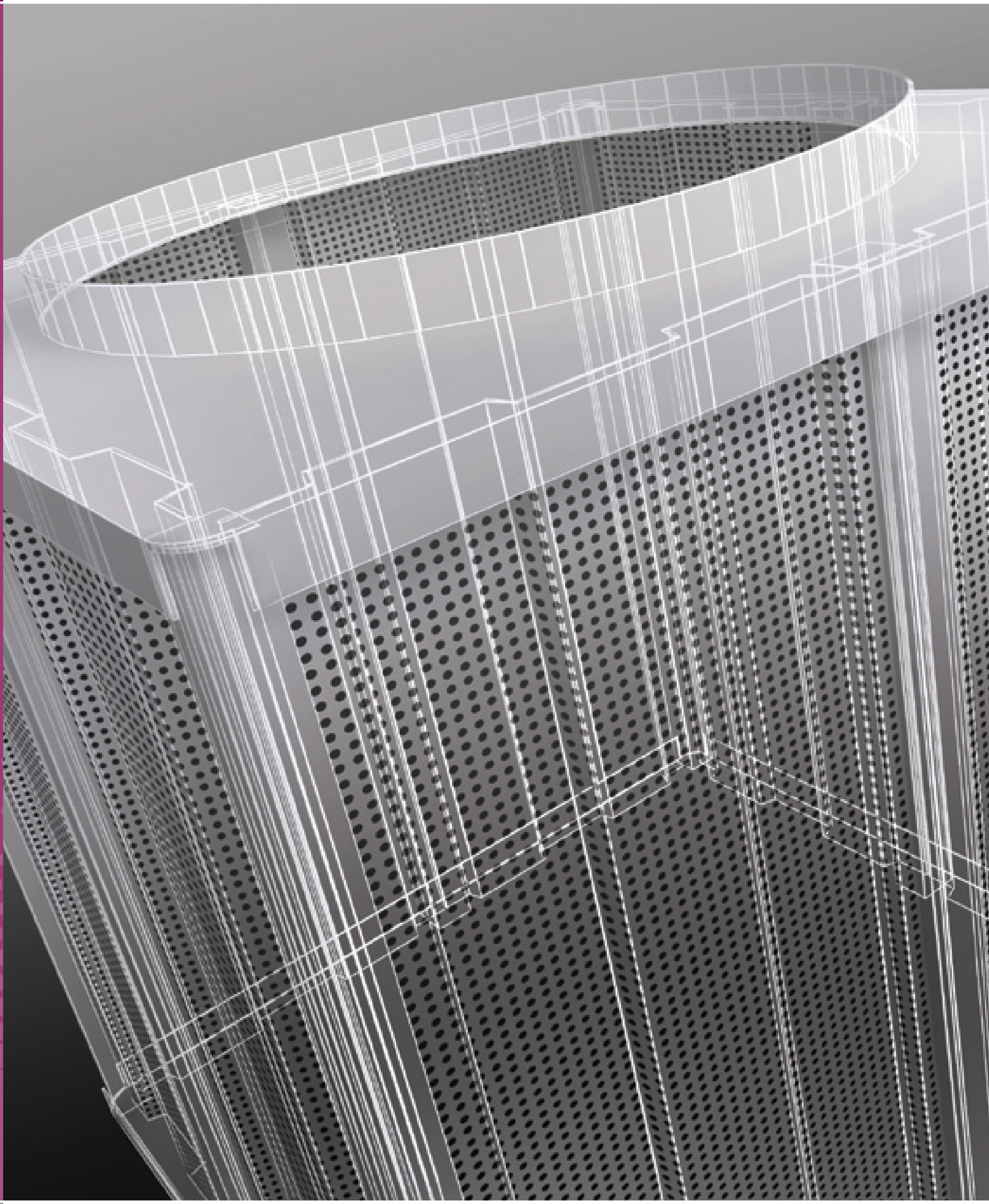
SECTION J



Engineering Guide
Displacement Ventilation

Please refer to the **Price Engineer's HVAC Handbook** for more information on Displacement Ventilation.

Introduction

Displacement ventilation is an air distribution technology that introduces cool air into a zone at low velocity, usually also at a low level. Buoyancy forces ensure that this supply air pools near the floor level, allowing it to be carried up into the thermal plumes that are formed by heat sources. This type of air distribution is effective at delivering fresh air to occupants and removing many of the contaminants associated with heat sources, while creating a comfortable environment. This chapter focuses on the main design criteria for displacement ventilation systems as well as introduces its common applications. The following pages will go further into depth on the specific requirements of schools, theaters, health care and industrial spaces.

