CHAPTER 56

ROOM AIR DISTRIBUTION

ROOM air distribution systems, like other HVAC systems, are intended to achieve required thermal comfort and ventilation for space occupants and processes. Although air terminals (inlets and outlets), terminal units, local ducts, and the rooms themselves may affect room air distribution, this chapter addresses only air terminals and their effect on occupant comfort. This chapter is intended to help HVAC designers apply air distribution systems to occupied spaces, providing information on characteristics of various air distribution strategies, and tools and guidelines for applications and system design. Naturally ventilated spaces are not addressed; see Chapter 27 of the 2005 *ASHRAE Handbook—Fundamentals* for details. Also see Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* for more information on space air diffusion; Chapter 17 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment* for information on room air distribution equipment; and Chapter 47 of this volume for sound and vibration control guidance.

Room air distribution systems can be classified by (1) their primary objective and (2) the method by which they attempt to accomplish that objective. The objective of any air distribution system can be classified as one of the following:

- Conditioning and/or ventilation of the space for occupant thermal comfort
- Conditioning and/or ventilation to support processes within the space
- A combination of these

As a general guideline, the **occupied zone** in a space is any location where occupants normally reside, and may differ from project to project; it is application-specific, and should be carefully defined by the designer. The occupied zone is generally considered to be the room volume between the floor level and 6 ft above the floor. Standards and guidelines, such as ANSI/ASHRAE *Standards* 55 and 62.1, further define the occupied zone (e.g., *Standard* 55 exempts areas near walls).

Occupant comfort is defined in detail in ANSI/ASHRAE *Standard* 55-2004. Figure 5.2.1.1 of the *Standard* shows acceptable ranges of temperature and humidity for spaces. As a general guide, 80% of occupants in typical office spaces can be satisfied with thermal environments over a wide range of temperatures and relative humidities. Designers often target indoor dry-bulb temperatures between 73 and 77°F, relative humidities between 25 and 60%, and occupied zone air velocities below 50 fpm. ANSI/ASHRAE *Standard* 113 describes a method for evaluating effectiveness of various room air distribution systems in achieving thermal comfort.

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

- **Mixed systems** (e.g., overhead distribution) have little or no thermal stratification of air in the occupied and/or process space.
- **Full thermal stratification systems** (e.g., thermal displacement ventilation) have little or no air mixing in the occupied and/or process space.
- **Partially mixed systems** (e.g., most underfloor air distribution designs) provide limited air mixing in the occupied and/or process space.
- **Task/ambient air distribution** (e.g., personally controlled desk outlets, spot conditioning systems) focuses on conditioning only part of the space for thermal comfort and/or process control.

Because task/ambient design requires a high degree of individual control, it is not covered in this chapter; see Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* for details. Limited design guidance is also provided by Bauman and Daly (2003).

Figure 1 illustrates the spectrum between the two extremes (full mixing and full stratification) of room air distribution strategies.

SYSTEM DESIGN

Design Constraints

Space design constraints affect room air distribution system choices and how air inlets and outlets are used. Space constraints may include the following:

- Dimensions
- Heat gain and loss characteristics
- Use
- Acoustical requirements
- Available locations for air inlets and outlets

Inlet and outlet characteristics are discussed in Chapter 17 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*. This chapter discusses more specific application considerations for air inlets and outlets.

Sound

Sound emitted from inlets and outlets is directly related to the airflow quantity and free area velocity. The airflow sound intensity in a space also depends on the room's acoustical absorption and the observer's distance from air distribution devices, For more information, see Chapter 47 of this volume and Chapter 7 in the 2005 *ASHRAE Handbook—Fundamentals*.

Inlet Conditions to Air Outlets

The way an airstream approaches an outlet is important. For good air diffusion, the inlet configuration should create a uniform discharge velocity profile from the outlet, or the outlet may not perform as intended.

The outlet usually cannot correct effects of improper duct approach. Many 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, devices for directing and equalizing the airflow may be necessary for proper direction and diffusion of the air.

Return Air Inlets

The success of a mixed air distribution system depends primarily on supply diffuser location. Return grille location is far

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

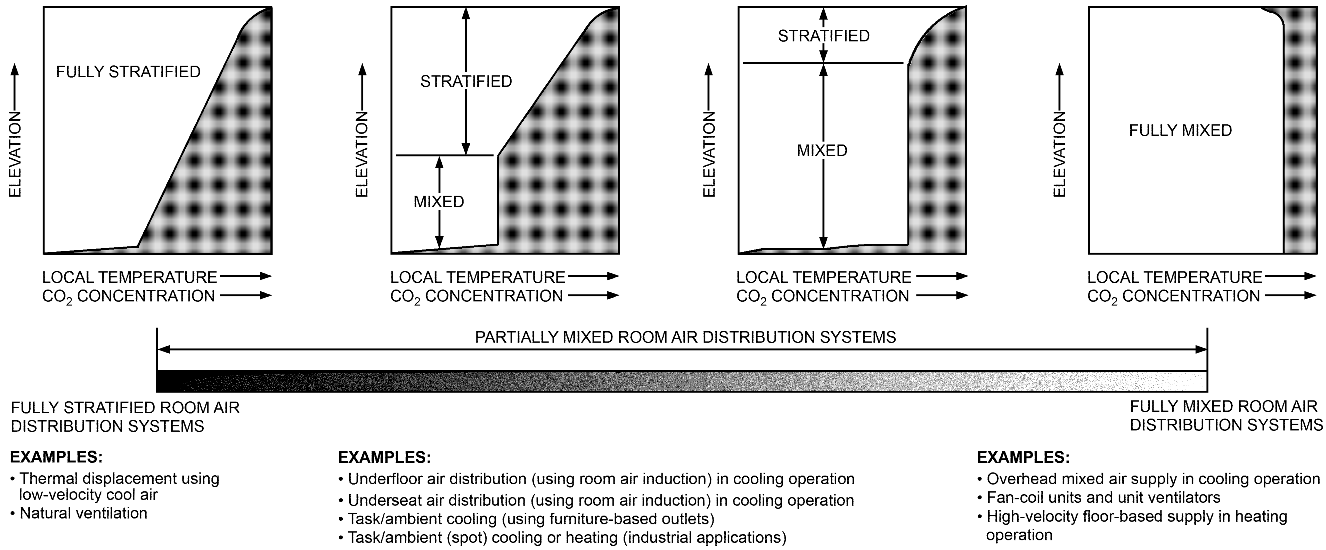

Fig. 1 Classification of Air Distribution Strategies

Table 1 Recommended Return Inlet Face Velocities

Inlet Location	Velocity Across Gross Area, fpm
Above occupied zone	>800
In occupied zone, not near seats	600 to 800
In occupied zone, near seats	400 to 600
Door or wall louvers	200 to 300
Through undercut area of doors	200 to 300

less critical than with outlets. In fact, the return air intake affects room air motion only immediately around the grille. Measurements of velocity near a return air grille show a rapid decrease in magnitude as the measuring device is moved away from the grille face. Table 1 shows recommended maximum return air grille velocities as a function of grille location. Every enclosed space should have return/transfer inlets of adequate size per this table.

