air Motion Characteristics of Group D Outlets
(Straub et al. 1956)

because the air entrained in the immediate vicinity of 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 would occur with Group A outlets.

While the temperature gradients for both outlet groups are about the same, the stagnant layer for Group B is lower than that 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 6).

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 formed is larger during cooling and smaller during heating. Diffusion of the primary air usually causes the total air to fold back on the primary and total air during cooling, instead of following the ceiling. This diffusing action of the outlets makes it more difficult to project the cool air, but it also provides a greater area for induction of room air. This action is beneficial during heating because the 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 7) that discharge the primary air in single or multiple jets. During cooling, because the 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 of warm air. The temperature variations are uniform, except in the total air region.

Group E Outlets. This group includes ceiling diffusers, linear-type grilles, sidewall diffusers and grilles, and similar outlets mounted or designed for vertical downward air projection. Figure 8 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.

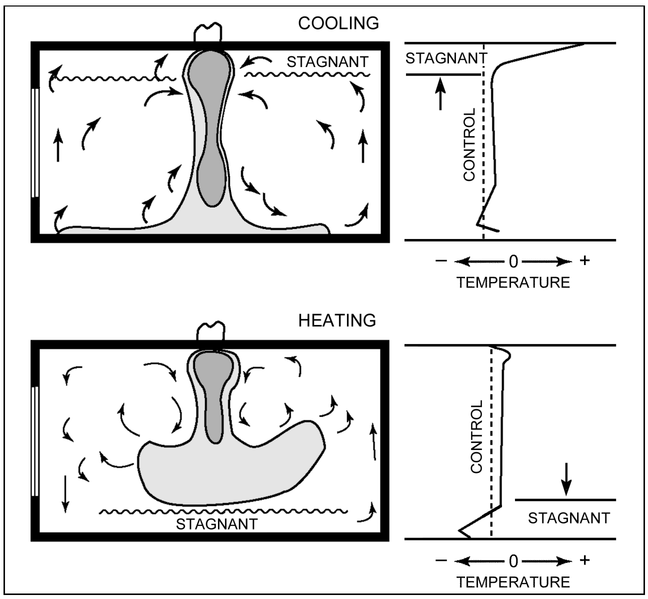

Fig. 8 Air Motion Characteristics of Group E Outlets
(Straub et al. 1956)

Factors Affecting Outlet Performance

Vanes. Vanes affect grille performance if their depth is at least equal to the distance between the vanes (vane ratio ≥ 1). If the vane ratio is less than unity, effective control by the vanes of the airstream discharged from the grille is impossible. Increasing the vane ratio above two has little or no effect, so vane ratios should be between one and two.

A grille discharging air uniformly forward (vanes in straight position) has a spread of 14 to 24°, depending on the type of outlet, the duct approach, and the discharge velocity. Turning the vanes influences the direction and throw of the discharged airstream.

A grille with diverging vanes (vertical vanes with uniformly increasing angular deflection from the centerline to a maximum at each end of 45°) has a spread of about 60° and reduces the throw considerably. With increasing divergence, the quantity of air discharged by a grille for a given upstream total pressure decreases.

A grille with converging vanes (vertical vanes with uniformly decreasing angular deflection from the centerline) has a slightly higher throw than a grille with straight vanes, but the spread is approximately the same for both settings. The airstream converges slightly for a short distance in front of the outlet and then spreads more rapidly than air discharged from a grille with straight vanes.

In addition to vertical vanes that normally spread the air horizontally, horizontal vanes may spread the air vertically. However, spreading the air vertically risks hitting beams or other obstructions or blowing primary air into the occupied zone at excessive velocities. On the other hand, vertical deflection may increase adherence to the ceiling and reduce the drop.

Beamed Ceilings and Obstructions. In spaces with exposed beams, the outlets should be located below the bottom of the lowest beam level, preferably low enough to employ an upward or arched air path. The air path should be arched sufficiently to miss the beams and prevent the primary or induced airstream from striking furniture and obstacles and producing objectionable drafts (Wilson 1970). Obstructions influence airflow patterns and can reduce air distribution efficiency. Obstructions can reduce jet throw, increase air velocities in portions of the occupied zone, and create stagnant zones.

Variable Air Volume (VAV) Systems. The design of air distribution systems is usually based on the full load (heating/cooling). When only a partial load exists, VAV systems reduce the supply airflow, which in turn reduces the air velocity at the outlet. Therefore, the different operation modes of the system (airflow and initial temperature difference) should be considered in designing a VAV system air distribution.

