Basics of Well-Mixed Room Air Distribution

BY DAN INT-HOUT, FELLOW ASHRAE

This article is the first of three I have written for the Fundamentals at Work series. This one will cover air distribution for well-mixed systems, the most common application in commercial and institutional buildings in the U.S. It will be followed by articles on air terminals and acoustics. All three topics are interdependent, meaning that there must be an understanding of the relationship between air distribution, air delivery rates, and acoustics to properly design an HVAC system that will provide an acceptable indoor thermal environment for occupants.

Air-Distribution Design Goals

To begin, there are four basic performance mandates for an HVAC system: acceptable indoor air temperature, air movement, humidity, and quality. The parameters for these are defined mostly in two design standards, which are commonly referenced and/or integrated into local building codes. These standards are ASHRAE Standard 62.1, *Ventilation for Acceptable Indoor Air Quality,* and ASHRAE Standard 55, *Thermal Environmental Conditions for Human Occupancy*. Some non-ASHRAE international standards covering these parameters also exist, but they often have similar requirements. ASHRAE/IES Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings,* defines limits on energy consumption for creating these indoor conditions.

The three common room air-diffusion patterns are: fully mixed; partially mixed; and fully stratified (*Figure 1*).

Rules for Fully Mixed Systems

The first basic "rule" is that space temperatures in well-mixed spaces should be uniform, within a couple degrees, at all points within a defined occupied zone. The occupied zone is a definable space normally occupied by people, and is defined by Standard 55-2013 in *Figure 2.*

Temperatures also should remain the same throughout the day, and the room must be relatively free from objectionable air currents or drafts that are generated by improperly selected or installed air supply devices. *Figure 2* depicts the "comfort zone" as defined in Standard 55-2013.

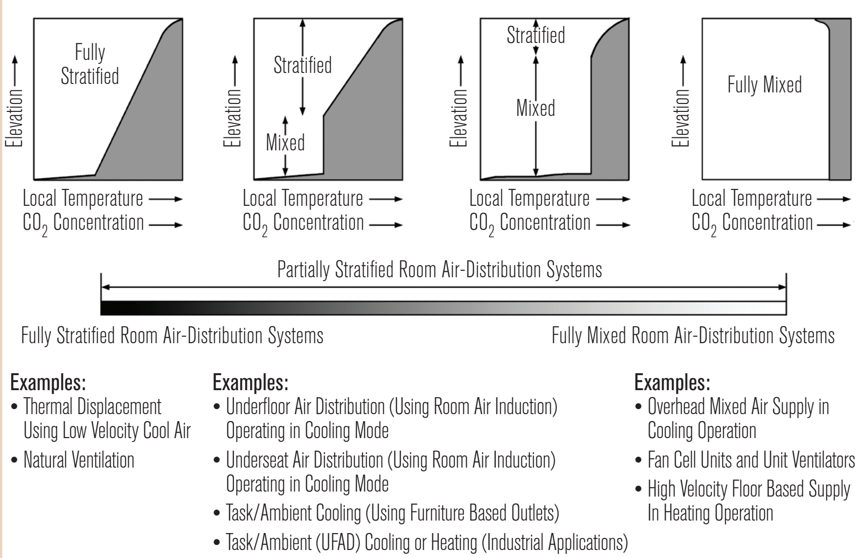
The second requirement is that the space's relative humidity be controlled. Standard 62.1-2013 limits the

Dan Int-Hout is a chief engineer at Krueger in Richardson, Texas. He is a member of SSPC 55-2013, SPC 129 and consultant to SSPC 62.1-2013.

relative humidity to 65% maximum. This means that when in cooling and dehumidification mode with an all-air HVAC system, the supply air must have a dew-point temperature low enough to keep the indoor airborne moisture controlled. Typically, the dew point of the supplied air is no greater than 55°F (13°C) for conventional all-air systems.

Finally, a minimum quantity of "ventilation" air needs to be introduced into the space. This typically refers to outdoor air, but it may contain some cleaned, recycled air if the requirements of ASHRAE Standard 62.1 are met. The required volumetric flow rate, in cfm or L/s, is based on several factors in the standard, which is typically around 0.15 cfm/ft^2 (1.3 L/s·m^2) in office spaces.

FIGURE 1 Characteristics of air-distribution patterns (2011 *ASHRAE Handbook—Fundamentals*).