For stratified and partially mixed air distribution systems, there are advantageous locations for return air inlets. For example, an intake can be located to return the warmest air in cooling season.

If the outlet is selected to provide adequate throw and directed away from returns or exhausts, supply short-circuiting is normally not a problem. The success of this practice is confirmed by the availability and use of combination supply and return diffusers.

Indoor Air Quality and Sustainability

Air distribution systems affect not only IAQ and thermal comfort, but also energy consumption over the entire life of the project; choices made early in the design process are important. ANSI/ASHRAE/IESNA *Standard* 90.1 provides energy efficiency requirements that affect supply air characteristics. The U.S. Green Building Council's (USGBC) Leadership in Energy and Efficient Design (LEED™) rating system was originally created in response to indoor air quality concerns, and the latest versions of their *New Construction and Major Renovations Guide* include prerequisites and credits for increasing ventilation effectiveness and improving thermal comfort. These requirements and optional points are relatively easy to achieve if good room air distribution design principles, methods, and standards are followed.

Environmental tobacco smoke (ETS) control is a LEED prerequisite. Banning indoor smoking is a common approach, but if indoor smoking is to be allowed, ANSI/ASHRAE *Standard* 62.1-2004 requires that more than the base non-ETS ventilation air be provided where ETS is present in all or part of a building. Rock (2006) provides additional advice on dealing with ETS.

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

Because of the difficulty in predicting ventilation effectiveness, ANSI/ASHRAE *Standard* 62.1 provides a table of typical values that were determined through the experiences of its Standard Project Committee and reviewers or extracted from research literature; for example, well-designed ceiling-based air diffusion systems produce near perfect air mixing, when in cooling mode, and as such they yield an air change effectiveness of almost 1.0. More information on ANSI/ASHRAE *Standard* 62.1 is available in its user's manual (ASHRAE 2004).

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

Air terminals, such as diffusers or grilles, may become unsightly over time because of accumulation of dirt on their faces (smudging). Instead of replacing air terminals, and thus requiring new materials and energy for manufacturing, they can often be cleaned in place to restore their appearance. Those that cannot be cleaned and must be replaced should be recycled, not discarded, to recover the various metals and other desirable materials of construction.

MIXED AIR DISTRIBUTION

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

Maintaining velocities less than 50 fpm in the occupied zone is often overlooked by designers, but is critical to maintaining comfort. The outlet's selection, location, supply air volume, discharge velocity, and air temperature differential determine the resulting air motion in the occupied zone.

Principles of Operation

Mixed systems generally provide comfort by entraining room air into discharge jets located outside occupied zones, mixing supply and room air. Ideally, these systems generate low-velocity air motion (less than 50 fpm) throughout the occupied zone to provide uniform temperature gradients and velocities. Proper selection of an air outlet is critical for proper air distribution; improper selection can result in room air stagnation, unacceptable temperature gradients, and unacceptable velocities in the occupied zone that may lead to occupant discomfort.

The location of a discharge jet relative to surrounding surfaces is important. Discharge jets attach to parallel surfaces, given sufficient velocity and proximity. When a jet is attached, the throw increases by about 40% over a jet discharged in an open area. This difference is important when selecting an air outlet. For detailed discussion of the surface effect on discharge jets, see Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals*.

Space Ventilation and Contaminant Removal

These systems are intended to maintain acceptable indoor air quality by mixing supply and room air (dilution ventilation). Supply air is typically a conditioned mixture of ventilation and recirculated air. Outlet type and discharge velocity determine the mixing rate of the space and should be a design consideration. The room's return or exhaust air carries away diluted air contaminants. Space air ventilation rates are mandated under ASHRAE *Standard* 62.1-2004, but supply airflow rates are often higher because of thermal loads.

Benefits and Limitations

Benefits of fully mixed systems include the following:

- Most office applications can use lower supply dry-bulb temperatures, for smaller ductwork and lower supply air quantities.
- Air can be supplied at a lower moisture content, possibly eliminating the need for a more complex humidity control system.
- Vertical temperature gradients are lower for cooling applications with high internal heat gains, which may improve thermal comfort.
- Mixed systems are the most common design for distribution systems, because designers and installers are familiar with the required system components and installation.

Limitations of mixed systems include the following:

- Partial-load operation in variable-air-volume (VAV) systems may reduce outlet velocities, reducing room air mixing and compromising thermal comfort. Designers should consider this when selecting outlets.
- Cooling and heating with the same ceiling or high-sidewall diffuser may cause inadequate performance in heating mode and/or excessive velocity in cooling mode.
- Ceilings more than 12 ft high may require special design considerations to provide acceptable comfort in the occupied zone. Care should be taken to select the proper outlet for these applications.
- Because mixed systems typically use high-velocity jets of air, any obstructions in the space (e.g., bookshelves, wall partitions, furniture) can reduce comfort.
- Lighter-than-air contaminants are uniformly mixed in the space and typically result in higher contaminant concentrations, which may compromise indoor air quality.

Mixed air systems typically use either ceiling or sidewall outlets discharging air horizontally, or floor- or sill-mounted outlets discharging air vertically. They are the most common method of air distribution in North America.

Horizontal Discharge Cooling with Ceiling-Mounted Outlets

Ceiling-mounted outlets typically use the surface effect to transport supply air in the unoccupied zone. The supply air projects across the ceiling and, with sufficient velocity, can continue down wall surfaces and across floors, as shown in Figure 2. In this application, supply air should remain outside the occupied zone until it is adequately mixed and tempered with room air. Air motion in the occupied zone is generated by room air entrainment into the supply air (Nevins 1976).

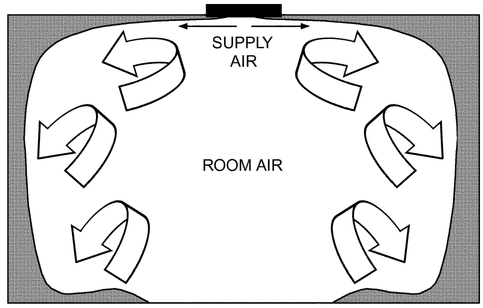

Fig. 2 Air Supplied at Ceiling Induces Room Air into Supply Jet

Overhead outlets may also be installed on exposed ducts, in which case the surface effect does not apply. Typically, if the outlet is mounted 1 ft or more below a ceiling surface, discharge air will not attach to the surface. The unattached supply air has a shorter throw and can project downward, resulting in high air velocities in the occupied zone. Some outlets are designed for use in exposed duct applications. Typical outlet performance data presented by manufacturers are for outlets with surface effect; consult manufacturers for information on exposed duct applications.