DISPLACEMENT VENTILATION

In displacement ventilation, conditioned air with a temperature slightly lower than the desired room air temperature in the occupied zone is supplied from air outlets at low air velocities (0.5 m/s or less). The outlets are located at or near the floor level, and the supply air is introduced directly to the occupied zone. Returns through which the warm room air is exhausted from the room are located at or close to the ceiling. The supply air is spread over the floor and then rises as it is heated by the heat sources in the occupied zone. Heat sources (e.g., person, computer) in the occupied zone create upward convective flows in the form of thermal plumes. These thermal plumes tend to remove heat and contaminants within the plume from the occupied zone (Figure 9).

The air volume in the plumes increases as they rise because the plumes entrain ambient air. A stratification level exists where the airflow rate in the plumes equals the supply airflow rate. Two distinct zones are thus formed within the room: one lower zone below the stratification level and with no recirculation flow (close to displacement flow), and one upper zone, with recirculation flow (Figure 9). The height of the lower zone depends on the supply airflow rate and the characteristics of heat sources and their distribution across the floor area. In a properly designed displacement ventilation system, the upper boundary of the lower zone is above the occupied zone so that the occupied zone can be ventilated effectively. For this type of system to function properly, a stable vertically stratified temperature field is essential.

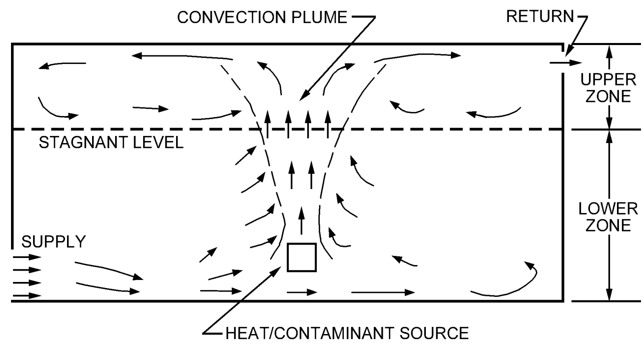

Fig. 9 Schematic of Displacement Ventilation

In contrast to mixing ventilation, displacement ventilation is designed to minimize mixing of air within the occupied zone. The objective of the displacement ventilation is to create conditions close to supply air conditions in the occupied zone. This type of ventilation was originally used in industrial buildings as an effective method for removing contaminants in the occupied zone. It is now also used for ventilating and cooling office buildings. However, local discomfort due to draft and vertical temperature gradient may be critical (Melikov and Nielsen 1989). Sandberg and Blomqvist (1989) suggest that the maximum convective cooling load in office buildings with displacement ventilation not exceed about 25 W/m^2 so that the maximum vertical temperature gradient in the occupied zone will not be larger than 3 K. This is equivalent to 5 L/(s·m^2) at a maximum cooling differential of 4 K. Kegel and Schulz (1989) and Svensson (1989) suggest somewhat higher cooling load limits of 30 to 40 W/m^2.

One way of increasing the cooling capacity of displacement ventilation systems is to recirculate some of the room air in the occupied zone through an induction circuit; that is, the room air is induced into the supply air and is mixed before discharge through the low-velocity air terminal device into the room. This reduces the room air temperature gradient for a given cooling load, thus allowing a cooling load limit of up to 50 W/m^2 (Jackman 1991).

Air diffusers with a large outlet area are used to supply air at low velocity. Displacement ventilation has been compared with conventional mixing ventilation (Svensson 1989, Seppanen et al. 1989, Stymne et al. 1991). Design guidelines for displacement ventilation can be found in Scaret (1985), Jackman and Appleby (1990), Jackman (1991), and Shilkrot and Zhivov (1992).

UNIDIRECTIONAL AIRFLOW VENTILATION

In this type of ventilation, air is either (1) supplied from the ceiling and exhausted through the floor, or vice versa; or (2) supplied through the wall and exhausted through returns at the opposite wall. The outlets are uniformly distributed over the ceiling, floor, or wall to provide a low-turbulence "plug"-type flow across the entire room. This type of system is mainly used for clean room ventilation, in which the main objective is to remove contaminant particles from the room. Details about clean room ventilation are given in Chapter 15 of the *ASHRAE Handbook—Applications*. Unidirectional flow ventilation is also used in other areas, such as computer rooms and paint booths.

