

CHAPTER 58. 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 2021 ASHRAE Handbook—Fundamentals](#) for details. Also see [Chapter 20 of the 2021 ASHRAE Handbook—Fundamentals](#) for more information on space air diffusion; [Chapter 20 of the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#) for information on room air distribution equipment; and [Chapter 49](#) 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** of a space is any location where occupants normally reside, and may differ from project to project; it is application-specific, and should be carefully defined by the designer. The occupied zone is generally considered to be the room volume between the floor level and 6 ft above the floor. Standards and guidelines, such as 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 ASHRAE *Standard* 55-2020. Figure 5.3.1 of the standard shows acceptable ranges of temperature and humidity for spaces. As a general guide, a majority of occupants in typical office spaces can be satisfied with thermal environments over a wide range of temperatures and relative humidities. Designers often target indoor dry-bulb temperatures between 73 and 77°F, relative humidities between 25 and 60%, and occupied zone air velocities below 50 ft/min.

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:

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

Because task/ambient design requires a high degree of individual control, it is not covered in this chapter; see [Chapter 20 of the 2021 ASHRAE Handbook—Fundamentals](#) for details. Guidance is also provided by ASHRAE (2013).

[Figure 1](#) shows the spectrum between the two extremes (full mixing and full stratification) of room air distribution strategies.

1. APPLICATION GUIDELINES

Design Considerations

Architectural and Spatial Constraints. Air distribution products must fulfill both the functional requirement of conditioning space and the visual aesthetic determined by the architect. Architectural constraints may limit placement of air outlets and ductwork.

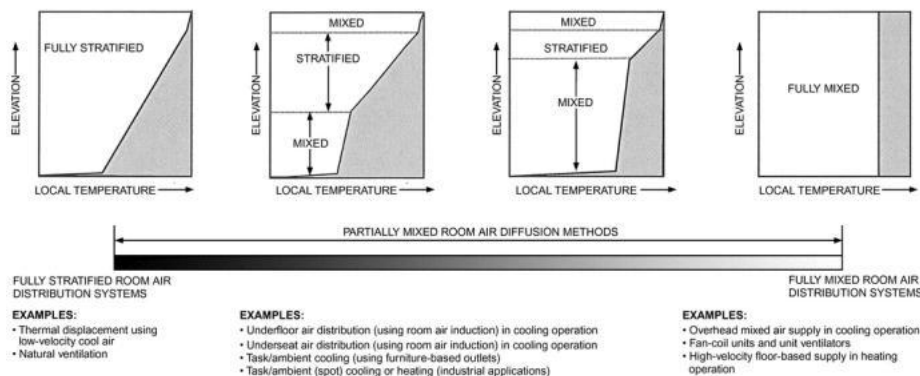


Figure 1. Classification of Air Distribution Strategies

Heat Gain and Loss Characteristics. Large sensible heat loads can drive air movement in a space. Warm air is more buoyant and rises; cooler air is denser and descends to the ground. A zone may experience different heat loads depending on the season or time of day. The air distribution system must meet the varying heating and cooling needs throughout the building's operation. Care should be taken to ensure cool surfaces, such as exterior windows, do not reach temperatures below dew point, or condensation will occur.

Thermal Comfort. Occupant thermal comfort depends on several variables, including air velocity, air temperature, thermal radiation, humidity, occupant metabolic rate, and occupant clothing. Air distribution systems that use higher air velocities and temperature differentials may create a greater risk of draft. Likewise, exterior windows with warm or cold surfaces can produce undesired thermal radiation to nearby occupants. For more information, see ASHRAE *Standard* 55-2020 and its user's manual (ASHRAE 2016).

Acoustical Requirements. 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 49](#) of this volume and [Chapter 8 in the 2021 ASHRAE Handbook—Fundamentals](#).

Available Locations for Air Inlets and Outlets. Inlet and outlet characteristics are discussed in [Chapter 20 of the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#). This chapter discusses more specific application considerations for air inlets and outlets.

Code Requirements. Some applications (e.g., operating rooms) may require compliance with special or local codes that limit the selection and placement of certain types of air outlets.

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 the location of air outlets except in applications where contaminant control and removal are a primary concern. In fact, the return air intake affects room air motion only in the area 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 (to avoid excessive noise) return inlet face 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, it can be advantageous to place the return air inlet in the ceiling. ASHRAE *Standard* 62.1 allows ventilation effectiveness greater than 1.0 for some stratified and partially mixed air distribution systems in cooling mode if the return air inlet is located in the ceiling.

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

Indoor Air Quality, Sustainability, and Airborne Contaminants

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. ASHRAE *Standard* 90.1 provides energy efficiency requirements that affect supply air characteristics.

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 (new.usgbc.org/leed). 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, ASHRAE *Standard* 62.1 requires that more than the base non-ETS ventilation air be provided where ETS is present in all or part of a building. Rock (2006) provides additional advice on dealing with ETS.

Table 1 Recommended Return Inlet Face Velocities

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

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 ASHRAE (2013).

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 construction materials.

The number and location of supply diffusers and return grilles and their operation can have a major effect on room airflow patterns and the distribution of airborne contaminants. As such, the following recommendations apply to all room air distribution systems (Khankari et al. 2020):

- Ensure that all supply diffusers and return grilles are performing as per the design intent and are consistent with contaminant control intent
- Ensure that all supply and return devices are unobstructed by furniture, architectural features, or other obstacles.
- Review the sequence of operation of the HVAC systems to ensure that the sequence matches design intent and is also consistent with contaminant control intent.
- Review room air set points for temperature, humidity, and pressure and the relevant sensor locations, and if necessary, make adjustments or replace malfunctioning devices so that the space can be operated as intended.
- Create a path of least resistance for the contaminated air to exit the space by increasing the size and number of return grilles at appropriate locations.
- Promote sweeping airflow patterns that move the contaminated air away from the occupants by placing the return grilles away from the occupants.
- Relocate supply outlets to eliminate air jet collisions that result in drafts (high velocities above the ASHRAE *Standard* 55-2017 recommendation of 40 fpm) in the occupied zone. If relocation is not possible, replace them with high induction diffusers with shorter throws.
- Ensure that air motion is provided to all occupied areas of the space to avoid any stagnant zones with low air motion.
- Return grilles should be located outside the path of supply air jets to avoid any short-circuiting of clean supply air.
- In the case of fully stratified systems, since there is no recirculation of air from the upper part of the space back to the occupied zone, the use of upper-room UVGI or other upper-level cleaning devices may be redundant in such cases.
- In the case of fully stratified systems, a fan-assisted portable filter unit must be used with caution because the air discharged by such devices can disturb the thermal stratification in the space.
- Follow source control recommendations such as mask use, social distancing, and install lids on all toilets and instruct occupants to close them before flushing to reduce aerosol escape.
- Each space is unique in its layout, airflow distribution, and in the resulting airflow patterns. Therefore, seek professional guidance before implementing any substantial retrofit of the airflow distribution system.

2. FULLY 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 average room air velocities of less than 50 ft/min are maintained in the occupied zone.

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

Principles of Operation

Mixed systems generally provide comfort by entraining room air into discharge jets located outside occupied zones, mixing supply and room air. Ideally, these systems generate low-velocity air motion (less than 50 ft/min) 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, possibly leading to occupant discomfort or poor air quality.

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 30% 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 2021 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, 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 air distribution systems, because designers and installers are familiar with the required system components and installation.

Limitations of mixed systems include the following:

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

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

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.

Table 2 Forward Throw Asymmetry

| Diffuser Type | Vertical Inlet Duct Height | | | | | |
|---------------|----------------------------|---------|--------------------|---------|------------------|---------|
| | 0 Duct Diameter | | 1.5 Duct Diameters | | 3 Duct Diameters | |
| | No Damper | Damper | No Damper | Damper | No Damper | Damper |
| Square | 1.4 | 1.2-1.4 | 1.1 | 1.0-1.1 | 1.0 | 1.2 |
| Round | 1.4 | 1.2-1.4 | 1.1 | 1.0-1.1 | 1.0 | 1.2 |
| Plaque | 1.2-1.3 | 1.1-1.3 | 1.1 | 1.0-1.1 | 1.0 | 1.1 |
| Perforated | 1.3 | 1.2-1.3 | 1.1 | 1.0-1.1 | 1.0 | 1.1-1.2 |
| Modular core | 1.4 | 1.2-1.4 | 1.1 | 1.0-1.1 | 1.0 | 1.0 |
| Louvered | 1.4 | 1.2-1.4 | 1.1 | 1.0-1.1 | 1.0 | 1.0 |

Source: data from Landsberger et al. (2011).

Note: Multipliers valid for round hard duct and flex duct.

Multipliers represent proportional increase of throw distance in forward direction of ducted air motion.

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.

ASHRAE research project RP-1335 (Landsberger et al. 2011) determined that a wide-open damper installed in the neck of a diffuser could add up to 8 NC to the cataloged NC value, depending on diffuser and damper types. Significantly closed balancing dampers can add more than 10 NC, depending on duct pressure and how far upstream it is installed. Table 2 gives forward throw asymmetries for various diffuser types, and Figure 2 compares the total pressure ratio of a diffuser with no damper versus a neck-mounted damper that is increasingly throttled.

Effects of Typical Field Installations on Common Ceiling Diffusers.

Ceiling air outlets are tested using ideal installation conditions described in ASHRAE *Standard 70*: a minimum of three straight equivalent duct diameters before the diffuser inlet.

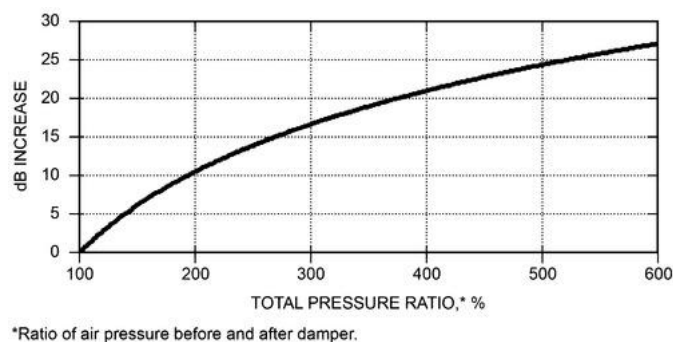


Figure 2. Effects of Neck-Mounted Damper on Air Outlet

Field installations are often not ideal because of ceiling plenum limitations, architectural design, structural supports of the building, etc. ASHRAE research project RP-1335 (Landsberger et al. 2011) developed some useful application data that can be used by design engineers to predict throw, pressure drop, and sound caused by non-ideal installations. Tables 2, 3, and 4 give a general summary of application data from RP-1335. The multipliers provided in these tables may be applied to diffuser performance data measured from tests compliant with ASHRAE *Standard 70-2006*.

Table 3 Total Pressure Increase

| Diffuser Type | Vertical Inlet Duct Height | | | | | |
|---------------|----------------------------|---------------------|------------------------|---------------------|------------------------|---------------------|
| | 0 Duct Diameter | | 1.5 Duct Diameters | | 3 Duct Diameters | |
| | No Damper ^a | Damper ^b | No Damper ^a | Damper ^b | No Damper ^a | Damper ^b |
| Square | 1.4-1.7 | 1.7-2.2 | 1.3-1.5 | 1.7-2.1 | 1.1-1.2 | 1.5-1.8 |
| Round | 1.4-1.6 | 1.6-2.0 | 1.2-1.4 | 1.5-1.9 | 1.1-1.2 | 1.2-1.7 |
| Plaque | 1.4-1.7 | 1.7-2.2 | 1.3-1.5 | 1.7-2.1 | 1.1-1.2 | 1.5-1.8 |
| Perforated | 1.4-1.6 | 1.6-2.0 | 1.2-1.4 | 1.5-1.9 | 1.1-1.2 | 1.2-1.7 |
| Modular core | 1.2-1.4 | 1.4-1.6 | 1.1-1.2 | 1.3-1.5 | 1.1 | 1.3-1.4 |
| Louvered | 1.5-1.8 | 1.8-2.4 | 1.3-1.6 | 1.8-2.3 | 1.2-1.3 | 1.6-2.0 |

Source: data from Landsberger et al. 2011.

^a Multipliers based on round hard duct; for flex duct, add 0.2-0.4.

^b Multipliers based on round sliding damper; if round OBD, subtract 0.3.

Table 4 NC Increase

| Diffuser Type | Vertical Inlet Duct Height | | | | | |
|---------------|----------------------------|---------------------|------------------------|---------------------|------------------------|---------------------|
| | 0 Duct Diameter | | 1.5 Duct Diameters | | 3 Duct Diameters | |
| | No Damper ^a | Damper ^b | No Damper ^a | Damper ^b | No Damper ^a | Damper ^b |
| Square | 7 | 8 | 3 | 10 | 0 | 7 |
| Round | 9 | 10 | 4 | 12 | 0 | 8 |
| Plaque | 7 | 8 | 3 | 10 | 0 | 7 |
| Perforated | 5 | 5 | 2 | 7 | 0 | 5 |
| Modular core | 3 | 3 | 1 | 3 | 0 | 2 |
| Louvered | 8 | 8 | 3 | 9 | 0 | 6 |

Source: data from Landsberger et al. 2011.

^a NC based on round hard duct; for flex duct, add 1 NC.

^b NC based on round sliding damper; if round OBD, −2 NC (except for 0 duct diameter).

Space Temperature Gradients and Airflow Rates

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

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

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

(1)

where

Q = required supply airflow rate to meet sensible load, cfm

q_s = net sensible heat gain in the space, Btu/h

t_r = return or exhaust air temperature, °F

t_s = supply air temperature, °F

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

Methods for Evaluation

The objective of air diffusion is to create the proper combination of room air temperature, humidity, and air motion to provide thermal comfort and acceptable indoor environmental quality in the occupied zone. 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 can also 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

Design Procedures

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. Consider both cooling and heating mode requirements.
2. Determine acceptable outlet noise criterion (NC); consult [Chapter 49](#) of this volume, or [Chapter 8 in the 2021 ASHRAE Handbook—Fundamentals](#).
3. Locate a range of products from manufacturers’ catalogs that meet the airflow and NC requirements. Multiple outlets in a space at the same cataloged NC, and other design considerations, may result in actual sound levels greater than cataloged values. Manufacturers’ data are obtained using ideal inlet conditions, and may vary from field installations. From experience,
 - For identical outlets 10 ft or more apart, the cataloged NC rating applies.
 - Identical outlets within 10 ft of each other add no more than 3 dB to the sound pressure level.
 - For continuous linear outlets, only the sound produced by the closest 10 ft need be considered.
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.

Selection by Isovels and Mapping. Using manufacturers’ catalog throw data, a designer can predict the path of an outlet’s discharge jet. Most manufacturers’ catalogs list the distance a jet travels to reach a terminal velocity of 150 to 50 ft/min. With this information, the designer can map the path of the discharge jet for a given outlet. This evaluation can prevent problems such as excessively high air velocities in the occupied zone, or stagnation in a given area. Note that most manufacturers’ throw data are based on isothermal supply air; the supply jet temperature is equal to the room air temperature. When using this mapping method, consider the positive or negative buoyancy of nonisothermal (heated

or cooled) supply air. In both heating and cooling, a discharge jet should travel the distance shown in the catalog to a terminal velocity of 150 ft/min without much influence from buoyancy. When evaluating a jet at lower terminal velocities (e.g., 100 or 50 ft/min), consider buoyancy's effect on the distance the jet will travel.

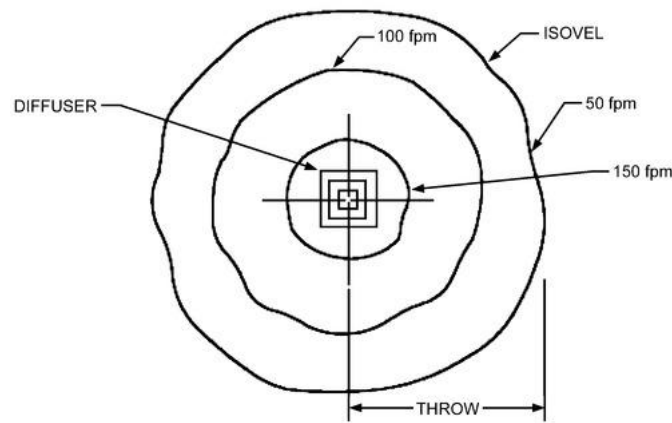


Figure 3. Throw Isovels at Different Terminal Velocities (Adapted from ASHRAE Standard 70-2006)

Horizontal Throw. A cooled confined air jet projecting along a horizontal surface travels a shorter distance than the equivalent volumetric isothermal air jet traveling the same path. If an outlet is selected so that the cooled 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.

A heated confined air jet projecting along a horizontal surface travels farther than the equivalent volumetric isothermal jet traveling the same path. If an outlet is selected so that the heated, horizontal jet does not have enough velocity to reach a vertical surface, the jet can pool at the ceiling level, causing stagnation (no mixing) within the occupied space and resulting in high levels of contaminants, thermal stratification, and discomfort.

In a free jet scenario, expect similar results as for a confined jet, but if isothermal performance is based on confined jet installation, a 30% reduction in throw length from isothermal performance will occur, along with buoyancy effects.

Vertical Throw. A cooled air jet projecting vertically upward travels a shorter distance than the equivalent volumetric isothermal air jet traveling the same path. A heated air jet projecting upward travels farther than the equivalent volumetric isothermal air jet traveling the same path. For both cooled and heated air jets, the opposite is true for downward projection.

Mapping with acoustical selection. Combining selection by isovels 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, as defined by ASHRAE Standard 55-2020.
2. Select outlet(s) that meet design NC, pressure drop, and flow rate requirements. Identify the supply jet location using cataloged throw data.
3. Evaluate air jet mapping to ensure terminal velocities in the occupied zone do not exceed 50 ft/min.
4. For overhead heating applications, $\Delta t \times 15^\circ\text{F}$ (see [Chapter 20 of the 2021 ASHRAE Handbook—Fundamentals](#)), evaluate the diagram to ensure that jet velocities 4.5 ft from the floor are at least 150 ft/min.

Other design considerations include the following:

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

Example 1. For a 20 by 40 ft zone, with a 9 ft ceiling, with uniform loading of 10 Btu/h·ft² or 2400 Btu/h, and air volumetric flow of 1 cfm/ft² or 800 cfm for a dual-outlet configuration, use isovel mapping to select an appropriate diffuser option that meets a sound criteria of NC 35.

Solution:

1. The occupied zone is defined as a height of 6 ft off the floor, 1 ft from interior walls, and 3.3 ft from exterior walls.
See ASHRAE Standard 55 for the most current definition of occupied zone.
2. One manufacturer suggests that a single 2 × 2 ft plaque diffuser with an 8 in. inlet at 400 cfm yields an NC 30 and an isothermal throw of 11 ft at a terminal velocity of 50 ft/min.
3. With an air jet characteristic length of 10 ft, and 3 ft between ceiling and start of the occupied zone, the velocity of the colliding jets can be assumed to not exceed 50 ft/min in the occupied zone. Verify that resulting diffuser pressure drop and aesthetics meet requirements.
4. If one or more of the walls are exterior in this example, even with additional throw length due to warmer supply air, additional air volume may be required to meet 4.5 ft above the floor at 150 ft/min.

Evaluate other diffuser types for similar or better performance criteria.

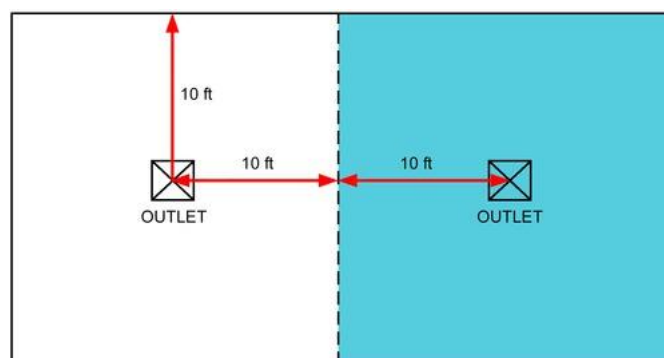


Figure 4. Schematic for Example 1

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

The comfort criteria T_{50}/L method was developed to predict occupant comfort using manufacturers' isothermal catalog throw data (T , usually for 50 ft/min terminal velocity) and the dimensions available for throw L on the plan view of a mechanical drawing. By using the ratio of T_{50}/L , the designer can predict the level of comfort with a single rating number, ADPI, which 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 in heating and cooling for a space conditioned by a mixed air system. ADPI uses the effective draft temperature collected at an array of points taken within the occupied zone to predict comfort. ADPI is the percentage of points in a space where the effective draft temperature is between -3 and $+2^{\circ}\text{F}$ for cooling and -4 and $+3.6^{\circ}\text{F}$ for heating. The acceptable air velocity is less than 70 ft/min for heating and cooling. In addition, the acceptable vertical temperature gradient should be less than $1.6^{\circ}\text{F}/\text{ft}$. High ADPI values generally correlate to high space thermal comfort levels with the maximum obtainable value of 100. Selecting outlets to provide a minimum ADPI value of 80 generally results in a well-mixed space.

Table 5 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 |
| Adjustable blade | |
| Fixed blade | |
| Linear bar | |
| Nozzle | |
| Horizontal-throw ceiling diffuser | Distance to closest wall, midplane between outlets or intersecting air jet |
| Round | |
| Square | |
| Perforated | |
| Louvered | |
| Plaque | |
| Swirl | |
| Sill grille | Length of room in direction of jet flow |
| Ceiling slot diffuser | Distance to wall perpendicular to jet or midplane between outlets |
| Light troffer diffusers | Distance to midplane between outlets plus distance from ceiling to top of occupied zone |

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 for the cooling condition, 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):

$$t_{ed} = (t_x - t_c) - 0.07(V_x - 30) \text{ [Cooling Only]} \quad (2)$$

where

t_{ed} = effective draft temperature, $^{\circ}\text{F}$

t_x = local airstream dry-bulb temperature, $^{\circ}\text{F}$

t_c = average (control) room dry-bulb temperature, $^{\circ}\text{F}$

V_x = local airstream centerline velocity, ft/min

Liu and Novoselac (2015) developed an effective draft temperature t_{ed} for the heating condition based on predicted mean vote (PMV) model specified in ASHRAE *Standard* 55. A high vertical temperature gradient occurs during heating when throws are insufficient, resulting in stagnation of supply air and unacceptable ventilation to the breathing zone.

$$t_{ed} = (t_x - t_c) - 0.08(V_x - 30) \text{ [Heating Only]} \quad (3)$$

where

t_{ed} = effective draft temperature, $^{\circ}\text{F}$

t_x = local airstream dry-bulb temperature, $^{\circ}\text{F}$

t_c = average (control) room dry-bulb temperature, $^{\circ}\text{F}$

V_x = local airstream centerline velocity, ft/min

T_{50}/L Selection Method. This method uses the ratio of cataloged isothermal throw data at 50 ft/min to the characteristic length for a given device (Table 5).

Each type of diffuser has different performance characteristics and therefore may provide a different ADPI value for the same conditions. Calculating T_{50}/L for a given outlet can predict the level of comfort for a space. Using Tables 6A and 6B, the designer can optimize not only the type of diffuser to select but also the size and capacity for both cooling and heating. The maximum ADPI listed in Tables 6A and 6B reflects the optimal value expected for the given outlet type and location. The maximum ADPI T_{50}/L represents the ratio corresponding with that maximum ADPI value.