Vertical-Discharge Cooling or Heating with Ceiling-Mounted Outlets

Vertically projected outlets are typically selected for high-ceiling applications that require forcing supply air down to the occupied zone. It is important to keep cooling supply air velocity below 50 fpm in the occupied zone. For heating, supply air should reach the floor.

There are outlets specifically designed for vertical projection and it is important to review the manufacturer's performance data notes to understand how to apply catalog data. Throws for heating and cooling differ and also vary depending on the difference between supply and room air temperatures.

Cooling with Sidewall Outlets

Sidewall outlets are usually selected when access to the ceiling plenum is restricted. Sidewall outlets within 1 ft of a ceiling and set for horizontal or a slightly upward projection the sidewall outlet provide a discharge pattern that attaches to the ceiling and travels in the unoccupied zone. This pattern entrains air from the occupied zone to provide mixing.

In some applications, the outlet must be located 2 to 4 ft below the ceiling. When set for horizontal projection, the discharge at some distance from the outlet may drop into the occupied zone. Most devices used for sidewall application can be adjusted to project the air pattern upwards toward the ceiling. This allows the discharge air to attach to the ceiling, increasing throw distance and minimizing drop. This application provides occupant comfort by inducing air from the occupied zone into the supply air.

Some outlets may be more than 4 ft below the ceiling (e.g., in high-ceiling applications, the outlet may be located closer to the occupied zone to minimize the volume of the conditioned space). Most devices used for sidewall applications can be adjusted to project the air pattern upward or downward, which allows the device's throw distance to be adjusted to maximize performance.

When selecting sidewall outlets, it is important to understand the manufacturer's data. Most manufacturers offer data for outlets tested with surface effect, so they only apply if the device is set to direct supply air toward the ceiling. When the device is 4 ft or more below a ceiling, or supply air is directed horizontally or downward, the actual throw distance of the device is typically shorter. Many sidewall outlets can be adjusted to change the spread of supply air, which can significantly change throw distance. Manufacturers usually publish throw distances based on specific spread angles.

Cooling with Floor-Mounted Air Outlets

Although not typically selected for nonresidential buildings, floor-mounted outlets can be used for mixed system cooling applications. In this configuration, room air from the occupied zone is induced into the supply air, providing mixing. When cooling, the device should be selected to discharge vertically along windows, walls, or other vertical surfaces. Typical nonresidential applications include lobbies, long corridors, and houses of worship.

It is important to select a device that is specially designed for floor applications. It must be able to withstand both the required dynamic and static structural loads (e.g., people walking on them, loaded carts rolling across them). Also, many manufacturers offer devices designed to reduce the possibility of objects falling into the device. It is strongly recommended that obstructions are not located above these in-floor air terminals, to avoid restricting their air jets.

Long floor-mounted grilles generally have both functioning and nonfunctioning segments. When selecting air outlets for floor mounting, it is important to note that the throw distance and sound generated depend on the length of the active section. Most manufacturers' catalog data include correction factors for length's effects on both throw and sound. These corrections can be significant and should be evaluated. Understanding manufacturers' performance data and corresponding notes is imperative.

Cooling with Sill-Mounted Air Outlets

Sill-mounted air outlets are commonly used in applications that include unit ventilators and fan coil units. The outlet should be selected to discharge vertically along windows, walls, or other vertical surfaces, and project supply air above the occupied zone.

As with floor-mounted grilles, when selecting and locating sill grilles, consider selecting devices designed to reduce the nuisance of objects falling inside them. It is also recommended that sills be designed to prevent them from being used as shelves.

Heating and Cooling with Perimeter Ceiling-Mounted Outlets

When air outlets are used at the perimeter with vertical projection for heating and/or cooling, they should be located near the perimeter surface, and selected so that the published 150 fpm isothermal throw extends at least halfway down the surface or 5 ft above the floor, whichever is lower. In this manner, during heating, warm air mixes with the cool downdraft on the perimeter surface, to reduce or even eliminate drafts in the occupied space.

If a ceiling-mounted air outlet is located away from the perimeter wall, in cooling mode, the high-velocity cool air reduces or overcomes the thermal updrafts on the perimeter surface. To accomplish this, the outlet should be selected for horizontal discharge toward the wall. Outlet selection should be such that isothermal throw to the terminal velocity of 150 fpm should include the distance from the outlet to the perimeter surface. For heating, the supply air temperature should not exceed 15°F above the room air temperature.

Space Temperature Gradients and Airflow Rates

A fully mixed system creates homogeneous thermal conditions throughout the space. As such, thermal gradients should not be expected to exist in the occupied zone. Improper selection, sizing, or placement may prevent full mixing and can result in stagnant areas, or having high-velocity air entering the occupied zone.

Supply airflow requirements to satisfy space sensible heat gains or losses are inversely proportional to the temperature difference between supply and return air. The following equation can be used to calculate space airflow requirements (at standard conditions):

$$Q = \frac{q_s}{1.08(t_r - t_s)} \tag{1}$$

where

Q = required supply airflow rate to meet sensible load, cfm
q_s = net sensible heat gain in the space, Btu/h
t_r = return or exhaust air temperature, °F
t_s = supply air temperature, °F

For fully mixed systems with conventional ceiling heights, the return (or exhaust) and room air temperatures are the same; for example, a room with a set-point temperature of 75°F has, on average, a 75°F return or exhaust air temperature.

Methods for Evaluation

The objective of air diffusion is to create the proper combination of room air temperature, humidity, and air motion in the occupied zone to provide thermal comfort and acceptable indoor environmental quality. There are three recommended methods of selecting outlets for mixed air systems using manufacturers' data:

- By appearance, flow rate, and sound data
- By isovels (lines of constant velocity) and mapping
- By comfort criteria

These selection methods are not meant to be independent. It is the designer's choice as to which to start with, but it is recommended that at least two methods be used for any design.

Variation from accepted thermal limits (ASHRAE *Standard* 55), lack of uniform thermal conditions in the space, or excessive fluctuation of conditions in one part of the space may produce discomfort. Thermal discomfort also can arise from any of the following conditions:

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

Selection

By Appearance, Flow Rate, and Sound Data. For a given appearance, flow rate, pressure drop, and sound level criteria, designers can select outlets from manufacturers' catalogs, using the following steps:

1. Determine 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. Determine acceptable outlet noise criterion (NC); consult Chapter 47 of this volume, or Chapter 7 in the 2005 *ASHRAE Handbook—Fundamentals*.
3. Locate a range of products from manufacturers' catalogs that meet the airflow and NC requirements. Multiple outlets in a space at the same cataloged NC, and other design considerations, may result in actual sound levels greater than cataloged values. Manufacturers' data are obtained using ideal inlet conditions, and may vary from field installations. From experience,
   - For identical outlets 10 ft or more apart, the cataloged NC rating applies.
   - Identical outlets within 10 ft of each other add no more than 3 dB to the sound pressure level.
   - For continuous linear outlets, only the sound produced by the closest 10 ft need be considered.