UNDERFLOOR AIR DISTRIBUTION AND TASK/AMBIENT CONDITIONING

Underfloor air distribution systems are installed with a raised floor through which conditioned air is delivered to the space through floor grilles or as part of the workstation furniture and partitions. Sometimes called localized ventilation, these systems

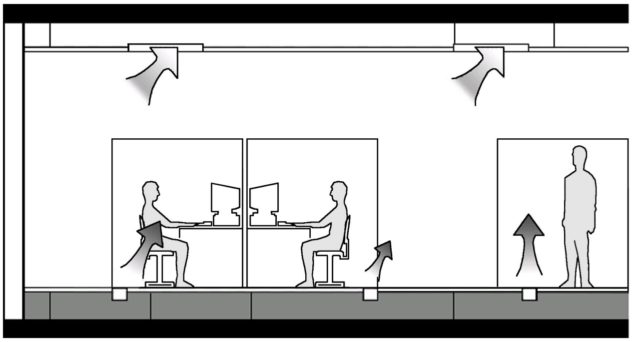

Fig. 10 Underfloor Air Distribution System

supply air to local areas that are often near building occupants or other specific locations in the space. In comparison to conventional ceiling-based air diffusion, underfloor air distribution systems generally have a larger number of supply diffusers directly in the occupied zone of the building (e.g., in floors, desks, workstation partitions, or theater seats). Air typically returns at or close to ceiling level, so that localized systems benefit from the same overall upward movement of air in the room as displacement ventilation systems. In cooling applications, this air movement efficiently removes heat and contaminant sources from the room.

Underfloor air distribution differs from displacement ventilation in that (1) it generally uses higher supply volumes, which enable higher cooling loads to be met; and (2) it supplies air at a higher velocity through smaller diffusers. Because air is delivered directly to the occupied zone, the supply air temperature is usually warmer (above about 17.5°C) than that maintained for conventional ceiling distribution in order to avoid occupant discomfort due to drafts. Bauman et al. (1999), Hanzawa and Nagasawa (1990), Houghton (1995), Loudermilk (1999), McCarry (1995, 1998), Shute (1992, 1995), Sodec and Craig (1990), Spoormaker (1990), and Tanabe and Kimura (1996) describe the results of laboratory studies, case studies of actual installations, field experiments of system performance, and present design guidelines.

A well-designed underfloor air distribution system also requires less energy and is more flexible in providing and maintaining building services than traditional overhead systems. Extremely low operational static pressures in the underfloor air supply plenum can reduces central fan energy use. Thermal storage in the exposed structural mass in the underfloor plenum (e.g., concrete slab) can save energy and reduce peak cooling loads. The use of raised flooring provides maximum flexibility and significantly lowers costs associated with reconfiguring building services, particularly in buildings having high churn rates. (Churn rate is defined as the annual percentage of workers and their associated work spaces in a building that are reconfigured or undergo significant changes.) Figure 10 shows a schematic diagram of an underfloor air distribution system.

Task/Ambient Conditioning (TAC)

Task/ambient conditioning (TAC) is most commonly installed with underfloor air distribution (Arens et al. 1991; Bauman et al. 1991, 1993, 1995, 1998; Bauman and Arens 1996; 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 the incoming air supply. TAC systems have been most frequently installed in open-plan offices in which they provide supply air and, in some cases, radiant heating directly into workstations. Figure 11 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).

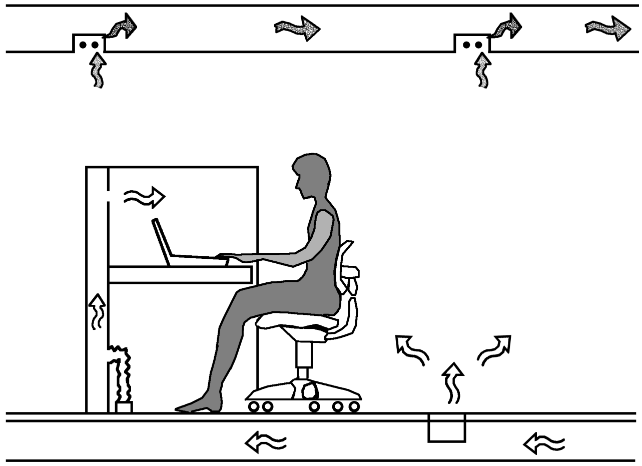

Fig. 11 Underfloor TAC and Personal HVAC System
(Matsunawa et al. 1995)

As further evidence of the benefits of providing personal control, field research has 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 (de Dear and Brager 1999, Bauman et al. 1998).

METHODS OF EVALUATION

Standards for Satisfactory Conditions

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 1.8 m above floor level (Miller 1989). The effective draft temperature combines the effects of air temperature, air motion, and relative humidity in terms of their physiological effects on a human body. Variation from accepted standard limits (ASHRAE *Standard* 55) may cause occupant discomfort. Lack of uniform conditions within the space or excessive fluctuation of conditions in the same part of the space also produces discomfort. Discomfort can arise due to 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 the load requirements at different locations
- Overly rapid fluctuation of room temperature

Draft. Koestel and Tuve (1955) and Reinmann et al. (1959) studied the effect of air motion on comfort and defined **draft** as any localized feeling of coolness or warmth of any portion of the body due to 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 24°C dry-bulb at the center of the room, 0.75 m above the floor, with air moving at about 0.15 m/s.