Table 6A Air Diffusion Performance Index (ADPI) Selection Guide for Typical Cooling Loads

| Terminal Device in Cooling Mode | Installation | Load, Btu/h · ft ² | Max. ADPI T_{50}/L | Max. ADPI | T/L Low Limit for ADPI > 80% | T/L High Limit for ADPI > 80% |
|--|-------------------------------------|-------------------------------|----------------------|-----------|--------------------------------|---------------------------------|
| Adjustable-blade grilles | 45° upward blades, High sidewall | 8 | 0.8 | 98 | 0.4 | 1.3 |
| | | 16 | 0.9 | 96 | 0.5 | 1.2 |
| | 0° horizontal blades, High sidewall | 8 | 1.7 | 94 | 1.2 | 2.2 |
| | | 16 | 1.8 | 88 | 1.4 | 2.2 |
| | 45° downward blades, High sidewall | 8 | 0.9 | 76 | NA | NA |
| | | 16 | 1 | 70 | NA | NA |
| Fixed-blade grilles (high sidewall installation) | 15° upward blades, High sidewall | 8 | 1.4 | 96 | 0.5 | 2.4 |
| | | 16 | 2.1 | 94 | 1.2 | 2.9 |
| | 15° downward blades, High sidewall | 8 | 1.9 | 85 | 1.5 | 2.2 |
| | | 16 | 2 | 82 | 1.8 | 2.2 |
| Linear-bar grilles (high sidewall installation) | High sidewall | 8 | 1.3 | 92 | 0.7 | 1.8 |
| | | 16 | 1.3 | 88 | 1.0 | 1.6 |
| | Sill | 8 | 1.3 | 94 | 0.9 | 1.7 |
| | | 16 | 1.3 | 90 | 1.0 | 1.6 |
| Nozzles (high sidewall installation) | High sidewall | 8 | 0.7 | 96 | 0.4 | 2.0 |
| | | 16 | 1 | 89 | 0.4 | 1.9 |
| Round ceiling diffuser | Ceiling | 8 | 1.6 | 99 | 0.4 | 3.2 |
| | | 16 | 1.9 | 98 | 0.5 | 3.2 |
| Square ceiling diffuser | Ceiling | 8 | 1.8 | 100 | 0.8 | 2.8 |

| | | | | | | |
|---|------------------------------|----|-----|-----|-----|-----|
| | | 16 | 1.8 | 100 | 0.6 | 3.1 |
| Perforated diffusers, round pattern | Ceiling | 8 | 1.9 | 95 | 0.5 | 3.3 |
| | | 16 | 2.1 | 95 | 0.9 | 3.4 |
| Perforated diffusers, directional pattern (4-way) | Ceiling | 8 | 2.1 | 100 | 1.2 | 3.1 |
| | | 16 | 2 | 95 | 1.0 | 2.9 |
| Louvered face diffusers, with lip on deflector blade | Ceiling | 8 | 2.5 | 100 | 0.5 | 4.4 |
| | | 16 | 2.6 | 100 | 0.6 | 4.5 |
| Louvered face diffusers, without lip on deflector blade | Ceiling | 8 | 2 | 100 | 0.5 | 3.6 |
| | | 16 | 1.8 | 100 | 0.4 | 3.4 |
| Plaque face diffusers | Ceiling | 8 | 1.6 | 100 | 0.3 | 3.0 |
| | | 16 | 1.6 | 100 | 0.4 | 3.2 |
| Linear-slot diffusers | Ceiling | 8 | 1.8 | 100 | 0.5 | 3.0 |
| | | 16 | 1.8 | 100 | 0.5 | 3.1 |
| T-bar slot diffusers | Ceiling, periphery of a wall | 8 | 1.3 | 96 | 0.7 | 1.9 |
| | | 16 | 1.5 | 90 | 1.1 | 1.9 |
| Swirl diffusers | Ceiling | 8 | 1.3 | 100 | 0.4 | 2.4 |
| | | 16 | 1.3 | 98 | 0.4 | 2.4 |
| N-slot diffusers | Ceiling | 8 | 1.8 | 100 | 1.3 | 2.4 |
| | | 16 | 1.8 | 95 | 1.3 | 2.3 |

Source: Data developed by Liu et al. (2016) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (70 fpm) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 12 ft ceiling.

Table 6B Air Diffusion Performance Index (ADPI) Selection Guide for Typical Heating Loads

| Terminal Device in Heating Mode | Installation | Load, Btu/h · ft ² | Max. ADPI T ₅₀ /L | Max. ADPI | T/L Low Limit for ADPI > 80% | T/L High Limit for ADPI > 80% |
|---|-------------------------------------|-------------------------------|---------------------------------|-----------|------------------------------|-------------------------------|
| Adjustable-blade grilles | 45° upward blades, High sidewall | 10 to 12 | 1.1 | 95 | 0.6 | 1.9 |
| | 0° horizontal blades, High sidewall | 10 to 12 | 1.6 | 94 | 1.1 | 2.4 |
| | 45° downward blades, High sidewall | 10 to 12 | 0.7 | 84 | 0.6 | 0.8 |
| Fixed-blade grilles | 15° upward blades, High sidewall | 10 to 12 | 1.8 | 96 | 1.2 | 2.8 |
| | 15° downward blades, High sidewall | 10 to 12 | 1.4 | 88 | 0.6 | 2.2 |
| Linear-bar grilles | High sidewall | 10 to 12 | 1.2 | 94 | 0.6 | 1.7 |
| | Sill | 10 to 12 | 1.2 | 100 | 0.7 | 1.8 |
| Nozzles (high sidewall installation) | High sidewall | 10 to 12 | 1.5 | 92 | 1.0 | 2.0 |
| Round ceiling diffuser | Ceiling | 10 to 12 | 1.4 | 93 | 1.0 | 2.3 |
| Square ceiling diffuser | Ceiling | 10 to 12 | 1.7 | 91 | 2.5 | 3.4 |
| Perforated diffusers, round pattern | Ceiling | 10 to 12 | 2.1 | 90 | 2.0 | 2.8 |
| Perforated diffusers, directional pattern (4-way) | Ceiling | 10 to 12 | 2.5 | 87 | 2.5 | 3.4 |
| Louvered face diffusers, with lip on deflector blade | Ceiling | 10 to 12 | 2.6 | 88 | 2.5 | 4.4 |
| Louvered face diffusers, without lip on deflector blade | Ceiling | 10 to 12 | 2.1 | 88 | 2.1 | 3.2 |
| Plaque face diffusers | Ceiling | 10 to 12 | 2.1 | 93 | 2.1 | 3.0 |
| Linear-slot diffusers | Ceiling | 10 to 12 | 1.7 | 90 | 1.7 | 3.1 |
| T-bar slot diffusers | Ceiling, periphery of a wall | 10 to 12 | 1.6 | 91 | 1.3 | 2.0 |
| Swirl diffusers | Ceiling | 10 to 12 | 1.4 | 100 | 1.4 | 2.1 |
| N-slot diffusers | Ceiling | 10 to 12 | 1.9 | 100 | 1.5 | 2.4 |

Source: Data developed by Liu and Novoselac (2015) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (70 fpm) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 12 ft ceiling.

Table 7A Air Diffusion Performance Index (ADPI) Selection Guide for Typical Cooling Loads

| Terminal Device in Cooling Mode | Installation | Load, Btu/h · ft ² | Max. ADPI T ₅₀ /L | Max. ADPI | T/L Low Limit for ADPI > 80% | T/L High Limit for ADPI > 80% |
|--|-------------------------------------|-------------------------------|---------------------------------|-----------|------------------------------|-------------------------------|
| Adjustable-blade grilles | 45° upward blades, High sidewall | 8 | 0.8 | 98 | 0.4 | 1.3 |
| | | 16 | 0.9 | 96 | 0.5 | 1.2 |
| | 0° horizontal blades, High sidewall | 8 | 1.7 | 94 | 1.2 | 2.2 |
| | | 16 | 1.8 | 88 | 1.4 | 2.2 |
| | 45° downward blades, High sidewall | 8 | 0.9 | 76 | NA | NA |
| | | 16 | 1 | 70 | NA | NA |
| Fixed-blade grilles (high sidewall installation) | 15° upward blades, High sidewall | 8 | 1.4 | 96 | 0.5 | 2.4 |
| | | 16 | 2.1 | 94 | 1.2 | 2.9 |
| | 15° downward blades, High sidewall | 8 | 1.9 | 85 | 1.5 | 2.2 |
| | | 16 | 2 | 82 | 1.8 | 2.2 |
| Linear-bar grilles (high sidewall installation) | High sidewall | 8 | 1.3 | 92 | 0.7 | 1.8 |
| | | 16 | 1.3 | 88 | 1.0 | 1.6 |
| | Sill | 8 | 1.3 | 94 | 0.9 | 1.7 |
| | | 16 | 1.3 | 90 | 1.0 | 1.6 |
| Nozzles (high sidewall installation) | High sidewall | 8 | 0.7 | 96 | 0.4 | 2.0 |
| | | 16 | 1 | 89 | 0.4 | 1.9 |
| Round ceiling diffuser | Ceiling | 8 | 1.6 | 99 | 0.4 | 3.2 |
| | | 16 | 1.9 | 98 | 0.5 | 3.2 |

| | | | | | | |
|---|------------------------------|----|-----|-----|-----|-----|
| Square ceiling diffuser | Ceiling | 8 | 1.8 | 100 | 0.8 | 2.8 |
| | | 16 | 1.8 | 100 | 0.6 | 3.1 |
| Perforated diffusers, round pattern | Ceiling | 8 | 1.9 | 95 | 0.5 | 3.3 |
| | | 16 | 2.1 | 95 | 0.9 | 3.4 |
| Perforated diffusers, directional pattern (4-way) | Ceiling | 8 | 2.1 | 100 | 1.2 | 3.1 |
| | | 16 | 2 | 95 | 1.0 | 2.9 |
| Louvered face diffusers, with lip on deflector blade | Ceiling | 8 | 2.5 | 100 | 0.5 | 4.4 |
| | | 16 | 2.6 | 100 | 0.6 | 4.5 |
| Louvered face diffusers, without lip on deflector blade | Ceiling | 8 | 2 | 100 | 0.5 | 3.6 |
| | | 16 | 1.8 | 100 | 0.4 | 3.4 |
| Plaque face diffusers | Ceiling | 8 | 1.6 | 100 | 0.3 | 3.0 |
| | | 16 | 1.6 | 100 | 0.4 | 3.2 |
| Linear-slot diffusers | Ceiling | 8 | 1.8 | 100 | 0.5 | 3.0 |
| | | 16 | 1.8 | 100 | 0.5 | 3.1 |
| T-bar slot diffusers | Ceiling, periphery of a wall | 8 | 1.3 | 96 | 0.7 | 1.9 |
| | | 16 | 1.5 | 90 | 1.1 | 1.9 |
| Swirl diffusers | Ceiling | 8 | 1.3 | 100 | 0.4 | 2.4 |
| | | 16 | 1.3 | 98 | 0.4 | 2.4 |
| N-slot diffusers | Ceiling | 8 | 1.8 | 100 | 1.3 | 2.4 |
| | | 16 | 1.8 | 95 | 1.3 | 2.3 |

Source: Data developed by Liu et al. (2016) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (70 fpm) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 12 ft ceiling.

Table 7B Air Diffusion Performance Index (ADPI) Selection Guide for Typical Heating Loads

| Terminal Device in Heating Mode | Installation | Load, Btu/h · ft ² | Max. ADPI T ₅₀ /L | Max. ADPI | T/L Low Limit for ADPI > 80% | T/L High Limit for ADPI > 80% |
|---|-------------------------------------|-------------------------------|---------------------------------|-----------|------------------------------|-------------------------------|
| Adjustable-blade grilles | 45° upward blades, High sidewall | 10 to 12 | 1.1 | 95 | 0.6 | 1.9 |
| | 0° horizontal blades, High sidewall | 10 to 12 | 1.6 | 94 | 1.1 | 2.4 |
| | 45° downward blades, High sidewall | 10 to 12 | 0.7 | 84 | 0.6 | 0.8 |
| Fixed-blade grilles | 15° upward blades, High sidewall | 10 to 12 | 1.8 | 96 | 1.2 | 2.8 |
| | 15° downward blades, High sidewall | 10 to 12 | 1.4 | 88 | 0.6 | 2.2 |
| Linear-bar grilles | High sidewall | 10 to 12 | 1.2 | 94 | 0.6 | 1.7 |
| | Sill | 10 to 12 | 1.2 | 100 | 0.7 | 1.8 |
| Nozzles (high sidewall installation) | High sidewall | 10 to 12 | 1.5 | 92 | 1.0 | 2.0 |
| Round ceiling diffuser | Ceiling | 10 to 12 | 1.4 | 93 | 1.0 | 2.3 |
| Square ceiling diffuser | Ceiling | 10 to 12 | 1.7 | 91 | 2.5 | 3.4 |
| Perforated diffusers, round pattern | Ceiling | 10 to 12 | 2.1 | 90 | 2.0 | 2.8 |
| Perforated diffusers, directional pattern (4-way) | Ceiling | 10 to 12 | 2.5 | 87 | 2.5 | 3.4 |
| Louvered face diffusers, with lip on deflector blade | Ceiling | 10 to 12 | 2.6 | 88 | 2.5 | 4.4 |
| Louvered face diffusers, without lip on deflector blade | Ceiling | 10 to 12 | 2.1 | 88 | 2.1 | 3.2 |
| Plaque face diffusers | Ceiling | 10 to 12 | 2.1 | 93 | 2.1 | 3.0 |
| Linear-slot diffusers | Ceiling | 10 to 12 | 1.7 | 90 | 1.7 | 3.1 |
| T-bar slot diffusers | Ceiling, periphery of a wall | 10 to 12 | 1.6 | 91 | 1.3 | 2.0 |
| Swirl diffusers | Ceiling | 10 to 12 | 1.4 | 100 | 1.4 | 2.1 |
| N-slot diffusers | Ceiling | 10 to 12 | 1.9 | 100 | 1.5 | 2.4 |

Source: Data developed by Liu and Novoselac (2015) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (70 fpm) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 12 ft ceiling.

Table 8A Air Diffusion Performance Index (ADPI) Selection Guide for Typical Cooling Loads

| Terminal Device in Cooling Mode | Installation | Load, Btu/h · ft ² | Max. ADPI T ₅₀ /L | Max. ADPI | T/L Low Limit for ADPI > 80% | T/L High Limit for ADPI > 80% |
|--|-------------------------------------|-------------------------------|---------------------------------|-----------|------------------------------|-------------------------------|
| Adjustable-blade grilles | 45° upward blades, High sidewall | 8 | 0.8 | 98 | 0.4 | 1.3 |
| | | 16 | 0.9 | 96 | 0.5 | 1.2 |
| | 0° horizontal blades, High sidewall | 8 | 1.7 | 94 | 1.2 | 2.2 |
| | | 16 | 1.8 | 88 | 1.4 | 2.2 |
| | 45° downward blades, High sidewall | 8 | 0.9 | 76 | NA | NA |
| | | 16 | 1 | 70 | NA | NA |
| Fixed-blade grilles (high sidewall installation) | 15° upward blades, High sidewall | 8 | 1.4 | 96 | 0.5 | 2.4 |
| | | 16 | 2.1 | 94 | 1.2 | 2.9 |
| | 15° downward blades, High sidewall | 8 | 1.9 | 85 | 1.5 | 2.2 |
| | | 16 | 2 | 82 | 1.8 | 2.2 |
| Linear-bar grilles (high sidewall installation) | High sidewall | 8 | 1.3 | 92 | 0.7 | 1.8 |
| | | 16 | 1.3 | 88 | 1.0 | 1.6 |
| | Sill | 8 | 1.3 | 94 | 0.9 | 1.7 |
| | | 16 | 1.3 | 90 | 1.0 | 1.6 |
| Nozzles (high sidewall installation) | High sidewall | 8 | 0.7 | 96 | 0.4 | 2.0 |
| | | 16 | 1 | 89 | 0.4 | 1.9 |
| Round ceiling diffuser | Ceiling | 8 | 1.6 | 99 | 0.4 | 3.2 |

| | | | | | | |
|---|------------------------------|----|-----|-----|-----|-----|
| | | 16 | 1.9 | 98 | 0.5 | 3.2 |
| Square ceiling diffuser | Ceiling | 8 | 1.8 | 100 | 0.8 | 2.8 |
| | | 16 | 1.8 | 100 | 0.6 | 3.1 |
| Perforated diffusers, round pattern | Ceiling | 8 | 1.9 | 95 | 0.5 | 3.3 |
| | | 16 | 2.1 | 95 | 0.9 | 3.4 |
| Perforated diffusers, directional pattern (4-way) | Ceiling | 8 | 2.1 | 100 | 1.2 | 3.1 |
| | | 16 | 2 | 95 | 1.0 | 2.9 |
| Louvered face diffusers, with lip on deflector blade | Ceiling | 8 | 2.5 | 100 | 0.5 | 4.4 |
| | | 16 | 2.6 | 100 | 0.6 | 4.5 |
| Louvered face diffusers, without lip on deflector blade | Ceiling | 8 | 2 | 100 | 0.5 | 3.6 |
| | | 16 | 1.8 | 100 | 0.4 | 3.4 |
| Plaque face diffusers | Ceiling | 8 | 1.6 | 100 | 0.3 | 3.0 |
| | | 16 | 1.6 | 100 | 0.4 | 3.2 |
| Linear-slot diffusers | Ceiling | 8 | 1.8 | 100 | 0.5 | 3.0 |
| | | 16 | 1.8 | 100 | 0.5 | 3.1 |
| T-bar slot diffusers | Ceiling, periphery of a wall | 8 | 1.3 | 96 | 0.7 | 1.9 |
| | | 16 | 1.5 | 90 | 1.1 | 1.9 |
| Swirl diffusers | Ceiling | 8 | 1.3 | 100 | 0.4 | 2.4 |
| | | 16 | 1.3 | 98 | 0.4 | 2.4 |
| N-slot diffusers | Ceiling | 8 | 1.8 | 100 | 1.3 | 2.4 |
| | | 16 | 1.8 | 95 | 1.3 | 2.3 |

Source: Data developed by Liu et al. (2016) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (70 fpm) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 12 ft ceiling.

Table 8B Air Diffusion Performance Index (ADPI) Selection Guide for Typical Heating Loads

| Terminal Device in Heating Mode | Installation | Load, Btu/h · ft ² | Max. ADPI T_{50}/L | Max. ADPI | T/L Low Limit for ADPI > 80% | T/L High Limit for ADPI > 80% |
|---|-------------------------------------|-------------------------------|-------------------------|-----------|--------------------------------|---------------------------------|
| Adjustable-blade grilles | 45° upward blades, High sidewall | 10 to 12 | 1.1 | 95 | 0.6 | 1.9 |
| | 0° horizontal blades, High sidewall | 10 to 12 | 1.6 | 94 | 1.1 | 2.4 |
| | 45° downward blades, High sidewall | 10 to 12 | 0.7 | 84 | 0.6 | 0.8 |
| Fixed-blade grilles | 15° upward blades, High sidewall | 10 to 12 | 1.8 | 96 | 1.2 | 2.8 |
| | 15° downward blades, High sidewall | 10 to 12 | 1.4 | 88 | 0.6 | 2.2 |
| Linear-bar grilles | High sidewall | 10 to 12 | 1.2 | 94 | 0.6 | 1.7 |
| | Sill | 10 to 12 | 1.2 | 100 | 0.7 | 1.8 |
| Nozzles (high sidewall installation) | High sidewall | 10 to 12 | 1.5 | 92 | 1.0 | 2.0 |
| Round ceiling diffuser | Ceiling | 10 to 12 | 1.4 | 93 | 1.0 | 2.3 |
| Square ceiling diffuser | Ceiling | 10 to 12 | 1.7 | 91 | 2.5 | 3.4 |
| Perforated diffusers, round pattern | Ceiling | 10 to 12 | 2.1 | 90 | 2.0 | 2.8 |
| Perforated diffusers, directional pattern (4-way) | Ceiling | 10 to 12 | 2.5 | 87 | 2.5 | 3.4 |
| Louvered face diffusers, with lip on deflector blade | Ceiling | 10 to 12 | 2.6 | 88 | 2.5 | 4.4 |
| Louvered face diffusers, without lip on deflector blade | Ceiling | 10 to 12 | 2.1 | 88 | 2.1 | 3.2 |
| Plaque face diffusers | Ceiling | 10 to 12 | 2.1 | 93 | 2.1 | 3.0 |
| Linear-slot diffusers | Ceiling | 10 to 12 | 1.7 | 90 | 1.7 | 3.1 |
| T-bar slot diffusers | Ceiling, periphery of a wall | 10 to 12 | 1.6 | 91 | 1.3 | 2.0 |
| Swirl diffusers | Ceiling | 10 to 12 | 1.4 | 100 | 1.4 | 2.1 |
| N-slot diffusers | Ceiling | 10 to 12 | 1.9 | 100 | 1.5 | 2.4 |

Source: Data developed by Liu and Novoselac (2015) for this chapter from ASHRAE research project RP-1546 (Liu 2016), and air speed limit (70 fpm) extrapolated from data. Additional data point used to create new regressions for ADPI curves to better represent current diffusers/grilles. Table applies to spaces with maximum 12 ft ceiling.

Using T_{50}/L helps designers maximize space cooling comfort; however, this method is not meant to, nor may it be practical to, evaluate T_{50}/L values for each outlet on a project. The design guidelines in [Tables 6A](#) and [6B](#) were developed from laboratory experiments in chambers lower than 10 ft, with test diffusers symmetrically distributed (Liu and Novoselac 2014; Liu et al. 2016; Miller and Nash 1971). Therefore, attention should be paid to ceiling height of buildings (e.g., airport terminals) or highly asymmetric diffuser layouts.

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

1. Determine air volumetric flow requirements based on load and room size. For VAV systems, evaluation should include both minimum occupied and maximum design flow rates.
2. Select tentative diffuser type and location in room.
3. Determine room's characteristic length L ([Table 5](#)).
4. Select recommended T_{50}/L ratio from [Tables 6A](#) and [6B](#).
5. Calculate throw distance T_{50} by multiplying recommended T_{50}/L ratio from [Tables 6A](#) and [6B](#) 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 2. For a 20 by 12 ft room, with 9 ft ceiling, with uniform loading of 10 Btu/h·ft² or 2400 Btu/h and air volumetric flow of 1 cfm/ft² or 240 cfm for one outlet, find the size for a 0° deflection horizontal blade, high sidewall grille located at center of 12 ft end wall, 9 in. from ceiling.

Solution:

1. Constant-volume system, 240 cfm, high sidewall grille, located at center of 12 ft end wall, 9 in. from ceiling

2. Characteristic length $L = 20$ ft (length of room: [Table 5](#))
3. Recommended maximum,

$$T_{50}/L = 1.8 \text{ in cooling mode (Table 6A)}$$

$$T_{50}/L = 1.6 \text{ in heating mode (Table 6B)}$$

4. Throw to 50 ft/min,

$$T_{50} = 1.8 \times 20 = 36 \text{ ft in cooling mode}$$

$$T_{50} = 1.6 \times 20 = 32 \text{ ft in heating mode}$$

To satisfy both modes of operation, choose one or find a common throw distance that resides within the overall ADPI range of both modes and base product selection off chosen criteria. Further evaluation may be required if air volumetric flow changes based on mode of operation.

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

There are many other considerations to be made when designing a fully mixed system. For more information, see Rock and Zhu (2002).

Typical Applications

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 5](#). 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 1 ft or more below a ceiling surface, discharge air will not attach to the surface. The unattached supply air has a shorter throw and can project downward, resulting in high air velocities in the occupied zone. Some outlets are designed for use in exposed duct applications. Typical outlet performance data presented by manufacturers are for outlets with surface effect; consult manufacturers for information on exposed duct applications.

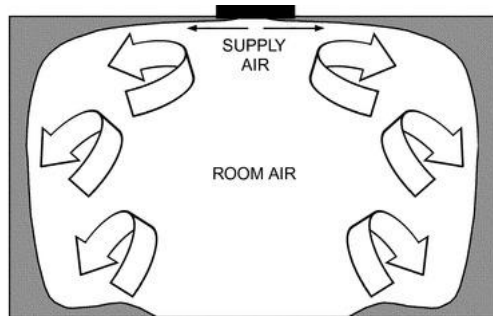


Figure 5. Air Supplied at Ceiling Induces Room Air into Supply Jet

Vertical-Discharge Cooling or Heating with Ceiling-Mounted Outlets. Vertically projected outlets are typically selected for high-ceiling applications that require forcing supply air down to the occupied zone. It is important to keep cooling supply air velocity below 50 ft/min in the occupied zone.

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 that are within 1 ft of a ceiling and set for horizontal or a slightly upward projection provide a discharge pattern that attaches to the ceiling and travels in the unoccupied zone. This pattern entrains air from the occupied zone to provide mixing.

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

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

When selecting sidewall outlets, it is important to understand the manufacturer's data. Most manufacturers offer data for outlets tested with surface effect, so they only apply if the device is set to direct supply air toward the ceiling. When the device is 4 ft or more below a ceiling, or supply air is directed horizontally or downward, the actual throw distance of the device is typically 30% 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 fully and partially 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. See also the description of underfloor air distribution in the Partially Mixed Air Distribution section of this chapter.

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, loaded carts rolling across). 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 in a way that prevents their use as shelves.

Perimeter Control Techniques. In many cases, it is advantageous to decouple perimeter and interior supply air sources, especially as new buildings trend toward full glazing exteriors. Often, interior spaces require cooling all year round due to personnel, electronic equipment, and lighting load requirements. Perimeter spaces, however, are more susceptible to the outside environmental conditions and can have major effects on interior space loads if not properly managed.

Heating and Cooling with Perimeter Ceiling-Mounted Outlets. When air outlets are used at the perimeter with vertical projection for heating and/or cooling, they should be located near the perimeter surface, and selected so that the published 150 ft/min isothermal throw extends at least halfway down the surface or 4.5 ft above the floor, whichever is lower. In this manner, during heating, warm air mixes with the cool downdraft on the perimeter surface, to reduce or even eliminate drafts in the occupied space.