Table 2 Effect of Neck-Mounted Damper on Air Outlet NC

Total Pressure Ratio*	100%	150%	200%	400%
dB Increase	0	4.5	8	16

*Ratio of air pressure before and after damper.

- A wide-open damper installed in the neck of a diffuser can add 4 to 5 NC to the cataloged NC value.
- Significantly closed balancing dampers can add more than 10 NC, depending on duct pressure and how far upstream it is installed. Table 2 gives an example.

4. Select air terminals from manufacturers' catalogs that meet aesthetic and physical needs.

Although these selections may meet the sound requirements for a project, the results do not fully address occupant comfort. Without evaluating the throw of the outlets or room air mixing, this selection method may result in excessive air velocities in the occupied zone, or limited mixing and resultant stagnation. It is recommended that the designer consider selection by isovel mapping or by comfort criteria in addition to selection by appearance, flow rate, and sound data. Either of these methods addresses resulting air motion in the occupied zone and occupant comfort.

By Isovels and Mapping. Using manufacturers' catalog throw data, a designer can predict the path of an outlet's discharge jet. Most manufacturers' catalogs list the distance a jet travels to reach a terminal velocity of 150 to 50 fpm. With this information, the designer can map the path of the discharge jet for a given outlet. This evaluation can prevent problems such as excessively high air velocities in the occupied zone, or stagnation in a given area. Note that most manufacturers' throw data are based on isothermal supply air; the supply jet temperature is equal to the room air temperature. When using this mapping method, consider the positive or negative buoyancy of nonisothermal (heated or cooled) supply air. In both heating and cooling, a discharge jet should travel the distance shown in the catalog to a terminal velocity of 150 fpm without much influence from buoyancy. When evaluating a jet at lower terminal velocities (e.g., 100 or 50 fpm), consider buoyancy's effect on the distance the jet will travel.

A cool air jet travels less far along a horizontal surface than an isothermal jet does. If an outlet is selected so that the horizontal jet does not have enough velocity to reach a vertical surface, the jet can separate from the horizontal surface and project down into the occupied zone, causing drafts and discomfort. Manufacturers' tables show the drop of a cool air jet so the designer can predict the resulting path.

When evaluating heated air, the designer should consider the positive buoyancy of the discharge jet. A heated jet projecting along a horizontal surface or in an upward vertical pattern travels farther than an isothermal jet. In downward vertical discharge, a heated jet travels a shorter distance than an isothermal jet.

Combining selection by isovels and mapping with acoustical selection allows discharge jet location and intensity in a space to be predicted. Outlet selection should be evaluated at the space's typical operating points (i.e., maximum heating and cooling, and minimum heating and cooling). The following steps may be used:

1. Identify the occupied zone for the space.
2. Select outlet(s) that meet design NC, pressure drop, and flow rate requirements. Identify the supply jet location using cataloged throw data.
3. Evaluate air jet mapping to ensure terminal velocities in the occupied zone do not exceed 50 fpm.
4. For overhead heating applications, Δt < 15°F (see Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals*), evaluate the diagram to ensure that jet velocities 5 ft from the floor are at least 150 fpm.

Other design considerations include the following:

- In multiple-outlet applications, jets should not collide to cause a downward projection of air resulting in velocities greater than 50 fpm in the occupied zone.
- For VAV applications, consider both minimum and maximum flow conditions.

Selection by Comfort Criteria T_{50}/L. Selection by isovels and mapping is effective at predicting the path of the discharge jet from an outlet and evaluating resultant occupant comfort. However, there is an established method to quantify occupant comfort during cooling conditions, based on space dimensions and isothermal catalog throw data. This method can be used to predict a space's resulting air diffusion performance index (ADPI).

The comfort criteria T_{50}/L method was developed to predict occupant comfort during cooling conditions, using manufacturers' isothermal catalog throw data (T, usually for 50 fpm terminal velocity) and the dimensions available for throw (L) on the plan view of a mechanical drawing. By using the ratio of T_{50}/L, the designer can predict the level of comfort with a single rating number: ADPI. ADPI can provide further information about the comfort level in a space for results obtained from the NC and mapping selection methods.

Air Distribution Performance Index (ADPI). The air distribution performance index was developed as a way to quantify the comfort level for a space conditioned by a mixed air system in cooling. ADPI uses the effective draft temperature collected at an array of points taken within the occupied zone to predict comfort. ADPI is the percentage of points in a space where the effective draft temperature is between –3 and +2°F and the air velocity is less than 70 fpm. A high percentage of people have been found to be comfortable in cooling applications for office-type occupations where these conditions are met. High ADPI values generally correlate to high space thermal comfort levels with the maximum obtainable value of 100. Select outlets to provide a minimum ADPI value of 80.

The effective draft temperature provides a quantifiable indication of comfort at a discrete point in a space by combining the physiological effects of air temperature and air motion on a human body. The effective draft temperature t_{ed} (the difference in temperature between any point in the occupied zone and the control condition) can be calculated using the following equation proposed by Rydberg and Norback (1949) and modified by Straub (Straub et al. 1956; Straub and Chen 1957) in discussion of a paper by Koestel and Tuve (1955):

$$t_{ed} = (t_x - t_c) - 0.07(V_x - 30) \qquad (2)$$

where

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

T_{50}/L *Selection Method*. This method uses the ratio of cataloged isothermal throw data at 50 fpm to the characteristic length for a given device (Table 3).

Each type of diffuser has different performance characteristics and therefore may provide a different ADPI value for the same cooling application at the same conditions. Calculating T_{50}/L for a given outlet can predict the level of cooling comfort for a space. Using Table 4, the designer can optimize not only the type of diffuser to select but also the size and capacity.

Using T_{50}/L helps designers maximize space cooling comfort; however, this method is not meant to, nor may it be practical to, evaluate T_{50}/L values for each outlet on a project.

Design Procedures. T_{50}/L can be used as a general tool to evaluate cooling comfort levels in a space, at the beginning of design to optimize outlet selection (as shown in the following steps), or at the end of the process to predict cooling comfort levels in spaces designed using NC and mapping methods:

1. Determine air volumetric flow requirements based on load and room size. For VAV systems, evaluation should include both minimum occupied and maximum design flow rates.
2. Select tentative diffuser type and location in room.
3. Determine room's characteristic length L (Table 3).
4. Select recommended T_{50}/L (or T_{100}/L) ratio from Table 4.
5. Calculate throw distance T_{50} by multiplying recommended T_{50}/L (T_{100}/L for linear slots) ratio from Table 4 by available length L.
6. Locate appropriate outlet size from manufacturer's catalog.
7. Ensure that this outlet meets other imposed specifications (e.g., noise, static pressure loss).