To define the **effective draft temperature** θ (the difference in temperature between any point in the occupied zone and the control condition), the investigators used the following equation proposed by Rydberg and Norback (1949) and modified by Straub in discussion of a paper by Koestel and Tuve (1955):

$$\theta = (t_x - t_c) - 8(V_x - 0.15) \tag{16}$$

where

θ = effective draft temperature, K
t_x = local airstream dry-bulb temperature, °C
t_c = average (control) room dry-bulb temperature, °C
V_x = local airstream centerline velocity, m/s

Equation (16) accounts for the feeling of coolness produced by air motion and is used to establish the neutral line in Figure 12. 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 0.15 m/s, and they both add to the feeling of coolness. If, in winter, t_x is above t_c, any air velocity above 0.15 m/s 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 that make it possible to interpret statistically the percentage of room occupants that will object to a given draft condition. Figure 12 presents the data in the form used by Koestel and Tuve (1955). The data show that a person tolerates higher velocities and lower temperatures at ankle level than at neck level. Because of this, conditions in the zone extending from approximately 0.75 to 1.5 m above the floor are more critical than conditions nearer the floor.

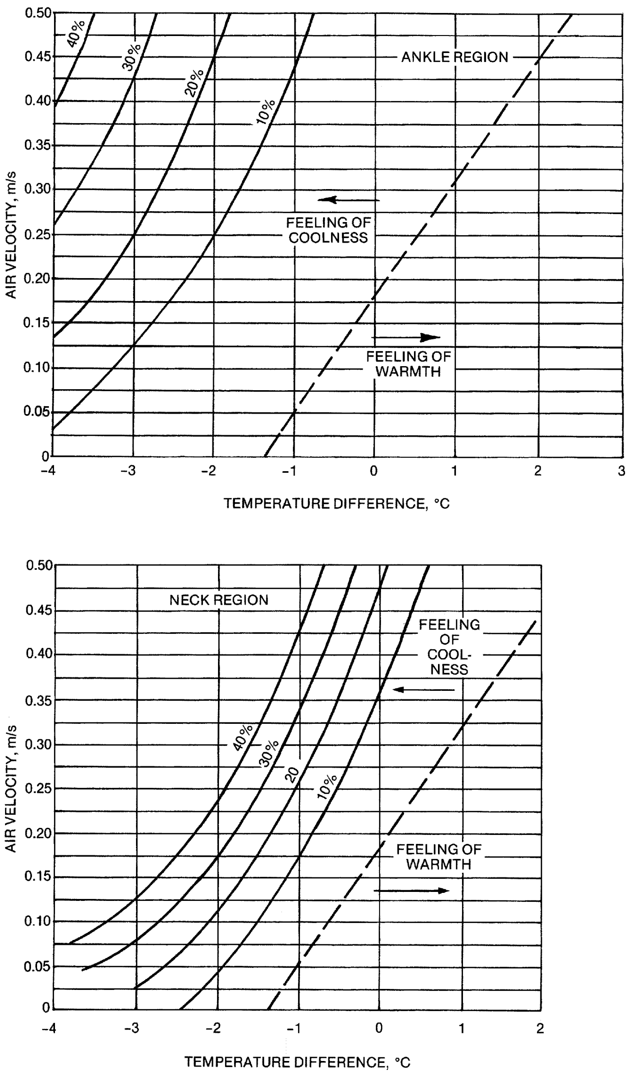

Fig. 12 Percentage of Occupants Objecting to Drafts in Air-Conditioned Room

Air Velocity. Room air velocities less than 0.25 m/s are generally preferred; however, Figure 12 shows that even higher velocities may be acceptable to some occupants. ASHRAE *Standard* 55 recommends elevated air speeds at elevated air temperatures. No minimum air speeds are recommended for comfort, although air speeds below 0.1 m/s are usually imperceptible.

Temperature Gradient. Figure 12 also shows that up to 20% of occupants will not accept an ankle-to-sitting-level gradient of about 2 K. Poorly designed or operated systems in a heating mode can create this condition, which emphasizes the importance of proper selection and operation of perimeter systems. The section on Outlet Classification and Performance describes possible regions of high room air velocities caused by various outlets; the section on Outlet Location and Selection describes how to evaluate acceptable air diffusion.

