CHAPTER 57

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 16 of the 2013 *ASHRAE Handbook—Fundamentals* for details. Also see Chapter 20 of the 2013 *ASHRAE Handbook—Fundamentals* for more information on space air diffusion; Chapter 20 of the 2012 *ASHRAE Handbook—HVAC Systems and Equipment* for information on room air distribution equipment; and Chapter 48 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 1.8 m 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 22 and 25°C, relative humidities between 25 and 60%, and occupied zone air velocities below 0.25 m/s. 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.

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

- **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 20 of the 2013 *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.

1. INDOOR AIR QUALITY AND SUSTAINABILITY

Air distribution systems affect not only indoor air quality (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 Environmental Design (LEED®) Green Building Rating System™ was originally created in response to indoor air quality concerns, and has evolved to include prerequisites and credits for increasing ventilation effectiveness and improving thermal comfort (USGBC 2009). 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* 62.1-2010 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, 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 in cooling mode, and yield an air change effectiveness of almost 1.0. More information on ASHRAE *Standard* 62.1 is available in its user's manual (ASHRAE 2011).

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).

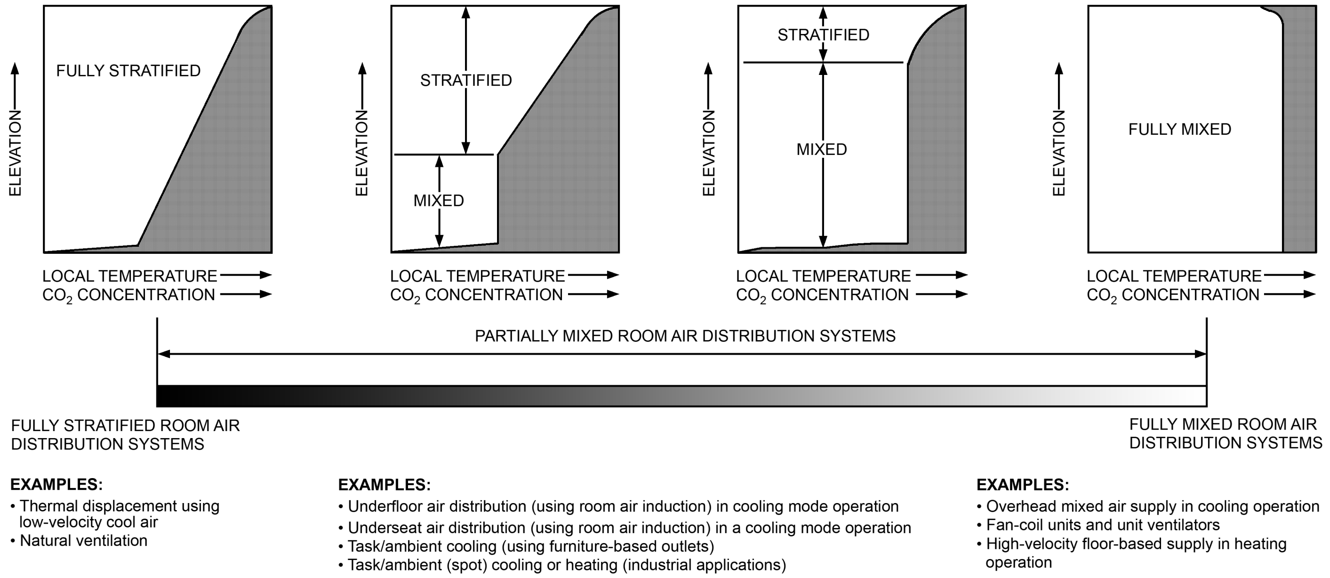

Fig. 1 Classification of Air Distribution Strategies

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.

2. APPLICATION GUIDELINES

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 20 of the 2012 *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 48 of this volume and Chapter 8 in the 2013 *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.

Table 1 Recommended Return Inlet Face Velocities

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

Return Air Inlets

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

3. 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 0.25 m/s are maintained in the occupied zone.

Maintaining velocities less than 0.25 m/s 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 0.25 m/s) 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 20 of the 2013 *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* 62.1-2010, 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 4 m 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.

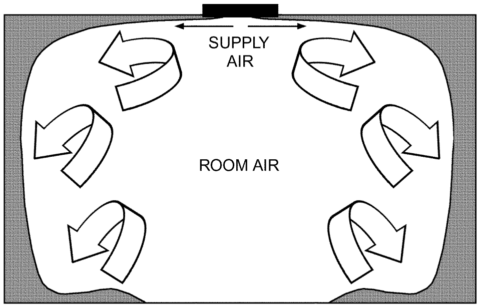

Fig. 2 Air Supplied at Ceiling Induces Room Air into Supply Jet

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).

Overhead outlets may also be installed on exposed ducts, in which case the surface effect does not apply. Typically, if the outlet is mounted 300 mm 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 0.25 m/s 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 300 mm 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 0.5 to 1.25 m 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 1.25 m 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 1.25 m 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 0.75 m/s isothermal throw extends at least halfway down the surface or 1.5 m 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 0.75 m/s should include the distance from the outlet to the perimeter surface. For heating, the supply air temperature should not exceed 8.5 K 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):

$$Q = \frac{q_s}{1.2(t_r - t_s)} \tag{1}$$

where

Q = required supply airflow rate to meet sensible load, L/s
q_s = net sensible heat gain in the space, W
t_r = return or exhaust air temperature, °C
t_s = supply air temperature, °C

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 24°C has, on average, a 24°C 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 48 of this volume, or Chapter 8 in the 2013 *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 3 m or more apart, the cataloged NC rating applies.
   - Identical outlets within 3 m 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 3 m need be considered.
   - 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 0.75 to 0.25 m/s. 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 0.75 m/s without much influence from buoyancy. When evaluating a jet at lower terminal velocities (e.g., 0.5 to 0.25 m/s), 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.

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.

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 0.25 m/s.
4. For overhead heating applications, $\Delta t < 8.5$ K (see Chapter 20 of the 2013 *ASHRAE Handbook—Fundamentals*), evaluate the diagram to ensure that jet velocities 1.5 m from the floor are at least 0.75 m/s.

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 0.25 m/s in the occupied zone.
- For VAV applications, consider both minimum and maximum flow conditions.

Selection by Comfort Criteria $T_{0.25}/L$. 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 $T_{0.25}/L$ method was developed to predict occupant comfort during cooling conditions, using manufacturers' isothermal catalog throw data (T, usually for 0.25 m/s terminal velocity) and the dimensions available for throw (L) on the plan view of a mechanical drawing. By using the ratio of $T_{0.25}/L$, 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 –1.5 and +1°C and the air velocity is less than 0.35 m/s. 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 and Chen 1957; Straub et al. 1956) in discussion of a paper by Koestel and Tuve (1955):

$$t_{ed} = (t_x - t_c) - 7.73(V_x - 0.15) \tag{2}$$

where

t_{ed} = effective draft temperature, °C
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

$T_{0.25}$/L *Selection Method*. This method uses the ratio of cataloged isothermal throw data at 0.25 m/s 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 $T_{0.25}/L$ 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 $T_{0.25}/L$ helps designers maximize space cooling comfort; however, this method is not meant to, nor may it be practical to, evaluate $T_{0.25}/L$ values for each outlet on a project.

Design Procedures. $T_{0.25}/L$ 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:

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, 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 to 2.2
	125	1.6	78	70	1.2 to 2.3
	60	1.5	85	80	1.0 to 1.9
Circular ceiling diffusers	250	0.8	76	70	0.7 to 1.3
	190	0.8	83	80	0.7 to 1.2
	125	0.8	88	80	0.5 to 1.5
	60	0.8	93	90	0.7 to 1.3
Sill grille, straight vanes	250	1.7	61	60	1.5 to 1.7
	190	1.7	72	70	1.4 to 1.7
	125	1.3	86	80	1.2 to 1.8
	60	0.9	95	90	0.8 to 1.3
Sill grille, spread vanes	250	0.7	94	90	0.6 to 1.5
	190	0.7	94	80	0.6 to 1.7
	125	0.7	94	—	—
	60	0.7	94	—	—
Ceiling slot diffusers for $T_{0.50}/L$)	250	0.3	85	80	0.3 to 0.7
	190	0.3	88	80	0.3 to 0.8
	125	0.3	91	80	0.3 to 1.1
	60	0.3	92	80	0.3 to 1.5
Light troffer diffusers	190	2.5	86	80	<3.8
	125	1.0	92	90	<3.0
	60	1.0	95	90	<4.5
Perforated, louvered ceiling diffusers	35 to 160	2.0	96	90	1.4 to 2.7
				80	1.0 to 3.4

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 $T_{0.25}/L$ (or $T_{0.50}/L$) ratio from Table 4.
5. Calculate throw distance $T_{0.25}$ by multiplying recommended $T_{0.25}/L$ ($T_{0.50}/L$ 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 6 by 3.7 m room, with 2.7 m ceiling, with uniform loading of 31.5 W/m² or 700 W, and air volumetric flow of 5 L/s per square metre or 111 L/s for one outlet, find the size for a high sidewall grille located at the center of 3.7 m end wall, 225 mm from ceiling.

Solution:

Characteristic length L = 6 m (length of room: Table 3)
Recommended $T_{0.25}/L$ = 1.5Table 4
Throw to 0.25 m/s $T_{0.25}$ = 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 at 0.25 m/s: 400 by 100 mm, 300 by 125 mm, or 250 by 150 mm.