Example 1. For a 20 by 12 ft room, with 9 ft ceiling, with uniform loading of 10 Btu/h·ft^2 or 2400 Btu/h, and air volumetric flow of 1 cfm/ft^2 or 240 cfm for one outlet, find the size for a high sidewall grille located at the center of 12 ft end wall, 9 in. from ceiling.

Table 3 Characteristic Room Length for Several Diffusers (Measured from Center of Air Outlet)

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, Btu/h·ft^2	T_{50}/L for Maximum ADPI	Maximum ADPI	For ADPI Greater Than	Range of T_{50}/L
High sidewall grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5 to 2.2
	40	1.6	78	70	1.2 to 2.3
	20	1.5	85	80	1.0 to 1.9
Circular ceiling diffusers	80	0.8	76	70	0.7 to 1.3
	60	0.8	83	80	0.7 to 1.2
	40	0.8	88	80	0.5 to 1.5
	20	0.8	93	90	0.7 to 1.3
Sill grille, straight vanes	80	1.7	61	60	1.5 to 1.7
	60	1.7	72	70	1.4 to 1.7
	40	1.3	86	80	1.2 to 1.8
	20	0.9	95	90	0.8 to 1.3
Sill grille, spread vanes	80	0.7	94	90	0.6 to 1.5
	60	0.7	94	80	0.6 to 1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling slot diffusers (for T_{100}/L)	80	0.3	85	80	0.3 to 0.7
	60	0.3	88	80	0.3 to 0.8
	40	0.3	91	80	0.3 to 1.1
	20	0.3	92	80	0.3 to 1.5
Light troffer diffusers	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Perforated, louvered ceiling diffusers	11 to 50	2.0	96	90	1.4 to 2.7
				80	1.0 to 3.4

Solution:

Characteristic length L = 20 ft (length of room: Table 3)
Recommended T_{50}/L = 1.5 (Table 4)
Throw to 50 fpm T_{50} = 1.5 × 20 = 30 ft

Refer to the manufacturer's catalog for a size that gives this isothermal throw to 50 fpm. One manufacturer recommends the following sizes, when vanes are straight, discharging at 240 cfm: 16 by 4 in., 12 by 5 in., or 10 by 6 in.

More information on conventional mixing systems, and many more design examples, can be found in Rock and Zhu (2002).

FULLY STRATIFIED AIR DISTRIBUTION

Systems that discharge cool air at low sidewall or floor locations with very little entrainment of (and thus mixing with) room air create (vertical) thermal stratification throughout the space. These **displacement ventilation** systems have been popular in northern Europe for some time. Floor-based outlets in underfloor applications may also be used to provide fully stratified air distribution.

Principles of Operation

Thermal displacement ventilation (TDV) systems (Figure 3) use very low discharge velocities, typically 50 to 70 fpm, to deliver cool supply air to the space. The discharge temperature of the supply air is generally above 60°F, although lower temperatures may be used in industrial applications, exercise or sports facilities, and transient areas. The cool air is negatively buoyant compared to ambient air and drops to the floor after discharge. It then spreads across the lower level of the space.

As convective heat sources (Figure 3) in the space transfer heat to the cooler air around them, natural convection currents form and rise along the heat transfer boundary. Without significant room air movement, these currents rise to form a convective heat plume around and above the heat source. As the plume rises, it expands by entraining surrounding air. Its growth and ascent are proportional to the heat source's size and intensity and temperature of ambient air above it. Ambient air from below and around the heat source fills the void created by the rising plume. If the heat source is near the floor (e.g., an occupant), the plume entrains cool, conditioned air from the floor level, which is drawn to the respiration level, and serves as the source of inhaled air. Exhaled air rises with the escaping heat plume, because it is warmer and more humid than the ambient air. Convective heat from sources located above the occupied zone has little effect on occupied-zone air temperature.

At a certain height, where plume temperature equals ambient temperature, the plume disintegrates and spills horizontally. Two distinct zones are thus formed in the room: a lower occupied zone with little or no recirculation flow (close to displacement flow), and an upper zone with recirculation flow. The boundary between these two zones is called **shift zone**. The shift zone height is calculated as the height

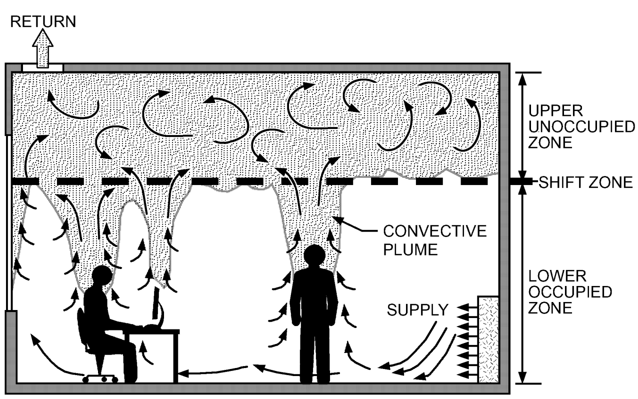

Fig. 3 Displacement Ventilation System Characteristics

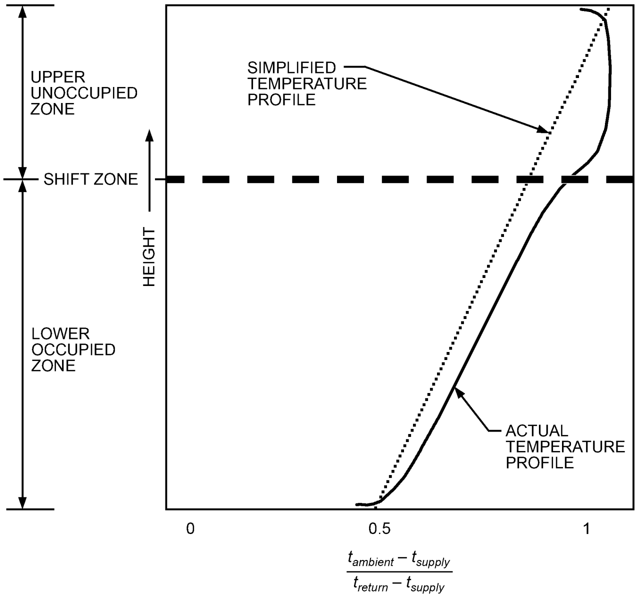

Fig. 4 Temperature Profile of Displacement Ventilation System

above the floor where the total amount of air carried in convective plumes above heat sources equals the supply airflow distributed through displacement diffusers. Actual and simplified representations of the temperature gradient in the space are shown in Figure 4.