AIR DIFFUSION PERFORMANCE INDEX (ADPI)

A high percentage of people are comfortable in sedentary (office) occupations where the effective draft temperature θ, as defined in Equation (16), is between −1.5 and +1 K and the air velocity is less than 0.35 m/s. If several measurements of air velocity and air temperature are made throughout the occupied zone of an office, the ADPI is the percentage of locations where measurements were taken that meet these specifications for effective draft temperature and air velocity. If the ADPI is maximum (approaching 100%), the most desirable conditions are achieved (Miller and Nevins 1969, 1970, 1972, 1974; Miller 1971; Miller and Nash 1971; Nevins and Ward 1968; Nevins and Miller 1972).

The ADPI is based only on air velocity and the effective draft temperature (a combination of local temperature variations from the room average) and is not directly related to the 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.

The ADPI is 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. The ADPI technique uses isothermal throw data determined under ASHRAE *Standard* 70 (see Table 4) to predict what will happen under cooling conditions.

ADPI Selection Guide

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 0.25 m/s, except in the case of ceiling slot diffusers, where it was selected as 0.5 m/s. Each manufacturer gives data for the throw of a jet from various diffusers for isothermal conditions and without a boundary wall interfering with the jet.

The throw distance of a jet is denoted by T_V, where subscript V indicates the terminal velocity for which the throw is given. The **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, the characteristic length is one-half the distance between diffusers plus the distance the mixed jet travels downward to reach the occupied zone. Table 3 summarizes definitions of characteristic length for various diffusers.

The midplane between diffusers also can be considered the module line when diffusers serve equal modules throughout a

Table 3 Characteristic Room Length for Several Diffusers

Diffuser Type	Characteristic Length L
High sidewall grille	Distance to wall perpendicular to jet
Circular ceiling 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
Perforated, louvered ceiling diffusers	Distance to wall or midplane between outlets

Table 4 Air Diffusion Performance Index (ADPI) Selection Guide

Terminal Device	Room Load, W/m²	$T_{0.25}/L$ for Maximum ADPI	Maximum ADPI	For ADPI Greater than	Range of $T_{0.25}/L$
High sidewall grilles	250	1.8	68	—	—
	190	1.8	72	70	1.5–2.2
	125	1.6	78	70	1.2–2.3
	65	1.5	85	80	1.0–1.9
Circular ceiling diffusers	250	0.8	76	70	0.7–1.3
	190	0.8	83	80	0.7–1.2
	125	0.8	88	80	0.5–1.5
	65	0.8	93	90	0.7–1.3
Sill grille, straight vanes	250	1.7	61	60	1.5–1.7
	190	1.7	72	70	1.4–1.7
	125	1.3	86	80	1.2–1.8
	65	0.9	95	90	0.8–1.3
Sill grille, spread vanes	250	0.7	94	90	0.6–1.5
	190	0.7	94	80	0.6–1.7
	125	0.7	94	—	—
	65	0.7	94	—	—
Ceiling slot diffusers (for T_{100}/L)	250	0.3	85	80	0.3–0.7
	190	0.3	88	80	0.3–0.8
	125	0.3	91	80	0.3–1.1
	65	0.3	92	80	0.3–1.5
Light troffer diffusers	190	2.5	86	80	<3.8
	125	1.0	92	90	<3.0
	65	1.0	95	90	<4.5
Perforated, louvered ceiling diffusers	35–160	2.0	96	90	1.4–2.7
				80	1.0–3.4

space, and a characteristic length consideration can be based on module dimension d.

Load Considerations. The recommendations in Table 4 cover cooling loads of up to 250 W per square metre of floor surface. The loading is distributed uniformly over the floor up to about 22 W/m², lighting contributes about 30 W/m², 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 highest loads; however, the optimum design condition changes only slightly with the 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) cone-type circular ceiling diffusers, (3) sill grilles, (4) two- and four-slot ceiling diffusers, (5) light troffer diffusers, and (6) square-faced perforated and louvered ceiling diffusers. Table 2 summarizes the results of the recommendations on values of T_V/L by giving the value of T_V/L at which the ADPI is a maximum for various loads, as well as a range of values of T_V/L for which ADPI is above a minimum specified value.

SYSTEM DESIGN

DESIGN CONSIDERATIONS

Noise

The noise generated by diffusers transmits to the occupied space directly and cannot be attenuated. Therefore, the diffusion system design should meet the sound level criteria specified in Chapter 46 of the *ASHRAE Handbook—Applications*.

Duct Approaches to Diffuser Outlets

The manner in which the airstream approaches the diffuser outlet is important. For correct air diffusion, the velocity of the airstream must be as uniform as possible over the entire cross-sectional area of the connecting duct and must be perpendicular to the outlet face. Effects of improper duct approach generally cannot be corrected by the diffuser.