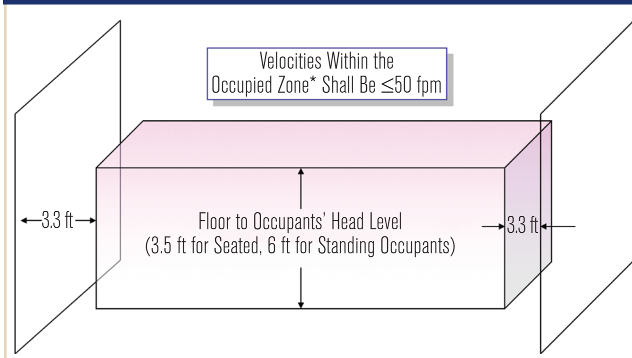


There are two basic behaviors of air as it is introduced into a space. The first is that cold air (being denser) tends to fall and hot air (being lighter) tends to rise. This is known as buoyancy. The second is that air wants to move from high pressure to low pressure. A jet of moving-typically air has lower pressure than the air surrounding it. In addition, air is viscous, and the high velocity supply air, introduced via supply air outlets, will transfer momentum to the much lower velocity air already in the room, thus inducing room air movement and enhancing mixing and tempering of the supply air into the room's air. These factors are what drive the science of room air distribution.

The goal of conventional fully mixed air distribution is to create a "well-mixed" environment. In most offices today, air is supplied from overhead air outlets, typically via diffusers because they are intended to diffuse the supply air into the space, but supply grilles are also used. The entering air temperature from such devices is often designed to be around 55°F (13°C) with conventional systems, which is colder than the desired room air temperature. To understand how the air will behave and prevent the air from falling into the space, design engineers must not only draw upon the above behaviors of room air diffusion, but also understand the related effects and terminology when developing their designs.

A jet of air flowing next to a solid surface will create a slightly lower pressure region when compared to that in the room. Called the Coanda effect, also referred to as the surface or Bernoulli effect, this is what causes a negatively buoyant jet of cool air to stay along the surface of the ceiling. As this attached jet of air moves along a ceiling, two things happen.

FIGURE 2 Thermal comfort zone in a room per ASHRAE Standard 55-2013.



*Occupied Zone – the region normally occupied by people within a space. In the absence of known occupant locations, the occupied zone is to be between the floor and 1.8 m (6 ft) above the floor and more than 1.0 m (3.3 ft) from outside walls/windows or fixed heating, ventilating, or air-conditioning equipment and 0.3 m (1 ft) from internal walls.

First, the temperature difference between the room and the supply air is reduced as surrounding higher pressure room air mixes into the jet. This increases the volumetric flow rate of the moving airstream, which, in turn, increases its mass. As no additional kinetic energy is introduced, the air velocity will slow as momentum is conserved. This then decreases the jet's ability to entrain more air, thereby reducing the Coanda effect.

FIGURE 3 Primary and secondary airstreams.

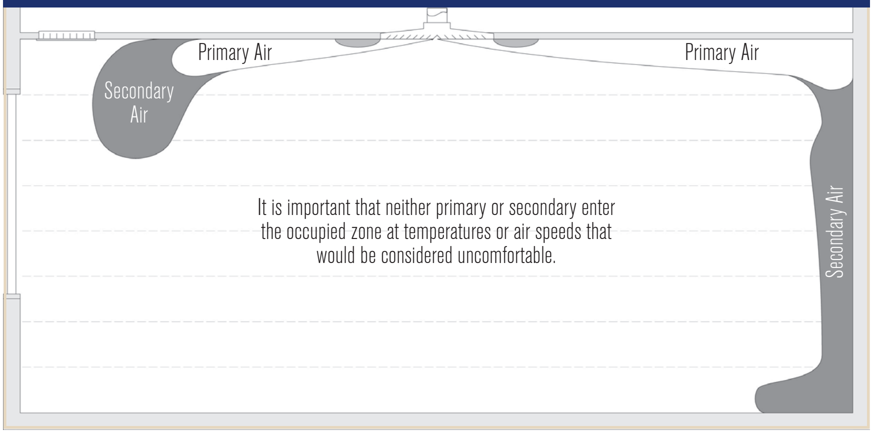
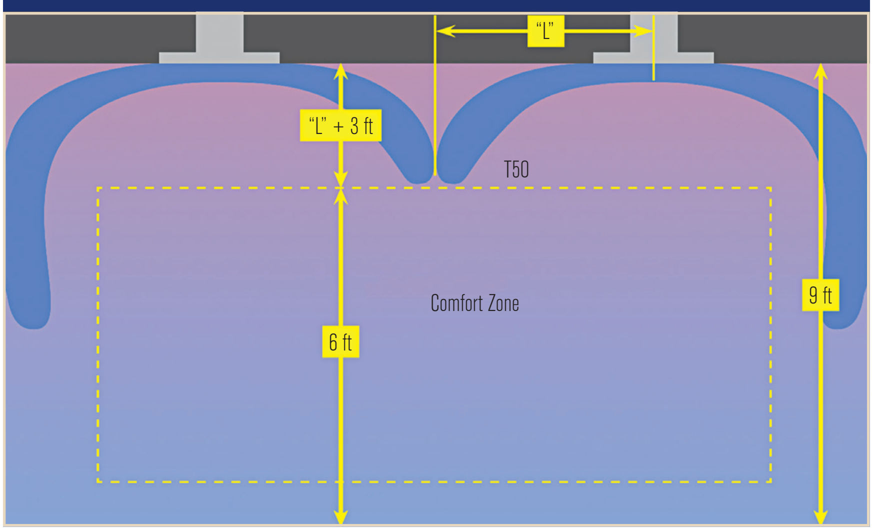