Thermal displacement ventilation systems can be modeled as shown in Figure 4. A thin layer of conditioned supply air lies adjacent to the floor. Next is a lower zone in which both ambient air temperature and contaminant concentration levels increase with height. Finally, a pool of warm used or contaminated air (the upper zone) may form next to the ceiling, depending on the supplied airflow rate in proportion to the volume of thermal plumes rising through the space.

Space Ventilation and Contaminant Removal

Thermal displacement ventilation is very effective at removing airborne contaminants that are equal to or lighter than the ambient air (e.g., respiratory-produced contaminants, tobacco smoke). According to ASHRAE *Standard* 62.1-2004, these systems have zone air distribution effectiveness E_Z values of 1.2, compared to maximum values of 1.0 for mixed air systems.

Typical Applications

Thermal displacement ventilation systems typically have higher return air temperatures than mixed systems. Thus, they may allow extended periods of air- or water-side economizer operation, especially in mild, relatively dry climates.

Thermal displacement ventilation systems are commonly used in applications such as

- Restaurants
- Casinos
- Large open-plan offices, classrooms, lecture halls, and meeting rooms
- Theaters and auditoriums
- Hospitals and clean rooms
- Other spaces with high ceilings

Benefits and Limitations

Benefits of displacement ventilation systems include the following:

- Removal of airborne contaminants is more effective.
- In mild climates, significantly less energy may be used to maintain the same space occupied zone air temperature in cooling mode.
- Air distribution effectiveness is high: less outside air is required to meet ASHRAE *Standard* 62.1-2004 requirements.
- Diffuser noise level is lower.
- Lower turbulence intensity can reduce draft-related complaints.

Some applications do not favor use of thermal displacement ventilation. Small offices, especially with perimeter exposures, often do not have room for the large outlets that may be required. The following types of areas may be better served by a mixed system:

- Spaces with ceiling heights less than 9 ft
- Spaces with exceptionally high occupied zone heat loads (Bauman and Daly 2003)
- Spaces with ceiling heights below 10 ft that are subjected to significant room air disturbances
- Applications where contaminants are heavier and/or colder than ambient air

When thermal displacement systems are used in humid climates, it may be necessary to dehumidify and possibly reheat supply air to maintain desired space conditions. As with all HVAC air systems' design, a psychrometric analysis is advised.

Outlet Characteristics

Displacement outlets are designed for average face velocities between 50 and 70 fpm, and are typically in a low sidewall or floor location. Return or exhaust air intakes should always be located above the occupied zone for human thermal comfort applications.

Displacement outlets are available in a number of configurations and sizes. Some models are designed to fit in corners or along sidewalls, or stand freely as columns. It is important to consider the degree of flow equalization the outlet achieves, because use of the entire outlet surface for air discharge is paramount to minimizing clear zones and maintaining acceptable temperatures at the lower levels of the space.

Stationary occupants should not be subjected to discharge velocities exceeding about 40 fpm because air at the ankle level within this velocity envelope tends to be quite cool. As such, most outlet manufacturers define a **clear zone** in which location of stationary, low-activity occupants is strongly discouraged, but transient occupancy, such as in corridors or aisles, is possible. Occupants with high activity levels may also find the clear zone acceptable.

Space Temperature Gradients and Airflow Rates

Figure 5 illustrates thermal temperature gradients that might be expected for a classroom with a 10 ft ceiling, served by thermal displacement ventilation. If loads are typical to the application and proper space airflow is supplied, Skistad et al. (2002) indicate that approximately 50% of the total temperature difference between supply air and return or exhaust air is dissipated in clear zone(s) next to the outlet(s). The other half of the temperature gradient is the **space temperature gradient (STG)**, assumed to be linear with air temperature, increasing gradually from floor to ceiling.

For stationary, low-activity occupants, keep supply air temperatures above 60°F. When occupants are very near outlets (e.g., in underseat delivery), keep supply air temperatures at or above 64°F.

Methods of Evaluation

Unlike mixed systems, outlets in thermal displacement systems discharge air at very low velocities, resulting in very little mixing. As such, design of these systems primarily involves determining a supply airflow rate to manage the thermal gradients in the space in accordance with ASHRAE comfort guidelines. ASHRAE *Standard* 55 recommends that the vertical temperature difference between the ankle and head levels of space occupants be limited to no more than 5.4°F to maintain a high degree (>95%) of occupant satisfaction.

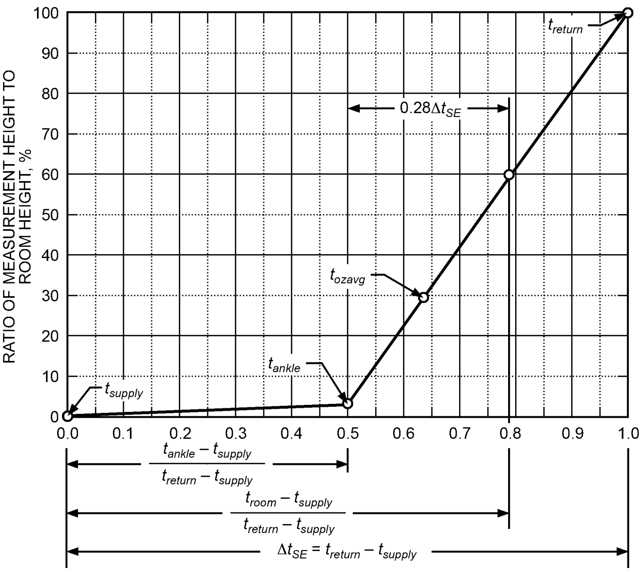

Fig. 5 Temperature Gradient Relationships for Thermal Displacement Ventilation System in Typical Classroom or Office with 10 ft Ceiling

Design Procedures

Displacement ventilation system design is somewhat different than for mixing ventilation. For mixing ventilation systems, where air is mixed relatively evenly throughout the space, the return/exhaust air temperature is assumed to equal the space temperature. In displacement ventilation systems, the space is divided into two vertical zones. The desired space air temperature is maintained only in the lower zone and is always higher in the upper zone because of the temperature stratification created by natural convection.

Depending on space requirements, two types of design methods are used. The most common is temperature-based, and is used when heat removal is the main objective of the air-conditioning system design (e.g., in schools, offices, auditoriums, sport facilities). The other, the shift-zone method, is used when contaminant removal should be considered (e.g., smoking rooms and other facilities with emissions of gaseous, equal- or lighter-than-air contaminants). The objective of the temperature-based method is satisfying thermal comfort in the occupied zone; the shift-zone design addresses this concern and also ensures that contaminants rise above the occupants' breathing level.