If the system is designed carefully, a wall grille installed at the end of a horizontal duct and a ceiling outlet at the end of a vertical duct receive the air perpendicularly and at uniform velocity over the entire duct cross section. However, few outlets are installed in this way. Most sidewall outlets are installed either at the end of vertical ducts or in the side of horizontal ducts, and most ceiling outlets are attached either directly to the bottom of horizontal ducts or to special vertical takeoff ducts that connect the outlet with the horizontal duct. In all these cases, special devices for directing and equalizing the airflow are necessary for proper direction and diffusion of the air.

The influence of the duct approach on outlet performance has been investigated for vertical stack heads with plain openings (Nelson et al. 1940) or equipped with grilles (Nelson et al. 1942) and side outlets on horizontal ducts (Nelson and Smedberg 1943). In tests conducted with the stack heads, splitters or guide vanes in the elbows at the top of the vertical stacks are needed, regardless of the shape of the elbows (rounded, square, or expanding). Cushion chambers at the top of the stack heads are not beneficial. Figure 13 shows the direction of flow, diffusion, and velocity (measured 300 mm from the opening) of the air for various stack heads tested, expanding from a 350 mm by 150 mm stack to a 350 mm by 230 mm opening, without a grille. The air velocity for each was 2.5 m/s in the stack below the elbow, but the direction of flow and the diffusion pattern indicate performance obtained with nonexpanding elbows of similar shapes for velocities from 1 to 2 m/s.

In tests conducted with 75 mm by 250 mm, 100 mm by 230 mm, and 150 mm by 150 mm side outlets in a 150 mm by 510 mm horizontal duct at duct velocities of 1 to 7 m/s in the horizontal duct section, multiple curved deflectors produced the best flow characteristics. Vertical guide strips in the outlet were not as effective as curved deflectors. A single scoop-type deflector at the outlet did not improve the flow pattern obtained from a plain outlet and, therefore, was not desirable.

Return and Exhaust Openings

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

Velocity. Airflow patterns and room air movement are not influenced by the location of the return and exhaust outlets beyond a distance of one characteristic length of the return or exhaust opening (e.g., square root of the opening area). Air handled by the opening approaches the opening from all directions, and its velocity decreases rapidly as the distance from the opening increases. Therefore, drafty conditions rarely occur near return openings. Table 5 shows recommended return opening face velocities.

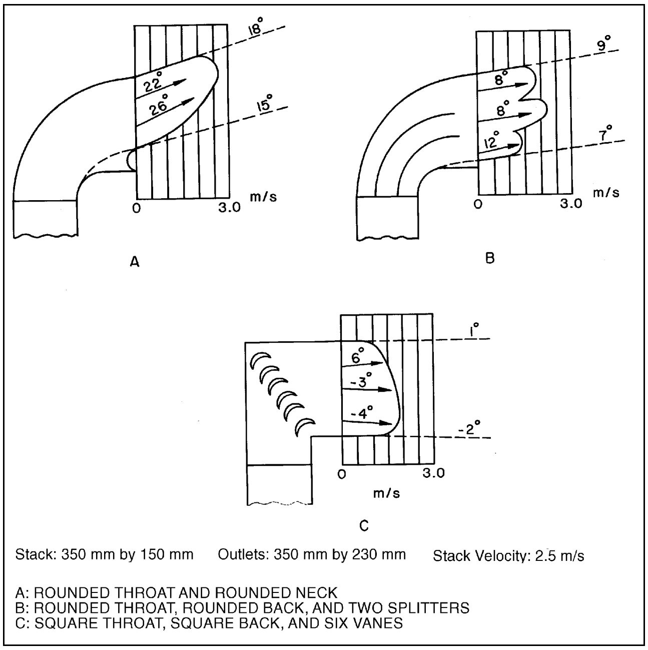

Fig. 13 Outlet Velocity and Air Direction Diagrams for Stack Heads with Expanding Outlets

Table 5 Recommended Return Inlet Face Velocities

Inlet Location	Velocity Across Gross Area, m/s
Above occupied zone	> 4
Within occupied zone, not near seats	3 to 4
Within occupied zone, near seats	2 to 3
Door or wall louvers	1 to 1.5
Through undercut area of doors	1 to 1.5

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

Noise. The problem of noise in return openings is the same as that in supply outlets. In computing room noise levels resulting from the operation of an air-conditioning system, the return opening must be included as part of the total grille area.