FIGURE 4 Jet collisions.



It is reduced the farther away the jet is from the air outlet. If the jet is cooler, and thus denser, than the room air, it will detach from the ceiling when the forward momentum of the supply jet is less than the downwards buoyancy force. If the buoyancy force is great enough, the supply jet may enter the occupied zone with an objectionable temperature or velocity. This effect is predictable and should be considered by the designer.

Throw, defined as the projection of an air device, is the distance the air travels before it slows to a specified air speed. Often manufacturers report throws to air speeds of 50 fpm (0.25 m/s), 100 fpm (0.5 m/s), and 150 fpm (0.76 m/s). These throws are most often reported for isothermal air, which is simply supply air that is the same temperature as the room air, as isothermal throw is more easily measured and is used for air diffusion performance index (ADPI) calculations, as we will see later.

If a diffuser has too short a throw, such as at low airflow rates in cooling mode, it may detach from the ceiling before it has entrained enough air to warm up and mix enough with the room air. The result is that the cold air will fall into the space. This is often referred to as "dumping," although the more correct term is "excessive drop." But if opposing jets have too high of airflows, they may collide and enter the occupied zone at air speeds that may be perceived as a draft. For design engineers, it is a matter of finding the right balance between preventing excessive drop and maintaining the entrainment rate.

Figure 3 shows the total airstream from a ceiling-mounted air outlet. The shaded areas are secondary air, or air that has been significantly mixed with room air. *Figure 4* shows what happens when opposing jets collide.

It is important that the jet not enter the occupied zone until sufficiently tempered and slowed. Standard 55-2013 requires the use of an elevated air speed analysis when air speeds exceed 40 fpm (0.20 m/s). The *ASHRAE Handbook—Fundamental's* Space Air Diffusion chapter recommends not exceeding 50 fpm (0.25 m/s), and the ADPI method excludes points in excess of 70 fpm (0.35 m/s). Using manufacturer's catalog data, engineers are able to select devices that will have long enough throws at minimum airflows to avoid excessive drop while spacing diffusers far enough apart to avoid excessive collisions. While manufacturers usually report throws to 50 fpm (0.25 m/s) in their catalogs, in selection software the desired terminal velocity is usually a selectable parameter.

The distance from a diffuser to the edge of the space it is designed to serve is designated as the "characteristic length" or "L." Therefore, identical diffusers with the same flow rates are spaced 2 L apart. To avoid drafts in the occupied zone, a good rule of thumb is that diffusers should be

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spaced so that at the designed maximum airflow, a 50 fpm throw (T_{50}) is no greater than L plus the distance from 6 ft to the room height, as shown in *Figure 4*.

Furthermore, while it might appear that higher discharge velocities equal greater induction rates and longer throws, this is not always the case. Mass effect must be considered because at some point mass will overcome velocity and determine the throw. Different supply air outlets will perform in different ways. Higher discharge velocities will mean greater required system air pressures and higher fan energy, and usually produce greater sound levels. We will talk more about when more sound production can actually be a good thing.

In addition to aerodynamic considerations, there are two other performance issues to be considered: pressure and acoustics.