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

4. 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 0.25 to 0.35 m/s, to deliver cool supply air to the space. The discharge temperature of the supply air is generally above 16°C, 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 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* 62.1-2010, these systems have zone air distribution effectiveness E_Z values of 1.2, compared to maximum values of 1.0 for mixed air systems.

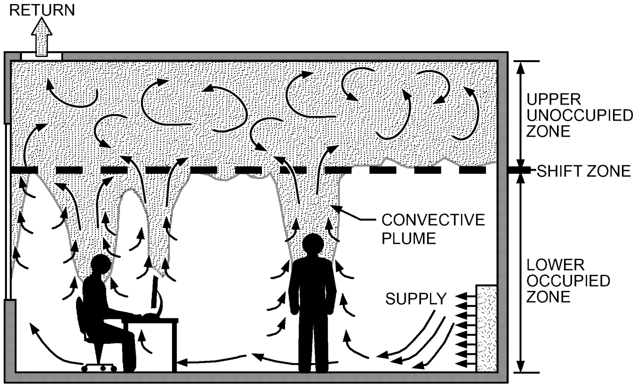

Fig. 3 Displacement Ventilation System Characteristics

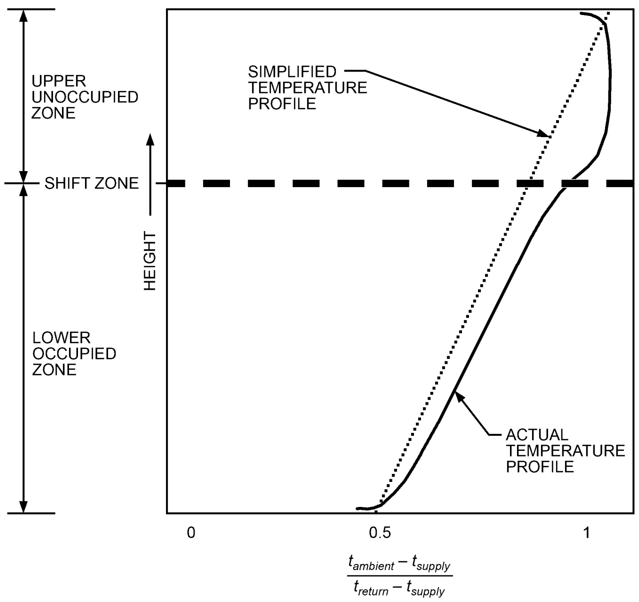

Fig. 4 Temperature Profile of Displacement Ventilation System

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 outdoor air is required to meet ASHRAE *Standard* 62.1-2010 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 2.7 m
- Spaces with exceptionally high occupied zone heat loads (Bauman and Daly 2003)
- Spaces with ceiling heights below 3 m 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 0.25 to 0.35 m/s, 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 0.2 m/s 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.

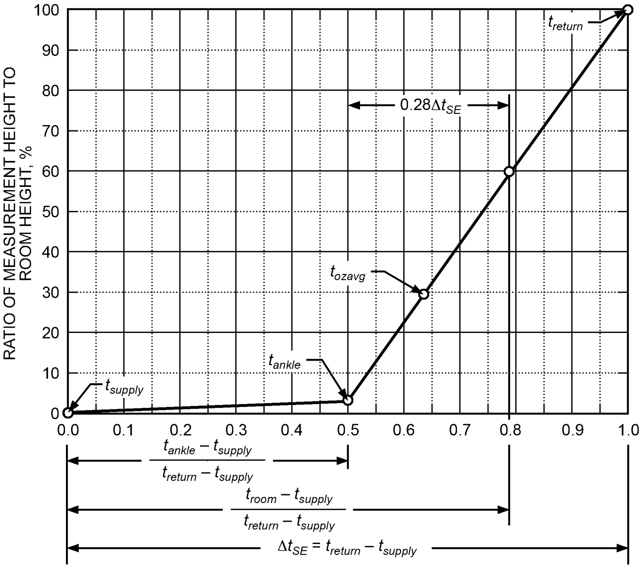

Fig. 5 Temperature Gradient Relationships for Thermal Displacement Ventilation System in Typical Classroom or Office with 3 m Ceiling

Space Temperature Gradients and Airflow Rates

Figure 5 illustrates thermal temperature gradients that might be expected for a classroom with a 3 m 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 16°C. When occupants are very near outlets (e.g., in underseat delivery), keep supply air temperatures at or above 18°C.

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 3 K to maintain a high degree (>95%) of occupant satisfaction.

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 3.7 m, such as classrooms, office spaces, and meeting rooms. When designing more complex spaces, computational fluid dynamics (CFD) software programs may be used (see Chapter 13 of the 2013 *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 3 m and the occupied zone is 1.5 m 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 3 K 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 3 m ceiling is to be cooled by thermal displacement ventilation. The supply air temperature is 16.5°C and room temperature is maintained at 24°C at 1.5 m level. The total sensible heat gain of the space is 8200 W.

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:

$$\Delta t_{SR} = (t_{room} - t_{supply})/0.75 = (24 - 16.5)/0.75 = 10 \text{ K}$$

$$t_{return} = t_{supply} + \Delta t_{SR} = 16.5 + 10 = 26.5°\text{C}$$

To ensure a high level of thermal comfort, the occupied-zone temperature gradient Δt_{oz} should not exceed 3 K. For this application, the occupied zone gradient is acceptable:

$$\Delta t_{oz} = \Delta t_{SR} \times 0.25 = 10 \times 0.25 = 2.5°\text{C}$$

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

$$Q = 8200/(1.2 \times 10) = 683 \text{ L/s}$$

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 2 K 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.

5. 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 (>0.75 m/s), 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 0.25 m/s. 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 1.5 to 2.5 K temperature difference from head to ankle height, which satisfies ASHRAE *Standard* 55-2010.
- By increasing airflow and mixing in the occupied zone, excessive stratification can be avoided, thereby improving thermal comfort.

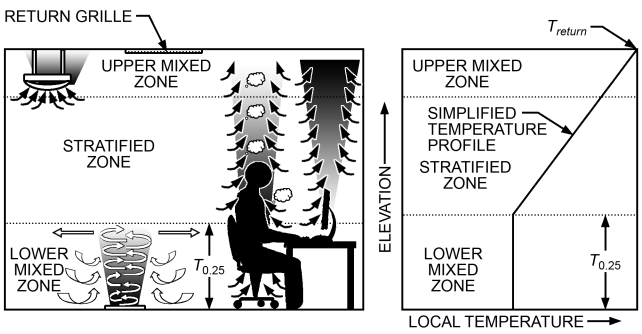

Fig. 6 UFAD System in Partially Stratified Application

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 0.25 m/s 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.

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 outdoor 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 (<3 K) 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 3 K 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 16°C 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.

6. TERMINAL UNITS

System Selection

Designers have various systems (terminal units and their associated controls) to choose from when designing a building. The owner's needs must be met for installation, application, and cost of operation. The designer must consider performance, capacity, reliability, energy consumption, sustainability, and spatial requirements and restrictions. The following guidelines describe different types of equipment and their general uses, restrictions, and limitations. Table 5 summarizes the different types of terminal units

Table 5 Suitability of Terminal Units for Various Applications

	Facility Type													
	Office Space, Educational, and Institutional Buildings					Hospitals, Cleanrooms, and Laboratories[a]			Noise-Sensitive Applications[b]			Other Facilities		
	Large			Small										
Terminal Types	Interior Zone	Exterior Zone	Low Temperature	Interior Zone	Exterior Zone	Patient Areas	Operating Areas	Laboratory Space	Broadcast Studios	Theaters	Libraries	Public Use	Shopping Centers	Mixed Use
Single-Duct														
VAV without reheat	•	•	•	•	•	•	N	•	•	•	•	•	•	•
VAV with reheat	•	•	•	•	•	•	P	P	•	•	•	•	•	•
Dual-Duct														
VAV no mixing	•	•	N	N	N	N	N	N	•	•	•	•	•	•
VAV with mixing	•	•	N	N	N	•	•	N	•	•	•	•	•	•
Constant volume	•	•	N	N	N	P	P	•	P	•	•	•	N	•
Exhaust Terminal	•	•	N	N	N	P	P	P	•	•	•	•	•	•
Induction Terminal														
VAV with heat	•	•	•	•	•	•	N	•	•	•	•	•	•	•
VAV without heat	•	•	•	•	•	N	N	N	•	•	•	•	•	•
Fan-Powered														
Parallel with heat	N	•	•	N	•	N	N	N	N	•	•	P	•	•
Series without heat	P	•	•	•	•	N	N	N	P	P	P	P	P	P
Series with heat	•	P	•	•	•	•	N	•	P	P	P	P	P	P
Low-temperature	N	N	P	N	N	N	N	N	N	N	N	N	N	N
Bypass	N	N	N	•	•	N	N	N	N	N	N	N	N	N

P = Preferred for this application.
• = Used for this application.
N = Not recommended for this application.
[a]Sealed lining is recommended to minimize entrainment of airborne fibers from liner to occupied spaces.
[b]Special consideration should be given to selecting very quiet operating equipment and use of attenuators or silencers.

and their suitability for particular commercial building applications.