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

Be conscious of room design. If there are shelves or desks against or near the perimeter surface the high velocity air jet could disturb items on the surfaces or result in drafts.

Perimeter Heating with Floor/Sill-Mounted Outlets. The outlets used in this configuration must be located near the perimeter surface. In heating mode, it is important to select a vertical throw such that it overcomes the down draft produced by a cold perimeter surface. The amount of heat generated from the outlet must be enough to overcome or mix with this down draft airstream to avoid ankle-level draft issues in the occupied space.

3. FULLY STRATIFIED AIR DISTRIBUTION

Fully stratified air distribution systems are characterized by a vertical temperature gradient throughout the space, where the coolest temperature is at the floor level, and the warmest temperature is at the ceiling height. Displacement ventilation (DV) systems are the most common example of a fully stratified air distribution system. DV systems typically use floor or low sidewall diffusers delivering low-velocity, cool air across the floor. The low-velocity air, in conjunction with room loads and buoyancy effects, creates the characteristic vertical thermal stratification.

Displacement diffusers may be mounted above the occupied zone but are generally intended for spot-cooling applications and perform similar to laminar flow diffusers.

Principles of Operation

DV systems (Figure 6) use very low discharge velocities, with diffusers typically sized to provide outlet velocities between 40 and 70 ft/min. In addition to the low velocity discharge, the temperature of the supply air is also different from a fully mixed system, with temperatures generally above 60°F; lower temperatures may be used in industrial applications, exercise or sports facilities, and transient areas where comfort concerns are minimal. This cool supply air is more dense than the ambient air and drops to the floor after discharge, whether from floor, low sidewall, or ceiling mounted locations, spreading across the lower level of the space (typically less than 8 in. in height).

As convective heat sources (Figure 6) 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 (thermal plume) around and above the heat source; as the plume rises, it expands by entraining surrounding air. Its growth and velocity are proportional to the heat source's size and sensible load, as well as the temperature of the ambient air above it. As the plume rises, ambient air from below and around the heat source fills the void. An occupant in a DV system entrains the cool, conditioned air directly into their breathing zone. As the occupant exhales, the spent air, being warmer and more humid than the ambient air, is pulled out of the breathing zone by the rising plume. Convective heat from sources located above the occupied zone has little effect on occupied-zone air temperature.

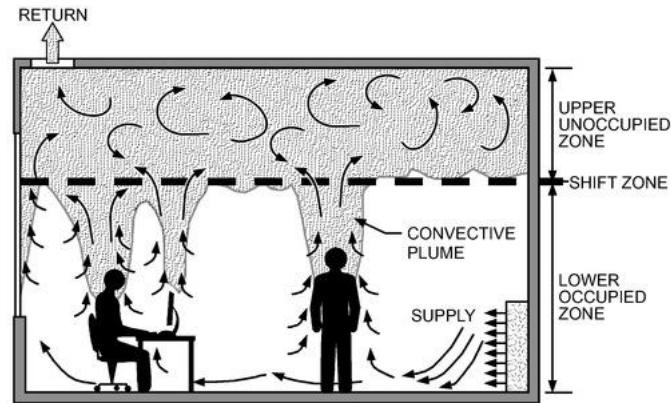


Figure 6. Displacement Ventilation System Characteristics

At a certain height, where plume temperature equals ambient temperature, the plume dissipates 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 often called the **shift zone** (or **stratification height**). The height of this boundary layer between the upper and lower zones is determined based on the convection flow rates of the thermal plumes in relation to the airflow rate supplied by the diffusers. In a DV system, increasing the airflow rate reduces stratification and raises the boundary layer height, with a decrease in airflow providing the opposite effect. Actual and simplified representations of the temperature gradient in the space are shown in Figure 6.

DV systems can be modeled as shown in Figure 7. A thin layer of conditioned supply air (typically between 4 and 8 in.) lies adjacent to the floor. Directly above this layer of conditioned air is the lower zone, in which both ambient air temperature and contaminant concentration levels increase with height; this relationship is mostly linear. As the air transitions through the boundary layer, the upper, unoccupied zone contains a pool of warm, used, and/or contaminated air. This upper zone may or may not form, depending on the supplied airflow rate in proportion to the volume of thermal plumes rising through the space.

Space Ventilation and Contaminant Removal

Thermal plumes created by heat sources, in conjunction with thermal stratification within the space, allow DV to be very effective at removing airborne contaminants that are equal to or lighter than the ambient air (e.g., respiratory-produced contaminants, tobacco smoke). The upward momentum of room air created by thermal plumes from heat sources drive an overall upward momentum of air, which displace contaminants out of the breathing zone. This typically results in a concentration of contaminants above the occupied zone greater than that in the breathing zone. This has been recognized in ASHRAE Standard 62.1-2022, which allows DV systems to have a zone air distribution effectiveness E_z value of 1.2, compared to maximum values of 1.0 for mixed air systems. Thus, the designer can decrease the required outdoor air by 17% when using a DV system, which may result in energy savings for the system. Care should be taken when selecting DV systems when it is known that there will be hazardous airborne contaminants that are heavier than the ambient air. In these systems, a fully mixed system or dilution ventilation may be recommended.

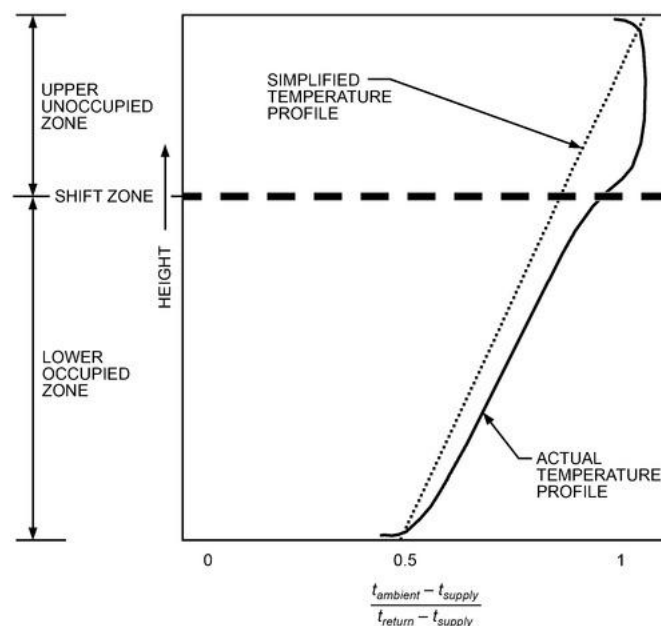


Figure 7. Temperature Profile of Displacement Ventilation System

Figure 8 shows thermal temperature gradients that might be expected for a classroom with a 10 ft ceiling, served by DV. 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 applications where contaminant control and removal are a primary concern, refer to the Application Guidelines section of this chapter.

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

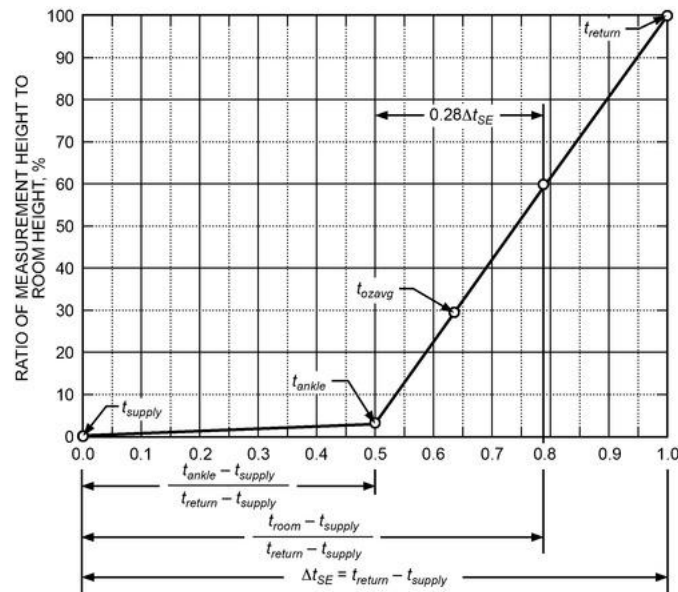


Figure 8. Temperature Gradient Relationships for Thermal Displacement Ventilation System in Typical Classroom or Office with 10 ft Ceiling

Outlet Characteristics

Displacement outlets are designed for average outlet velocities between 40 and 70 ft/min, and are typically mounted in a low sidewall or floor location. Returns or exhausts should be located as high as possible in the space to promote upward momentum of the air, and should efficiently remove as much of the stratified zone as possible; they also should be equally distributed throughout the space to encourage equal air movement and discourage cross flow in the stratified zone. Intensive heat sources may benefit from having returns located directly above them to quickly remove generated heat from the room.

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 around 50 ft/min because air at the ankle level within this velocity envelope tends to be quite cool. As such, most outlet manufacturers define an **adjacent zone** (also called a **clear zone**) in which locating stationary, low-activity occupants is strongly discouraged, but transient occupancy, such as in corridors or aisles, is allowed. Occupants with high activity levels may also find the clear zone acceptable.

For a typical space, a single displacement diffuser can penetrate around 26 to 30 ft. Spaces that are wider than this or have high load densities should look at providing displacement diffusers on multiple walls, or free standing (e.g., around columns) for even airflow distribution. A computational fluid analysis (CFD) may be warranted for more complex spaces to validate the design. Contact the diffuser manufacturer for further diffuser selection assistance.

Benefits and Limitations

Benefits of displacement ventilation systems include the following:

- **Flexibility:** as load distribution changes in the space, the buoyancy forces drive/pull where the conditioned air will go. Thus, a DV system can work effectively in an evenly distributed load space, as well as a space with concentrated loads. A mixing system may cause drafts, or have stagnant zones in similar loading circumstances.
- **Low noise:** due to the low velocities associated with DV diffusers, the generated noise is typically very low (<20 NC for most applications).
- **Fewer drafts:** lower turbulence intensity can reduce draft-related complaints.
- **Indoor air quality improvements:** properly designed DV systems displace higher concentrations of contaminants above the occupied zone than in the breathing zone.
- **Energy savings:** due to a supply air temperature typically above 60°F (more typically 65°F), significantly less energy may be used in certain climates due to an increase in free cooling hours. Less outdoor air is required to meet ASHRAE *Standard* 62.1 requirements, meaning less outdoor air needs to be conditioned.

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

- Spaces with ceiling heights less than 9 ft
- Some spaces with exceptionally high occupied zone heat loads.
- Spaces in hot and humid climates that may result in higher dehumidification and reheat energy usage
- Spaces with ceiling heights below 10 ft that are subjected to significant room air disturbances
- Applications where hazardous contaminants are heavier and/or colder than ambient air.

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 seated occupants be no more than 5.4°F and 7.2°F for standing occupants to maintain an acceptable degree of occupant satisfaction.

Inlet Conditions

Similar to mixing system diffusers, inlet conditions (including presence of dampers, elbows, flex duct, etc.) can have a negative effect on performance, primarily noise (NC) and pressure drop. Inlet conditions should match ASHRAE *Standard* 70 testing conditions as closely as possible to mimic catalogue performance data. DV diffusers are typically designed with internal baffles or nozzles, which should not be affected by inlet conditions at relatively low plenum velocities.

Design Procedures

DV 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.

ASHRAE research project RP-949 (Chen et al. 1999) developed a calculation method for determining supply air volume, air temperature, and other design parameters specifically for DV systems. Using these calculations, the maximum temperature stratification, as outlined in ASHRAE *Standard* 55, is not exceeded. Example 4 uses the calculations outlined in RP-949 and the procedure presented by Chen & Glicksman (2003). This research project is intended for use with typical office and classroom spaces; for larger spaces such as atriums and theaters, careful consideration is necessary, and a computational fluid dynamic (CFD) analysis is recommended to verify design. The Federation of European Heating and Air Conditioning Associations (REHVA) also developed two procedures for calculation of air volume in DV systems; see the REHVA design guide for further information on these methods.

Example 4. Determine the supply air temperature, supply air flow rate, and face area for a displacement diffuser for a small office room ($10 \times 12 \times 9$ ft) with the following characteristics:

q_o = occupant load, 500 Btu/h

q_e = computer/equipment load, 410 Btu/h

$q_{oe} = q_o + q_e = 910$ Btu/h

q_l = lighting load, 615 Btu/h

q_{ex} = exterior load, 1200 Btu/h

$t_{set\ point} = 75^\circ\text{F}$

Step 1. Determine total cooling load:

$$q_t = q_{oe} + q_l + q_{ex} = 910 + 615 + 1,200 = 2,725 \text{ Btu/h}$$

Step 2. Determine airflow rate to meet cooling load. ASHRAE RP-949 (Chen et al. 1999) allows applying factors to each heat load based on how much they contribute to room stratification. The factors identified in their research are as follows:

Equipment and occupant loads F_{oe} : 0.295

Lighting loads F_l : 0.132

External loads F_{ex} : 0.185

The individual space heat gains can then be multiplied by these factors to establish an effective sensible heat gain (ESHG) for the space.

$$\text{ESHG} = (q_{oe} \times F_{oe}) + (q_l \times F_l) + (q_{ex} \times F_{ex}) = (910 \times 0.295) + (615 \times 0.132) + (1200 \times 0.185) = 269 + 81 + 222 = 572 \text{ Btu/h}$$

$$Q_{DV} = \frac{0.295 q_{oe} + 0.132 q_l + 0.185 q_{ex}}{60 \rho c_p \Delta t_{hf}} \quad (4)$$

$$Q_{DV} = \frac{0.295(800) + 0.132(825) + 0.185(450)}{60(0.075)(0.24)(5.4)} = 73 \text{ cfm}$$

where

c_p = specific heat, 0.24 Btu/lb \cdot $^\circ\text{F}$

ρ = density of air, 0.075 lb/ft³

Δt_{hf} = maximum head-to-foot temperature differential for a seated occupant, 5.4 $^\circ\text{F}$ (per ASHRAE *Standard* 55-2020)

Step 3. Determine minimum outdoor air requirement per ASHRAE *Standard* 62.1-2022:

$$Q_{oz} = \frac{R_p P_z + R_a A_z}{E_z} \quad (5)$$

$$Q_{oz} = \frac{(5)(2) + (0.06)(10 \times 12)}{1.2} = 14 \text{ cfm}$$

where

R_p = outdoor air rate per person, 5 cfm/person per ASHRAE *Standard* 62.1-2022

P_z = zone population (number of people in zone during typical usage), 2

R_a = area outdoor air rate, 0.06 cfm/ft² per *Standard* 62.1

A_z = zone floor area, 120 ft²

E_z = ventilation effectiveness, 1.2 (per *Standard* 62.1 for DV systems)

Therefore, the total supply air volume Q_s for cooling is determined by the cooling load and is 98 cfm.

Step 4. Calculate supply air temperature. The following formula assumes thermostat is located 42 in. from the floor:

$$t_s = t_{sp} - \Delta t_{hf} - \frac{A_z q_T}{2.456 Q_{DV}^2 + 1.08 A Q_s} \quad (6)$$

$$t_s = 75 - 5.4 - \frac{(120)(2075)}{(2.456)(98)^2 + (1.08)(120)(98)} = 60.6^\circ\text{F}$$

Step 5. Determine return air temperature:

$$t_r = t_s + \frac{q_T}{1.08 Q_s} \quad (7)$$

$$t_r = 60.6^\circ\text{F} + \frac{2725}{(1.08)(98)} = 86.3^\circ\text{F}$$

Step 6. Adjust airflow for new supply temperature. DV systems typically use warmer supply air temperatures than mixing systems. The supply air temperature should be at least 60.6 $^\circ\text{F}$, or 10 $^\circ\text{F}$ less than the room set point, whichever is higher. Supply airflow should maintain the same return temperature:

(8)

$$t_s = 60.6^\circ\text{F}$$

$$Q_s = \frac{q_t}{60 \rho c_p (t_r - t_s)}$$

$$Q_s = \frac{2725}{(60)(0.075)(0.24)(86.3 - 60.6)} = 98 \text{ cfm}$$

Therefore, in order to condition this particular office using a DV system, it is necessary to deliver 63°F, air at an airflow rate of 73 cfm.

Step 7. Size diffuser. Assume a maximum diffuser face velocity of 50 fpm to obtain acceptable thermal comfort. This recommendation is based on the nominal diffuser size.

$$A_{\text{diffuser}} = \frac{Q_s}{50}$$

(9)

$$A_{\text{diffuser}} = \frac{98}{50} = 1.96 \text{ ft}^2$$

A single diffuser that is 18 × 24 in. will meet this minimum area requirement to maintain comfort. Check with the manufacturer's performance data to ensure diffuser selection meets noise and comfort requirements.

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 8](#)). This method is applicable for typical spaces with a ceiling height up to 12 ft, such as classrooms, office spaces, and meeting rooms. When designing more complex spaces, computational fluid dynamics (CFD) software programs may be used (see [Chapter 13 of the 2021 ASHRAE Handbook—Fundamentals](#) for more information).

The thermal gradient relationships illustrated in [Figure 8](#) 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 located at height 10 ft and the occupied zone is 4 ft high, its gradient comprises 40% of the space temperature gradient. The temperature difference between room air at the top of the occupied zone and the supply air is therefore 70% 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 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 supply air temperature, as shown in Example 5.

The supply airflow rate Q to achieve a given Δt_{SR} is calculated from [Equation \(1\)](#).

Example 5. A classroom with a 10 ft ceiling is to be cooled by displacement ventilation. The supply air temperature is 63°F and room temperature is maintained at 75°F at 4 ft level. The total sensible heat gain of the space is 22,000 Btu/h.

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

Solution: Using the relationships in [Figure 8](#), the supply-to-return temperature differential Δt_{SR} and return air temperature can be predicted as follows:

$$\Delta t_{SR} = (t_{\text{room}} - t_{\text{supply}})/0.70 = (75 - 63)/0.70 = 17.1^\circ\text{F} \quad (10)$$

The value is then added to the supply air temperature to estimate a return air temperature.

$$t_{\text{return}} = t_{\text{supply}} + \Delta t_{SR} = 63 + 17.1 = 80.1^\circ\text{F} \quad (11)$$

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

$$\Delta t_{oz} = \Delta t_{SR} \times 0.25 = 17.1 \times 0.25 = 4.3^\circ\text{F}$$

From [Equation \(1\)](#), the airflow required to maintain this gradient is

$$Q = 22,000/(1.08 \times 17.1) = 1191 \text{ cfm}$$

The ankle-level temperature will be midway between the supply and return temperatures, or 71.6°F. This results in a head-to-foot temperature gradient Δt_{hf} of 3.4°F, which is well within the recommendations of *ASHRAE Standard 55*.

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
- Classrooms/education facilities
- Large open-plan offices, classrooms, lecture halls, and meeting rooms
- Theaters and auditoriums
- Industrial spaces
- Hospitals and cleanrooms
- Other spaces with high ceilings

Perimeter Control

DV systems rely on buoyancy of the cool supply air and thermal plumes of the heat sources to drive stratification in the space. Due to this, DV is typically a cooling-only method of room air distribution. When heating is required, the warmer (more buoyant) air, in conjunction with the low discharge velocity, can mean that the air may not effectively condition the occupied zone, or may completely bypass the occupied zone to the returns. This can have a negative effect on both the thermal comfort and indoor air quality in the space. Limited heating differentials may be acceptable depending on a number of factors. *ASHRAE Standard 62.1-2019* provides effectiveness guidance, and ASHRAE research project RP-1373 (Jiang and Chen 2009) explores the ventilation effectiveness of various DV systems in heating and cooling scenarios.

For heating in cold climates, a separate decoupled system is generally recommended. Displacement ventilation can be used successfully in combination with perimeter fan coils, hydronic systems, or radiators and convectors installed at exterior walls to offset space heat losses. Radiant heating panels and heated floors can also be used. Using this hybrid approach, the DV diffusers can provide adequately cool ventilation air, with the auxiliary devices handling the heating load.

Displacement diffusers with integrated heat typically have a separate heating plenum featuring a heating coil and are often used for perimeter applications. Heat/cool changeover diffusers can also be used. For this design, the heating portion of the diffuser will not provide a displacement pattern, so in this case ventilation effectiveness is compromised.

Care is also needed when using DV systems in high perimeter cooling load applications. DV diffuser mounting locations should be carefully considered to minimize the risk of conditioned air being drawn toward the strong thermal plume of the exterior surface. It may be better to place diffusers along an interior surface to ensure that the conditioned air reaches interior occupants before conditioning the exterior envelope loads.

Considerations Unique to Displacement Ventilation Systems

Thermal displacement ventilation systems can be either constant (CV) or variable air volume (VAV). 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* 62.1-2019 are met, intermittent on/off airflow control can be used.

Avoid using DV and mixed air systems in the same space, because mixing destroys the natural stratification that drives the thermal displacement ventilation system. DV 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.

It is often beneficial to couple DV systems with hydronic products. Displacement chilled beams are available that integrate sensible heating and cooling coils. For more information about chilled beams, see the Chilled Beams section of this chapter.

Another common application is to use DV systems to provide only the space ventilation and cooling, while a complementary hydronic system satisfies the majority of the cooling load. In this case, passive (not active) sensible cooling devices (e.g., passive beams, radiant ceiling panels) should be used to minimize the disruption of the natural stratification that drives the DV system.

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

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

4. PARTIALLY MIXED AIR DISTRIBUTION

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

Principles of Operation

Supply air is discharged, usually vertically, at relatively high velocities and entrains room air in a similar fashion to outlets used in mixed air systems. This entrainment, as shown in [Figure 9](#), reduces the temperature and velocity differentials between supply and ambient room air. This discharge results in a vertical plume that rises until its velocity is reduced to about 50 fpm. At this point, its kinetic energy is insufficient to entrain much more room air, so mixing stops. Because air in the plume is still cooler than the surrounding air, the supply air spreads horizontally across the space, where it is entrained by rising thermal plumes generated by nearby heat sources.

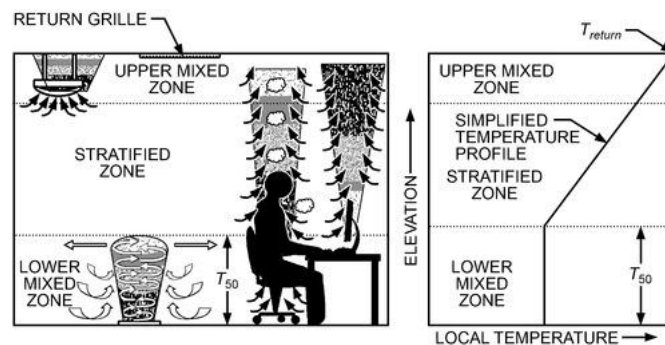


Figure 9. UFAD System in Partially Stratified Application

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

- By reducing airflow and mixing in the occupied zone, fan energy can be reduced and stratification can be increased, approaching a reasonable target at 3 to 4°F temperature difference from head level to ankle level, which satisfies ASHRAE *Standard* 55-2020.
- By increasing airflow and mixing in the occupied zone, excessive stratification can be avoided, thereby improving thermal comfort.

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

[Figure 9](#) shows one example of the resulting room air distribution in which the room air is mixed in the **lower mixed zone**, which is bounded by the floor and the elevation (**throw height**) at which the 50 fpm terminal velocity occurs. At this elevation, stratification begins to occur and a linear temperature gradient, similar to that found in thermal displacement systems, forms and extends through the **stratified zone**. As with thermal displacement ventilation, convective heat plumes from space heat sources draw conditioned air from the lower (mixed) level through the stratified zone and to the overhead return location. A third zone, referred to as the **upper mixed zone**, may exist where the volume of rising heat plumes terminate. Although velocities in this area are quite low, the air tends to be mixed.

Space Ventilation and Contaminant Removal

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

According to ASHRAE *Standard* 62.1, these systems may have zone air distribution effectiveness E_z values that exceed those of fully mixed systems. For applications where contaminant control and removal are a primary concern, see the section on Application Guidelines.

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 to automatically throttle 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. Keeping 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.

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 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.
- Possibility of lowering deck-to-deck dimensions.
- 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.

Limitations to consider are

- 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 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.
- Uncontrolled air can leak from pressurized underfloor plenums. Proper design and installation can minimize or eliminate this.
- Plenum air temperature can degrade in relation to distance traveled from the supply air source.

Methods of Evaluation

As with 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; that is, the vertical temperature difference between the ankle and head level of space occupants should be no more than 5.4°F if a high degree (>95%) of occupant comfort is to be maintained.