Location. The openings should be located to minimize short-circuiting of supply air. If air is supplied by the jets attached to the ceiling, exhaust openings should be located between the jets or at the side of the room away from the supply air jets. In rooms with vertical temperature stratification, such as foundries, computer rooms, theaters, bars, kitchens, dining rooms, and club rooms, exhaust openings should be located near the ceiling to collect warm air, odors, and fumes.

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

Exhaust outlets 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 the pressure drop through door returns is excessive, the air diffusion to the room may be seriously unbalanced by opening or closing the doors. Outward leakage through doors or windows cannot be counted on for dependable results.

System Balancing

Ducts and diffusers in a system should be sized so that the supply of air is distributed properly. However, for flexibility, use standard sizes and allow for future redistribution; the system as designed may not be self-balancing. Chapter 36 of the *ASHRAE Handbook—Applications* describes the procedures used to balance air distribution systems.

DESIGN PROCEDURE

1. Determine the air volumetric flow requirements based on load and room size. For VAV systems, evaluation should include the range of flow rates from minimum occupied to design load.
2. Select the tentative diffuser type and location within room.
3. Determine the room's characteristic length L (Table 3).
4. Select the recommended T_V/L ratio from Table 4.
5. Calculate the throw distance T_V by multiplying the recommended T_V/L ratio from Table 4 by the room length L.
6. Locate appropriate outlet size from the manufacturer's catalog.
7. Ensure that this outlet meets other imposed specifications, such as for noise and for static pressure.

Example 2.

Specifications:

Room size	6 m by 4 m with 2.5 m ceiling
Loading	Uniform, 30 W/m^2 or 720 W
Air volumetric flow	5×10^{-3} m^3/(s·m^2) or 0.12 m^3/s for the one outlet
Device	High sidewall grille, located at center of 4 m endwall, 230 mm from ceiling

Calculations:

Characteristic length L	= 6 m (length of room: Table 3)
Recommended T_V/L	= 1.5 (Table 4)
Throw to 0.25 m/s T_{50}	= 1.5 × 6 = 9 m

Refer to the manufacturer's catalog for a size that gives this isothermal throw to 0.25 m/s. One manufacturer recommends the following sizes, when vanes are straight, discharging 0.12 m^3/s: 400 mm by 100 mm, 300 mm by 125 mm, or 250 mm by 150 mm.

OUTLET LOCATION AND SELECTION

No criteria have been established for choosing among the six types of outlets to obtain an optimum ADPI. All outlets tested, when used according to these recommendations, can have ADPI values that are satisfactory (greater than 90% for loads less than 130 W/m^2).

The design of an air distribution and air diffusion system is influenced by the same factors that influence the design of an air-conditioning plant—building use, size, and construction type. Location and selection of the 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 the 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 at the occupancy level of the room, the heating effect can be counteracted (1) by directing cool air toward the heat source or (2) by locating an exhaust or return grille adjacent to the heat source. The second method is more economical for cooling applications, since heat is withdrawn at its source rather than dissipated into the conditioned space. Where lighting loads are heavy (50 W/m^2) and ceilings relatively high (above 4.5 m), 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 temperature of the air 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. Enclosed lights are more economical than exposed lights because a considerable portion of the energy is radiant.

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

Group A Outlets

Outlets mounted in or near the ceiling with horizontal air discharge should not be used with temperature differentials exceeding 15 K during heating. Researchers have recommended that temperature differentials not exceed 8 K during heating (Hart and Int-Hout 1980, Lorch and Straub 1983). Consequently, such outlets should be used for heating buildings located 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.

The performance of these outlets 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 the throw and decreases the drop. To prevent smudging, the 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 may be better in a room than one large outlet.

With the data in the section on Principles of Jet Behavior, the throw may be selected for a portion of the distance between the outlet and wall or, preferably, for the entire distance. For outlets in opposite walls, the 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, it may be necessary to select a longer throw. In no case should the drop exceed the distance from the outlet to the 1.8 m 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 will be desirable—the highest induction can be maintained at reduced flows. Adequate induction by a ceiling-mounted diffuser prevents short-circuiting of 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 the air high enough for proper cooling in the occupied zone. An increase in supply air velocity improves air diffusion during both heating and cooling. Also, a terminal velocity of about 0.75 m/s is found at the same distance from the floor during both heating and cooling. Therefore, outlets should be selected from the data given in the section on Principles of Jet Behavior, with throw based on a terminal velocity of 0.75 m/s.

With outlets installed near the exposed wall, the 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 the supply air should be deflected at an angle away from the wall. However, to prevent the 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 150 mm and an angle of 15° is satisfactory.