Overview

Air flow in ventilated spaces generally can be classified by two different types: mixing (or dilution) ventilation and displacement ventilation. Mixing ventilation systems **(Figure 1)** generally supply air in a manner such that the entire room volume is fully mixed. The cool supply air exits the outlet at a high velocity, inducing room air to provide mixing and temperature equalization. Since the entire room is fully mixed, temperature variations throughout the space are small, while the contaminant concentration is uniform throughout the zone.

Displacement ventilation systems **(Figure 2)** introduce air into the space at low velocities, which causes minimal induction and mixing. Displacement outlets may be located almost anywhere within the room, but have been traditionally located at or near floor level. The system utilizes buoyancy forces in a room, generated by heat sources such as people, lighting, computers, electrical equipment, etc., to remove contaminants and heat from the occupied zone. By so doing, the air quality in the occupied zone is generally superior to that achieved with mixing ventilation.

Benefits

Flexibility - As load distribution changes within the space, a displacement system will be able to compensate. For example, if the space was designed to have a fairly even load distribution and now has the loads concentrated to one side, the system is able to compensate as the buoyant forces drive the supply system and will draw the supply air towards the loads.

IAQ - Because fresh supply air is pooling at the floor level, personal thermal plumes draw fresh air up the body. All of the warm and polluted air is extracted at the high return. When properly designed, there should always be a greater amount of fresh air in the breathing zone when compared to a conventional dilution system, leading to higher ventilation efficiency.

Green building rating systems, such as the LEED® program and Green Globes® have credits that are applicable to displacement ventilation systems. See the Green Tips for further information.

Energy Savings - Displacement systems present many potential opportunities for energy savings. The lower pressure drop associated with displacement ventilation outlets and the corresponding selection of smaller fan components may allow for a reduction in fan energy. The supply air temperature is typically higher for displacement systems than for overhead mixing systems, and can lead to free cooling from increased economizer hours. Combined with a higher return temperature than overhead systems, the warmer supply temperature of DV systems can cause an increase in chiller efficiency. Due to a high ventilation effectiveness, the amount of outdoor air that must be conditioned can also be decreased when compared with a mixing system. This is especially significant in humid climates, where dehumidification of outdoor air is a significant cost.

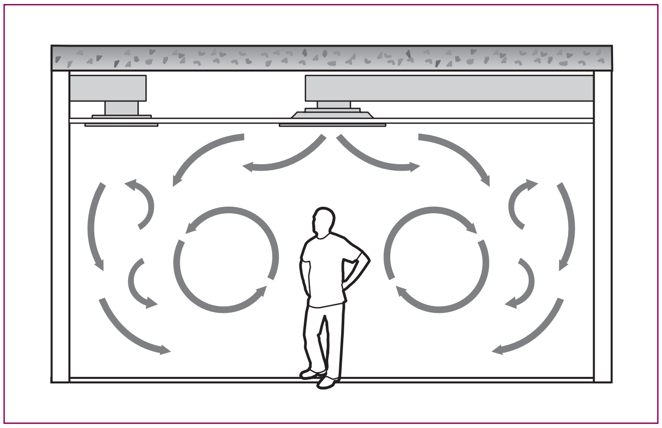


Figure 1: Mixing (Dilution) Ventilation

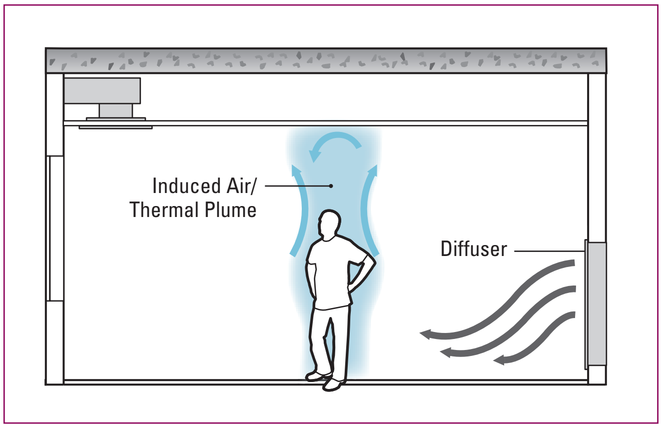


Figure 2: Displacement ventilation

Limitations

The size of displacement outlets can make selecting and locating diffusers difficult in areas where there is limited wall area. Ceiling and floor mounted diffusers may help alleviate this issue, where appropriate.

DV systems are limited in their maximum cooling capacity, primarily due to stratification limits set by ASHRAE (2004a) and ISO (2005). The Price Engineer's Handbook contains more information on how stratification affects the maximum cooling capacity of DV systems. Chen, Glicksman, Yuan, Hu, & Yang (1999) indicate a maximum cooling capacity of 38 Btu/hft2 [119 W/m2] while ensuring thermal comfort.