Building Use. Before specifying equipment types, the designer must consider the building's intended use. Office buildings with daily operational schedules frequently use fan-powered terminal units. Usually, fan-powered terminals with auxiliary heaters (supplementary heat) are used in perimeter zones; these units allow the greatest flexibility for individual zones while also allowing the central system to be turned off during unoccupied periods. During unoccupied periods, the fan-powered terminal units maintain the minimum or setback temperature levels without the help of the central air-conditioning equipment.

In institutional, medical, or campus buildings, systems that provide pressurization differences between interior areas may be required.

Buildings with centralized heating and cooling plants sometimes use dual-duct terminal units.

Building Size. In large buildings, central air handlers deliver large quantities of air to many zones with different needs. Interior zones may not require heat; therefore, they can be served either by single duct or fan-powered units with no supplemental heat. Unless the building is located in a tropical climate, the perimeter zones require heat, typically electric or hot water. These are usually included with the terminal units, but sometimes separate heating systems are used, such as dual-duct or baseboard. In buildings where the owner desires low operating costs, the static pressure in the ducts should be lowered in accordance with ASHRAE *Standard* 90.1 to the minimum pressure, which sets at least one VAV damper to near-full open. Interior zones in these buildings can use fan-powered terminal units to keep the static pressure low. Buildings with parallel-type fan-powered terminal units usually use single-duct units in the interior zones and require higher system static pressures.

In small buildings, such as shopping malls and other low-rise buildings where each tenant area is small, it is common to use small packaged air conditioners. If terminal units are used on these systems, usually single duct or bypass units are selected. A variation of this system, variable-volume variable-temperature (VVT), uses pressure-dependent single-duct units with a main bypass damper in the supply duct. The bypass damper is regulated by static pressure in the supply duct. A nearly constant pressure can be maintained, allowing the packaged units to operate at constant volume and the individual zones to be pressure-dependent VAV.

Building Controls. The type of control system to be used is somewhat dependent on the size and type of building. Controls can be electric, pneumatic, analog electronic, or digital electronic.

- Electric controls are pressure dependent where the damper responds to a single control input. For example, the thermostat sends a signal to the damper to open or close based solely on room sensible temperature.
- Pneumatic controls are usually used for building renovation or buildout of existing buildings where the base building already has a pneumatic system installed. They can be pressure dependent or pressure independent. Pneumatic controls require regular system maintenance and may need to be periodically rebalanced.
- Analog controls are often applied to smaller buildings that do not have a building operation staff. Typically, these controls do not communicate with other zones or other equipment in the building.
- Digital controls are typically used on buildings that have a building operation staff. These controls provide individual zone control and communication to the building management system.

Acoustical Considerations. Terminal units and room air distribution devices are typical equipment sound sources for the room.

However, they are not the only sources that affect room acoustics. Refer to Chapter 48 as well as AHRI *Standard* 885 and other standards for guidance on space acoustics. Broadcast studios, theaters, and libraries require very low noise levels. Equipment selection and location is important here, and careful examination of the equipment sound performance is imperative. Radio-frequency interference (RFI) and electromagnetic interference (EMI) should also be considered when designing television studios.

Environmental Factors. Environmental factors play an important role in system selection. They include the climate and air conditions inside as well as outside the building. They also include legislative requirements such as outdoor air ventilation rates and local building codes. When high ventilation rates are required, reheat is often required to maintain human comfort. Fan-powered terminal units are usually used where the thermal load changes significantly and heating is required. Single-duct terminal units are usually applied in the interior where the thermal load is normally stable.

Contamination Considerations. Hospitals, cleanrooms, and laboratories pose special challenges. Protective isolation spaces such as operating rooms, bone marrow transplant patient rooms, AIDS patient areas, and cleanrooms require positively pressurized environments. Infectious isolation spaces such as tuberculosis patient rooms require negatively pressurized environments. See ASHRAE *Standard* 170 for details on room pressurization, Chapter 8 for specifics on health care requirements, and Chapter 18 for details on clean spaces. Hospital rooms and cleanrooms frequently also require constant high ventilation rates, which tend to favor single- or dual-duct terminal units. Pressure-independent, variable-speed motor technology has lead to the development of fan-powered pressurization units.

To minimize entrainment of fibers into the airstream, either do not use internal insulation or use special liners in the terminal units and duct systems. Insulations can be isolated from the airstream by metal, foil, or polymer liners inside silencers and terminal units. All of these liners have different thermal, acoustic, and other physical properties and should be evaluated for each job.

Maintenance and Accessibility.

Typical Applications. Terminal units are typically not easily accessible after building occupation; they should be selected and located with consideration for required maintenance. Review the applicable building codes [e.g., ICC (2009)] for required access.

Critical Environments. Some applications, such as cleanrooms and operating theaters, require high levels of reliability from terminal units because of the difficulty and cost associated with servicing or maintaining the equipment. In a cleanroom, for example, if the ceiling must be opened, the space may require disinfection before it can be used again. Associated costs might include lost production time as well as the cost for wipedown and/or disinfecting the room and/or equipment. In cases like these, the designer should consider locating the equipment outside of the clean space or consider highly reliable, very-low-maintenance, very basic equipment and controls. Terminal units may require access to internal components for cleaning in the case of contamination.

Cost Factors. Costs should be considered before system selection is finalized. Installation, operation, and maintenance all contribute to total cost. Often, one of these costs overrides the others. Electric heaters usually have a lower installed cost than hot water coils, but they may have a higher operating cost. Local utility rates and building codes should be researched to arrive at the correct decision before making the final selection.

Applications

Single Duct. The basic single-duct unit consists of an airflow regulator and may also include an actuator, an airflow-measuring device, and selected controls. Accessory discharge attenuators and multiple outlet attenuators are also frequently used.

Typical applications include

- Where the supply air system is not tasked with the space heating requirements
- With VVT or other auto-changeover controls
- Constant air volume
- Constant pressure control

A **single duct with reheat** has an added heating coil (hot water or electric). They are typically applied in zones where heat losses create a need for heating. The terminal usually reheats at the minimum airflow setting. An auxiliary higher heating setting may be available as an option with additional controls.

Interior zones, where ventilation requirements may be larger than the desired heating airflow, may require additional reheat.

A **single-duct exhaust** consists of an airflow regulator and may also include an actuator, an airflow-measuring device, and selected controls. Accessory inlet attenuators are sometimes used. Specialty materials may need to be considered for corrosion resistance. The designer should consider pressure drop and inlet versus outlet designs.

Single-duct exhaust units are typically used in isolation wards, operating theaters, laboratories, cleanrooms, and systems that require supply and exhaust tracking.

Dual Duct. A **nonmixing dual duct** is effectively two single-duct terminal units side by side. The basic unit incorporates separate cold and hot (sometimes neutral) air inlets. They are usually applied in exterior zones in buildings where overhead heating and cooling are desired but not auxiliary heat, and zero minimum flow is acceptable during changeover between heating and cooling.

A **mixing dual duct** is the same, with an integral mixing/attenuator section on the downstream end of the terminal unit to minimize temperature stratification in the discharge airstream. Interior and exterior zones in buildings (e.g., hospitals) where overhead heating and cooling are desired but an auxiliary heating coil is not, and zero minimum flow is unacceptable during changeover between heating and cooling. Mixing performance should be evaluated to fit the application.

Fan-Powered Parallel Flow. Sometimes called **variable-volume** or **intermittent-fan units**, these consist of a single duct unit, blower/motor, backdraft damper, and selected controls where the motor and primary damper are arranged such that mixing occurs downstream of the blower. Supplemental heating coils (either hot water or electric) are generally required. Electric heaters are typically located on the discharge of the unit. Water coils may be on the discharge or the induction port, although location on the discharge adds to the supply air system's static pressure requirements and increases leakage through the backdraft damper, as shown in ASHRAE research project RP-1292 (Davis et al. 2007). Heating coils on the induction port increase ambient temperature at the motor and decrease motor life.

Fan-powered parallel-flow units are used in exterior zones where heating and cooling loads may vary considerably, and in buildings where heating is needed when the central system is shut down during unoccupied hours.

Fan-Powered Series Flow. The basic unit consists of a single-duct unit, blower/motor, and selected controls where the motor and primary damper are arranged such that mixing occurs upstream of the blower. These units are also called **constant-volume** or **constant-fan**. Supplemental heating coils (either hot water or electric) are generally required. Electric heaters are typically located on the discharge of the unit; water coils may be on the discharge or the induction port. Heating coils on the induction port increase ambient temperature at the motor and decrease motor life.

Fan-powered series-flow units generally are used in the following situations:

- Exterior zones where heating and cooling loads may vary considerably
- Buildings where heating is desired when the central system is shut down during unoccupied hours
- To allow lower central system static pressure
- Where occupant comfort can be optimized, because the high- (sometimes constant-) volume variable-temperature air delivery produces consistent air distribution, acoustics, and ventilation

Low-Profile Fan-Powered Series or Parallel Flow. Similar in construction to the standard series or parallel flow terminal described earlier, these units are typically less than 300 mm high for all sizes, to minimize the depth of ceiling space required. Unlike standard fan-powered terminals, the fan/ motor assembly is installed flat on its side with the wheel rotating in a horizontal plane.

Typical applications are the same as for regular series or parallel units, but low-profile versions are commonly used where zoning requirements limit building height and the architect wishes to maximize the number of floors, because these units fit in a shallow ceiling plenum. Designers should pay special attention to available space and unit heights.