Pressure

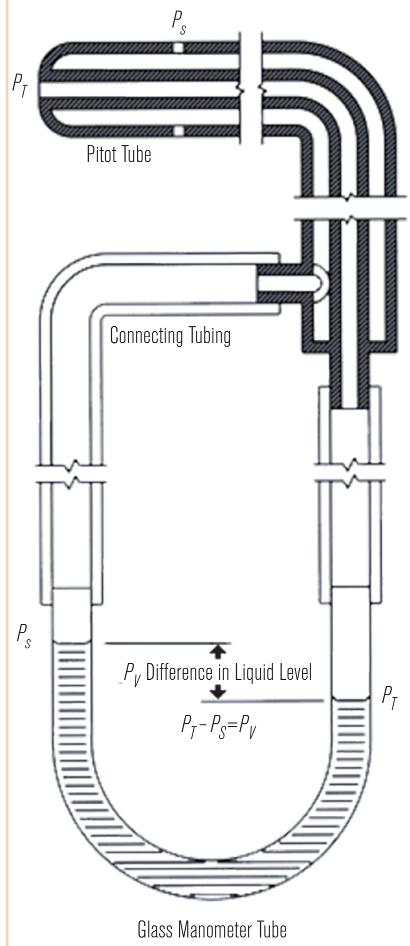
The HVAC system's fan needs to provide enough pressure to deliver air to the outlets as well as overcome the resistance created by the diffuser. In addition, the fan energy also creates room air motion, and pushes air back to the air handler through the air inlets and the return ducts.

Air pressure is characterized by three values:

- Static pressure is the force per unit area exerted by a fluid. It is measured at a 90 degree angle to the flow's direction, traditionally with a static tube connected to a water-, mercury-, or red gauge oil-filled manometer.
- Velocity pressure is the kinetic energy of the fluid flow. It is measured by subtracting the static pressure from the stagnation (total) pressure, traditionally with a concentric pitot-static tube.
- Total pressure is the sum of the static and velocity pressures.

Total pressure is the measure of the energy in moving air within a duct, remaining fairly constant, less friction effects. Most air diffuser manufacturers present catalog data on static and total pressure losses over a range of airflows (with velocity pressure being the difference between the two). In a duct, velocity pressures are traditionally measured with a pitot-static tube and manometer as depicted in *Figure 5*. The pitot tube's opening on the end reads total pressure (sometimes called stagnation pressure), while the openings on the side measure static pressure. By connecting each of the connections on the pitot tube to opposite ends of a manometer, it will display the velocity pressure, which is easily converted to air velocity. Air velocity multiplied by duct area gives the airflow rate.

FIGURE 5 Pitot tube.



Acoustics

We will discuss acoustics in detail in a following article, "Basics of Applied Acoustics." For the moment, however, we need to understand that in reporting sound produced by air outlets that manufacturers are required to use 10 diameters of straight supply duct attached to the air device. In the real world, this almost never happens. Studies (reported in "Effects of Typical Inlet Conditions on Air Outlet Performance," *ASHRAE Journal,* April 2012) show the effect that real-life inlet conditions have on acoustic performance. Using flex duct or hard elbows can increase the sound produced by an outlet (on the average of a noise criteria [NC] of 5) compared to manufacturer's catalog data obtained per the above.

Sound generated by a diffuser is created by mixing air. As a general rule, the louder an air device is, the better it works. By selecting the quietest possible diffuser, it may mean it is not mixing as well as it could. With variable volume designs, which we will describe in detail in a following article, cooling airflow rates are at their highest only when the space is warm, but this only happens a small percent of the time. When it is warm indoors, occupants are less likely to complain about the sound of a well-selected diffuser at a higher flow rate, recognizing they are being cooled.

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FIGURE 6 Dumping diffuser.