Inlet Conditions

UFAD systems usually pressurize the open plenum space under the access floor, this way the diffusers do not need to have an inlet. If linear floor diffusers with plenum boxes are utilized, they are usually made with very large rectangular inlets that are the same size as the linear length of those plenums. For ducted installations, use the same inlet criteria as the other air distribution systems.

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, 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. This subject is discussed further in ASHRAE's *UFAD Guide* (2013).
- 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.

Perimeter Control

There are several ways to provide perimeter air distribution using a partially mixed air system. Air distribution in the perimeter zone is highly influenced by weather and solar loads at the building envelope; this means that different amounts of cooling or heating may be needed at different times, independent of interior zone load needs. The most common scenario is when cooling is needed at the interior zone and heating is needed at the perimeter zone; both internal and perimeter loads can be met using a zone divider or a dedicated plenum system. See ASHRAE (2013) for more information on how to handle perimeter zone applications.

Space Temperature Gradients and Airflow Rates

The objective of partially mixed systems is to condition the air in the occupied zone while allowing stratification to naturally occur. By allowing this stratification, some of the space heat gain can be removed by return or exhaust instead of by supply air delivery to the space. If the supply airflow rate and sensible heat gains affecting the lower zone are balanced, an acceptable temperature gradient (<5°F) can be achieved in the occupied zone. Supply airflow beyond that required by these heat gains reduces the degree of stratification shown in [Figure 9](#). 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.

Considerations Unique to Underfloor Air Distribution Systems

The ASHRAE *UFAD Guide* (2013) includes a thorough discussion of issues involved in the design, application, and commissioning of UFAD systems. Some considerations include the following:

- Supply temperatures in the access floor cavity should be kept at 60°F or above, to minimize the risk of condensation and subsequent mold growth.
- Most UFAD outlets can be adjusted automatically by a space thermostat or other control system, or manually by the occupant. In the latter case, outlets should be located within the workstation they serve.

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

5. AIR DISPERSION SYSTEMS

Principles of Operation

The design methodologies provided in this section cover textile air dispersion systems, though the same principles can be applied to systems constructed with metal. Textile air dispersion systems are low pressure extended plenum systems with pressurized tubing and air distributed along the path of least resistance. Suggestions are provided for material selection and dispersion style, support and structure of the textile air dispersion system, and sizing/venting locations of the system. Consult with the manufacturer and specific product data and performance information when designing and specifying textile air dispersion systems.

These systems can also be found designed into the plenum space of raised floor applications. In these applications, the designer must consider that the system is exposed in the space that it is conditioning and delivering air to. Air outlet throws of the textile air dispersion system must be designed for the plenum space in which they are located.

A version of these systems is also being used for return air (negative pressure). This requires a sturdy framework that holds the outer walls of the product open to prevent the restriction from becoming too great and restricting airflow beyond design intent. Maximum internal velocity of 1000 fpm and -0.125 in. of water static pressure

Air Dispersion System Supply Air Outlet Styles

Various supply air outlet styles are available: porous fabric weave used as an outlet, microperforations, linear vents, orifices, and nozzles. Many of these air outlet styles can be used together to achieve specific results.

Porous Fabric Weave (Figure 10). Air is delivered through the weave of the fabric. This can result in air velocities of less than 30 fpm at the material surface. The ability to achieve such a low face velocity makes this the preferred dispersion style where displacement ventilation is needed. This dispersion style is ideal for food processing, cleanrooms, and laboratory environments where elimination of drafts and uniform air distribution is required, and can be used in combination with other types of air outlets to achieve the specified room airflow. Cool air drops to the floor and then spreads across the lower level of the space.

Porous fabric ducts can be used in combination with most linear vents and orifice applications. Air throw distance is generally lower than with linear vents, and may not be appropriate for heated supply air applications.

Microperforation Outlets (Figure 11). Air is delivered through laser-cut microperforations, generally smaller than 0.4 mm. As with porous fabric, air velocities of less than 30 fpm can be realized 2 ft from the duct. Microperforations offer low-velocity distribution in the occupied zone without forcing the majority of the air through the weave of the material. Further, the microperforations can be uniformly located along 360°, or within a specific area or side of the air dispersion device, allowing designers to disperse air exactly where it is needed. This method can be used for displacement ventilation and isothermal/makeup air, but is limited for heating applications (when using directed microperforations for heating, consult manufacturer for throw data). This dispersion style is also ideal for food processing, cleanrooms, and laboratory environments where elimination of drafts and lower-velocity air is required. Cool air can be more evenly distributed in the space using directed microperforations rather than dropping or “dumping” due to buoyancy, directly below the diffuser. Microperforations can be used in combination with most other flow models to modify throw and velocity to meet design requirements.

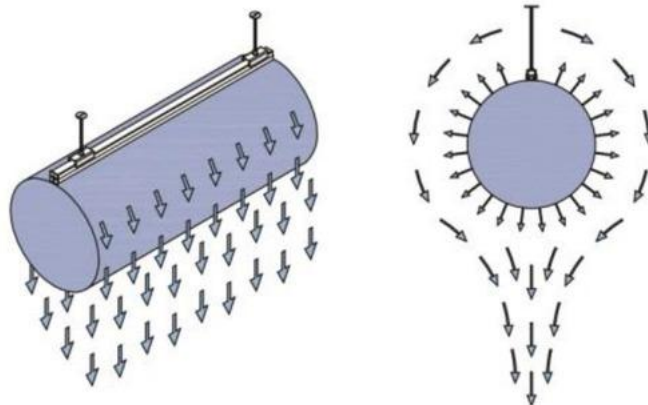


Figure 10. Porous Fabric Weave Used as Outlet

Linear Vent Outlet (Figure 12). Air is delivered through a linear vent outlet, which generally consists of many small outlets in a linear pattern. This provides uniform airflow with throw suitable for commercial and retail spaces, schools, and theaters. Generally, medium air throw distance can be achieved.



Figure 11. Microperforations Used as Outlet

Orifice and Nozzle Outlets (Figure 13). Air is delivered through orifices or venturi-shaped nozzles providing jet-type air distribution for applications such as gymnasiums, pools and manufacturing facilities. This style generally has a higher air throw distance than other outlet styles. Nozzles can direct air perpendicularly away from the surface of the duct. Adjustable nozzles allow for changing the direction and/or flow rate and throw. Nozzles generally have the least entrainment of longer-throw outlets. Manufacturers have many specific options for this style of air outlet, and the manufacturer's data must be consulted.

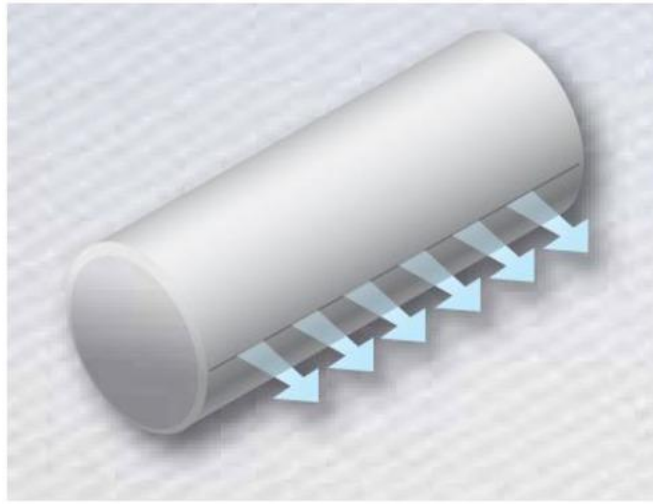


Figure 12. Fabric with Linear Vent Outlet

Air Dispersion System Shapes

Three shapes are commonly available: cylindrical, half circle (D-shape), and the quarter circle, as shown in [Figure 14](#). Cylindrical systems are typical for open ceiling spaces and are mounted using a tension cable or suspended aluminum track suspension system. The half circle can be installed against a ceiling or wall. Manufacturers also have many custom shapes that can be used to solve installation challenges.

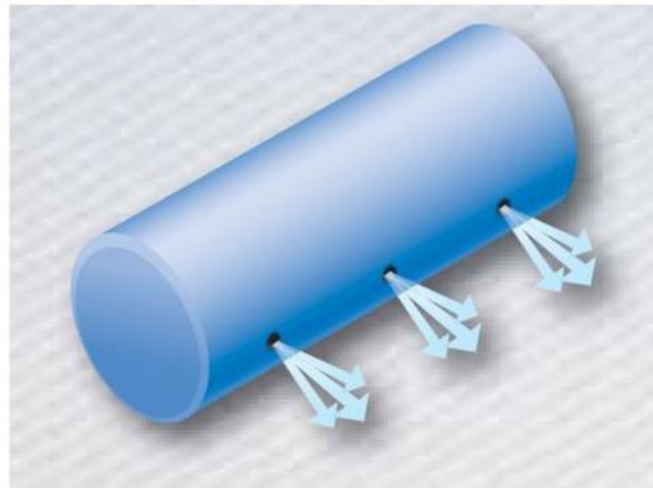


Figure 13. Fabric with Orifice Outlets

Material Selection

Air dispersion system ducts, classified by Underwriters Laboratories as an air distribution device, should have a Class 1 rating per UL *Standard* 723. The maximum flame spread/smoke developed index is currently 25/50.

Additional important properties in selecting a material for air dispersion systems are durability and aesthetics. Durability includes not only the environment, but the design. Supports and static pressure must hold fabric air dispersion system materials in tension to prevent fabric wear and tear. The maximum velocity in a system should not exceed the manufacturer's rated velocity; excessive air velocities cause fabrics to fail prematurely. The *International Mechanical Code*[®] (ICC 2018) and the *Uniform Mechanical Code* (IAPMO 2018) require that air dispersion systems be listed and labeled in compliance to UL *Standard* 2518.

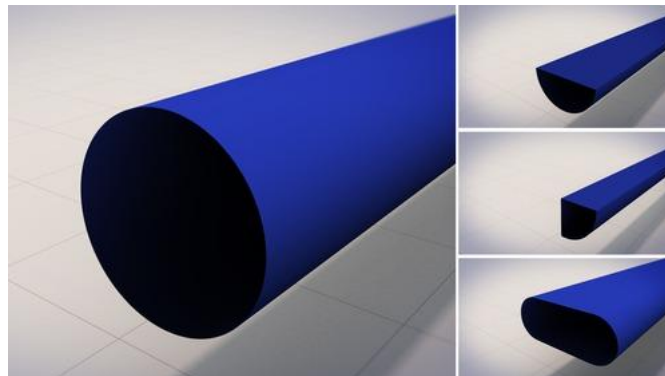


Figure 14. Common Shapes of Air Dispersion Systems

The use of a slightly permeable material allows dispersion of air below dew point without the risk of condensation forming on the fabric air dispersion system. The duct is pressurized, and air is forced through the surface of the material, forming an insulating barrier of cooler air around the duct, preventing warm moist air from contacting the cold surface of the air dispersion system and thereby preventing water droplets from forming on the surface. The permeated air is often induced by outlets on the system and entrained into the supply air exiting the outlets.

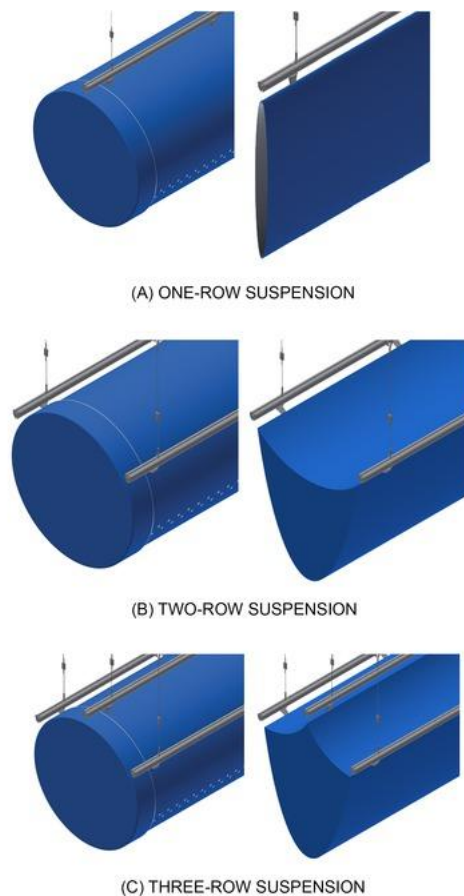


Figure 15. Inflated and Deflated Suspension System

Nonpermeable materials can be used where risk of condensation is low and applied in a way similar to uninsulated single wall metal ducts. These are often used in dryer climates or for extensive systems.

Polyester is typically the preferred material for fabric air dispersion systems because it has very low hygroscopic properties, is a synthetic material that is not a food source for organisms, and is easy to launder. Several manufacturers offer materials with complementary properties, such as antimicrobial treatment (to prevent microbial growth), antistatic for use around sensitive electronics and in explosion proof facilities, and fabrics that do not “shed” filament particles for use in cleanrooms and critical laboratories.

Suspension Systems

Textile air dispersion system manufacturers offer many different types of suspension systems to address the various applications in which they are used ([Figure 15](#)). Typical types include

- Clips sewn to the top of the air dispersion system and clipped onto a tensioned horizontal cable
- Sliders or continuous keder cord sewn to the top of the air dispersion system and slid into a metal track (the track could be mounted directly to the ceiling or suspended a distance below it)
- Sliders or continuous keder cord sewn to the top corners of a half or quarter circle air dispersion system and slid into a metal track (the track could be mounted directly to the ceiling or suspended a distance below it)
- Direct suspension from an internal frame system

Hold-Open and Fabric Retention Systems ([Figure 16](#)). Textile air dispersion systems can also be held open (restricting their potential to collapse onto themselves when the HVAC unit is not providing airflow) with rings, hoops, and arcs.

Fabric Tension Systems ([Figure 17](#)). Fabric tensioning systems hold the fabric in place regardless of whether the system is pressurized. These systems increase fabric longevity by reducing movement of the fabric. Some hold the fabric in such a way that air velocities in the product can be increased to a higher maximum (which results in smaller diameter sizes) without the fabric wall being unstable and fluttering, which could lead to premature failure.

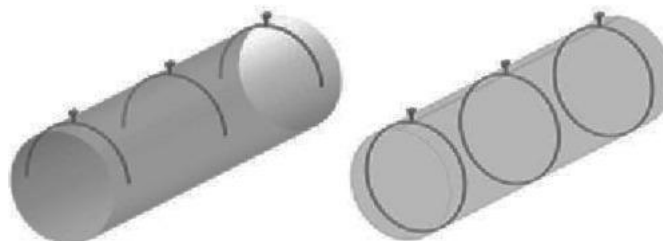


Figure 16. Ring and Arc Style Hold-Open Retention Systems

Layout

A fabric air dispersion system performs as both a duct and a diffuser. When designing a textile air dispersion system, keep the layout as simple as possible, preferably with straight runs. Because porous fabric duct, linear vents, and orifice outlets can be integrated into all sections, system design may vary significantly while providing adequate air dispersion.

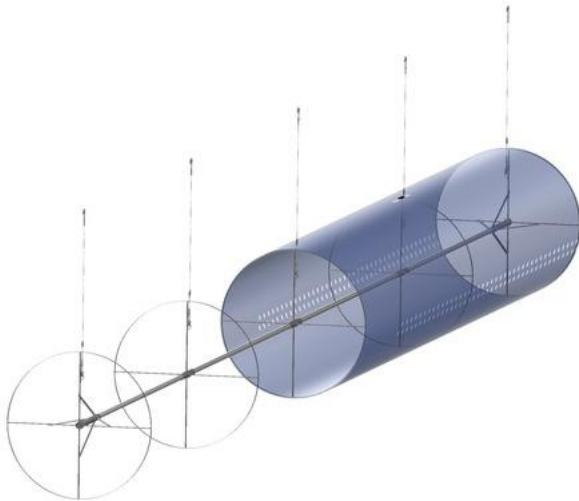


Figure 17. Direct Suspension from Fabric Tensioning System

Metal ductwork before the fabric inlet should have the same air velocity as the fabric duct, and, when possible, it is recommended to use 3× the diameter of straight metal duct. If there is a transition preceding the fabric inlet, it is recommended to use 1.5× the diameter radius elbows and turning vanes in sheet metal transitions before fabric. High velocity (+1600 fpm) and turbulent conditions immediately before the system can cause excessive fabric movement and premature wear.

There is little need to reduce diameters because air dispersion systems are essentially extended plenums that can maintain static pressure due to continuously occurring static regain. Custom fittings should be coordinated with the manufacturer. It is recommended that end caps be one diameter from a wall to maintain clearance for systems and allow for movement and ease of installation.

Fittings. Straight duct lengths and fittings are connected together using a circumference zipper, which is affixed with its start/stop typically located at the top center, and includes a circumferential fabric overlap to conceal the zipper. Typical lengths of zippered sections are sized for easy laundering and installation. Longer sections are broken into multiple lengths.

Elbows. The typical centerline radius of an elbow is figured by multiplying the cross-sectional diameter by 1.5. For example, a 24 in. diameter air dispersion system elbow would have a centerline radius of 36 in. The number of gores depends on the angle of the turn, as shown by [Figure 18](#).

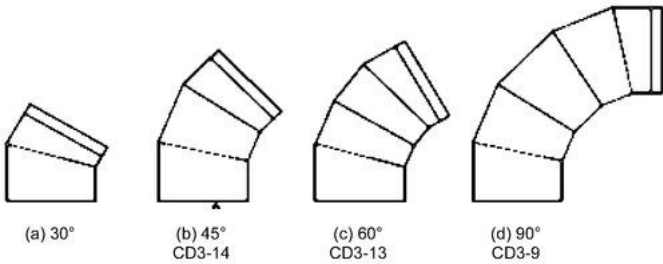


Figure 18. Number of Elbow Gores Based on Turn Angle

Transitions. Reducing transitions are available in concentric, top flat, or bottom flat configurations ([Figure 19](#)). Transition length varies and is based on the change in diameter, such that the total angle does not exceed 30°.

Tees. Tees are saddle type, and the branch requires a zipper for attachment. Typical tee arrangement is shown in [Figure 20](#). It is recommended that tees be located at least 1.5× the outlet diameter from end caps ([Figure 21](#)).

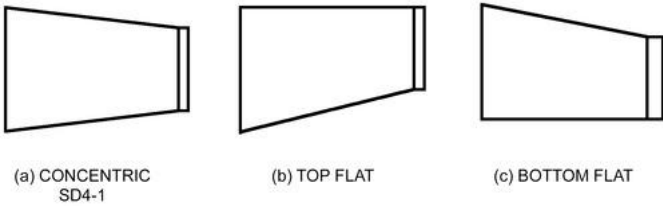


Figure 19. Styles of Fabric Duct Transitions

Cross, Capped. See [Figure 22](#) for the capped cross, which is SD5-20 in the ASHRAE (2017a) *Duct Fitting Database* (DFDB).

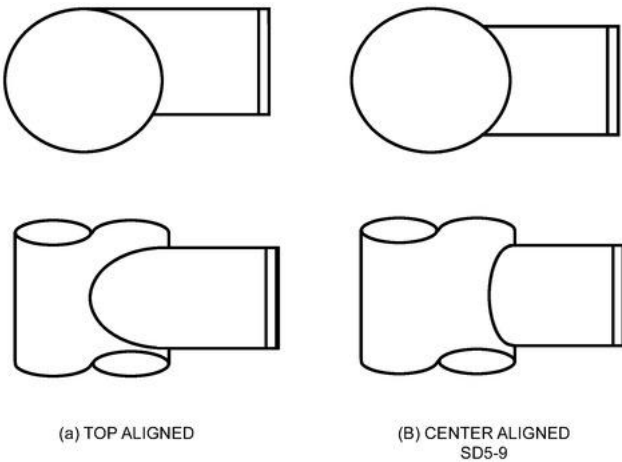


Figure 20. Common Tee Types for Fabric Duct

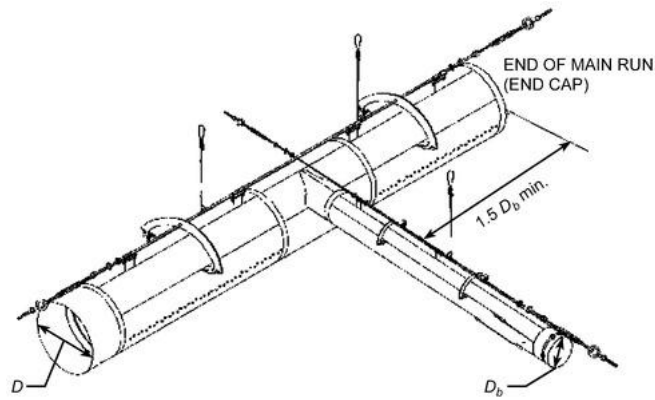


Figure 21. Relationship of End Caps to Tees

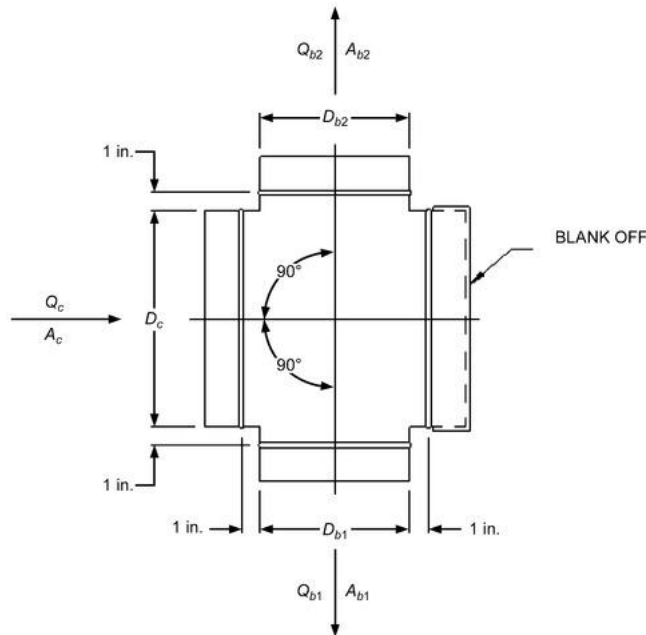


Figure 22. Capped Cross, Fabric (SD5-20)

Dampers and Static Regain Devices. These fittings are used to address multiple objectives. The fittings offer engineered resistance to balance airflow in multiple runs and branches, reduce turbulence, reduce inflation pop, reduce noise, reduce movement from abrupt start-ups, and balance static regain. Locate flow devices at (1) inlet collars, (2) one-third point of a straight section, and (3) after take-offs and elbows. A flow device at the inlet collar reduces fluttering due to turbulent airflow entering the air dispersion system. At the first zipper, the flow device reduces the force of initial inflation and reduces “pop;” a fabric tensioning system can also eliminate any movement, deflation, or noise due to the HVAC system cycling on and off. Straightening out velocity profiles also reduces wear in a fabric duct system. Examples of dampers and static regain devices are shown in [Figure 23](#).

Sizing

Cylindrical Air Dispersion Systems. The recommended design velocity is 1600 fpm for a system without fittings, and 1400 fpm for a system with fittings. Use lower velocity (1000 to 1200 fpm) for noise sensitive areas. If the diameter is too large, design the system for multiple runs.

Half Circle (D-Shape) Air Dispersion Systems.

D-shape air dispersion systems should be sized so that the fabric duct velocity does not exceed the rigid supply duct velocity. The maximum velocity for a straight run is 1000 fpm. For a system with a fitting, the recommended velocity is 800 fpm. A lower inlet velocity to the fabric duct reduces stress and noise.

Flow can be any split (e.g., for a top inlet in the middle, split is 50/50; for a top inlet a third of the distance from one end, split is 33/67), including flow from the end. The D-shape diameters and the duct diameters are good for a split to 38 to 62%. For a greater split, the inlet duct diameter needs to be reduced so the branch duct velocity does not exceed the design inlet duct velocity (800 or 1000 fpm).

Design Procedure

To design air dispersion systems, follow the steps below:

Step 1. Lay out the system. Keep as simple as possible.

Step 2. Given the system airflow requirement, determine the fabric duct size.

Step 3. Assuming an inlet static pressure (ISP) to each section of the air dispersion system, calculate the average static pressure.