Ventilation Air Inlet Fan-Powered Series Flow. These units are similar in construction to the standard series-flow terminal, but have an added secondary air inlet that provides a direct connection to the terminal unit for ventilation air. They are commonly used in buildings where ventilation air is piped in a dedicated ventilation duct system to each terminal unit; this is generally done where it is desirable to monitor ventilation air quantities to each zone.

Low-Temperature Fan-Powered Series Flow. These units are the same as fan-powered series flow, but have special construction to minimize the potential for condensation. They can be used with cold-water/ice storage systems that provide low-temperature central system air distribution to the zone terminals when there is potential for condensation, or where standard terminals may be exposed to high humidity.

Underfloor Fan-Powered Series Flow. This unit is a fan-powered series flow terminal designed to fit between the pedestal support grids of a raised- or access-floor HVAC system, without modification to the floor. They are available in several unit sizes, but with limited height and width.

Primary and induction ports, if any, to the unit may or may not be ducted. Typically, air under the raised floor is cool air supplied directly to the space, although heated air may also be ducted to the unit. In these cases, a control system is required to select the proper damper sequence to control room air distribution to maintain the proper ambient conditions in the occupied space.

Bypass Terminals. The basic bypass terminal consists of a diverter-type damper, actuator, bypass port, and selected pressure-dependent controls. A balancing damper is recommended ahead of the inlet. Use of reheat coils is discouraged, and electric reheat should be prohibited because of the potential fire hazard.

Bypass terminals are used primarily with packaged rooftop air conditioning (PTAC) equipment with a direct-expansion (DX) coil where zoning is desired, but relatively constant airflows across the system components (e.g., coils, fans) are required to minimize the potential for freeze-up. The system offers an economical VAV supply design with low first cost. It does not provide the energy-saving advantages of variable fan volume, but avoids the expense of a more sophisticated system.

Comparison of Series- and Parallel-Flow Fan-Powered Terminal Units

Series (constant-volume) units have a continuously operating fan during occupied mode, and are typically installed in the ceiling plenum. Primary air is ducted from a central air handler. Induction air is either from the ceiling plenum or occasionally ducted from the conditioned space. Fan air is a combination of plenum and primary air supplied to the zone.

Parallel (variable-volume) units should have an intermittently operating fan during the occupied mode and are typically installed in the ceiling plenum. Primary air is ducted from the central air handler and should flow directly to the zone in a variable volume cooling mode. Fan air is induced from the ceiling plenum or may be ducted from the conditioned space. The fan should run in the dead band and heating modes. When the fan runs, the zone air is induced air or a combination of primary and induced air:

- **Primary air** is that delivered to the space for the purposes of satisfying ventilation, latent, and all or part of the sensible load.
- **Secondary air** is that circulated from the return air plenum.
- **Supply air** is a mixture of primary and secondary air delivered to the space.

Configuration.

Series. The fan and VAV damper are aligned so that all conditioned air and all induced air independently enter the mixing section and must go through the fan to exit the unit. The mixing section is between the VAV damper and the fan.

Parallel. The fan and VAV damper are arranged so that all induced air enters the fan, and conditioned air bypasses the fan. Any mixing of conditioned air with the induced plenum air occurs on the discharge side of the fan. A backdraft damper inhibits air from exiting the unit through the fan when not running.

Terminal Fan Selection.

Series. In occupied mode, the fan runs continuously, supplying either a constant or modulating volume to the space. Some direct digital controls (DDCs) provide an optional output that may be used for controlling fan airflow using the building management system (BMS). This allows dynamic fan volume control, which may be either modulating or multiple-speed operation from a single speed motor. The fan must be sized to match the maximum airflow to be supplied to the zone. These terminal fan airflows are usually larger than those with parallel units for similar zones. When modeling energy consumption for this unit, it is important to model the energy consumption and heat generated by the fan motor for both unoccupied and occupied periods. The motor heat should also be included in the heating mode calculation (Davis et al. 2007).

Parallel. Typically, the fan runs only in heating or deadband modes, supplying a relatively constant induced air volume to the space. The fan must be sized to supply the required heating airflow to the zone, which requires overcoming the pressure created in the mixing chamber caused by the inclusion of primary air. These units usually have smaller terminal fan airflows than series units for similar zones. When modeling energy consumption for this unit, it is important to model casing and backdraft damper leakage as well as fan energy consumption for both the terminal unit and the air handler during operation. Motor heat should also be included in the heating mode calculation (Davis et al. 2007).

VAV Cooling and Inlet Static Pressure Requirements.

Series. Additional savings above that for the single- or dual-duct units can be realized because of the low inlet static pressure requirement of the series fan powered terminal unit. The pressure at the air handler can be reduced to that required to push the conditioned air through the ducts to the unit and across the VAV damper into the mixing section.

Parallel. Like the single or dual duct unit, the air handler must push the conditioned air through the ducts to the parallel unit, across the VAV damper, into the mixing section, through the discharge duct from the unit and across the diffuser(s) into the room.

Control Sequence.

Series. The fan runs constantly during occupied periods. On a call for cooling, the controls modulate the VAV damper toward maximum airflow, delivering primary air to the mixing chamber. If the fan is set at the same airflow as the primary air, no air is induced from the plenum. If the fan is set at a higher airflow than the primary air (e.g., as in a low-temperature application), air is induced from the plenum to meet the fan's set point. The primary and induced air are blended before they enter the fan. Constant-volume, variable-temperature air is then discharged into the downstream duct and into the conditioned space.

As cooling demand decreases, the VAV damper modulates toward minimum airflow, reducing the primary air into the mixing chamber. This increases the volume of warmer induced air into the mixing chamber. Fan air can also be reduced [typically with an electronically commutated motor (ECM)] as load changes. The increased percentage of plenum air causes the discharge temperature to rise to approach the plenum temperature, taking advantage of recaptured heat from lights, people, and machinery.

After a further decrease in zone temperature, the controls automatically energize the supplemental heat (optional equipment), which can be either electric or hot water coils. The discharge temperature increases as heat is applied. As the temperature increases in the zone, the sequence reverses.

Parallel. On a call for cooling, the controls modulate the VAV damper toward maximum airflow while the fan is off. Variable-volume, constant-temperature air is then discharged into the downstream ducts and into the conditioned space.

On a decreasing call for cooling, the VAV damper modulates toward minimum airflow. The unit delivers variable-volume, constant-temperature air to the zone.

In deadband, the controls energize the fan. Fan air and primary air are blended in the mixing chamber on the fan's discharge side. The increased plenum air causes the discharge temperature to rise. Constant-volume, constant-temperature air is delivered to the zone.

On a call for heating, the heat is energized. The heater may be staged or modulating. Constant-volume, constant- or variable-temperature air can be delivered to the zone.

Because the largest energy-consuming characteristic of parallel boxes is leakage and not motor energy, and because modulating fan air in the heating mode increases reheat, ECMs are not recommended for parallel boxes.

Fan Interlocks.

Series. Typically, series units are designed to run continuously. Usually, they are energized only during occupied periods or when needed for emergency heating during unoccupied periods. Care should be taken to interlock the unit fan with air handlers in the building to ensure that the terminal unit fans start during occupied periods. Series-unit fans should be started ahead of the air handler to prevent backflow into the plenum and backward rotation of the fan.

Parallel. Fans in parallel units are designed to be energized in the deadband and heating modes throughout the day. Primary air enters the mixing chamber at the fan discharge. When the fan is not energized, there is a positive pressure at the fan's discharge. Typically, this would cause the blower and motor to rotate backwards; however, parallel units are equipped with backdraft dampers at the fan, which inhibit backward airflow through the fan into the plenum.

Acoustics.

Series. Series fans are sized to match the maximum airflow required in the zone. The fan runs constantly during occupied periods. There are two sound sources in the unit: the fan motor and the VAV damper. Although both contribute to the overall discharge and radiated sound emitted from the unit, the fan is primarily responsible for discharge noise, and both the damper and the fan are responsible for radiated sound. Usually, sound radiated into the occupied space is the greater and usually more difficult to attenuate.

Comparing the sound level between a series and a parallel unit in similar zones, the fan in the series unit might generate a slightly higher sound power level. In the series unit, the fan and damper would be at their peak when the unit operates at full cooling capacity, the loudest point in the sequence of operation for sound generation. As the primary air decreases, the sound generated by the fan typically masks the sound from the damper. Sequences that also command the fan to modulate in this condition create lower ambient sound levels.

Series fan power sound levels are more consistent compared to ambient background sound levels than the rapid cyclic nature of the parallel unit.

Parallel. Parallel fans are sized to match the heating airflow required in the zone. The fan runs intermittently when the mode changes from cooling to deadband. There are two sound sources in the unit: the fan/blower and the VAV damper. Both damper and fan are responsible for radiated and discharge sound. Usually, sound radiated into the occupied space is the greater and usually more difficult to attenuate.

Comparing the sound level between a parallel and a series unit in similar zones, the parallel unit may generate slightly less sound. In the parallel unit the fan and damper would normally not peak simultaneously. When the unit operates at full cooling capacity, the damper is at its peak sound generation and the fan is off. In heating mode, the fan peaks while the damper is at minimum sound generation.

Damper sound must be considered as the sound increases with increasing inlet static pressure. Parallel units require significantly higher inlet static pressures at the unit. Fan sound is consistent into the zone when the fan is running; however, the fan is intermittent during much of the day. This rapid fan cycling can create a variation in sound levels and subsequent noise in the space, which can be more annoying than a higher consistent sound level.

Energy Consumption.