Space temperature gradient (STG) is affected by the strength and location of heat sources in the space, heat exchange by radiation between surfaces in the space, and supply airflow. The design procedure presented in this section is based on Skistad et al.'s (2002) simplified method of estimating temperature gradient (Figure 5). This method is applicable for typical spaces with a ceiling height up to 12 ft, such as classrooms, office spaces, and meeting rooms. When designing more complex spaces, computational fluid dynamics (CFD) software programs may be used (see Chapter 34 of the 2005 *ASHRAE Handbook—Fundamentals* for more information).

The thermal gradient relationships illustrated in Figure 5 can be used to establish an acceptable supply-to-return air temperature differential Δt_{SR} from which the supply airflow rate is calculated. Because the space temperature gradient is assumed to be linear, the occupied gradient in the occupied zone is proportional to the volume of the space it represents. For example, if return height is 10 ft and the occupied zone is 5 ft high, its gradient comprises 50% of the space temperature gradient, or 25% of Δt_{SR}. The temperature difference between room air at the top of the occupied zone and the supply air is therefore 75% of Δt_{SR}.

Determining an acceptable Δt_{SR} should consider both the room-to-supply temperature differential and the occupied zone temperature gradient (as limited to 5.4°F by ASHRAE *Standard* 55).

In general, high-ceiling applications allow larger supply-to-return air temperature differentials, because the occupied zone is a smaller percentage of total room air volume. However, the differential may be reduced by limitations on the supply air temperature, as shown in Example 2.

The supply airflow rate Q to achieve Δt_{SR} is calculated from Equation (1).

Example 2. A classroom with a 10 ft ceiling is to be cooled by thermal displacement ventilation. The supply air temperature is 62°F and room temperature is maintained at 76°F at 5 ft level. The total sensible heat gain of the space is 28,000 Btu/h.

Calculate the (1) overall temperature differential between supply and return airflow and (2) required space airflow. Identify return air temperature and temperature at occupants' ankle level.

Solution: Using the relationships in Figure 5, the supply-to-return temperature differential Δt_{SR} and return air temperature can be predicted as follows:

$$\Delta t_{SR} = (t_{room} - t_{supply})/0.75 = (76 - 62)/0.75 = 18.7°\text{F}$$

$$t_{return} = t_{supply} + \Delta t_{SR} = 62 + 18.7 = 80.7°\text{F}$$

To ensure a high level of thermal comfort, the occupied-zone temperature gradient Δt_{oz} should not exceed 5.4°F. For this application, the occupied zone gradient is acceptable:

$$\Delta t_{oz} = \Delta t_{SR} \times 0.25 = 18.7 \times 0.25 = 4.7°\text{F}$$

From Equation (1), the airflow required to maintain this gradient is

$$Q = 28{,}000/(1.08 \times 18.7) = 1386 \text{ cfm}$$

Application Considerations

Displacement ventilation is a cooling-only method of room air distribution. For heating, a separate system is generally recommended. Displacement ventilation can be used successfully in combination with radiators and convectors installed at the exterior walls to offset space heat losses. Radiant heating panels and heated floors also can also be used with displacement ventilation. To maintain displacement ventilation, outlets should supply ventilation air about 4°F lower than the desired room temperature.

Thermal displacement ventilation systems can be either constant or variable air volume. A thermostat in a representative location in the space or return plenum should determine the delivered air volume or temperature. If the time-averaged requirements of ASHRAE *Standard* 62.1-2004 are met, intermittent on/off airflow control can be used.

Avoid using thermal displacement and mixed air systems in the same space, because mixing destroys the natural stratification that drives the thermal displacement ventilation system. Thermal displacement systems can be complemented by hydronic systems such as chilled floors. Use caution when combining chilled ceilings, beams, or panels with fully stratified systems, because cold surfaces in the upper zone of the space may recirculate contaminants stratified in the upper zone back into the occupied zone.

Chen and Glicksman (2003) provide additional information on fully stratified air distribution systems.

PARTIALLY MIXED AIR DISTRIBUTION

A partially mixed system's characteristics fall between a fully mixed system and a fully stratified system. It includes both a high-velocity mixed air zone and a low-velocity stratified zone where room air motion is caused by thermal forces. For example, floor-based outlets, when operating in a cooling mode with relatively high discharge velocities (>150 fpm), create mixing, thus affecting the amount of stratification in the lower portions of the room. In the upper portions of the room, away from the influence of floor outlets, room air often remains thermally stratified in much the same way as displacement ventilation systems.

Principles of Operation

Supply air is discharged, usually vertically, at relatively high velocities and entrains room air in a similar fashion to outlets used in mixed air systems. This entrainment, as shown in Figure 6, reduces the temperature and velocity differentials between supply and ambient room air. This discharge results in a vertical plume that rises until its velocity is reduced to about 50 fpm. At this point, its kinetic energy is insufficient to entrain much more room air, so mixing stops. Because air in the plume is still cooler than the surrounding air, the supply air spreads horizontally across the space, where it is entrained by rising thermal plumes generated by nearby heat sources.

Research and experience have shown that the amount of room air stratification varies depending on design, commissioning, and operation. Control of stratification includes the following considerations:

- By reducing airflow and mixing in the occupied zone, fan energy can be reduced and stratification can be increased, approaching a reasonable target at 3 to 4°F temperature difference from head to ankle height, which satisfies ASHRAE *Standard* 55-2004.
- By increasing airflow and mixing in the occupied zone, excessive stratification can be avoided, thereby improving thermal comfort.

In practice, successful installation requires an optimal balance of these issues (Webster and Bauman 2006).

Figure 6 shows one example of the resulting room air distribution in which the room air is mixed in the **lower mixed zone**, which is bounded by the floor and the elevation (**throw height**) at which the 50 fpm terminal velocity occurs. At this elevation, stratification begins to occur and a linear temperature gradient, similar to that found in thermal displacement systems, forms and extends through the **stratified zone**. As with thermal displacement ventilation, convective heat plumes from space heat sources draw conditioned air from the lower (mixed) level through the stratified zone and to the overhead return location. A third zone, referred to as the **upper mixed zone**, may exist where the volume of rising heat plumes terminate. Although velocities in this area are quite low, the air tends to be mixed.

Space Ventilation and Contaminant Removal

Partially mixed systems' ventilation and contaminant removal efficiencies vary considerably. Restricting mixed conditions to below the breathing level results in most respiratory-associated contaminants being conveyed directly to the overhead return by heat plumes rising from occupants. If the lower mixed zone extends above the breathing level, contaminants are entrained and horizontally transmitted across occupied levels of the space, as occurs in mixed air (dilution ventilation) systems.