Step 4. In each section, use [Equation \(12\)](#), and use the DFDB (ASHRAE 2017a) to calculate Δp_{tx} . [Equation \(12\)](#) is an empirical equation based on experience gained by air dispersion system designers. Leverette et al. (2014) state that this equation is approximately correct. Most often, the ISP to the air dispersion system is approximately 0.5 in. of water.

$$p_{sx,avg} = p_{sx} + 0.65(p_{vx} - \Delta p_{tx}) \quad (12)$$

where

$p_{sx,avg}$ = average static pressure in section x , in. water

p_{sx} = ISP to section x , in. water

p_{vx} = inlet velocity pressure to section x , in. water

Δp_{tx} = total pressure loss of section x , in. water

The component of total pressure drop Δp_{tx} of a fabric duct that is dispersing air equally along its length can be estimated using the DFDB (ASHRAE 2017a; fitting CD11-1) and 35% of the duct length, where 35% is an approximation (because typical systems have a constant diameter from inlet to endcap). The absolute roughness for a system with an

internal support frame is 0.0056 ft, and for an unsupported system (no internal frame) is 0.0004 ft (Kulkarni et al. 2012). These values are based on a 14.5 in. nonporous polyester fabric duct with an acrylic/urethane coating.

Step 5. Airflow through porous material of length L is calculated by Equation (13):

$$Q_{material} = P \left[\pi \left(\frac{D}{K_1} \right) L \right] \left[\frac{p_{sx,avg}}{K_2} \right] \quad (13)$$

where

$Q_{material}$ = airflow diffused through porous material, cfm

D = system duct diameter, in.

P = material porosity, cfm per ft² at 0.5 in. water

L = length of system duct, ft

$p_{sx,avg}$ = average static pressure in section x , in. water

K_1 = conversion to feet, 12

K_2 = reference SP, 0.5

Step 6. Calculate the airflow required through the fabric duct outlets (vents or orifices):

$$Q_{outlet} = Q_{system} - Q_{material} \quad (14)$$

where

Q_{outlet} = total outlet airflow, cfm

Q_{system} = system airflow, cfm

$Q_{material}$ = airflow through material, cfm

Step 7. Determine the length of vent or number of orifices, orientation of outlets (Figure 24), and throw. Typically there is a 4 ft void (no outlets) near the inlet or after any fitting within a system to reduce wear. This is manufacturer dependent. Consider the following when selecting the orientation of air outlets:

- 11 and 1, 10 and 2, and 3 and 9 o'clock (Figure 24): Primarily chosen for cooling or ventilating, these locations direct the exiting air upward and/or outward from the air dispersion system. Throw should reach the exterior walls or fill the gaps between parallel air dispersion ducts.
- 4 and 8, 5 and 7, and 6 o'clock (Figure 24): Primarily chosen for applications with heating, but can also be used for cooling or ventilating, these location direct the exiting air downward and/or outward from the air dispersion system. Throw requirements can be critical in these locations because the air is directed towards the occupied space.

Determine throw:

$$T = K_5(H - K_T) \quad (15)$$

where

T = required throw, ft

H = distance between bottom of duct and floor, ft

K_T = 6

K_5 = 2.0 for 4 and 8 o'clock

1.15 for 5 and 7 o'clock

1.0 for 6 o'clock

For linear vent outlets, select vent size using manufacturer data. Terminal velocity is the maximum airstream velocity at end of throw.

For orifice outlets, select orifice diameter using manufacturer data. Terminal velocity is the maximum airstream velocity at end of throw.

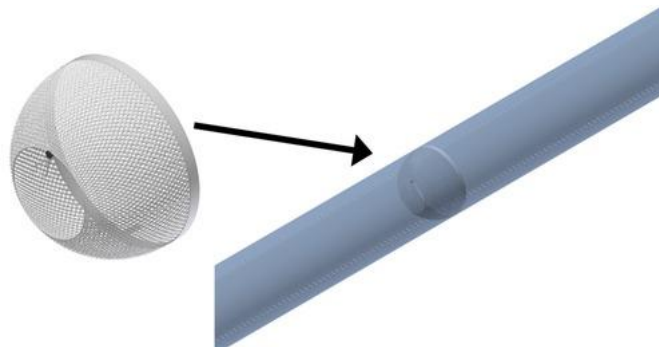


Figure 23. Fabric Adjustable Flow Devices

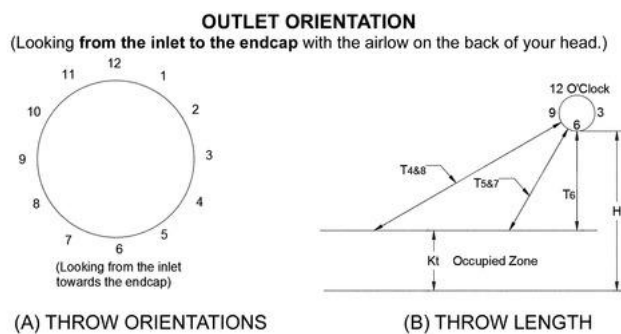


Figure 24. Throw: Directional Airflow/Distance

Operation

Filtration. It is essential to filter incoming air before it reaches the textile air dispersion system. An efficient filtration system means less cleaning, resulting in a longer product life. Higher filtration (MERV 8) is recommended for low-velocity outlets (porous fabric and microperforations). Textile air distribution systems are launderable.

Pressure Required for Inflation. For proper inflation, the static pressure at the inlet to a fabric air distribution system should be no less than 1.3× the inlet velocity pressure.

6. AIR TERMINAL UNITS (ATUS)

Designers have various systems (ATUs 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 9](#) summarizes the different types of ATUs and their suitability for particular commercial building applications.

Principles of Operation

Single-Duct ATUs. Single-duct ATUs are used to regulate airflow to a conditioned space. Primary air is ducted from an air handler. The basic single-duct unit consists of an airflow regulator and may also include an actuator, an airflow-measuring device, selected controls, and heating coils.

Table 9 Suitability of Terminal Units for Various Applications

| Terminal Types | Facility Type | | | | | | | | | | | | | |
|---|--|------------------|--------------------|------------------|------------------|--|--------------------|---------------------|---|----------|-----------|------------------|---------------------|--|
| | Office Space, Educational, and Institutional Buildings | | | | | Hospitals, Cleanrooms, and Laboratories ^a | | | Noise-Sensitive Applications ^b | | | Other Facilities | | |
| | Large | | | Small | | Patient Areas | Operating Areas | Laboratory Space | Broadcast Studios | Theaters | Libraries | Public Use | Shopping Centers | |
| | Interior Zone | Exterior Zone | Low Temperature | Interior Zone | Exterior Zone | | | | | | | | | |
| 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 | |
| Series with heat | • | P | • | • | • | • | N | • | P | P | P | P | P | |
| Low- temperature | • | • | P | • | • | 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.

Single-duct ATUs can be applied to pressure-dependent or -independent systems. **Pressure-dependent** units consist, at minimum, of an airflow regulator, thermostat, and actuator. The thermostat dictates the damper position based on room demand, which is independent of system pressure. Using pressure-dependent ATUs can cause system pressure fluctuation, which must be considered. Current ventilation requirements may preclude the use of pressure-dependent ATUs.

Pressure-independent units consist of, at minimum, an airflow regulator, airflow measuring device, controller, room sensor, and actuator. This is a closed-loop control system that measures and regulates airflow based on room demand. Pressure-independent ATUs allow a minimum airflow set point to be established to enable proper ventilation to the zone.

Pressure-independent single-duct ATUs can be used for constant- or variable-volume applications. Variable-volume applications often include controls that limit maximum and minimum airflow set points determined by load, acoustical, and zone ventilation requirements.

Dual-Duct ATUs. Dual-duct ATUs function similarly to single-duct terminals but are composed of two sets of airflow regulators merging two airstreams into a single mixed airstream. These are typically used to mix cold and hot primary air ducts to a single warm discharge air duct.

Series Fan-Powered ATUs. These fan-powered ATUs have a continuously operating fan during occupied mode, supplying either a constant or variable volume of air to the space, and they are typically installed in the ceiling plenum. Primary air is ducted from an air handler. Induction air is either from the ceiling plenum or ducted from the space. The fan and damper are aligned so that all primary air and all induced air independently enter the mixing section and go through the fan to be delivered to the space. The mixing space is between the VAV air valve and the fan.

Parallel Fan-Powered ATUs. These ATUs have an intermittently operating fan and are typically installed in the ceiling plenum. Primary air is ducted from an air handler and flows directly to the zone in a variable-volume cooling mode. Fan air is induced from the ceiling plenum or may be ducted from the space. The fan should run in heating and dead-band mode. In dead band and heating mode, the supply air is a combination of primary and induced air. The fan and damper are arranged so that all induced air enters the fan, and primary air bypasses the fan. Any mixing of primary 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 it is not running.

Fan-powered ATUs applied with underfloor air distribution systems differ from conventional overhead systems in that the ATUs are mounted in a supply air plenum rather than a return air plenum. The ASHRAE UFAD Guide (2013) addresses these ATUs in detail.

DOAS ATUs. ATUs may be used in conjunction with dedicated outdoor air systems (DOAS) to provide ventilation air (and in many cases latent cooling) to individual zones. The ATUs could be single-duct VAV, dual-duct, or fan-powered. Single ATUs can work in parallel with sensible cooling devices (e.g., passive chilled beams, radiant cooling ceilings) to handle latent loads and meet ventilation requirements. Dual-duct ATUs can blend ventilation air with conditioned building air to meet ventilation requirements. Series-fan-powered ATUs designed with chilled-water sensible cooling coils on the induction inlets are specifically designed to work with DOAS applications. The chilled-water coils are intended to perform sensible cooling only, thus the entering chilled-water temperature should be warm enough to prevent condensate formation. The entire zone latent load is satisfied by an outdoor air handler providing preconditioned air to the fan-powered chilled-water ATU.

Benefits and Limitations

Single-Duct ATUs. *Benefits:* Typically produce very low sound levels and can be used for both VAV and CV applications. With the addition of an inlet-mounted thermistor, these units can also provide VAV heating and cooling from a central air handler.

Limitations: The central air handler must provide adequate inlet static pressure to supply air to the box, overcome any pressure drop of the unit and any integral coils, and move air to the zone. Any heating coils integral to the unit must be treated as reheat. Only remote heating devices located in the occupied space such as finned-tube elements or radiant panels that could be operated by unit controls would be treated as heat.

Dual-Duct ATUs. *Benefits:* Allow centralized heating and cooling in the facility. They can provide very high comfort at relatively low sound levels.

Limitations: Dual-duct systems generally have higher initial costs and operating costs.

Series Fan-Powered ATUs. *Benefits:* Make use of free heat in the form of return air for reduced heating requirement. Typically provide constant air motion and low noise levels in the zone for improved air circulation and occupant comfort. The constant fan operation recirculates return air from the plenum, returning unvitiated air and reducing outdoor air requirements. Low inlet static pressure requirements can reduce fan energy at the air handler. Energy savings can be maximized with electronically commutated motor (ECM). Fans with ECMs can modulate the supply air based on zone sensible load, reducing the amount of plenum air sent to the occupied space during cooling and nearly eliminating motor heat in the supply air stream (John et al. 2018). Unit fan allows night setback heating capability on a zone-by-zone basis without need to operate the air handler during unoccupied mode.

Limitations: Some states still allow permanent split capacitor (PSC) motors to be used. Such applications send excess plenum heat to the space in cooling. PSC motors are not efficiently modulated and are difficult to control when attempting to track part-load conditions.

Parallel Fan-Powered ATUs. *Benefits.* Make use of free heat in the form of return air for reduced heating requirement. Fan only runs during heating and dead band modes, and fan is sized to only handle heating airflow. Unit fan allows night setback heating capability on a zone-by-zone basis without need to operate the air handler during unoccupied mode.

Limitations: VAV turndown must be considered when selecting diffusers. Sudden changes in airflow cause noticeable changes in sound levels in the space, which is generally annoying to occupants. Casing and backdraft damper air leakage during cooling severely affects overall efficiency of these units (Sardoueinassab et al. 2018) and measurably limits the amount of heat to be reclaimed in the ceiling plenum.

Fan-Powered Chilled-Water ATUs with Sensible Cooling Coils. *Benefits:* Make use of free heat in the form of return air for reduced heating requirement. Typically provides constant air motion and sound level in the zone for improved air circulation and occupant comfort. Low inlet static pressure requirements can reduce fan energy at the air-handling unit. DOAS allows for reduced ventilation air volumes. Energy savings can be maximized with ECMs. Unit fan allows night setback heating capability on a zone-by-zone basis without need to operate the air handler during unoccupied mode.

Limitations: Units must be selected and operated to keep local cooling 100% sensible. Units should not be allowed to condense water on the coils.

Selection Considerations

Sizing ATUs. For primary air inlets, select an inlet size that meets the minimum and maximum airflows desired from the recommended primary airflow range table in the manufacturer's catalog. Selecting terminals near the top of their range increases velocity, sound, and air handler fan energy. Oversizing single-duct terminals reduces control accuracy and stability, and may result in higher sound levels. Transducer fidelity at the low end varies by manufacturer and must be considered in combination with the airflow sensor signal at the minimum airflow rate when selecting an inlet size. To maximize performance, size the terminal's maximum airflow limit for 70 to 85% of its rated capacity (approximately 2000 ft/min) in accordance with the catalog recommendations. For accurate control, the minimum setting guideline should not be lower than 400 ft/min inlet neck velocity for units using inlet velocity sensors. Other minimum guidelines may apply for units with specialty controls.

Select terminals based on recommended air volume ranges. A 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 near-closed 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 reliable 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.

Oversizing the discharge duct may create low static conditions, requiring the fan to operate outside its recommended operating range.

Oversizing terminals with electric heat can lead to insufficient total pressure, which can occasionally trip the airflow safety switch.

Environmental Factors. Environmental factors play an important role in system selection. They include the climate and air conditions both indoors and outdoors. They also include legislative requirements such as outdoor air ventilation rates and local building codes. When high ventilation rates are required, such as in critical hospital spaces and clean rooms, reheat is often required to maintain human comfort. Fan-powered ATUs are usually used in noncritical spaces where the thermal load changes significantly and heating is required. Single-duct ATUs are frequently applied in the interior, where the thermal load is normally stable.

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 ATUs. Usually, fan-powered ATUs 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 ATUs 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 ATUs.

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 may be served by single-duct or fan-powered ATUs with no heat. Unless the building is located in a mild heating climate, the perimeter zones require heat, typically electric or hot water. These are usually included with the ATUs, but sometimes separate heating systems are used (such as baseboard heat). The static pressure in the ducts should be lowered to the minimum pressure in accordance with ASHRAE *Standard* 90.1, which allows at least one VAV damper to remain near full open. Interior zones in these buildings can use fan-powered ATUs to keep the static pressure low. Buildings with parallel-type fan-powered ATUs usually use single-duct ATUs 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 ATUs are used on these systems, single-duct or bypass units are usually selected. A variation of this system, variable-volume variable-temperature (VVT), uses pressure-dependent single-duct units with a main bypass air valve in the supply duct. The bypass damper is regulated by static pressure in the supply duct. A near 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 depends somewhat on the size and type of building.

- **Electric** controls are pressure dependent where the air valve responds to a single control input. For example, the thermostat sends a signal to the air valve to open or close based solely on room sensible temperature.
- **Pneumatic** controls are usually used for building renovation or expansion where the base building already has a pneumatic system installed. They can be pressure dependent or independent. They 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 automation system. 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 automation system. These controls provide individual zone control and communication to the building management system.

Cost Factors. Consider costs before finalizing system selection. 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. Research local utility rates and building codes to arrive at the correct decision before making the final selection.

Acoustical Considerations. ATUs and room air distribution devices are typical sound sources. However, they are not the only equipment affecting room acoustics. See [Chapter 49](#) of this volume, Chapter 4 of ASHRAE (2017b), AHRI *Standard* 885, and other standards for guidance on space acoustics. Broadcast studios, theaters, libraries, and other acoustically sensitive applications require careful consideration, because equipment selection and location are important. Radio-frequency interference (RFI) and electromagnetic interference (EMI) should also be considered when designing broadcast studios.

Sound levels are affected by primary air valve and/or fan-generated sound. The maximum sound generated by a given primary air valve size is determined by the difference between the highest inlet static pressure and external static pressure at the design cooling airflow for a non-fan-powered ATU and parallel fan-powered ATUs. For series fan-powered ATUs, the highest air valve sound level usually occurs at the highest inlet static pressure; however, the fan usually sets the room noise levels for series ATUs. To determine fan noise levels, fan airflow (adjusted within its range by the speed controller) and external static pressure conditions are required.

Acoustical performance data are presented in formats for both the parallel and series ATUs, because their sequence of operation differs. With a parallel ATU, air valve and fan operation are evaluated separately. With a series ATU, air valve and fan are evaluated together. Series fan-powered ATU sound levels are more consistent compared to ambient background sound levels than those of the parallel fan-powered ATU, which has a cycling fan.

From the performance data, determine the sound power levels and predicted room noise criteria for discharge and radiated paths under the appropriate operating conditions. Use care, because some published room noise criteria are based on certain path attenuation assumptions that may not correspond to specific applications. To minimize these differences, AHRI *Standard* 885, Appendix E, provides specific values to apply to sound power levels for NC calculations. These may not match any specific space, but yield comparable NC values between different manufacturers. For a complete description of these processes, see AHRI *Standard* 885 and Chapter 4 in ASHRAE (2017b). Both of these sources provide guides to apply specific attenuation values for a specific space to predict actual room noise criteria. However, the attenuation values depend on the room furnishing and finish, as well as the sound power levels generated by the ATU.

It is necessary for the architect and the engineer to recognize all factors in the building specifications that affect sound attenuation. This is necessary to properly evaluate the room noise criteria and ensure the finished levels do not exceed the design goal in the occupied space. An ideal specification identifies maximum allowed discharge and radiated sound power by octave band, rather than just catalog based NC values.

A sound-sensitive occupied space (e.g., conference room, private office, music studio, concert hall, classroom) may require more discharge sound attenuation than less sound-sensitive spaces. Usually some internal lining in the discharge duct can accomplish this. Occasionally, a discharge attenuator or silencer may be attached to the ATU. Care must be taken to not increase the discharge pressure enough to increase fan speed, generating additional noise that may be greater than the attenuated value.

Installation and Operational Considerations

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 internal diameter as the inlet should be provided. Flex duct runouts at the ATU inlets are generally good attenuators.
- 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. Using flow-straightening or velocity equalizing 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 as best as possible. 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 effect 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 1000 ft/min. Flex duct runouts at the diffuser inlets are generally a good attenuator.

Compliance with Applicable Building 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 (e.g., fan-powered terminals, single-duct terminals with electric heat) should be checked for compatibility with source. Voltage, phase, and frequency must match.

Fan Interlocks. Typically, series fan-powered ATUs are designed to run continuously. Usually, they are energized only during occupied periods or when needed for emergency heating during unoccupied periods. Use care to interlock the unit fan with building's air handlers to ensure that the ATU fans start during occupied periods. Series-unit fans should be started prior to the air handler to prevent backflow into the plenum and backward rotation of the fan.

Fan Shift in Fan-Powered ATUs. 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, pressure drop and changes in local jets may cause the fan to shift its performance as it rides the fan curve. Consequences 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 ATU.

Avoiding Excessive Air Temperature Rise. ATUs 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 supply air temperature and room air temperature. Hart and Int-Hout (1980) and Lorch and Straub (1983) recommend a maximum Δt of 15°F 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 15°F requires an increase of 25% in the ventilation air per ASHRAE *Standard* 62.1. Absolute maximum discharge air temperature is 120°F. 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 ATUs 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 56](#)).

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 the following precautions:

- Avoid locating terminals near return air openings or light fixtures to decrease the potential of direct paths for radiated sound to enter the space without the benefit of ceiling attenuation.
- To avoid possible aerodynamic noise, keep airflow velocities below 1000 ft/min in branch ducts, and below 800 ft/min in runouts to air outlet devices.
- 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 air valve. Between the ATU and the air outlet, flexible duct can be more effective than lined duct at reducing ATU noise. Flexible duct can also generate sound if bends or sagging are present. Sometimes, flexible couplers can reduce vibration passed from the ATU to the duct connections.
- Select air valves in ATUs to operate at or below 2000 fpm or less. Larger inlets reduce velocity (and therefore noise) in low-pressure applications, but may increase noise in higher-pressure applications. For fan-powered ATUs, lower fan speeds generally produce lower sound levels, but care must be taken to ensure that the minimum airflows are met. Sound emissions are sometimes lower when fan-speed controllers are used to reduce fan rotational speed rather than using mechanical dampers to restrict airflow. AC induction motors can generate pure tones that are unacceptable when thyristor controls are used. Mechanical dampers will not reduce fan rotational speeds or sound levels when applied to systems with ECMs.
- When required, 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. Locating fan-powered series ATUs closer to the mechanical room increases the amount of heat and unvitiated air that can be reclaimed.
- Locate terminals in the largest ceiling plenum space available to maximize radiated noise reduction. Install ATUs at the highest practical point above ceiling to optimize radiated sound dissipation.
- When required, locate ATUs to allow use of lined discharge ductwork to help attenuate discharge sound.
- 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.

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. Fan-powered ATUs do not require filters, per section 307.2 of International Code Council's (ICC) *International Mechanical Code*® (ICC 2018) and ASHRAE *Standard* 62.1. Many fan-powered terminals are manufactured with construction filters that must be removed after construction is complete and are not suggested to be replaced.

Critical Environments. Some applications, such as cleanrooms and operating theaters, require high levels of reliability from ATUs 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, consider locating the equipment outside of

the clean space or using highly reliable, very-low-maintenance, very basic equipment and controls. ATUs may require access to internal components for cleaning in the case of contamination.

Fan Airflow Control of Fan-Powered Terminal Units

Fan-powered ATUs nearly always use single-phase motors, either ECMs, PSC motors with electronic fan speed control (sometimes called **wave choppers**, **thyristor controllers**, or **silicon-controlled rectifiers [SCRs]**), or PSC motors with three-speed switches combined with electronic fan speed control.

Electronically Commutated Motor (ECM). Currently, ASHRAE *Standard* 90.1 and ICC's 2021 *International Energy Conservation Code*[®] (IECC) require ECMs in fan-powered ATUs. ECMs provide superior controllability. They can be pressure-independent devices that calculate airflow, and can respond to a signal from a local controller or a remote signal from the building management system (BMS).

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**. Some units may suffer from large changes in amp draw that significantly affect the motor efficiency and operating characteristics. The PSC motor is a pressure-dependent device with a single setting.

ECM versus PSC in Parallel and Series Fan-Powered ATUs

ECMs provide significant energy savings over PSC motors, especially in part-load conditions (Edmondson et al. 2011). The part-load savings are primarily achieved by reducing airflow and taking advantage of the increased fan efficiency at below-design airflows. PSC motors have a limited amount of turndown available, but they also lose efficiency at or sometimes above the savings that can be achieved by the blowers. ECMs have a very small decrease in efficiency when they are operating at reduced speeds, and consequently they allow for nearly all of the improved blower efficiency to be captured in the operating system costs. PSC motor inefficiency manifests itself in the motor heat that is added to the supply air. Generally, PSC motors in fan-powered VAV ATUs can add 1 to 3°F to the airstream (Davis et al. 2007).

Nameplate Ratings. UL *Standard* 1995 covers fan-powered ATU nameplate ratings. This standard relates to equipment manufacturers and not field issues (which are covered in international and local codes). Nameplate ratings on the unit usually do not match the nameplate ratings on the motor. Amperage can be above or below the motor nameplate. Differences between the motor label and the unit label may be significant in some cases. Refer to the unit nameplate ratings and not the motor nameplate ratings when determining supply circuit requirements. These ratings are set at the safest possible condition. Static pressure and set points vary on each unit, so performance may not match what is on the unit nameplate.

Series. At typical design airflows, the ECM uses about 70% less energy than the PSC (Edmondson et al. 2011). At reduced airflows during part-load conditions, the overall energy savings is greater. Additionally, the reduced plenum air during cooling saves as much or more energy than the motor due to reduced cooling requirements.

Parallel. Default programming in digital controllers run the fan in the parallel ATU during dead band and heating. Because both conditions can suffer from overcooling in the occupied zone due to cold air supplying the required outdoor airflows, the PSC motor heat is used to augment the heat in the induced air to the zone. If ECMs are used, heat may need to be supplied from some other source to offset the overcooling potential during dead band and short cycling during heating. The heat source may be more efficient than the PSC motor heat generation, but overall energy savings is nearly negated by the motor heat loss.

Control Strategy

Series. Most direct digital controllers (DDCs) provide an optional output that may be used for controlling fan airflow by the BMS if an ECM is used. 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.

The sequence for **series constant volume** is as follows: the fan runs constantly at maximum design fan airflow during all occupied periods. On a call for cooling, the controls modulate the primary air valve toward maximum airflow, delivering primary air to the mixing chamber. If the fan is set at the same airflow as the primary air at maximum cooling, no air is induced from the plenum. If the fan is at a higher airflow than the primary air (e.g., as in a low-temperature application), air is constantly induced from the plenum.