Fan-powered VAV terminal units take advantage of typical VAV savings at the air handler and chiller during cooling periods, and even more savings are realized when heating is required. Fan-powered terminals induce warm plenum air from the ceiling and blend it with the primary air at minimum ventilation requirements as required by ASHRAE *Standard* 90.1 during the heating sequence. This recaptures much of the heat created in the zone and plenum by lights, occupants, solar loading, and machinery or equipment such as computers, coffee machines, copiers, etc. The unit returns this heat as free heating rather than losing it at the air handler. If additional heating is required, supplemental heat is added to the sequence, but the unit still saves energy by warming blended air, for example, at 22°C rather than reheating primary cooled air at 13°C, saving the cost of 9 K at the heating airflow. According to ASHRAE research project RP-1292 (Davis et al. 2007), there is very little difference in total building energy use between series and parallel units when equipped with permanent split-capacitor (PSC) motors.

Series. Series fans run constantly during occupied periods, and the fan is sized for the full airflow to the zone. This causes the energy consumption from the fan to be higher than that of a parallel fan in a similar zone. ASHRAE RP-1292 identified the motor as the biggest energy user in a series unit (Davis et al. 2007). The fan energy raises the air temperature across the motor by 0.6 to 1.7 K. This means that total energy use can be reduced if the fan energy is reduced. Using electronically commutated motors (ECMs) can significantly increase motor lifetimes and provide significant energy savings in excess of 50%.

Series units are designed for very low inlet static pressures. This saves energy at the air handler compared to a parallel unit for a similar zone.

Casing leakage does not affect energy consumption of series units. All the primary air injected into the unit gets delivered to the occupied zone. This causes ceiling plenum temperatures to be warmer for installations with series units than those with parallel units, and more heat can be reclaimed from the ceiling plenum when the unit is in heating mode.

Because the motor heat is in the airstream, more heat is introduced into the occupied zone during cooling mode when compared to a parallel unit.

Parallel. Parallel unit fans operate only when required during heating and deadband modes. The fan is sized for the induced airflow required in heating mode, which may be much less than the zone's total airflow requirement.

Because the minimum primary air and induced air are mixed downstream of the terminal fan, the terminal inlet static pressure requirement is greater than for a series terminal. This adds operating cost at the air handler. When just one duct path is designated as the critical path for static pressure set point, this can cause the system pressure to increase in excess of 250 Pa compared to a building with series units. This can greatly increase energy usage. However, ASHRAE *Standard* 90.1 procedures reset the system static pressure to always drive at least one VAV damper to its nearly full-open position, making the increased static pressure required for parallel units not as great.

According to ASHRAE RP-1292, the largest single energy issue in both types of terminal unit, other than operating schedule, was leakage in the parallel unit (Davis et al. 2007). Leakage is typically between 10 and 15% of the total airflow through the parallel unit, and is highest at full cooling. This means that parallel units may need to be oversized to cover the total load to the occupied zone. It also means that the ceiling plenums are cooler with parallel units than those with series units, reducing the amount of free heat that can be reclaimed in heating mode.

Because the motor runs in deadband and heating modes, any heat generated by the motor is delivered to the occupied space. Consequently, the motor energy does not significantly affect the total energy consumed by the parallel unit. Because of this, using more efficient motors will not measurably improve the unit's energy consumption.

7. FAN SELECTION

When selecting a unit for a particular set of conditions, care should be taken so that the air delivery is designed to meet the room's sound criteria and system's static requirements. Specific sound data can be found in manufacturers' catalogs for various airflow deliveries for each unit. This should be the guiding factor in selecting unit sizes. A simple rule of thumb is that, when considering a unit selection for a typical office space, the fan should be selected for performance down from the high end performance by 20 to 25% of the distance to the low end of the fan curve at the specified external static requirement. This allows for low room sound levels while maintaining some flexibility for future changes in the zone. When selecting equipment for large, open areas where sound criteria may not be as critical, select equipment closer to the high end of the fan curve. For a meeting room or an executive office, select equipment slightly below the halfway point on the fan curve. For an auditorium, chief executive offices, conference room, or some similarly sensitive area, select operation nearer the low end of the fan curve.

Avoid selecting equipment right on the maximum or minimum curves. This leaves no flexibility, either in the equipment for future changes or for variations that may occur due to power variations or duct fittings.

Fan Airflow Control on Fan-Powered Terminals

When designing air systems and using fan-powered VAV terminal units, it is important to match the fan air and primary air capacities to the space requirements. Series units require precise adjustment of fan airflow in relation to the primary air. The parallel unit's fan airflow requires less critical adjustment. Fan-powered terminals nearly always use single-phase motors, most commonly with electronic fan speed control [sometimes called **wave choppers**, **thyristor controllers**, or **silicon-controlled rectifiers (SCRs)**] or with electronically commutated motors.

Fan Shift in Fan-Powered Terminal Units. Before adjusting the fan, the possibility of fan shift must be considered. This occurs when the blower is subjected to variations in pressure or airflow patterns. As the primary airflow changes, the pressure drop and changes in local jets may cause the fan to shift its performance as it rides the fan curve. Consequences from the phenomenon vary from building to building and zone to zone. Noise levels may change greatly as the volume changes, and this may be annoying. Design ventilation rates can also vary, sometimes by more than 20%. This can be aggravated by undersizing the terminal unit.

Electronic Fan Speed Control (PSC Motors). Electronic fan speed controls use a thyristor to adjust the fan's electrical input ac voltage. This is called **phase proportioning** or **wave chopping**. When the current sine wave crosses the zero point, the thyristor acts as a timing device, holding the voltage off the motor for some preset period of time. When the thyristor is turned on, the voltage seeks out the sine wave and then follows the curve to the next current zero crossing, where the process begins again on the opposite side of the sine wave. Basically, this reduces the root mean square (RMS) value of the voltage supplied to the motor. This in turn reduces the torque available to turn the rotor and lowers the motor speed. Amp draw is slightly affected during this process if the motors and blowers are sized properly. Some units may suffer from large changes in amp draw that significantly affect the motor's efficiency and operating characteristics. Reducing the voltage while keeping the amperage draw constant reduces the motor's power consumption.

Nameplate Ratings. The standard that covers fan-powered terminal unit nameplate ratings is ANSI/UL *Standard* 1995. This standard relates to equipment manufacturers and not field issues covered in international and local codes. Nameplate ratings on the unit usually do not match the nameplate ratings on the motor. Amp draw can be above or below the motor nameplate. Even voltage can vary. Differences between the motor label and the unit label may be significant in some cases, but these different ratings do not generally affect the performance or lifetime of the motor or unit. Be sure to refer to the unit nameplate ratings and not the motor nameplate ratings when sizing supply circuit requirements. These ratings are set at the safest possible condition. Because static and set points vary on each unit, performance may not be what is on the unit nameplate.

Electronically Controlled Motor (ECM) Technology. ECMs may provide significant energy savings and superior controllability. They can provide significant power savings on series units; however, they may not provide any savings on parallel units for reasons discussed in Davis et al. (2007).

Sizing Fan-Powered Terminals

Selection of fan-powered terminal units involves four elements: primary air valve, fan size, heating coil, and acoustics. How these elements are selected and their interactive effects determine the overall performance of the units.

Primary Air Valve. Identify the type of controller that is desired and select an inlet size that meets the minimum and maximum airflow desired from the recommended primary air airflow range table in the Performance Data section of the manufacturer's catalog. Selecting terminals near the top of their range may reduce cost, but increases velocity and noise. Selecting terminals toward the bottom may reduce noise, but may reduce controllability of the minimum airflow. Selecting the maximum airflow setting at between 70 and 85% of full capacity (approximately 10 m/s inlet velocity) is a good trade-off to avoid possible low-velocity control problems and sound problems at higher velocities.

Fan Size. Parallel fan size is determined by calculating the difference between the unit design heating airflow and minimum primary airflow. If minimum airflow is zero, then fan airflow is the heating airflow. In most cases, the fan can be downsized compared to a series terminal, reducing both first cost and operating cost because the fan only requires the capacity to handle the secondary airflow at reduced downstream static pressure compared to the maximum design airflow. In most applications of a parallel terminal, a minimum primary airflow is required to meet ventilation requirements. This primary airflow contributes to the total resistance experienced by the fan and should be accounted for along with all components downstream of the fan, such as heaters, ductwork, and diffusers. Hot-water coils may be positioned out of the primary airflow (i.e., on the inlet side of the fan, where they would not affect the primary airflow static pressure). In this configuration, heat generated by the water coil shortens the motor life.

Series fan terminal units require the fan to be sized to handle the maximum design airflow. The fan airflow must be at least equal to the primary airflow to ensure the mixing chamber in the terminal does not become pressurized, resulting in primary air spilling out into the ceiling plenum through the induction ports. The external static pressure requirements are the sum of the ductwork and diffusers downstream at design airflow plus an applicable hot-water coil or electric heater, if required. When fan airflow and downstream static pressure have been determined, select the fan size from the fan curves in the Performance Data section of the manufacturer's catalog. Selecting toward the upper end of the range reduces first cost and optimizes fan operating efficiency. Upsizing the fan and operating it at a reduced speed can result in quieter operation. When electric or hot-water coils are required, be sure to include the static pressure required for those items when referring to the fan curves.

Heating Coil. First, determine the heating supply air temperature to the space by calculation using the heat transfer equation:

$$q = 1.206 \times \text{m}^3/\text{s} \times \Delta t$$

where

q = design heat loss in space, kJ
Δt = supply air temperature (SAT) – room design temperature.