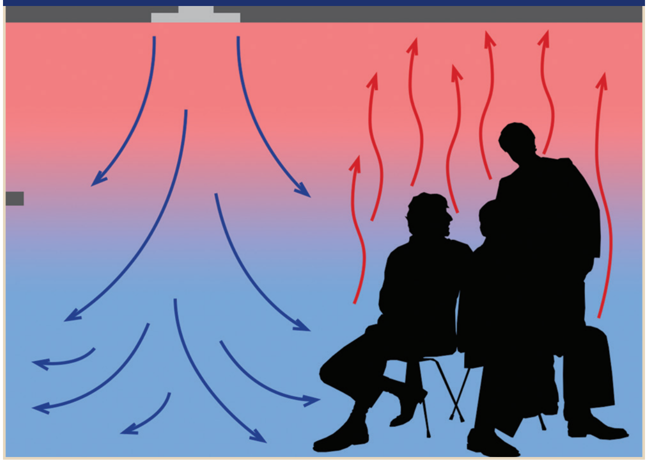
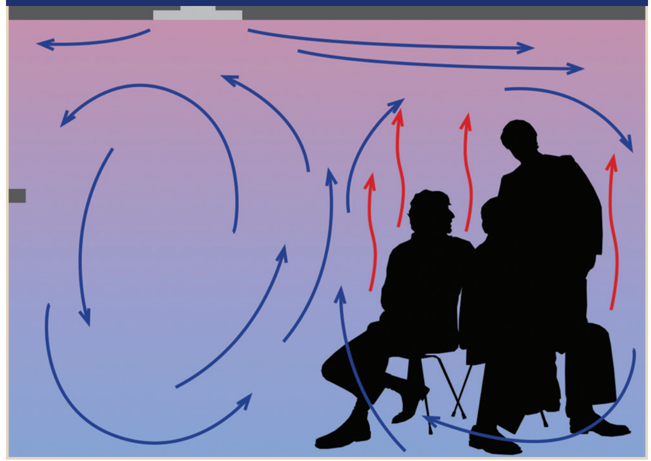


FIGURE 7 Properly selected air outlet for a mixing system.



Consequences of Poor Performance

At low airflows when in cooling mode, poorly selected or designed air outlets, including improperly or non-adjusted diffusers (see *ASHRAE Journal,* June 2013, "Slots are Adjustable"), may direct air downward into the occupied zone (*Figure 6*). Aside from occupants complaining of drafts, the room will tend to stratify, with cool air at the floor. A wall-mounted thermostat in this case does not sense the average room temperature, but rather it determines where the stratification layer will be, essentially becoming an "altitude controller." More importantly, the thermostat becomes much less sensitive to changes in load, further compromising temperature control.

In any occupied space, the local heat loads created by people and equipment create upward convection currents. These upward currents cause air to be drawn to the loads near the floor, helping to mix air in the occupied zone. With a properly selected air outlet, excellent zone temperature control (*Figure 7*) can be achieved at very low air delivery rates. In a recent ASHRAE research project (1515-RP), high occupant satisfaction rates were recorded at airflows as low as 0.2 cfm/ft^2 (1 L/s m^2).

One way of predicting temperature and air speed variations in a space when in cooling mode is to use the air diffusion performance index, or ADPI. Developed in the 1960s, and currently being expanded in an ASHRAE research project (1546-RP) at the University of Texas-Austin, the ratio of the isothermal throw to 50 fpm (0.25 m/s) and one-half of the spacing of opposed diffusers (or characteristic length) has been correlated to the percent of points in the occupied zone that meet thermal mixing requirements for draft and temperature deviation from the average in the space. This percentage is referred to as the ADPI, where 80% is considered a minimum acceptance value for most spaces.

Table 4 in Chapter 57 of the *2015 ASHRAE Handbook—HVAC Applications* presents predicted ADPI values for several types of air outlets over a range of space heat loads. Using this data, manufacturers have developed computer programs and charts showing the effective performance range of many types of air-distribution devices.

The plaque-type diffusers used in the space where ASHRAE research project 1515-RP was conducted predicted high ADPI at very low airflow rates using this type of analysis, the results of which were subsequently validated by occupant surveys in an operating space. Furthermore, the research confirmed the statement "there is no minimum airflow for comfort" that was made in earlier versions of Standard 55-2013.

Perimeter Zones

So far, our discussions have focused on interior cooling from the ceiling in open office applications. Around the outskirts is what is termed as the perimeter zone (often defined as the area within 15 ft of an exterior wall or window). This is where heating commonly takes place, typically through ceiling diffusers. Remembering that hot air rises, we know that if we supply heat from the ceiling, it is not without its challenges. If the temperature differential between the supply air and average room air (ΔT) is greater than 15°F (8.3°C), in many cases the room air will stratify. It is also probable that it will exceed Standard 55-2013's stratification limit of 5.4°F (3°C). Standard 62.1-2013 recognizes that the ventilation air is likely to stratify and exit the space (through ceiling returns) without mixing. As a result, it requires that ventilation

FIGURE 8 Improper ceiling heating.