As cooling demand decreases, the primary air valve modulates toward minimum airflow, reducing the flow of primary air into the mixing chamber. This increases the volume of warmer induced air into the mixing chamber. The increased percentage of induced plenum air causes the discharge temperature to rise to approach the plenum temperature, taking advantage of recaptured heat. [Figure 28](#) demonstrates operation of a typical series constant-volume ATU (Reid et al. 2016).

On a call for heating, 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.

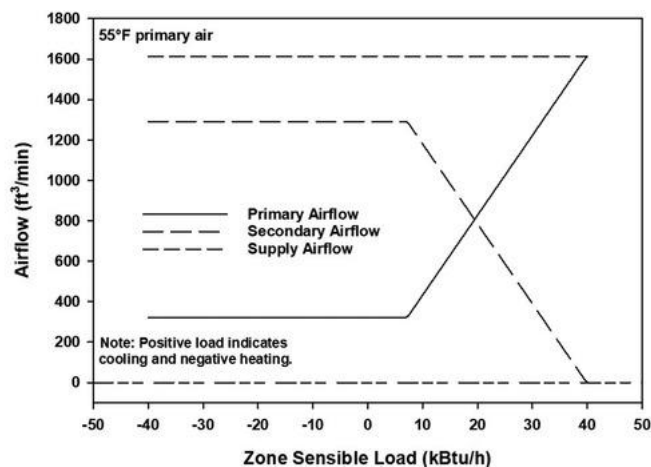


Figure 25. Typical Series Constant-Volume ATU

For **series variable volume**, the sequence is the same, except that the fan modulates.

As cooling demand decreases, the primary air valve and the fan modulate toward minimum airflow with the primary air valve reducing airflow more quickly than the fan.

On a call for heating, the fan modulates toward maximum heating airflow set point. On a further increase in heating demand, the controls energize the supplemental heat (optional equipment), which can be either electric or hot-water coils. [Figure 29](#) illustrates the operation of a typical series variable-volume ATU (Yin et al. 2018); see ASHRAE *Guideline* 36 for control sequence.

Parallel. In the heating and dead-band modes, the fan supplies a relatively constant volume of induced air 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. [Figure 30](#) shows the operation of a typical parallel constant-volume ATU (Yin et al. 2016).

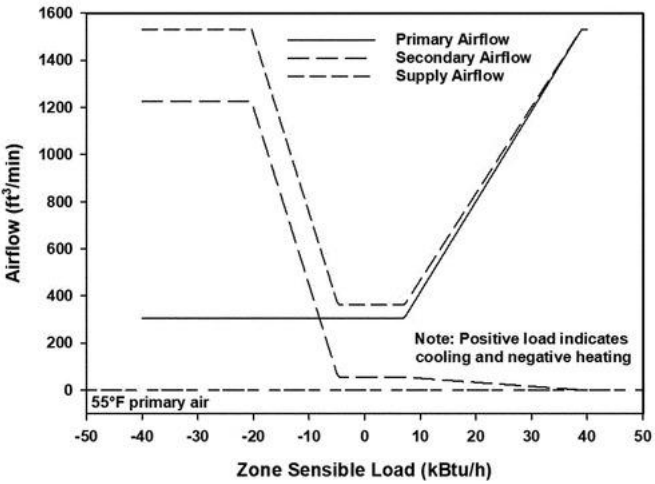


Figure 26. Typical Series Variable-Volume ATU

Parallel Variable-Volume Sequence. On a call for cooling, the controls modulate the air valve toward maximum airflow while the fan is off. Variable-volume, constant-temperature air is then discharged into the space. On a decreasing call for cooling, the sequence reverses.

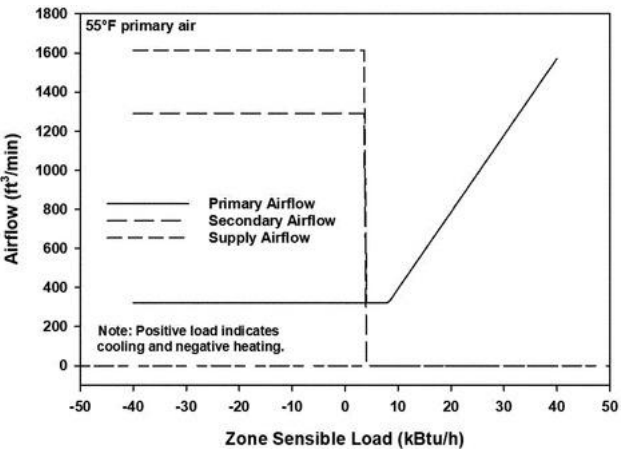


Figure 27. Typical Parallel Constant-Volume ATU

In dead band, 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 space.

On a call for heating, the controls automatically energize the supplemental heat (optional equipment), which can be either electric or hot-water coils. [Figure 31](#) shows the operation of a typical parallel variable-volume ATU (Yin and O’Neal 2018).

Energy Consumption

Fan-powered VAV ATUs 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. This recaptures much of the heat created in the zone and plenum. If additional heating is required, supplemental heat is added to the sequence, thereby complying with ASHRAE *Standard* 90.1 during the heating sequence. The unit saves energy by warming blended air, for example, at 72°F rather than reheating primary cooled air at 55°F, saving the cost of 17°F at the heating airflow. According to ASHRAE research project RP-1292 (Davis et al. 2007; Furr et al. 2007), there is very little difference in total building energy use between series and parallel units when both are equipped with permanent split-capacitor (PSC) motors. However, each type of unit consumes that energy in different ways. ASHRAE (2017b) explains these differences in greater detail.

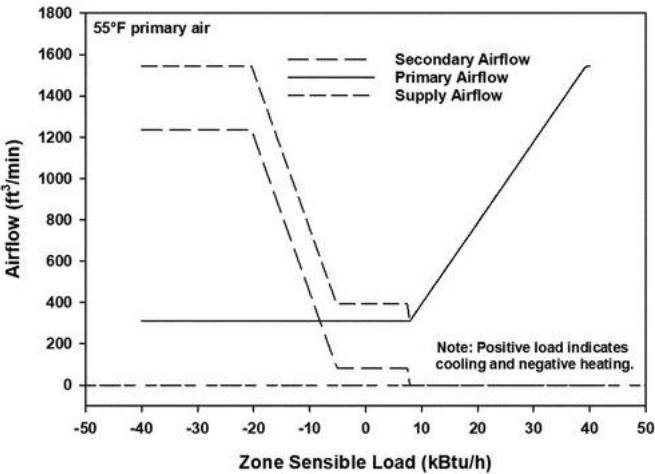


Figure 28. Typical Parallel Variable-Volume ATU

Series. 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. RP-1292 identified the PSC motor as the biggest energy user in a series unit (Davis et al. 2007; Furr et al. 2007). The fan energy raises the air temperature across the motor by 1 to 3°F. 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 (Davis et al. 2007; Furr et al. 2007). This means that total energy use can be reduced if the fan energy is reduced. Using ECMs can significantly increase motor lifetimes and provide significant

energy savings. Warm air induced from the plenum at part load conditions increases the cooling requirement because of the increased air temperature in the mixing chamber of the ATU. The combination of motor and plenum heat can equal or exceed the energy consumed by the PSC motor.

Because the fan cabinet is neutral or negative during operation, the casing leakage does not affect energy consumption of series units. All the primary air injected into the unit is 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.

Parallel. More energy is required at the air handler in a parallel ATU system due to the higher inlet static pressure requirement. When just one duct path is designated as the critical path for static pressure set point, system pressure (and thus energy usage) can increase compared to a building with series units. However, ASHRAE *Standard* 90.1 procedures reset the system static pressure to always drive at least one VAV air valve to its nearly full-open position, making the increased static pressure required for parallel units lesser.

According to RP-1292, the largest single energy usage is leakage in the parallel unit even with dynamic reset at the air handler (Davis et al. 2007; Furr et al. 2007). Leakage is typically between 5 and 12% of the total primary airflow through the parallel unit, and it is highest at full cooling. This does not account for other casing leakage like seams and penetrations. Thus, parallel units may need to be oversized to cover the total load to the occupied zone. Due to the leakage, ceiling plenums are cooler with parallel units than with series units, reducing or eliminating the amount of free heat that can be reclaimed in heating mode. Per ASHRAE (2017b), casing leakage was calculated at \$1.84/cfm per year. 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 ATU and the air handler during operation. Motor heat should also be included in heating mode calculation.

Since the motor runs in dead band and heating modes, any heat generated by the motor is delivered to the occupied space. Because of this, using more efficient motors (which run cooler) will not measurably improve the unit's energy consumption.

Inlet Static Pressure Requirements

Series. The inlet static pressure is the pressure required to push the conditioned air across the air valve into the mixing section.

Parallel. The inlet static pressure is the pressure required to push the conditioned air across the damper, through the mixing section, through the discharge duct, and across the diffuser(s) into the space.

Sizing Fan-Powered Terminals

Selection of fan-powered ATUs involves three elements: primary inlet, fan size, and heating coil. Selection of these elements and their interactive effects determine the overall performance of the units.

Primary Air Inlet Selection. See the section on Single-Duct Primary Air Inlet Selection. *Note:* casing leakage is an important issue for parallel fan-powered ATUs and must be taken into account when determining actual airflow supplied to and from the ATU.

Fan Size. Series fan-powered ATUs require the fan to be sized to handle the maximum design airflow. Fan airflow must be at least equal to primary airflow to ensure the mixing chamber in the terminal does not become positively pressurized, resulting in primary air short-circuiting into the ceiling plenum through the induction port(s). The external static pressure requirements are the sum of the ductwork and diffusers at design airflow plus an applicable hot-water coil or electric heater and optional filter, if required. When fan airflow and external static pressure have been determined, select the fan size from the fan curves provided by the manufacturer. Upsizing the fan and operating it at a reduced speed can result in quieter operation, but pay attention to fan stability at very low airflows.

Parallel fan airflow is determined by calculating the difference between the total discharge design heating airflow and minimum primary airflow. If minimum airflow is zero, then fan airflow is the heating airflow. In most applications of a parallel ATU, a minimum primary airflow is required to meet ventilation requirements. This primary airflow contributes to the total external static pressure experienced by the fan and should be accounted for along with all components, such as heaters, ductwork, diffusers, and optional filters.

Heating Coils

Heating coils on ATUs are generally either electric or hot water coils. Sometimes steam may be used. Electric heaters are always encased for safety. Water coils may or may not be encased.

Heating coils on single-duct ATUs are located on the discharge where the duct is attached. They are almost always not encased and are generally sized to be the same size as the discharge duct. If the ATU is internally lined, the internal diameter of the ATU will be smaller than the coil face area. Multiple outlet plenums or silencers may be mounted downstream of the coil. Sometimes the coils are mounted at the discharge end of the silencers. Access doors may be located upstream and/or downstream of the coils. Heating coils on single-duct ATUs are always reheat coils and must be designed to heat the air from the leaving air temperature at the air handler to the desired entering air temperature at the occupied space. Heating coils are usually either electric or hot water finned tube type.

Heating coils on series fan-powered ATUs are mounted on the discharge where the duct is attached. If the ATU is properly controlled, these coils provide supplemental heat and not reheat. Water coils may or may not be encased. Discharge ducts may have access doors to allow access to the coil. Coils should be designed to heat the mixed air in the ATU mixing chamber. The mixed air is a blend of the minimum primary airflow and the induced plenum air at the point where the heater is energized.

Heating coils on parallel fan-powered ATUs may be mounted on the discharge of the unit or at the induction inlet upstream of the fan. Heating coils on the induction port must be sized to heat the fan airflow to a temperature high enough to mix with colder primary air, achieving the required entering air temperature for the occupied space. The resulting air temperature must not exceed that of the motor(s) rated operating temperature. Coils mounted on the ATU discharge must be sized to heat the blended air from the mixing chamber to the required leaving air temperature. On ATUs with coils on their discharge, discharge ducts may have access doors to allow access to the coil. Coils may or may not be encased.

To avoid compromising zone ventilation effectiveness (ASHRAE *Standard* 62.1), the discharge temperature should not be more than 15°F warmer than the room temperature it serves.

Use manufacturers' catalog data or ASHRAE (2017b) for details on sizing coils.

Additional Fan Guidelines

When selecting a unit for a particular set of conditions, ensure that the air delivery is designed to meet the room's sound criteria and the system's static requirements. Specific sound data are provided by manufacturers for various airflow deliveries for each unit and should be the guiding factor in selecting unit sizes. Avoid selecting equipment near the maximum or minimum of the fan curves. Selecting fans at these points may limit flexibility for future changes. When designing air systems and using fan-powered ATUs, 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 to ensure proper discharge air temperatures and to protect against short circuiting of primary air into the plenum.

Special Applications

ATUs for Cold-Air Systems. These systems often involve ice storage and can provide supply air typically colder than 48°F, below that supplied by conventional systems. To reduce the chances of condensation on outer surfaces, keep the following in mind:

- Select insulating materials with vapor barriers to provide adequate thermal and moisture protection throughout the cold-air path. Internal and external insulation must be sufficient to prevent any outer exposed surfaces from dropping below the dew-point temperature of the surrounding environment.
- Single-duct, dual-duct, and parallel fan-powered ATUs have a higher potential for downstream condensation because they handle the coldest air during cooling operation.
- Series fan-powered ATUs have the lowest potential for condensation because, in most operations, they mix warmer return air with primary air (i.e., air delivered to the ATU through supply duct to satisfy all or part of ventilation, latent, and sensible load). Fan-powered series flow ATUs are sometimes used to mix return air during occupied mode to deliver supply air at more conventional temperatures to the occupied space. This may help to avoid occupant comfort issues caused by excessive vertical drop from air outlets, and possible condensation at outlet devices.

Several other design and operational considerations can also minimize the possibility of condensation and related issues:

- Ceiling plenum returns are recommended for cold-air systems. Return air circulating throughout the ceiling plenum can prevent uncontrolled humidity or temperature extremes that can occur in attic spaces of fully ducted systems.
- To control humidity and prevent condensation problems, it is always good practice to limit outdoor air infiltration. Design and construct buildings to operate under positive pressure, and limit the amount of uncontrolled outdoor air that infiltrates occupied and unoccupied spaces.
- Cold-air systems should follow start-up procedures that gradually reduce the supply air temperature. This prevents moisture trapped in new construction or previously unoccupied buildings from condensing and possibly causing damage to ceilings and other finished surfaces.

ATU Critical Environment Considerations. Hospitals, cleanrooms, and laboratories pose special challenges. Protective isolation spaces such as operating rooms, bone marrow transplant patient rooms, immunocompromised 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 9](#) for specifics on health care requirements, and [Chapter 19](#) for details on clean spaces. Hospital rooms and cleanrooms frequently require constant high ventilation rates, which tend to favor single- or dual-duct ATUs. Pressure-independent, variable-speed motor technology has led to the development of fan-powered pressurization units.

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

See Chapter 12 of ASHRAE (2017b) for more information on health care facilities.

7. ROOM FAN-COIL UNITS

Designers have various fan-coil systems to choose from when designing a building. Choosing which one to use depends on meeting the owner's needs for installation, application considerations, first cost, and cost of operation. The designer must consider performance, capacity, reliability and spatial requirements and restrictions. The following guidelines describe different types of fan-coil equipment and their general uses, restrictions, and limitations.

Principles of Operation

Fan-Coil Types. *Vertical Stack:* A vertical stack fan-coil uses a fan and a water coil to condition air within a space by regulating tempered water flow through the coil. Designed for free-blow or ducted, concealed, or painted cabinet applications. This fan-coil model usually uses a riser piping system to deliver the chilled and hot water to the coil. This type of unit is typically used in high-rise hotels, condominiums, dormitories, and residential buildings.

Vertical: A standard vertical fan-coil uses a fan and a water coil to condition air within a space by regulating tempered water flow through the coil. These units either are concealed in the wall or have a painted metal cabinet built around them. This type of fan-coil is commonly used in hallways, dormitories, small apartments, etc.

Horizontal (Blow-Through): A horizontal unit is normally mounted overhead and contains a fan and a water coil to condition air in a space by regulating tempered water flow through the coil. The fan blows air into the coil and then discharges into the space. These units may be built in or have decorative housings installed.

Horizontal (Draw-Through): A horizontal unit is normally mounted overhead and contains a fan and a water coil to condition air in a space by regulating tempered water flow through the coil. The fan draws air across the coil and then discharges into the space. These units may be built in or have decorative housings installed.

Water Piping Distribution Systems. For fan-coil units requiring chilled and/or hot water, the piping arrangement determines the performance quality, ease of operation, operating cost, and initial cost of the system.

Two-Pipe Changeover Without Electric Heat. In this system, either hot or cold water is supplied through the same piping. The fan-coil unit has a single coil. The most common system change-over scheme with the lowest initial cost is the two-pipe changeover scheduled by outdoor temperatures. The outdoor changeover temperature is set at some predetermined set point. If a thermostat is used to control water flow, it must reverse its action depending on whether hot or cold water is available.

The two-pipe system cannot simultaneously heat and cool, which is required for most projects during intermediate seasons when the morning hours may need heat and the afternoon hours likely require cooling. In intermediate seasons when the boiler is not on, two-pipe systems without supplemental electric heat cannot meet heating demands.

Two-Pipe Changeover with Partial Electric Strip Heat: This arrangement can provide electric heating in some zones and cooling in other zones in intermediate seasons by using a small electric strip heater in the fan-coil unit. The unit can handle heating requirements in mild weather while continuing to circulate chilled water to handle any cooling requirements. When the outdoor temperature drops sufficiently to require heating beyond the electric strip heater capacity, the water system must be changed over to hot water. The designer should consider the disadvantages of the two-pipe system carefully; many installations of this type increase operational cost, and can be unsatisfactory if climates temperatures vary widely from morning to afternoon.

Two-Pipe Nonchangeover with Full Electric Strip Heat: This system should be closely evaluated due to the escalated operational cost of electric heat compared to hot-water heat. It may be practical in areas with small heating requirements.

Three-Pipe Distribution: Three-pipe distribution uses separate hot- and cold-water supply pipes. A common return pipe carries both hot and cold water back to the central plant. The fan-coil unit control introduces hot or cold water to the common unit coil based on the need for heating or cooling. This type of distribution is not recommended because of its energy inefficiency and does not comply with most recognized energy codes.

Four-Pipe Distribution of Chilled and Hot Water: This system has dedicated supply and return pipes for chilled and hot water. It generally has a high initial cost compared to a two-pipe system. Four-pipe systems have better performance because of all-season availability of heating and cooling at each unit, no summer/winter changeover requirement, and simpler operation. It can be controlled to maintain a dead band between heating and cooling during shoulder season.

Variable-Refrigerant-Flow (VRF) Systems. The zone side of these systems is a fan-coil unit. Further information is available in [Chapter 18 of the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#).

Benefits and Limitations

Vertical Stack. *Benefits:* These units take advantage of the multi-story feature to plumb the water piping vertically floor to floor to conserve space.

Limitations: These units occupy floor space in the room. Because the units are traditionally located in the space, noise can be a problem if the fan and motor are not sized properly or if noise is not considered during design.

Vertical. *Benefits:* These units traditionally condition air directly in the space. Return air is typically drawn in the bottom and discharged out the top, directly back into the space.

Limitations: These units occupy floor space in the room. Because the units are traditionally located in the space, noise can be a problem if the fan and motor are not sized properly or if noise is not considered during design.

Horizontal (Blow-Through). *Benefits:* These units do not occupy floor space in the room, and can be manufactured with a lower profile.

Limitations: The velocity profile may not be evenly distributed across the cooling coil.

Horizontal (Draw-Through). *Benefits:* These units do not occupy floor space in the room, and generally have evenly distributed airflow across the cooling coil, resulting in greater heat transfer and lower noise.

Limitations: Any condensate blow-off caused by airflow over the coil has the potential to land on electrical components, including the fan motor.

Selection Considerations

Sizing Fan-Coil Units. How fan-coil units are selected and how they interact with the rest of the system determine the overall performance of the units.

General Sizing Guidelines: Select fan-coil based on recommended air volume ranges. A common misconception is that oversizing a fan-coil makes the unit operate more efficiently. On the contrary, an oversized fan-coil reduces run times, making the room air stratified and humidity more difficult to maintain at comfortable levels.

The recommended selection for maximizing performance is to size the fan-coil maximum airflow limit for 70 to 85% of its rated capacity in accordance with the catalog recommendations.

Do not exceed 500 ft/min air velocity through the coil section of a chilled-water fan-coil. Exceeding this limit could cause condensate blow-off from the fin material, potentially causing damage to surrounding materials and reduces the latent cooling capacity of the unit.

Select water flow rates that will not exceed a velocity of 8 ft/s in the coil tubing, because this velocity can erode the copper and cause pinholes, which could result in water leakage and potential damage to material and property.

Fan Size Selection: Fan-coil units require the fan to be sized to handle the maximum design airflow. The external static pressure requirements are the sum of the ductwork, diffusers, and filters at design airflow. When fan airflow and external static pressure have been determined, select the fan size from the fan curves or the selection software provided by the manufacturer. Vertical fan-coils with one return and one supply grille integral to the cabinet have no appreciable external static pressure requirements. This style of vertical fan-coil must deliver the design airflow for the application between 0 and 0.05 in. of water. Oversizing the fan and operating it at a reduced speed can result in quieter operation, but pay attention to fan stability at very low airflows.

ASHRAE research project RP-1741, Understanding Fan Coil Components and How They Relate to Energy Consumption and Energy Modeling, analyzed energy modeling of fan-coils. RP-1741 benchmarked the energy savings from air and water modulation control relative to conventional on-off control of hydronic fan-coil units. Similar to other HVAC equipment, hydronic fan-coil units are generally oversized to be able to meet peak cooling and heating loads. Consequently, fan-coil units rarely operate at their full load for an extended time. Running fans and pumps at part-load conditions could allow energy savings. The modulation control of air and water flow based on thermal loads can take advantage of oversized equipment and operate them at part-load conditions for energy savings. Building energy simulation conducted in RP-1741 showed that modulation control could save over 50% of the fan-coil units' fan energy compared with cycling on and off. Modulation control can also maintain an higher chilled-water ΔT at part-load conditions, leading to over 50% savings in pump energy and more efficient chiller staging. The total HVAC system savings ranges from 5 to 15% compared with the conventional on-off control.

Environmental Factors: Environmental factors include the climate and air conditions, indoors as well as outdoors. They also include legislative requirements such as outdoor air ventilation rates and local building codes. Fan-coil units receive ventilation air from a penetration in the outer wall or from a central air handler. Units that have outdoor air ducted to them from an aperture in the building envelope are not suitable for commercial buildings because wind pressure allows no control over the amount of outdoor air admitted. Ventilation rates can be affected by stack effect and by wind direction and speed. Also, freeze protection may be required in cold climates.

Fan-coils are, however, often used in residential construction because of their simple operation and low first cost, and because residential rooms are often ventilated by opening windows or by outer wall apertures, if not handled by a central system. Operable windows can cause imbalance in a ducted ventilation air system.

When outdoor air is introduced from a central ventilation system, it may be connected to the inlet plenum of the fan-coil or introduced directly into the space. If introduced directly, ensure that this air is pretreated, dehumidified, filtered, and held at the room’s temperature so as not to cause occupant discomfort when the fan-coil unit is off. One way to prevent air leakage is to provide a spring-loaded motorized damper that closes off ventilation air when the unit’s fan is off.

Building Size: In large buildings, where a central air handler(s) is not practical and where individual power costs are the responsibility of each space owner, a fan-coil is generally the system of choice for temperature and humidity control. These systems require a central water supply and a water transport system. If the initial cost and cost of operation of central chilled- and hot-water-generating equipment make more economical sense, the hydronic fan-coil comfort system is a good choice.

Cost Factors: Costs should be considered before the final system selection is made. Installation, operation, and maintenance all contribute to total cost. Sometimes one of these costs is more important than others. Electric heaters usually have a lower installed cost than hot-water coils, but may have a higher operating cost. Local rates and codes should be researched to arrive at the correct decision before making the final selection. See [Chapters 13 and 46 of the 2020 ASHRAE Handbook—HVAC Systems and Equipment](#), [Chapter 22 of the 2021 ASHRAE Handbook—Fundamentals](#), and [Chapter 51](#) of this volume for more information on piping systems.

Acoustical Considerations. Since fan coils are often located within the occupied space, extra consideration should be given to avoid generating noise near occupants. For fan coils located outside the occupied space, the acoustical considerations are very similar to those of terminal units. Refer to the Acoustical Considerations portion of the section on Terminal Units for more information.

Installation and Operational Considerations. Space Restrictions. During the design phase, ensure the fan-coils are located for ease of installation, optimum performance, and maintenance accessibility.

Optimizing Inlet Conditions. Fan-coils should be located to optimize the inlet return air and discharge grilles or duct work. Proper location and size of grille and duct contribute to the overall effectiveness of air mixing and efficiency.