The supply air temperature (SAT) to the space equals the leaving air temperature (LAT) for the terminal unit.

For a series unit, once the terminal LAT is determined, the heating requirements for the coil can be calculated. The leaving air temperature for the coil varies based on the coil design and terminal unit model. It is generally a good idea to limit leaving air temperature to 7 K above room temperature. The LAT is the temperature leaving heating coil for a series unit, but requires calculations using a mixed-air equation for the primary and fan air volumes and temperatures for a parallel unit with the coil located out of the primary airstream. These leaving air temperatures can be effectively used to warm the room, because their airstreams are not so buoyant that they cannot be driven to the floor in overhead applications, and are warm enough to not produce chills from drafts.

Once both coil entering air temperature (EAT) and LAT are calculated, the heat transfer q for the coil must be calculated, using the heat transfer equation. The required kilowatts and number of steps desired should be checked with availability from the charts in the Performance Data section of the manufacturer's catalog. For hot-water coils, reference the capacity charts in the Performance Data to select the appropriate coil.

$$\text{EAT of coil} = (T_1Q_1 + T_2Q_2)/QT$$

where

T_1 = plenum air temperature, °C
T_2 = primary air temperature, °C
Q_1 = plenum air quantity, m/s
Q_2 = primary air quantity, m/s
QT = total air moved by terminal fan, m/s

These processes are for calculating the total heat requirement based on the room load calculations. At part-load conditions, it may be desirable to modulate the airflow through the terminal unit as well as the heat output to maintain an acceptable discharge air temperature. This can be done with modulating valves on coils or SCRs on electric heaters. Staging the electric heaters can create similar results at a lower equipment cost. Modulating the heat causes the heaters to run longer, but at lower energy consumption. This can make the room more comfortable without increasing energy costs.

Acoustics. Sound levels are affected by air-regulator-generated noise and fan-generated noise. The maximum noise generated by a given air regulator size is determined by the difference between the highest inlet static pressure and external static pressure at the design cooling airflow for a parallel unit. For a series unit, it is the highest inlet static pressure. This represents the most extreme operating condition. To determine fan noise levels, fan airflow (adjusted within its range by the speed controller) and external static pressure conditions are required.

The acoustical performance data are presented in two formats for the parallel and series types, because their sequence of operation differs. With a parallel unit, air regulator and fan operation are evaluated separately because their operations are not simultaneous under most conditions. With a series unit, air regulator and fan are evaluated together, because they operate simultaneously, and fan-only for heating with no minimum primary airflow.

From the performance data, determine the sound power levels and predicted room noise criteria for both discharge and radiated path under the appropriate operating conditions. Radiated noise from the unit casing typically dictates the noise level when the terminal unit is installed above the occupied space.

Care should be taken because some published room noise criteria are based on certain path attenuation assumptions that may not be indicative of a specific design. The size of the terminal may be increased to reduce noise, but it is also preferable to evaluate the room noise criteria to ensure the necessary reductions are achieved and finished levels do not exceed the design goal in the occupied space. To do this properly, the engineer must specify all factors in the building specifications that affect sound attenuation. An ideal specification specifies maximum allowed discharge and radiated sound power by octave band rather than just catalog-based NC values.

Example 3. Parallel Terminal with Hot-Water Heat. Select a unit inlet for a maximum/minimum primary airflow at 0.472/0.118 m^3/s with 249 Pa inlet static pressure.

The heating airflow required is 0.283 m^3/s. Downstream resistance at 0.472 m^3/s is 99.5 Pa. Zone design heat loss is 5.85 kW, design room temperature is 22.2°C, plenum air temperature is 23.9°C, and primary air temperature is 12.8°C.

Solution:

Air Valve Selection. Based on a good design inlet velocity of 10.6 m/s, choose a 250 mm inlet.

Fan Selection. Fan heating airflow = Heating airflow (0.283 m^3/s) – Primary airflow (0.118 m^3/s) = 0.165 m^3/s. The fan must overcome the downstream static pressure of the fan airflow plus primary airflow

(283 m^3/s), and because this is less than maximum design airflow (0.472 m^3/s), fan downstream static pressure = $(0.283/0.472)^2 \times 99.5$ = 38 Pa. Refer to fan curves to select the proper unit. The correct unit will handle 0.165 m^3/s at 38 Pa static pressure with correct setting of the speed controller, and allows for the selection of a one- or two-row hot water coil.

Heating Coil Selection. For heating, the temperature difference (Δt) is the zone supply air temperature (SAT) minus the design set point temperature. Using the heat transfer equation,

$$5.853 \text{ kW} = 1.206 \times 0.283 \times (\text{SAT} - 22.2)$$

$$\text{SAT} = 39.4°\text{C}$$

As the heating coil is on the unit discharge, the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of plenum and minimum primary air.

Design heating flow × Coil EAT =
(Primary airflow × Primary air temperature)
+ [(Design heating airflow – Primary airflow) × Plenum temperature]

$$0.283 \times \text{Coil EAT} = 0.118 \times 12.8 + (0.283 - 0.118) \times 23.9$$

$$\text{Coil EAT} = 19.4°\text{C}$$

For the heating coil, the temperature difference is the coil LAT minus the coil EAT.

$$\text{Coil heat pickup } q = 1.206 \text{ m}^3/\text{s} \times \text{Design airflow (m}^3/\text{s)} \times (\text{Coil LAT} - \text{Coil EAT})$$

$$\text{Coil } q = 1.026 \times 0.283 \times (39.4 - 19.4) = 22\,788 \text{ kW}$$

From the hot-water coil data, select a two-row coil at 0.283 m^3/s to provide 22 788 kWh at about 0.051 L/s (based on a Δt of 61.1 K between entering air and entering water).

Note 1: The coil selection in this example produces a discharge air temperature that is too high for normal applications. A discharge air temperature limit of 30.6°C should be used. If additional heat is required, airflow should be increased.

Note 2: The mixed-air condition does not bring the EAT to room temperature. Additional induction or plenum air should be added to increase the mixed-air temperature to near room temperature to avoid reheat.

Note 3: When using a PSC motor, the motor will add 0.6 to 1.7 K to the airstream.

Example 4. Series Terminal with Electric Heat. Select a unit to supply a constant 0.708 m^3/s with 124 Pa inlet static pressure. Minimum primary airflow is 0.177 m^3/s and downstream resistance caused by ductwork and diffusers is 100 Pa. Zone design heat loss is 13.17 kWh, design room temperature is 22.2°C, plenum air temperature is 23.9°C, and primary air temperature is 12.8°C.

Solution:

Air Valve Selection. Based on a good design inlet velocity of 10.16 m/s, choose a 300 mm inlet

Fan Selection. Fan airflow equals design airflow with a series unit. Fan external static pressure equals downstream static pressure (ductwork and diffusers). The resistance of electric and hot-water heating coils and their associated additional pressure drop may or may not be taken into account on the fan curves. Be sure it is included in the final static needs. From the fan curves, select a unit that will handle 0.708 m^3/s at 124 Pa and falls in the middle of the fan range as recommended in the section on Fan Size.

Heating Coil Selection. For heating, the temperature difference (Δt) is the zone supply air temperature (SAT) minus the design set point temperature.

$$13.17 \text{ kWh} = 1.206 \times 0.708 \times (\text{SAT} - 22.2)$$

$$\text{SAT} = 37.8°\text{C}$$

Because the heating coil is on the unit discharge, the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of plenum and minimum primary air.

Design heating flow × Coil EAT =
(Primary airflow × Primary air temperature) +
[(Design heating airflow – Primary airflow) × Plenum temperature]

$$0.708 \times \text{Coil EAT} = 0.177 \times 12.8 + (0.708 - 0.177) \times 23.9$$

$$\text{Coil EAT} = 21.1°\text{C}$$

For the heating coil, the temperature difference is the coil LAT minus the coil EAT.

$$\text{Coil heat loss } q = 1.206 \times \text{Design airflow (m}^3/\text{s)} \times (\text{Coil LAT} - \text{Coil EAT})$$

$$\text{Coil } q = 1.206 \times 0.708 \times (37.8 - 21.1) = 14.3 \text{ kWh}$$

From the manufacturer's catalog, select an electric heater with the proper input voltage that could be available with up to three stages with pneumatic or digital control or two stages with electronic control.

Note 1: Although there are air-side pressure drop data in the catalog, it is only necessary to calculate the drop if it is not included in the fan curves.

Note 2: The coil selection in this example produces a discharge air temperature that is too high for normal applications. A discharge air temperature limit of 30.6°C should be used. If additional heat is required, airflow should be increased.

Note 3: The mixed-air condition did not bring the EAT to room temperature. Additional induction or plenum air should be added to increase the mixed-air temperature to near room temperature to avoid reheat.

Note 4: Using a PSC motor adds 0.6 to 1.6 K to the airstream.

Installation and Application Precautions: Avoiding Common Errors and Problems

Sizing Terminals.