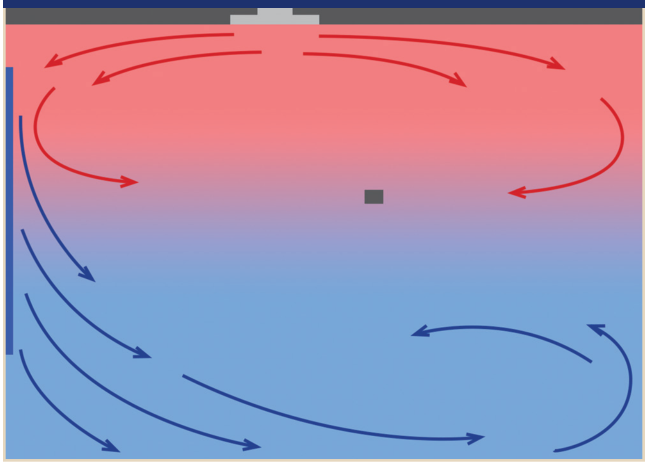
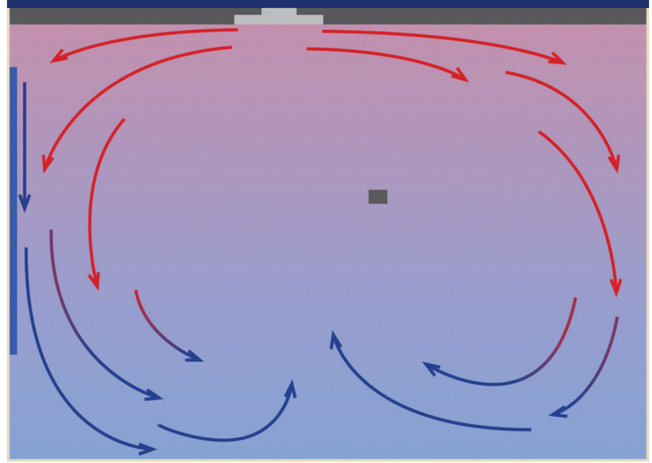


FIGURE 9 Properly designed ceiling heating.



be increased by 20% if the room to diffuser ΔT exceeds 15°F [8.3°C] or the 150 fpm (0.75 m/s) throw fails to come down the wall/window to at least 4.5 ft (1.4 m) from the floor (*Figure 8*).

Many engineers are not apparently aware of this requirement and continue to design systems with low flows and hot air. However, this may be the result of a prohibition in Standard 90.1-2013 that prevents reheating more than 30% of the design cooling airflow. In the next article, we will discuss how to overcome these limitations when using VAV terminals.

When an air outlet is properly located and there is a low ΔT (less than 15°F (8.3°C) temperature difference between supply air and room air), warm air will blend with cooler air and result in a well-mixed occupied zone. In fact, as shown in *Figure 9*, studies have shown that a slot diffuser located a couple feet from the window with air directed both toward and away from the window can work well in both heating and cooling applications.

Return Air Inlets

Return air in overhead mixed air systems goes back to the air handler either through the plenum or within ducted return air systems. At the airflow rates present in most well-mixed spaces, the location of the return air inlet has a negligible effect on air movement in the room. At higher flows, such as a hospital operating room, locating return air inlets near the floor may be beneficial (and required by ASHRAE Standard 170-2013) as it reduces backflow and room air mixing. Low returns also eliminate the Standard 62.1-2013 air change effectiveness penalty (described previously in the discussion of perimeter zones) for delivering hot air at the ceiling (but are difficult and/or expensive to install).

A return grille located next to a supply in the ceiling would seem to be a problem, but may be unavoidable in small spaces. Proper diffuser orientation may reduce the likely small degree of "short-circuiting" that may occur.

Summary

While the air terminal devices used in a well-mixed air-distribution system represent one of the lowest costs per unit area of any component in such a system, they have one of the greatest effects on building occupants' thermal comfort. It does not matter how elaborate, efficient, or clever the building's HVAC design is—if the air distribution is poor, it will not provide an acceptable space (as defined by ASHRAE Standard 55, for example) for occupants. It must fulfill their basic needs for comfort and acceptable air quality. Only by understanding how air-distribution devices can affect temperature, air movement, humidity, and ventilation and being mindful of the rules, limitations, and complex interactions between the performance mandate of various codes and standards are we able to design comfortable and safe environments that promote occupant productivity and health.

Acknowledgments

Many people, committees, companies, and organizations have contributed to the development of knowledge in room air diffusion and distribution. In ASHRAE, in particular, Technical Committees 4.3, Ventilation and Infiltration, and 5.3, Room Air Distribution, and the related standards' project committees, have direct interests in the topic of this article, and their members and research projects have helped improve our systems' performance. ■