Locating a return grille in an area that does not properly serve the entire conditioned space can lead to poor performance of the coil and hot or cold spots within the occupied space.

Zoning Requirements. Correctly sizing fan-coils with regard to the physical conditions of the occupied space is vital to ensure acceptable performance. One large fan-coil serving a space with divided work areas may result in the single thermostat only providing acceptable temperature control where the thermostat is located. The other space(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 effect on pressure drop. Avoid installing tees, transitions, or elbows close to the unit discharge. Avoid long runs of flex duct, and keep short flex runs as straight as possible. Make curves as shallow as possible, and ensure entrance condition to diffuser outlet is straight. Discharge ducts should be designed for a maximum velocity of 1000 ft/min.

Noncompliance with Local Electric Codes. Some local jurisdictions have more exacting codes than the minimum requirements of national codes and standards such as IBC, the National Fire Protection Association’s *National Electrical Code*® (NEC®; NFPA Standard 70), UL, and Canadian Standards Association (CSA) standards. One example is the primary fusing required of the power circuit in some areas.

Power Source Compatibility. Fan-coils 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 (4 wire wye) power source.

Fan Interlocks. When an electric heat source is used, a fan interlock is required to ensure the fan is on during the call for heat.

Avoiding Excessive Air Temperature Rise. Fan-coils with electric heaters or hot-water coils should be designed to satisfy load conditions, but attention should be paid to the temperature differential Δt between the supply air and room air. Exceeding Δt of 15°F requires an increase of 25% in the ventilation air per ASHRAE *Standard* 62.1. Absolute maximum discharge air temperature is 120°F per UL standards. Although this temperature will probably keep the equipment on line, it will probably not provide comfortable temperatures in the space.

Correctly Supporting Fan Coils. Fan-coil units should always be independently supported and secured to building structure, using the proper isolation mounting. Be careful not to block access panels with straps, thread rods, or trapeze supports. Where codes require seismic restraints, be sure to comply with all building and local codes.

Minimize 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

Acoustic Design and Installation. To help ensure an acceptable room noise level in the occupied space, engineers can minimize the sound contribution of a fan-coil by taking into account several design considerations and by using the following guidelines for good design practice:

- Design systems to operate at low (minimum) external static pressure. This reduces the generated sound level and provides more energy-efficient operation. Excessive static pressure generates noise.
- Between the fan-coil unit and the air outlets of a ducted system, flexible duct can be more effective than lined duct at reducing unit noise. Flexible duct can also generate sound if bends or sagging is present. Sometimes, flexible couplers can reduce vibration passed from the fan-coil unit to the duct connections.
- Select fan-coils to operate toward the middle area of their operating range. Larger return air inlets reduce velocity across coils and hence, noise. Lower fan speeds produce lower sound levels. Sound emissions can be lower when using an ECM motor blower combination.
- Whenever possible, locate horizontal fan-coils 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 fan-coils in the largest ceiling plenum space available to maximize radiated noise reduction. Install fan-coils at highest practical point above ceiling to optimize radiated sound dissipation.
- Avoid locating fan-coils 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 fan-coils to allow the use of lined discharge ductwork to help attenuate discharge sound.
- In large spaces, consider using a larger number of smaller air outlets to minimize outlet generated sound. Insulated flexible duct on diffuser run-outs provides excellent attenuation performance.
- Using ceilings with a high sound transmission loss classification will help reduce radiated sound.

Control of Fan Coil Units

Fan Airflow Control of Fan-Coil Units. Three-speed permanent split capacitor (PSC) motors are the standard motor option for most fan-coils. These motors have high, medium, and low speeds. These motors are likely be phased out because of minimum efficiency requirements in ASHRAE *Standard* 90.1.

The availability of **electronically commutated motors (ECMs)** for fan-coil units is rapidly growing, and most manufacturers offer a model. These motors often provide significant energy savings, improved controllability, and wider operating range in fan-coil units.

ECM versus PSC Motors in Fan-Coil Units. Three-Speed PSC: When a PSC motor is used in a fan-coil, the fan motor is switched on and off by a thermostat. This control scheme does not allow precise control of temperature and humidity levels in the conditioned space, and can cause comfort issues and increased energy cost.

Multispeed ECM: When a multispeed ECM is used in a fan-coil, the fan motor is switched on and off by a thermostat. The control scheme can be identical to that of a PSC but with the added benefit of a higher-efficiency motor, thus lower operating cost. Generally, an ECM is a 24 V control, and the motor can be energized directly from the thermostat or through a specially designed ECM fan control board, which could offer additional features.

Fully Modulating ECM: When used in the fan-coil with a more sophisticated control (e.g., analog control or DDC), this sequence of operation can provide a higher level of comfort and energy savings. Typically, an analog signal (0 to 10 V DC) from a controller is used to remotely vary the speed of the ECM. The dead-band differential between no call for heating/cooling and the energized state is generally much smaller in this type of control scheme. The water flow can also be modulated along with airflow modulation to avoid overcooling or overheating. This approach provides minimal swings in temperature and humidity levels in the conditioned space, which can provide superior comfort and reduced energy costs.

Table 10 Applications for Fan-Coil Configurations

| | Single Room | Multiple Room | Multifamily Low Rise | Multifamily High Rise | Office Buildings | Condominiums/Hospitality | University Dormitories | Schools |
|--------------------|-------------|---------------|----------------------|-----------------------|------------------|--------------------------|------------------------|---------|
| Vertical | | | | | | | | |
| Low profile (sill) | • | N | • | • | N | • | • | • |
| Stack units | • | • | • | P | N | P | • | N |
| High capacity | • | P | • | • | • | • | • | • |
| Horizontal | | | | | | | | |
| Low profile | | | | | | | | |

| | | | | | | | | | |
|--------|---------------|---|---|---|---|---|---|---|---|
| return | Free/plenum | • | N | • | • | • | • | • | • |
| | Exposed | • | N | N | N | N | N | N | • |
| | High capacity | | | | | | | | |
| return | Free/plenum | • | • | • | • | • | • | • | • |
| | Exposed | • | • | N | N | N | N | N | N |

P = Preferred for this application.
• = Used for this application.
N = Not recommended for this application

Basic Control Types. *Electric Thermostat:* Traditionally, the most common control type for fan-coils is a room thermostat. These thermostats come in both line voltage (i.e., same voltage as motor) and 24 V, which require a Class 2 step-down transformer (typically provided with the fan-coil).
When using a line voltage thermostat, the blower motor speed taps (three-speed) are connected directly to the thermostat. In this configuration, the water control valve is also line voltage. This type of thermostat is limited, and is generally used only on smaller motors because of ampacity limitations and thermostat ratings. In much of the United States, these thermostat leads must be run in a conduit between the fan-coil and thermostat.
When using a 24 V thermostat, fan relays, rated for the current of the motor, are used to switch each speed of the motor on and off. In this configuration, the water control valve is also typically 24 V. In most of the United States, these thermostat leads are not required to be run in a conduit between the fan-coil and thermostat.
Analog Stand-Alone: Another control type for fan-coils is an analog control combined with a room sensor. This control type is powered with 24 V thermostats, which require a Class 2 step-down transformer (typically provided with the fan-coil).
An analog controller is used when the fan-coil has modulating components such as an ECM and modulating water valves. The analog controller provides the appropriate outputs, based on room demand, to modulate the fan and water valves, if equipped.
This type of control can provide improved comfort over a simple electric thermostat. Depending on the features of the analog controller, the cost is generally higher than a room thermostat without set back features.
Direct Digital Control (DDC): This control type, paired with a room sensor, is powered with 24 V; these thermostats require a Class 2 stepdown transformer (typically provided with the fan-coil).
A DDC controller can be used for both types of fan-coil sequences described previously.
A DDC has both digital and analog inputs and outputs. The controller is programmable to provide many operation sequences, depending on the application. Generally, these controls can provide anything from a simple control sequence to a more sophisticated application. In some cases, DDC can control features and components outside the fan-coil, such as humidifiers, baseboard heat, bathroom ventilators, or lights, and can be programmed for occupied and unoccupied modes for energy savings.
Some DDCs can communicate through building automation systems (BASs) in several different protocols, such as BACnet™, Lonworks™, and Modbus®.
This type of control can provide improved comfort over a simple electric thermostat or stand-alone analog controls. Depending on the features, cost for a DDC is typically higher than for a room thermostat or analog controls.

Basic Control Sequences. *Three-Speed Motor with Two-Position Valves:* Thermostats are available with many features beyond the purview of this explanation, but the simplest sequence is as follows.
On a call for cooling, the fan is energized at the speed (high, medium, or low) selected by the occupant or by the thermostat. The chilled-water control valve opens fully. On reaching the room set point, the fan de-energizes and the chilled-water control valve closes.
On a call for heating, the fan is energized at the speed (high, medium, or low) selected by the occupant or by the thermostat. The hot-water control valve opens fully. On reaching the room set-point, the fan de-energizes and the hot-water control valve closes. For a fan-coil with electric resistance heat, a heater control, like a contactor or heater sequencer, is energized on and off by the thermostat based on room demand for heating.
Consult the manufacturers' catalog or installation and operating instructions for specific applications.
Modulating Motor with Modulating Valves: Controllers for fan-coils with modulating components are available with many additional features, but a basic sequence is as follows.
On a call for cooling, the chilled-water valve opens to a position consistent with the controls algorithm based on room demand, and the fan modulates toward the maximum cooling airflow set point. At a full call for cooling, the chilled-water valve is wide open and the fan is at the maximum cooling airflow set point. When approaching the room set point, the fan airflow reduces and the chilled-water control valve modulates toward the closed position. On reaching the room temperature setting, the valve closes completely and the fan modulates to the dead-band fan airflow set point.
Dead Band: Fan is off or running at a minimum airflow to circulate air through the space. On a call for heat, the hot-water valve opens to a position based on room demand, and the fan modulates toward the maximum heating airflow set point. At a full call for heating, the hot-water valve is wide open and the fan is at the maximum heating airflow set point. When approaching the room set point, the fan airflow reduces and the hot-water control valve modulates toward the closed position. Similar to the case of a fan-coil with electric resistance heat, a heater control, like a contactor or heater sequencer, is energized on and off by the room demand for heating. An SCR heater control can modulate the resistance heater by an analog output from the controller similar to the hot-water valve operation described above. [Figure 32](#) shows the operation of a typical fan-coil unit with hydronic cooling and electric heating in modulation control.

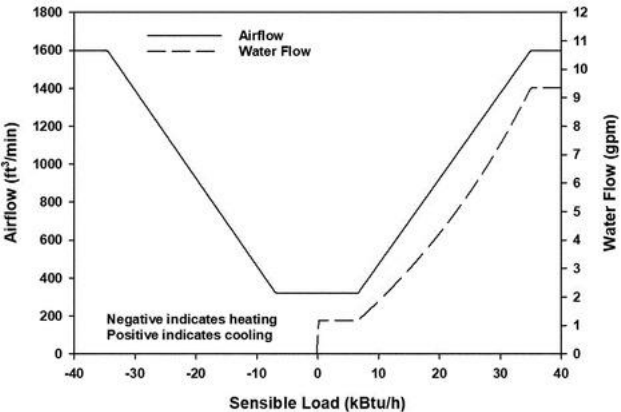


Figure 29. Typical Fan-Coil Unit with Hydronic Cooling and Electric Heating in Modulation Control

Consult the manufacturers' catalog or installation and operating instructions for specific needs.

Building Type

The designer must consider the intended building use when determining the type of fan-coil equipment to be used. Fan-coil systems are best applied where individual space temperature control or cross-contamination prevention is needed. Suitable applications are shown in [Table 10](#).

8. HEATING AND COOLING COIL SELECTION

Sensible Cooling and Heating Coil Selection

First, determine the air temperature rise by calculation using the heat transfer equation:

$$q = 1.085Q \Delta t$$

where

q = sensible coil capacity, Btu

(16)

Q = coil airflow rate, cfm

Δt = supply air temperature (SAT) – coil entering air temperature (EAT), °F

$$EAT = (T_1 Q_1 + T_2 Q_2) / Q_T \quad (17)$$

where

EAT = entering air temperature of coil, °F

T_1 = return air temperature, °F

T_2 = primary or ventilation air temperature, °F

Q_1 = return airflow rate, cfm

Q_2 = primary or ventilation airflow rate, cfm

Q_T = total airflow rate moved by fan, cfm

$$LAT = (T_1 Q_1 + T_2 Q_2) / Q_T \quad (18)$$

where

LAT = leaving air temperature of unit, °F

T_1 = coil leaving air temperature, °F

T_2 = primary or ventilation air temperature, °F

Q_1 = coil airflow rate, cfm

Q_2 = primary or ventilation airflow rate, cfm

Q_T = total airflow rate moved by fan, cfm

The supply air temperature (SAT) to the space equals the leaving air temperature (LAT) for the terminal unit or fan-coil. The terminal unit or fan-coil LAT and airflow rate should be known based on the load calculation. The coil EAT can be calculated using [Equation \(17\)](#) for all units mixing air upstream of the coil, or it should be equal to the return air temperature for all units mixing air downstream of the coil.

Once the coil EAT has been determined, the coil capacity can be calculated using [Equation \(16\)](#).

In applications for units mixing air downstream of the coil, the LAT of the unit can be calculated using [Equation \(18\)](#).

For electric heat, the capacity must be converted from Btu/h to kilowatts for selection. The required kilowatts and number of steps/stages desired should be checked with the manufacturer.

For hot-water coils, refer to the capacity charts in the manufacturer's performance data or selection software to select the appropriate coil.

In heating applications, it is good practice to limit supply air temperature to 15°F above room temperature.

In applications in which mixing occurs downstream of the coil, heat generated by the water coil shortens the motor life and may cause nuisance tripping of the motor thermal overload.

At part-load conditions, it may be desirable to modulate airflow through the terminal unit or fan-coil as well as the heat output to maintain an acceptable discharge air temperature. This can be done with modulating valves on coils or proportional control 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.

Example 3. Parallel Terminal with Discharge Hot-Water Heat. Select a unit inlet for a maximum/minimum primary airflow at 1000/250 cfm with 1 in. of water inlet static pressure.

The heating airflow required is 600 cfm at 103°F. Downstream resistance at 1000 cfm is 0.4 in. of water. Zone design heat loss is 20,181 Btu/h, design room temperature is 72°F, return air temperature is 75°F, and primary/ventilation air temperature is 55°F.

Solution:

Air Valve Selection. Based on a good design inlet velocity of 2000 ft/min, choose a 10 in. inlet.

Fan Selection. Fan heating airflow = Total heating airflow (600 cfm) – Primary airflow (250 cfm) = 350 cfm. The downstream static pressure the fan must overcome is the fan airflow plus primary airflow (600 cfm), and because this is less than maximum design airflow (1000 cfm), fan downstream static pressure = $(600/1000)^2 \times 0.4 = 0.144$ in. of water. Refer to fan curves to select the proper unit. The correct unit will handle 350 cfm at 0.144 in. of water 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. The heating coil is on the unit discharge in this example, so the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of return and minimum primary air. Using [Equation \(16\)](#), calculate the coil EAT.

$$EAT = [(350 \text{ cfm} \times 75^\circ\text{F}) + (250 \text{ cfm} \times 55^\circ\text{F})] / 600 \text{ cfm} = 67^\circ\text{F}$$

The coil capacity can be calculated using [Equation \(1\)](#),

$$q = 1.085 \times 600 \text{ cfm} \times (103^\circ\text{F} - 67^\circ\text{F}) = 23,436 \text{ Btu/h}$$

From the hot-water coil data, select a two-row coil at 600 cfm to provide 23,436 Btu/h at about 1.0 gpm.

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 87°F 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 considered to increase the mixed-air temperature.

Note 3: Using a PSC motor adds 1 to 3°F to the airstream.

Example 4. Series Terminal with Electric Heat. Select a unit to supply a constant 1500 cfm with 0.5 in. of water inlet static pressure. Minimum primary airflow is 375 cfm and downstream resistance caused by ductwork and diffusers is 0.4 in. of water. Zone design heat loss is 45,000 Btu/h, design room temperature is 72°F, return air temperature is 75°F, primary air temperature is 55°F, and supply air temperature is 100°F.

Solution:

Air Valve Selection. Based on a good design inlet velocity of 2000 ft/min, choose a 12 in. 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 1500 cfm at 0.4 in. of water and falls in the middle of the fan range as recommended in the section on Fan Size.

Heating Coil Selection. The heating coil is on the unit discharge, so the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of return and minimum primary air. Using [Equation \(16\)](#), calculate the coil EAT.

$$EAT = [(1125 \text{ cfm} \times 75^\circ\text{F}) + (375 \text{ cfm} \times 55^\circ\text{F})] / 1500 \text{ cfm} = 70^\circ\text{F}$$

Coil capacity can be calculated using [Equation \(1\)](#).

$$q = 1.085 \times 1500 \text{ cfm} \times (100^\circ\text{F} - 70^\circ\text{F}) = 48,825 \text{ Btu/h}$$

From the manufacturer's catalog, select an electric heater with the proper input voltage (120, 208, 240, 277 or 480 V electric coil) that could be available with a variety of stages.

Note 1: Although there are air-side pressure drop data for electric heaters 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 87°F should be used. Airflow should be increased if SAT is above the recommended 15°F Δt .

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.

Note 4: Using a PSC motor adds 1 to 3°F to the airstream.

Note 5: Reference manufacturer's recommendations for maximum temperature on electric heat to reduce the likelihood of nuisance tripping.

Total Cooling Coil Selection

Most manufacturers provide coil performance data and/or selection software. Coil capacity depends on coil design characteristics and the properties of the entering air and water. The enthalpy increase or decrease of the water through the coil multiplied by the airflow is the resultant capacity. The water temperature and the flow rate determine the inlet enthalpy; the airflow and coil characteristics determine the outlet enthalpy. The sensible to latent ratio is important to maintain occupant comfort. In the cooling mode, the resultant capacity should be the appropriate combination of sensible capacity for dry-bulb temperature control and latent capacity for humidity control. When these two components of capacity are met, the space being conditioned can stay in control and provide comfort for the occupants.

Do not exceed 500 ft/min through the coil section. Exceeding this limit could cause condensate blow-off, potentially causing damage to surrounding materials.

Select water flow rates that will not exceed a velocity of 8 ft/s within the coil tubing; this velocity can erode the copper and cause pin holes, resulting in water leakage and potential damage to material and property.

Example 5. Vertical Fan-Coil (Cooling Only). Select a unit at 400 cfm with a total capacity of 12,000 Btu/h, with 9000 Btu/h sensible capacity and 3000 Btu/h latent capacity. The entering air is 75°F db and 63°F wb. Consulting the manufacturer's coil capacity data, a three-row coil is adequate to provide the capacity at 4.0 gpm. The resultant sensible and latent capacities are met as follows:

Rating Conditions:

Entering air db: 75°F Entering air wb: 63°F

Airflow rate: 400 cfm Water flow rate: 4.0 gpm

Fluid temp. EWT: 45°F

Performance:

Total capacity: 12,519 Btu/h Sensible capacity: 9446 Btu/h Latent capacity: 3072 Btu/h

Leaving db: 53.2°F Leaving wb: 52.4°F Fluid pressure drop: 8.8 ft of water

Face velocity: 274 ft/min Air pressure drop: 0.16 in. of water

Fluid temperature out: 51.2°F Fluid velocity: 3.4 ft/s

9. CHILLED BEAMS

Principles of Operation

An **active chilled beam** is an air diffusion device that introduces conditioned air to the space for ventilation and temperature control purposes. Primary air, conditioned at the air handling unit to meet the ventilation and latent requirements of the space, is delivered through a series of nozzles, creating induction of room air through a unit-mounted sensible heat transfer coil. This primary air often also contributes to sensible cooling of the space and drives the induction function through the coil. Depending on their 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. When heating is required, warm water can be circulated through the coil.

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. On contact with the beam's integral sensible heat transfer coil, this air is cooled and falls back into the space. Primary air must be delivered to the space via a separate system for the purposes of ventilation and dehumidification.

Application Considerations

Chilled-beam systems must be designed to treat sensible and latent space heat loads, provide adequate space ventilation, conform to space acoustical requirements, 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 with beam coils being responsible for most of the sensible load in the zone and the primary air satisfying ventilation and latent load.

Benefits and Limitations

Benefits. Heat extraction or addition by the coil often allows for significant reduction in primary airflow requirements over all-air systems. Energy to transport cooling and/or heating media can be significantly 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 may make it possible to reduce the floor-to-floor height of a multistory building.

Water-side economizer opportunities may be extended as a result of the higher beam chilled-water temperatures, and provide an improved selection of available system options (e.g., geothermal, dry coolers, closed-circuit fluid coolers). Chilled-beam systems may offer opportunities to enhance chiller efficiencies and provide broader evaporator ranges.

Chilled beams require minimal maintenance. Vacuuming the beam coils is occasionally required and is typically guided by the needs of the space. Often, it is expected that service intervals could extend to three to five years. The lack of moving parts in chilled beams produces an inherently highly reliable system.

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

- Environments with moderate to high sensible heat ratios, such as offices, hotels, and other spaces with significant imbalances between sensible loads and ventilation requirements
- Heat-driven laboratories where ventilation (100% outdoor air) requirements are relatively low (4 to 8 ach) but sensible gains are often 40 to 60 Btu/h·ft², resulting in supply airflow rates in all-air systems that are significantly higher (12 to 18 ach).
- Hospital patient rooms where ventilation requirements severely limit the turndown ratio with all-air (VAV) systems, and thus require significant amounts of parasitic reheat to balance the cooling delivery with the actual room cooling demand. See ASHRAE *Standard* 170 for additional detail.
- Classrooms where outdoor air ventilation rates are significantly lower than the space sensible cooling requirements. Chilled beams typically provide at least half of the space sensible heat removal by way of their chilled-water coil, thus allowing ducted airflow rates to be reduced to a level near the classroom ventilation rate. This facilitates the use of dedicated outdoor air (DOAS) units for classroom applications.
- Retrofits, because of minimal mechanical space requirements, and in cases of suitable envelope construction.
- Passive chilled beams can provide a very efficient means of perimeter-area temperature control when coupled with underfloor air distribution (UFAD) systems.

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 inappropriate for the following applications/spaces:

- Environments with high latent gains such as kitchens, bathrooms, or locker rooms
- Natatoriums and sauna areas
- Poor building envelopes or buildings with unmanageable latent loads

- Mixed-mode ventilation (operable windows) without proper condensation safeguards

Design Considerations

The objective of chilled-beam system design is to minimize primary airflow rates. When chilled-beam systems are applied, the minimum primary airflow rate is typically the greater of that required for ventilation and for space dehumidification. In cases where these values are similar (differing by no more than about 25%), consider using a dedicated outdoor air system (DOAS). Laboratories and health care may require or benefit from the use of a DOAS, which uses chilled beams efficiently.

In cases where the sensible cooling requirements require significantly more than the minimum primary airflow rate, air-handling units that mix return and outdoor air volumes may be considered for use with chilled beams.

Chilled beams are generally 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 kept slightly (2 to 3°F) above the room dew-point temperature. In both cases, the chilled-water 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 and operated primary air system with 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. Beams with sensible-only cooling coils do not require filters per ASHRAE *Standard* 170 and section 307.2 of the International Code Council’s *International Mechanical Code*® (ICC 2018).

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 15°F 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.

Alternatively, resetting the primary air temperature with a duct-mounted heating 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, the beams should be located parallel to the curtain wall, to ensure air movement across these surfaces in order to promote a comfortable environment.

Six-port zone control valves may be used to eliminate a significant amount of zone runout piping and allow more efficient heating across active chilled beam coils.

When ceiling-based active beams are the primary space heating source, their primary air supply must be maintained to project the warm air into the space.

Thermal Comfort

Chilled-beam 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.

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 56 to 60°F; thus, reconditioned air leaving the coil is typically several degrees warmer than the primary air with which it is subsequently mixed. This results in beam design discharge air temperatures that are above 60°F, thus warmer than those normally used by conventional all-air systems. Because of warmer discharge temperatures, larger active beam discharge air volumes are required.

Control and Zoning

Chilled-beam system 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 active beams in applications where more precise space temperature control is required.

In applications where primary air is supplied at conventional temperatures (55 to 57°F) 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.

Although many chilled-beam applications involve a constant-volume supply of primary air, chilled beams can also be served by varying primary airflow rates. For example, classrooms and conference rooms where occupancy levels may vary considerably can be fitted with demand control ventilation (DCV) provisions. In such cases, the primary air supply to the beams can be varied according to occupancy, while a space dew point override ensures that the primary air volume reduction does not compromise room humidity levels. Water flow through the coil remains controlled by the space thermostat, resulting in the ability to control space temperature levels independent of the primary airflow rate.