- Select terminals based on recommended air volume ranges. The pressure-independent terminal's main feature is its ability to accept factory-recommended minimum and maximum airflow limits that correspond to the designer's space load and ventilation requirements for a given zone. A common misconception is that oversizing a terminal makes the unit's operation quieter. In reality, the oversized terminal damper must operate in a pinched-down condition most of the time, which may actually increase noise levels to the space. Control accuracy may suffer because the terminal is only using a fraction of its total damper travel or stroke. In addition, the low inlet velocities may be insufficient to produce a readable signal for the velocity pressure measuring device and reset controller. This means minimum settings may not hold with a resultant loss of control accuracy and undesirable hunting.
- To maximize performance, size the terminal's maximum airflow limit for 70 to 85% of its rated capacity (approximately 10 m/s) in accordance with the catalog recommendations. For accurate control, the minimum setting guideline should not be lower than 2 m/s inlet neck velocity for units using inlet velocity sensors. Other minimum guidelines may apply for unit with specialty controls.
- Oversizing the discharge duct may create low static conditions, requiring the fan to operate outside its recommended operating range.
- A problem associated with oversizing terminals with electric heat is insufficient total pressure, which can occasionally trip the airflow safety switch.

Space Restrictions. During design, try to ensure that terminals are located for ease of installation, optimum performance, and maintenance accessibility.

Optimizing Inlet Conditions.

- The type of duct and its approach may have a large and adverse impact on both pressure drop and control accuracy. Although multipoint velocity pressure measuring devices can compensate to a large degree, good design practice should always prevail. Wherever possible, a straight duct inlet connection with a minimum length of three duct diameters and the same diameter as the inlet should be provided.
- Terminal collars are undersized to suit nominal ductwork dimensions. The inlet duct slips over the terminal inlet collar and is fastened and sealed in accordance with job specifications. Never insert a duct inside the inlet collar, or control calibration will be adversely affected.
- Sometimes space restrictions make it impossible to provide an ideal inlet condition. In this case, field adjustment of the airflow settings may be required to compensate for error in the flow measurement. The use of flow-straightening devices (equalizing grids) is recommended after short-radius elbows that are immediately ahead of the terminal and where terminals are unavoidably tapped directly off the main duct. Use of these devices typically increases sound levels.
- The balancing contractor should validate flow rates. See ASHRAE *Standard* 111.

Zoning Requirements. Correctly sizing terminals with regard to the physical conditions of the occupied space is vital to ensure acceptable performance. One large terminal serving a space with divided work areas may result in the single thermostat only providing acceptable temperature control for the area where the thermostat is located. The other area(s) served may be too cold or too hot if they have differing space load requirements.

Optimizing Discharge Conditions. Poor discharge duct connections may have an adverse affect on pressure drop. Try to avoid installing tees, transitions, and elbows close to the unit discharge. Avoid long runs of flex, and keep short flex runs as straight as possible. Make curves as shallow as possible, and ensure that the entrance condition to diffuser outlet is straight. Discharge ducts should be designed for a maximum velocity of 5 m/s.

Noncompliance with Local Electrical Codes. Some local jurisdictions have more exacting codes than the minimum requirements of national codes and standards such as the International Code Council's (ICC) *International Building Code*® (IBC). One example is the primary fusing required of the power circuit in some areas.

Power Source Compatibility. Terminals with an electrical power supply, such as fan-powered terminals and single-duct terminals with electric heat, should be checked for compatibility with source. Voltage, phase and frequency must match. Where motor voltage differs, the single-phase voltage requirement may have to be tapped from a three-phase (four-wire wye) power source.

Avoiding Excessive Air Temperature Rise. Terminals with electric or hot-water reheat coils should be designed to satisfy load conditions, but attention should be paid to the temperature differential (Δt) between the entering room air and ambient temperature. Chapter 19 of the 2013 *ASHRAE Handbook—Fundamentals* recommends a maximum Δt of 8 K to avoid possible stratification when heating from overhead caused by the excessive buoyancy of the warm air. This ensures good room mixing and temperature equalization. Exceeding a Δt of 8 K requires an increase of 25% in the ventilation air per ASHRAE *Standard* 62.1. Absolute maximum discharge air temperature is 49 K. Although this temperature will probably keep the equipment on line, it will not provide comfortable temperatures in the space.

Correctly Supporting Terminals. Although the basic single-duct terminal is light enough that it usually can be supported by the ductwork in which it is installed, these units should be independently supported. When accessory modules such as heating coils, attenuators, or multiple-outlet plenums are included, the assembly must be supported independently. Larger terminals such as fan-powered terminal units should always be independently supported, secured to building structure, and may require isolation mounting. Be careful not to block access panels with straps, thread rods, or trapeze supports. Be sure to comply with all building and local codes regarding seismic restraints (see Chapter 55).

Minimizing Duct Leakage. To prevent excess air leakage and minimize energy waste, all joints should be sealed with a UL-approved duct sealer. Most leakage can be avoided by practicing good fabrication and installation techniques, particularly upstream of the terminal, which may be required to hold significantly higher pressures than downstream of the terminal.

Acoustic Design and Installation. To help ensure an acceptable room noise level in the occupied space, engineers can minimize the sound contribution of air terminals by taking into account several design considerations and by using the following guidelines.

- Design systems to operate at low (minimum) supply static pressure at the primary air inlet. This reduces the generated sound level, provides more energy-efficient operation, and allows the central fan to be downsized. Excessive static pressure generates noise.
- Use of metal ducts before the inlet can reduce breakout noise from the damper. Between the terminal unit and the air outlet, flexible duct can be more effective than lined duct at reducing terminal unit noise. Flexible duct can also generate sound if bends or sagging is present. Sometimes, flexible couplers can reduce vibration passed from the terminal unit to the duct connections.
- Select terminals to operate toward the middle area of their operating range. Larger inlets reduce velocity and hence noise in low-pressure applications, but may increase noise in higher-pressure applications. For fan-powered terminals, lower fan speeds generally produce lower sound levels. Sound emissions are lower when fan-speed controllers are used to reduce fan rotational speed rather than using mechanical dampers to restrict airflow.
- Whenever possible, locate terminals above noncritical areas that are less sensitive to noise, such as corridors, copy rooms, or storage/file rooms. This isolates critical areas from potential radiated noise.
- Locate terminals in the largest ceiling plenum space available to maximize radiated noise reduction. Install terminals at the highest practical point above ceiling to optimize radiated sound dissipation.
- Avoid locating terminals near return air openings or light fixtures. This decreases the potential for direct paths for radiated sound to enter the space without the benefit of ceiling attenuation.
- Locate terminals to allow use of lined discharge ductwork to help attenuate discharge sound.
- To avoid possible aerodynamic noise, keep airflow velocities below 5 m/s in branch ducts and below 4 m/s in runouts to air outlet devices.
- In large spaces, consider using a larger number of smaller air outlets to minimize outlet-generated sound.
- Insulated flexible duct on diffuser runouts reduces room noise levels
- Using ceilings with a high sound transmission loss classification helps reduce radiated sound.

See Chapter 8 in the 2013 *ASHRAE Handbook—Fundamentals* and Chapter 48 in this volume for more information on sound and vibration control.

8. CHILLED BEAMS

An **active chilled beam** is an air diffusion device that introduces conditioned air to the space for the purposes of temperature and latent control. Primary air (i.e., air delivered to meet ventilation, latent, and all or part of the sensible load) is delivered through a series of nozzles, creating induction of room air through a unit-mounted chilled-water coil, which conditions air before reintroducing it to the space. Depending on the nozzle size and configuration, active beams typically induce two to five parts of room air for every part of primary air they deliver to the space. Sensible heat removal by the beam's integral cooling coil complements the cooling effect of the primary air supply.

Passive chilled beams rely on the natural buoyancy of air currents associated with convective heat sources to transport warm air to the upper portion of the space. Upon contact with the beam's integral heat transfer coil, this air is cooled and falls back into the space. Primary air is delivered to the space for the purposes of ventilation and latent control via a separate system. Air circulated through the chilled-beam coil is called secondary air; supply air is comprised of mixed primary and secondary air.

Codes and Standards

Chilled-beam system designs should conform to the following building codes and standards, as well as any applicable local code requirements:

- ISO *Standard* 7730 and ASHRAE *Standard* 55
- ASHRAE *Standard* 62.1
- ASHRAE *Standard* 90.1
- IBC
- NFPA *Standard* 90A and 90B
- ANSI *Standard* S12.65

Application Considerations

Chilled-beam systems must be designed to treat sensible and latent space heat gains, provide adequate space ventilation, and maintain occupant comfort in conformance with ASHRAE *Standard* 55 and other applicable codes.

In general, chilled beams offer the opportunity to capitalize on the benefits of decoupled ventilation systems. They also offer the designer the opportunity to manage sensible loads in the space separately from ventilation and latent needs. Chilled beams work well with dedicated outdoor air and demand control ventilation systems.

Benefits. Heat extraction or addition by the coil allows for significant reduction in primary airflow requirements over all-air ducted systems. Energy to transport cooling and/or heating media is reduced because of water's high specific heat and density. As a result, chilled-beam systems require less space for the mechanical services, because of smaller ductwork and air-handling unit sizes. This reduction in mechanical service space requirements means it is possible to reduce the floor-to-floor height of a multistory building.

Effectively, a 25 mm diameter water pipe delivering 14°C water can transport as much energy as a 355 × 355 mm duct delivering air at 13°C. The transport energy required to deliver cooling by water is only 15 to 20% of that which would be required by air. Water provides 3500 times the heat-carrying capability of air.

Free cooling opportunities may be extended as a result of the lower secondary chilled-water temperatures, and provide an improved selection of available system options (e.g., geothermal, dry coolers, closed-circuit fluid coolers). Chilled-beam systems offer opportunities to enhance chiller efficiencies and provide broader evaporator ranges, because of higher chilled-water temperatures, and cascaded evaporator flows between primary and secondary chilled-water loops. Higher inlet water temperatures also make chilled-beam systems as an excellent choice for geothermal applications, where suitable electrical costs allow this as an option.