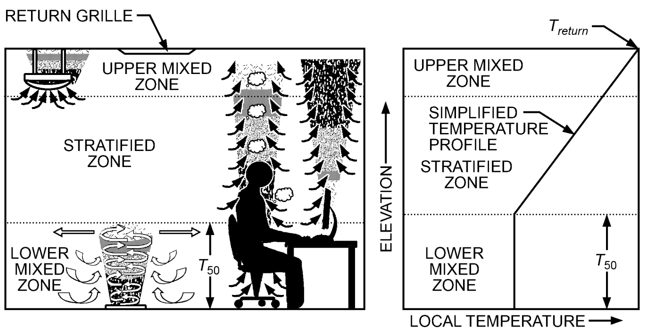

Fig. 6 UFAD System in Partially Stratified Application

According to ASHRAE *Standard* 62.1, these systems may have zone air distribution effectiveness E_Z values that exceed those of fully mixed systems.

Typical Applications

Partially mixed systems are commonly used in applications such as the following:

- Office buildings with raised floors
- Call centers
- Libraries
- Casinos
- Other spaces with open or high ceilings

Many underfloor air distribution (UFAD) systems can be classified as partially mixed systems. These systems are popular because of their relocation flexibility when used in conjunction with raised-access flooring systems. Outlet accessibility also allows easy occupant adjustment of space airflow delivery. The cavity beneath the access floor tiles is generally pressurized and used as a supply air plenum. Supply outlets placed in access floor tiles are commonly tapped directly into the pressurized plenum, but may be ducted from a fan-assisted terminal unit mounted beneath the floor.

Benefits and Limitations

Benefits of UFAD systems include the following:

- Using a raised floor system may substantially reduce air distribution ductwork and terminal requirements.
- Central fan energy consumption may be lower.
- The space service flexibility of the access floor platform is extended to include HVAC services as well. Nonducted outlets can be easily added or relocated.
- Because most outlets are sized to handle loads typical to an interior single-occupant office or workstation, they can be placed within the workstation to give occupants thermal control over their individual work environment. This makes higher individual occupant comfort levels possible.
- Air- and water-side economizer opportunities are extended, especially in mild and relatively dry climates.

Applications where contaminants are heavier and/or colder than ambient air may be better served by a mixed air system. As with thermal displacement systems, partially stratified systems in humid climates require that the outside air be sufficiently dehumidified to satisfy space latent requirements. The temperature of dehumidified air must often be increased before introduction to the occupied space.

Outlet Characteristics

One outlet type is a swirl diffuser with a high-induction core, which induces large amounts of room air to quickly reduce supply to ambient air velocity and temperature differentials. Supply air is injected into the room as a swirling vertical plume close to the outlet. Properly selected, these outlets produce a limited vertical projection of the supply air plume, restricting mixing to the lower portions of the space. Most of these outlets allow occupants to adjust the outlet airflow rate easily. Other versions incorporate automatically controlled dampers that are repositioned by a signal from the space thermostat and/or central control system.

Another category includes more conventional floor grilles designed for directional discharge of supplied airflow. These grilles may be either linear or modular in design, and may allow occupants to adjust the discharge air pattern by repositioning the core of the outlet. Most floor grilles include an integral actuated damper, or other means, that automatically throttles the volume of air in response to the zone conditioning requirements.

Room air induction allows UFAD diffusers to comfortably deliver supply air a few degrees cooler than possible with outlets used for thermal displacement ventilation outlets. The observance of clear, or adjacent, zones above and around the diffusers, where stationary occupants should not reside, is recommended. Outlet manufacturers typically identify such restrictive areas in their product literature.

Space Temperature Gradients and Airflow Rates

The objective of partially mixed systems is to condition the air in the occupied zone while allowing stratification to naturally occur. By allowing this stratification, some of the space heat gain can be removed by return or exhaust instead of by supply air delivery to the space. If the supply airflow rate and sensible heat gains affecting the lower zone are balanced, an acceptable temperature gradient (<5°F) can be achieved in the occupied zone. Supply airflow beyond that required by these heat gains reduces the degree of stratification shown in Figure 6. If the supply airflow rate is insufficient, excessive vertical space temperature gradients may occur.

Accurate calculation of the space design supply airflow rate requires analysis of all space sensible heat gains to determine their contribution to the lower zone. Although there is not yet a single recognized procedure for calculating these airflow rates, most UFAD equipment manufacturers offer guidance.

Methods of Evaluation

As for thermal displacement systems, design involves determining a supply airflow rate that limits thermal gradients in the occupied zone in accordance with ASHRAE *Standard* 55 guidelines. ASHRAE *Standard* 55 recommends that the vertical temperature difference between the ankle and head level of space occupants be limited to no more than 5.4°F if a high degree (>95%) of occupant comfort is to be maintained.

Design Procedures

The design of partially mixed air distribution systems requires identifying both thermal and contaminant removal objectives:

- The desired space temperature, the elevation to which it applies, and an appropriate supply air temperature must be identified.
- The supply air temperature for UFAD systems served by a pressurized or neutral pressure floor plenum should be limited to that which results in a relative humidity level below 80% in the floor cavity, to minimize the threat of mold or fungus growth.
- Supply air temperatures tend to rise as air moves through the floor cavity; therefore, the supply air temperature varies with its distance traveled. When determining space airflow requirements, supply temperatures should be modified accordingly to avoid undercooling the occupied space. Bauman and Daly (2003) discuss this subject further.
- If the objective is to provide displacement ventilation of respiratory contaminants in the stratified zone, mixing must be limited to below the breathing level of most space occupants.
- Outlets should be located far enough from stationary occupants to ensure that they are not subjected to drafts that might cause thermal discomfort. Outlet manufacturers generally prescribe clear zones that quantify this separation distance.

Application Considerations

ASHRAE's *Underfloor Air Distribution Design Guide* (Bauman and Daly 2003) includes a thorough discussion of issues involved in the design, application, and commissioning of UFAD systems. Some considerations include the following:

- Supply temperatures in the access floor cavity should be kept at 60°F or above, to minimize the risk of condensation and subsequent mold growth.
- Most UFAD outlets can be adjusted automatically by a space thermostat or other control system, or manually by the occupant. In the latter case, outlets should be located within the workstation they serve.
- Use of manually adjusted outlets should be restricted to open office areas where cooling loads do not tend to vary considerably or frequently. Perimeter areas and conference rooms require automatic control of supply air temperatures and/or flow rates because their thermal loads are highly transient.
- Heat transfer to and from the floor slab affects discharge air temperature and should be considered when calculating space airflow requirements. Floor plenums should be well sealed to minimize air leakage, and exterior walls should be well insulated and have good vapor retarders. Night and holiday temperature setbacks should likely be avoided, or at least reduced, to minimize plenum condensation and thermal mass effect problems. With air-side economizers, using enthalpy control rather than dry-bulb control can help reduce hours of admitting high moisture-content air, thus also reducing the potential for condensation in the floor plenums.
- Avoid using stratified and mixed air systems in the same space, because mixing destroys the natural stratification that drives the stratified system.
- Return static pressure drop should be relatively equal throughout the spaces being served by a common UFAD plenum. This reduces the chance of unequal pressurization in the UFAD plenum.

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