These outlets do not counteract natural convection currents unless sufficient outlets are installed around the perimeter of the space — 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. If installed correctly, outlets of this type 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 the required projection for cooling, the outlets should be used with temperature differentials of less than 8 K. With higher temperature differentials, supply air velocity is not sufficient to project the total air up to the desired level.

The 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 1.5 m/s, 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 have considerable influence on projection. Therefore, low values of each should be selected.

During heating, it is important to select the correct supply air velocity to project the warm air into the occupied zone. Temperature differential is also critical because a small temperature differential reduces variation of the throw during the 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 Nevins and Ward (1968) and Miller and Nevins (1969) 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 (65 W/m^2). For higher room loads (250 W/m^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 operation, the designer must provide for the cold wall effect, as with any ceiling supply diffusion system. The 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.

RETURN AIR DESIGN FOR OPTIMUM PERFORMANCE

An HVAC system operating in the cooling mode performs best when generated heat is removed at its source rather than distributed throughout the conditioned space. Heat from solar and miscellaneous loads such as machinery and floor or desk-mounted lamps is difficult to remove at the source. However, return air flowing over ceiling-mounted lighting fixtures keeps most of that heat from being distributed into the conditioned space. In addition to increasing HVAC system efficiency, return air lighting fixtures improve light output and extend the life of the lamps. The manufacturers of fixtures, ceiling grids, and grilles give performance information (airflow rate, pressure drop, and heat removal rate) of their product. Ball et al. (1971) found that the heat removal performance of return air fixtures covers a narrow range.

With a suspended ceiling, low operating static pressure across the ceiling must be maintained. Failure to do so can result in return air being forced around the edges of the ceiling panels or, in some cases, through the ceiling panels. The result is often a soiled ceiling and a mechanical system that is choked for return air. To avoid this, the static pressure difference across the ceiling should be as low as possible. If necessary, slotted tees or grilles can be used with return air fixtures to obtain the specified pressure drop. A maximum pressure drop of 5 to 7.5 Pa is acceptable under most conditions.

At the typical air supply rates found in office interior zone spaces [usually less than 7.5 $L/(s \cdot m^2)$] and with adequate induction at the supply diffusers, the location of the return diffuser has no effect on air patterns in the space. For most office spaces, it is only necessary that sufficient return outlets be provided to maintain inlet velocities within recommendations (see Table 5).

In spaces expected to operate in a cooling mode most of the time, returning the warmest air in the space can effectively reduce energy costs and increase circulation in the space. This is especially true in climates where economizer systems operate for long periods during the year. In spaces having very high ceilings, with atriums, skylights, or large vertical glass surfaces, and where the highest areas are unoccupied, air stratification may be used as an energy-saving measure by locating returns near the occupied zone.

CEILING-MOUNTED AIR DIFFUSER SYSTEMS

For the best thermal comfort conditions and highest ventilation effectiveness in an occupied space (i.e., 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. While throw data for individual diffusers are used in system design, an air diffuser 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 150 mm to 1.8 m above the floor).

Adequate ventilation requires that the selected diffusers effectively mix (by entrainment) the total air in the room with the supplied conditioned 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 Miller and Nevins (1970), Miller and Nash (1971), Miller (1979), and Hart and Int-Hout (1981) suggest that the air diffusion performance index (ADPI) can be improved by moving diffusers closer together (i.e., specifying more diffusers for a given space and air quantity) and by limiting the value of 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. The 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 has performed these tests and can provide the best choice of design options for a particular building. For a VAV system, the 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 the outlet air volume to ensure that ADPI values remain above a specified minimum.

Perimeter Spaces

All-air mechanical systems that handle both heating and cooling thermal loads are commonly used in modern office buildings instead of baseboards for heating and forced air for cooling. State energy codes (most based on 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 200 to 300 W per linear metre of wall.

A successful all-air heating/cooling mechanical system requires the designer to consider several design variables that have been the subject of research by Hart and Int-Hout (1980), Lorch and Straub (1983), and 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

The diffuser manufacturer is best able to recommend the use of equipment.

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 3 K.

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 under heating load conditions and chilled air in both directions under cooling load conditions. All researchers found less than optimum performance with high discharge temperatures (greater than 8 K 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. Under heating load conditions, the supply air temperature must be limited to avoid excessive thermal stratification.

To resolve any uncertainty about performance, a mockup should be constructed 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 load conditions, return air should exhaust from a location that takes advantage of any thermal stratification design. In many cases, this should be a high point in order to take advantage of rising warm air. Cooling supply air should be introduced as close to the heat sources as possible. Alternately, 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 3 K.
- For heating load conditions, thermal stratification should be discouraged. Heat should be introduced at points low in the large space. Ceiling-mounted fans may reduce stratification.

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