Chilled-beam systems are typically operated with a constant-(minimal) volume supply airflow to the space. Constant-volume systems offer enhanced thermal comfort because of their consistent room air movement, and an improved acoustic environment.

Maintenance of chilled beams is virtually nonexistent. Vacuuming the coils is occasionally required, and is typically guided by the needs of the space. Often, it is expected that service intervals could extend to 3 to 5 years. The lack of moving parts in chilled beams inherently produces a highly reliable system, and because most beams do not contain filters, servicing costs are minimal.

Passive chilled beams can provide a very efficient means of perimeter-area temperature control when coupled with underfloor air distribution (UFAD) systems. For more information on UFAD systems, see Chapter 20 of the 2013 *ASHRAE Handbook—Fundamentals*.

Limitations. Space humidity levels must be managed closely because of the limited dehumidification capability of the primary air. This is particularly important where operable windows are used or high infiltration levels are encountered in a humid climate.

Chilled beams may be considered beneficial for the following applications/spaces:

- Environments with moderate to high sensible heat ratios, such as offices, apartments, high rise condos, or classrooms
- Laboratories or other spaces with significant imbalances between sensible loads and ventilation requirements
- Retrofits, because of minimal mechanical space requirements, and in cases of suitable envelope construction

Chilled beams may be inappropriate for the following spaces:

- Environments with high latent gains, such as kitchens, bathrooms, or locker rooms
- Swimming pools, natatoriums, and sauna areas
- Poor building envelopes or buildings with unmanageable latent loads
- Mixed-mode ventilation (operable windows) without proper condensation safeguards

Chilled beams must be independently supported from ceiling grid systems, and subsequently positioned into the grid. Appropriate care must be used when beams are installed in seismic zones, to ensure compliance to all building and safety codes.

Passive beams are restricted to cooling-only applications, and require a separate heating system. Careful consideration is needed in the placement and control of passive chilled beams to ensure thermal comfort.

Cooling

The objective of chilled-beam design is to minimize primary airflow rates, ideally reducing them to the space minimum ventilation requirement. However, where zone cooling requirements cannot be achieved with the minimum primary airflow rate, chilled beams may be used with air-handling units that mix return and outdoor air volumes.

Active and passive chilled beams rely on the primary air supply for dew-point control. As such, the design conditions must be evaluated to confirm that the primary air is appropriately treated to manage the latent space loads. Chilled beams are intended to operate without condensation. Consequently, active chilled-beam supply water temperatures should be maintained at or above the room dew-point temperature to prevent condensation on the coil and its supply water piping. Passive chilled-beam water supply temperatures should be maintained slightly (0.6 to 1.1 K) above the room dew-point temperature. In both cases, the chilled-water supply piping must be adequately insulated to prevent condensation on the pipe itself. Where adequate control of space humidity levels cannot be ensured, higher supply water temperatures and/or condensation controls should be considered. This is discussed in the following sections.

Terminal filtration and condensate pans are not required with a properly designed primary air system and chilled-water temperatures maintained above the room dew point. Heating coils provide sensible heat only, and thus filtration and condensate capture devices are not necessary. Chilled-beam systems designed with noncondensing (dry) coils should be treated similarly.

Heating

Heating is limited to active chilled-beam systems; heating with overhead passive chilled beams is not effective.

The hot water serving the active beam's coil must be chosen to limit the discharge air temperature to less than 8 K above the room design set point. Additionally, to ensure proper room air distribution, the discharge velocity should be selected in accordance with guidance presented previously.

Resetting the primary air temperature with a duct-mounted booster coil allows the primary air serving the interior spaces to continue to provide cooling, while the perimeter duct adds heating capability through this reset. Assuming the active chilled beams are the primary heating system, caution is recommended in placing the equipment at ceiling level with respect to the curtain wall's orientation, to ensure air movement across these surfaces promotes a comfortable environment.

Thermal Comfort

Chilled-beams systems are designed to optimize delivery of cooling to the space, but the paramount consideration in sizing and locating beams in the room should focus on occupant thermal comfort. ASHRAE *Standard* 55 defines limits on local air temperatures and velocities that maintain acceptable levels of occupant thermal comfort. The standard defines the **occupied zone** as the one in which stationary occupants reside; the height of this zone is generally considered 1.7 m for standing occupants or 1.1 m for predominantly seated occupants. Velocities in the occupied zone should not exceed 0.25 m/s, and occupied-zone vertical temperature gradients should be maintained at 3 K or less.

Properly applied passive chilled beams have a limited effect on occupant thermal comfort; however, their complementary primary air supply system often does. Stratified or partially mixed air diffusion strategies are commonly used with passive beams because of their minimal influence on the natural buoyancy driven air patterns associated with the chilled-beam operation. The secondary air circulation through the passive beam transports upper-level air back to the occupied zone, possibly altering the level of stratification in the space.

Active chilled beams directly supply a mixture of primary and secondary air to the space and should therefore be treated like the other air distribution devices used in fully mixed air distribution systems. Because the temperature of the chilled water supplying the coil must be at (or above) the space dew-point temperature, it is typically 13 to 16°C; thus, reconditioned air leaving the coil is several degrees warmer than the primary air with which it is subsequently mixed. This results in beam design discharge air temperatures ranging from 14 to 16°C, warmer than those normally used by conventional all-air systems. Because the required supply airflow is inversely proportional to the room to supply air differential, active beams must discharge 15 to 25% more air to the space to satisfy its sensible heat gains.

Space Temperature Control and Zoning

Chilled-beam systems' primary airflow rates are much closer to the space ventilation rates than those of all-air systems, so primary control of the space temperature is normally accomplished by throttling the chilled-water flow. Simple on/off operation of two-position water valves provides adequate control of active chilled beams. Proportional valves are recommended for passive beams, and for active beams in applications where more precise space temperature control is required.

In applications where primary air is supplied at conventional temperatures (13 to 16°C) to spaces with significant sensible load variations, it may also be necessary to reset the primary airflow rate or temperature during low-load conditions. One approach is to vary the primary airflow rate in reaction to thermal demands and/or occupancy of the space.

Supply of primary air at or close to room temperature overcomes the potential for overcooling. This, however, results in reduced beam cooling capacities and necessitates the use of more or larger beams. It is also likely to require desiccants to provide adequate dehumidification of the primary air. Spaces with high ventilation requirements and significant cooling turndown rates, such as large conference rooms, may be better served by a VAV solution.

Thermal zoning of chilled-beam systems should be performed in a manner generally consistent with other HVAC systems. Each thermal zone consists of a space thermostat, a chilled- (and, where applicable, hot-) water control valve, and multiple chilled beams.

Selection and Location

Chilled beams may be exposed or integrated with an acoustical ceiling system. Active chilled beams may be of either open or closed design. Closed beams induce secondary air from below, whereas open beams induce through their top or sides. When passive or open active beams are applied, an adequate air path must be provided for secondary air to enter the beam.

For sizing and selection purposes, secondary air entering a chilled beam should generally be considered equal to that maintained within the occupied zone unless solid evidence indicates otherwise.

Most active beams suppliers offer various nozzle sizes and configurations. Nozzle configuration affects the beam's pressure requirement and acoustical performance as well as its induction function. Active beams with adjustable discharge or nozzle patterns may also allow for field alteration of the beam's air distribution characteristics. This may also affect the beam's cooling capacity, so changes should be made with caution.

Beam sizing and location must consider cooling capacity, acoustics, thermal comfort, and integration with other equipment and services. Active beams use a horizontal discharge of their supply air mixture through linear openings along their perimeter, and thus display room air diffusion characteristics similar to those of linear slot diffusers. As such, active beams should be selected and located such that velocities within the occupied zone are limited to 0.25 m/s or less if compliance with ASHRAE *Standard* 55 is the design intent. The use of mapping techniques (see the section on Mixed Air Distribution) and/or selection of active beam throw values within the ranges of Table 4 may be used to estimate compliance with these comfort recommendations.

Locating stationary occupants directly below passive beams can result in thermal discomfort. Care must be taken to ensure that the velocity and temperature of the descending airstream entering the occupied zone comply with the thermal comfort requirements of ASHRAE *Standard* 55.

Operational Considerations

Water supply service to active and passive beams should not be activated until space dew-point temperatures are at or below the chilled water's supply temperature.

Where maintenance of adequate space dew-point temperatures cannot be ensured, some type of condensation detection and mitigation strategy should be used. Designers have various methods of accomplishing this, including the following:

- Sensors, attached to the supply water pipe, that sense formation of surface moisture and discontinue chilled-water flow until the moisture has evaporated. This method is relatively inexpensive but also reactive, and results in termination of secondary air cooling while condensation exists.
- Dew-point calculation and reset of the chilled-water supply temperature is a proactive strategy that does not involve full suspension of secondary cooling. This method can be applied on a room-by-room basis, but calculation on a floor-by-floor basis is usually sufficient and less costly.
- For applications in spaces with operable windows or doors, occupants and staff should be educated on the effect this has on their thermal environment.
- In certain applications, condensate trays may be used to collect temporary and infrequent condensation. Where applied, adequate condensate removal means must be ensured; evaporation of the condensate should not be assumed. This may involve using condensate pumps when gravity drainage cannot be accomplished.

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