Table 11 Applications for Chilled Beams

| | Commercial Buildings | | | | | | Educational Facilities | | Laboratories and Health Care | | | | | Domiciliary | | |
|-----------------------------------|----------------------------|-------------------------------|------------------|---------------------------|-----------------------|--------------|------------------------|---------|-------------------------------|-----------------------|-----------|-----------------|---------|-------------|-----------------|-------------------|
| | Interior Open Office Zones | Interior Private Office Zones | Conference Rooms | Perimeter Overhead System | Perimeter UFAD System | Lobbies | Classrooms | Offices | Laboratories/Diagnostic Areas | General Patient Rooms | AI/ Rooms | Operating Rooms | Offices | Hotel Rooms | Dormitory Rooms | Multifam Resident |
| Radiant panels ^{a,b} | • | • | N | • | • | N | N | • | • | • | N | N | • | • | • | • |
| Sails ^{a,b} | • | • | N | • | • | N | N | • | • | • | N | N | • | N | N | N |
| Passive beams ^{a,b} | • | • | N | N | P | N | N | • | • | • | N | N | • | • | • | • |
| Active beams | | | | | | | | | | | | | | | | |
| Overhead or high-sidewall mounted | | | | | | | | | | | | | | | | |
| Constant volume | P | • | • | • | N | ^b | • | • | P | P | N | N | P | • | • | • |
| Variable volume | • | • | P | P | N | ^b | • | • | • | • | N | N | • | • | • | • |
| Floor or low-sidewall mounted | | | | | | | | | | | | | | | | |
| Constant volume | N | N | N | N | • | • | • | • | N | P | N | N | • | • | • | • |
| Variable volume | N | N | N | N | • | • | • | • | N | • | N | N | • | • | • | • |

- = Used
- N = Not Recommended
- [a](#) = requires decoupled ventilation to space
- [b](#) = requires decouple heating system, where applicable

When chilled-beam systems are used to condition spaces with widely varying sensible cooling requirements (e.g. perimeter spaces), consider using a variable-air-volume (VAV) terminal to vary the primary airflow rate to beams in the zone. This allows the primary airflow rate to be throttled in response to the zone's sensible cooling requirements. The minimum flow limit on the VAV terminal is set at the minimum primary airflow rate required to ensure proper space ventilation, adequate room air induction, and humidity control.

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 within a ceiling plenum. 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 an active chilled beam should generally be considered at an equal temperature to that maintained within the occupied zone, unless solid evidence indicates otherwise. For passive beams, the entering air temperature is higher than that in the occupied zone due to room air stratification. The actual placement of the beam with respect to space heat sources often affects the entering air temperature.

Most active chilled beam suppliers offer various nozzle sizes and configurations. Nozzle configuration affects the beam's primary air pressure requirement and acoustical performance as well as its induction rate. 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 50 ft/min or less if compliance with ASHRAE *Standard* 55 is the design intent. Mapping techniques (see the sections on Fully and Partially Mixed Air Distribution) and/or selecting active beam throw values from [Table 6](#) 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. There are various methods of accomplishing this, including the following:

- Sensors may be attached to the supply water pipe to detect formation of surface moisture and discontinue chilled-water flow until the moisture has evaporated. This method is relatively inexpensive but also reactive, and halts induced air cooling through the sensible water coil while conditions favoring the possibility of condensation formation exist.
- Dew-point calculation and reset of the chilled-water supply temperature is a proactive strategy that does not fully suspend 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.
- In spaces with operable windows or doors, occupants and staff should be educated on the effect these have on their thermal environment.
- In some applications, condensate trays may be used to collect temporary and infrequent condensation. When used, trays must have adequate condensate removal and/or water flow modification provisions.

Building Type

The designer must consider the intended building use when determining the type of chilled beams to be used. General recommendations for chilled beam applications are shown in [Table 11](#).

10. AIR CURTAIN UNITS

An air curtain unit acts as a controlled barrier for environmental and thermal separation and wind resistance when a building's doors or windows are opened. As an environmental separation barrier, it repels airborne dust, dirt, fumes, odors, and flying insects from entering a building or a protected indoor area. As a thermal barrier, it reduces cross migration of warm, lighter air flowing through the upper part of the opening and cold, heavier air flowing through the lower part of the opening. As a wind resistance barrier, it minimizes the effect of outdoor wind blowing into a building's openings.

A properly applied air curtain can maintain environmental integrity between two distinct areas while allowing unobstructed access between the areas. Energy savings are possible when the air curtain separates areas of different temperatures.

Principles of Operation

An air curtain unit operates on the principles of air entrainment, velocity vector, and pressure. Because the airstream entrains a volume of air as it travels across an opening, it can maintain separation of environments by returning these air volumes back to their respective areas when the airstream splits, thereby minimizing losses.

Air curtain unit energy effectiveness is defined by the amount of energy saved (i.e., the energy loss prevented through an opening with an air curtain), divided by the amount of energy that would have been lost without an air curtain. It is represented as a percentage, and the amount of energy saved is reduced by the energy consumed by the unit. Research (e.g., Pappas and Tassou [2003]) shows that air curtains have a range of effectiveness from 60 to 90%, depending on the type and application.

To fully realize benefits of using an air curtain unit, make sure that equipment is properly sized, installed, adjusted, and maintained.

Application Considerations

Air curtain unit selection depends on the opening's width and height. To maximize effectiveness, the air curtain unit must at least cover or slightly overlap the entire opening, and have a minimum velocity projection of 400 fpm at the target surface (Wang and Zhong 2014). The unit is usually mounted above or beside a door or window opening. When mounted above the opening, the horizontal air curtain unit discharges its air vertically down across the opening. When mounted next to the opening, the vertical air curtain unit discharges air horizontally across the opening. On wide openings, two vertically mounted, lower-air-velocity units can be used as an alternative to a single horizontally mounted, higher-air-velocity unit. The air curtain unit discharge must have a free and clear path to the entire opening for optimum performance. AMCA Publication 222 is an application manual for air curtains that provides detailed considerations for unit selection, installation, and construction.

Air curtain units are classified into two different types of construction: non-recirculating and recirculating. A non-recirculating system draws air into the unit directly from the surrounding environment in both horizontal and vertical applications (see [Figures 33, 34, 35, and 36](#)). An air curtain equipped with inlet ductwork, which draws air from outside the surrounding environment, is also considered to be non-recirculating (Figure 37). A recirculating system draws air from ductwork that primarily collects and returns the discharge air back to the inlet. Applications often use a plenum with a floor return connected to the inlet with ductwork (Figure 38). An alternative construction includes horizontal flow that discharges and returns from side to side (Figure 39).

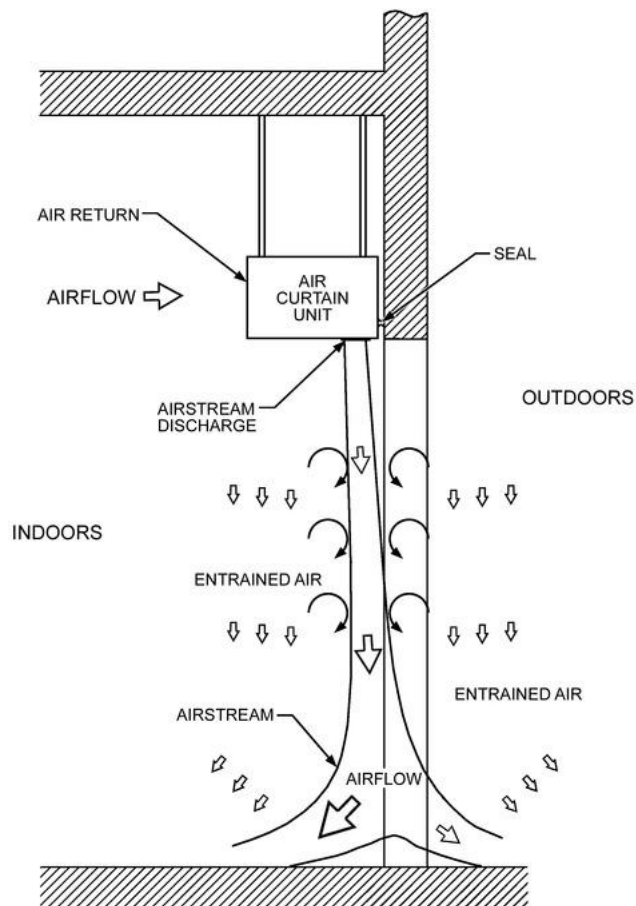


Figure 30. Non-Recirculating, Horizontal-Mount High-Velocity Air Curtain Unit (AMCA Standard 222)

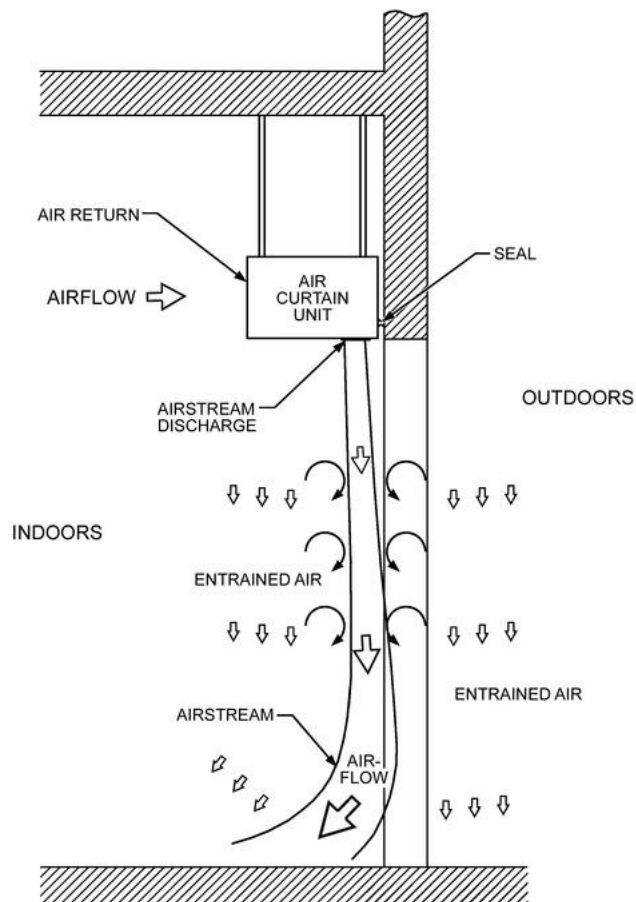


Figure 31. Non-Recirculating, Horizontal Mount Low-Velocity Air Curtain Unit (AMCA Standard 222)

Building Design Considerations

Non-recirculating air curtain units comprise the majority of systems built today. There are few limitations to where they can be installed, they have little effect on the construction or design of the building opening, and they are less expensive to install and maintain. In buildings with high ceilings, they can aid in destratification.

Recirculating air curtain systems have a higher effectiveness and are more expensive than non-recirculating systems. They are typically limited to commercial or cold storage applications where doorways have high traffic or are open for extended periods of time. They require careful integration into the building's construction and design and have door height limitations.

Types of Applications

Air curtain units use various electrical and mechanical means to control airflow rate and temperature to achieve environmental separation, mitigate wind conditions, and repel flying insects.

General applications of air curtain units include the following:

- **Exterior environmental separation:** protects an exterior opening from unwanted infiltration of outdoor air and the escape of indoor air caused by effects of natural wind and/or temperature differences.
- **Interior environmental separation:** provides protection between interior rooms connected by a common opening. This application is intended to prevent the unwanted infiltration of unconditioned air or the loss of conditioned air from one room to another caused by temperature differential. Typically, this can be controlled by an air curtain unit that has an air performance requirement much smaller than the air performance requirement for exterior applications.
- **Flying insect control:** protects an opening or doorway, usually exterior, from the unwanted entry of flying insects. This is a common requirement in facilities that produce, process, or serve foods (e.g., kitchens, cafeterias). This application typically requires an air curtain unit with a higher airstream velocity to repel flying insects. When units are selected with a higher airstream velocity to improve resistance against insect penetration, the energy effectiveness will be reduced.
- **Coolers/chill rooms and freezers/cold stores:** prevents loss of refrigerated air through openings and/or doorways in coolers and freezers. Three types of applications exist: cooler to freezer, ambient to cooler, and ambient to freezer. These types of installations are generally (but not limited to) indoor applications; therefore, the air curtain unit is only required to overcome airflow caused by temperature differential and not wind pressure. Air curtain units are typically horizontally mounted on the warm side of the doorway, so that the airstream split created can balance against the air trying to leave the cold room. Cold storage installations can be difficult to balance and may require a vertical or cold-side mount, dampers, and/or multispeed motors to effectively protect the opening.
- **Ovens:** protects against loss of heated air through openings and/or doorways in ovens. Air curtain units are normally mounted horizontally over the oven opening and angled slightly inward toward the oven to prevent hot air from escaping through the top of the opening. These types of installations are generally indoor applications; thus, the air curtain unit is only required to overcome airflow caused by temperature differential and not wind pressure. The heating process in ovens is typically designed to maintain a neutral pressure with the surrounding environment. The air curtain unit should be adjusted to only entrain and "turn back" the heated air to avoid creating an unbalanced condition by forcing air into the oven. The mounting location of the air curtain unit should also provide adequate protection from exposure to hot air that would escape the oven in the event the air curtain unit is shut down.
- **Negative building pressure:** Air curtain units installed on an opening where a negative pressure exists require special consideration. When the building is underpressurized, standard air curtain airflow rate will not be able to overcome the artificial deflection created by the negative condition. In special cases, airflow may be increased to overcome a slight negative condition. For proper operation of the air curtain, the building should be neutral or positively pressurized.
- **Special/custom:** other applications include protection against dust infiltration, water removal in drying processes, smoke and odor containment, and defrosting doorways. In these cases, effectiveness is defined by the application criteria.

Optional Features and Controls

Optional features for air curtain units include heating, cooling, filters, and special controls. A combination of special components, casing materials, and casing coatings may be required for outdoor mounting, hazardous locations, or harsh environmental applications. Supplemental heating/cooling can be provided by an air curtain unit to reduce the zone load on unitary HVAC equipment, but it should not be seen as the primary source for conditioning internal areas. The energy effectiveness of an air curtain unit is not enhanced when it provides conditioned air. Heating methods can include steam, hot water, electricity, and fuel gas. Cooling methods can include chilled water or direct expansion.

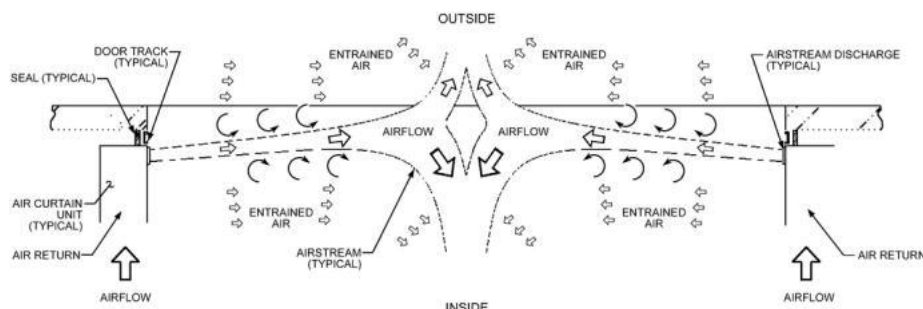


Figure 32. Two Non-Recirculating, Vertical-Mount Air Curtain Units (AMCA Standard 222)

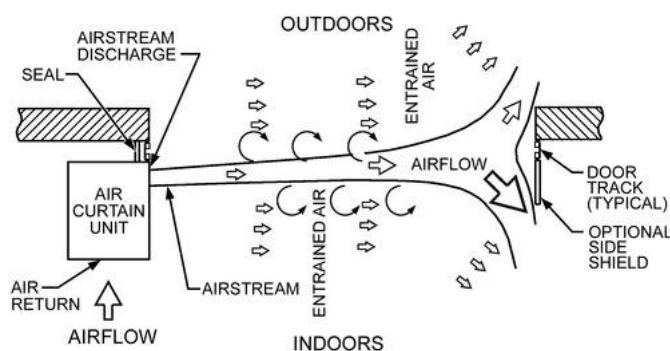


Figure 33. Non-Recirculating, Vertical-Mount Air Curtain Unit (AMCA Standard 222)

Applications in dusty or dirty areas may benefit from air curtain units equipped with inlet air filters to reduce maintenance and maintain optimal performance. Note that the aerodynamic performance of an air curtain unit will be reduced if filters are not properly sized, cleaned, or changed. Air curtain unit electrical controls that monitor door position and temperature, or BMS interfaces are required to provide the design velocity, temperature, and operation of the air curtain. They are also used to minimize unnecessary energy usage and overconditioning of the building opening.

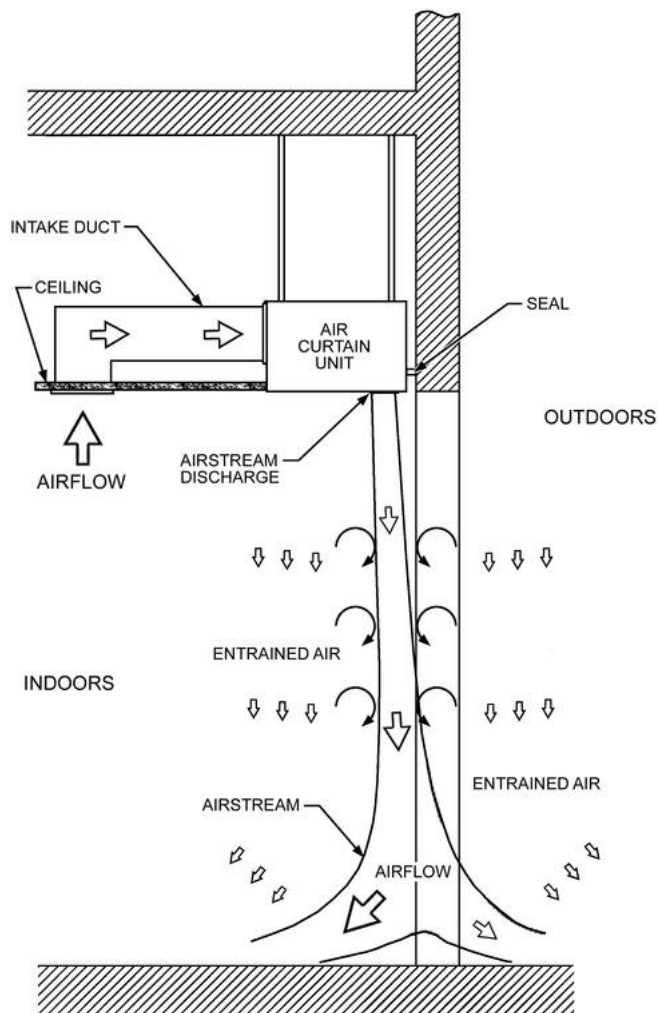


Figure 34. Non-Recirculating, Horizontal-Mount Air Curtain Unit with Ducted Inlet (AMCA Standard 222)

The discharge of an air curtain unit adjusts the direction of the air curtain for proper protection. This adjustment may include (but is not limited to) the following types:

- **Pivot mount:** the ACU cabinet is capable of pivoting on its mounting so that it may direct the air curtain at the proper angle to protect the opening.
- **Adjustable nozzle:** the ACU discharge nozzle is capable of pivoting within the ACU so that it may direct the air curtain at the proper angle to protect the opening.
- **Adjustable nozzle vane(s):** the ACU discharge nozzle employs a vane, or vanes, that are capable of pivoting within the nozzle so that it may direct the air curtain at the proper angle to protect the opening.
- **Both adjustable nozzle and vanes:** the ACU employs a combination of the adjustable nozzle and adjustable nozzle vanes.
- **Diverter nozzle:** the ACU employs an apparatus within the nozzle that is capable of diverting the air curtain at the proper angle to protect the opening.

Performance and Safety Standards

Air curtain unit performance data can be used to select and/or compare different products. ANSI/AMCA Standard 220 defines the test methods that can be used to generate data for the typical types of non-recirculating air curtain aerodynamic performance.

ANSI/AMCA Standards 300, 301, and 320 are sound standards that can be used to rate air curtains units are.

For insect control applications, use ANSI/NSF Standard 37 to determine criteria for air curtain unit air performance, construction, design, and material type. An air curtain unit that complies with this standard is considered by the food service industry to provide effective flying insect protection to an entryway by deterring flying insects from entering through the opening or nesting in the air curtain unit.

Safety standards that can be applied to air curtain units are UL Standards 507, 1995, and 2021.

Energy codes and standards such as ASHRAE Standard 189.1-2020 and the IECC define air curtain unit air performance and operation requirements for energy sustainable structures.

Maintenance and Accessibility

Whether the unit will be mounted horizontally or vertically, and inside or outside the opening, obstructions surrounding the opening will require special installation considerations. Typical obstructions may include beams, piping, ductwork, electrical conduit, door hardware, etc. Accessibility for maintenance should also be considered. Each manufacturer provides specific instructions for their products; carefully follow the specific instructions regarding safety, installer qualifications and recommended work practices.

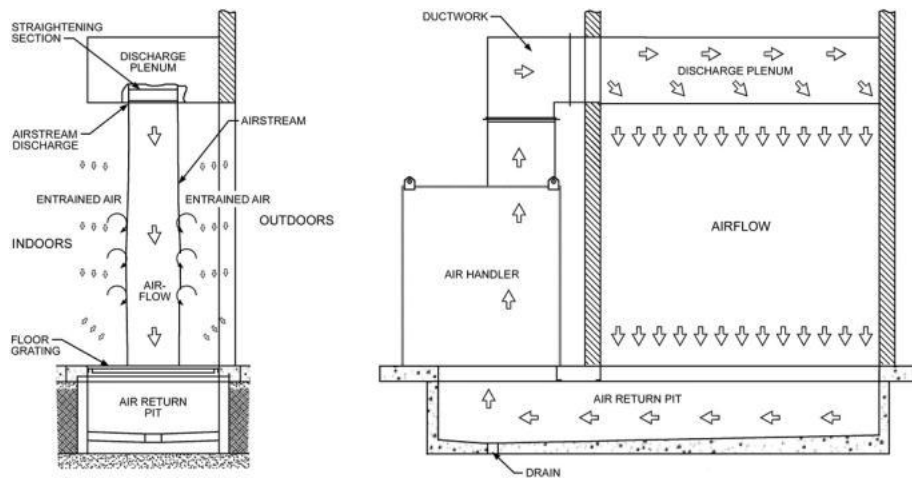


Figure 35. Recirculating, Horizontal-Mount Air Curtain Unit (AMCA Standard 222)

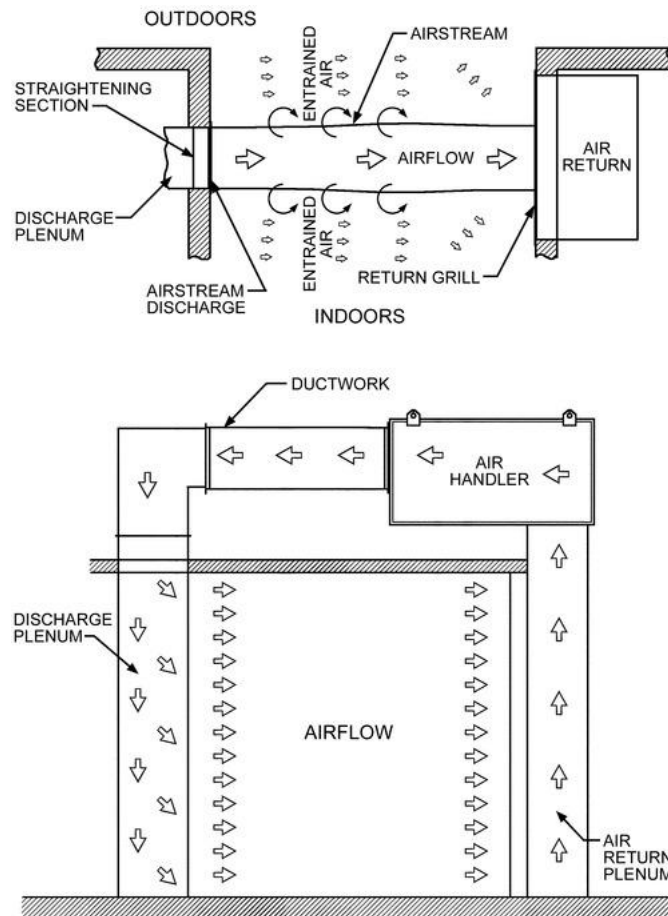


Figure 36. Recirculating, Vertical-Mount Air Curtain Unit (AMCA Standard 222)

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