Typical Applications

Displacement ventilation is an effective method of obtaining good air quality and thermal comfort in the occupied space. Spaces where displacement ventilation has been successfully applied include the following.

- Schools
- Theaters
- Hospitals
- Casinos
- Restaurants
- Industrial Spaces
- Supermarkets
- Open Offices

Concepts and Benefits

Displacement ventilation is usually a good choice if:

- The contaminants are warmer and/or lighter than the room air
- Supply air is cooler than the room air
- The room height is 9 ft [2.75 m] or more
- Low noise levels are desired

Overhead air distribution may be a better choice if:

- Ceiling heights are below 8 ft [2.4 m]
- Disturbances to room air flow are strong
- Contaminants are colder and/or denser than the ambient air
- Cooling loads are high and radiant cooling is not an option

Thermal Plume

A thermal plume is a convection current caused by buoyancy forces that causes local air to warm and rise above the heat source, entraining surrounding air and increasing in size and volume as it loses momentum, as depicted in **Figure 3**. The maximum height to which a plume will rise is dependent on the strength of the heat source, as the initial momentum of the plume will increase. Also, a room with more stratification will reduce the relative density of the plume and, as a result, limit the height to which the plume will rise.

The thermal plume generated from a point source acts differently than a thermal plume generated from large objects in the space. For example, a heated cylinder produces a boundary layer and the convective thermal plume takes a different shape than a point heat source. A point source type expansion of the thermal plume is still present, but at an altered height and with the thermal plume boundary layer included, shown in **Figure 4**. The cylinder is a better approximation of an occupant in the space than a point source.

Room Air Flow Pattern

Air flow patterns in a displacement ventilation system are quite different than in a mixing system. Because of the low discharge velocity of displacement outlets, the room air motion is largely driven by the convection flows created by heat sources such as people, equipment, and warm windows; or by heat sinks such as cold walls or windows. The convection flows within the room cause the formation of horizontal air layers. The warmest air layers are near the ceiling and the coolest air layers are near the floor.

Typical Design Parameters	IP	SI
Supply Temperature	63-68 °F	17-20 °C
Return Temperature	78-85 °F	26-29 °C
Supply Face Velocity – Mainly Sedentary Occupancy	40 fpm	0.2 m/s
Ventilation Effectiveness (ASHRAE, 2004b)	1.2	1.2

Table 1: Typical DV parameters

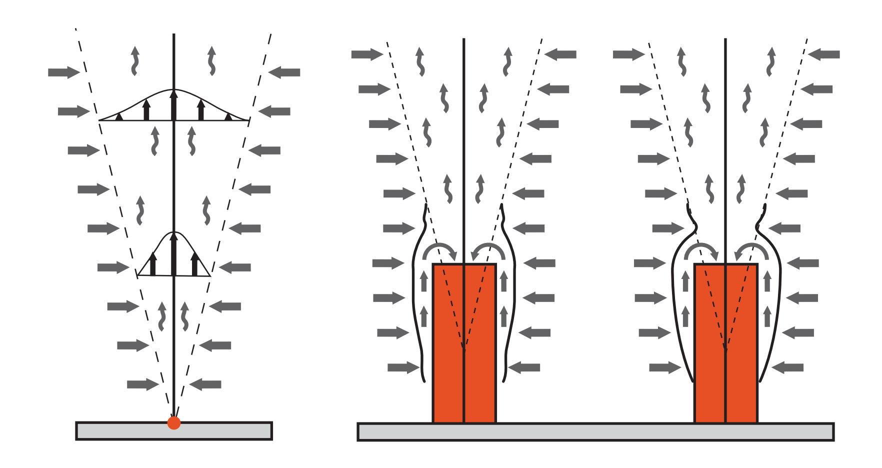


Figure 3 & 4: Thermal plume from a point heat source and of a heated cylinder

Figure 5: Air layers

Room air moves horizontally across the floor due to momentum from the supply outlet and suction from thermal plumes. It then passes vertically through the thermal plumes to a high level in the room where it is returned or exhausted.

Vertical air movement (see **Figure 5,** next page) between layers is caused by stronger convection forces associated with heat sources or cold sinks. Heat sources such as people, computers, lights, etc. create a rising convection flow known as a thermal plume. The strength of the thermal plume is dependent on the power and geometry of the heat source. The strength of the thermal plume will determine how high the convection flows can rise before the momental is fully dissipated. Cold sinks such as exterior walls or windows can generate convection flows down the wall and across the floor.

Displacement Ventilation Characteristics

Air Flow Penetration

A displacement system supplying cool air through a diffuser will deliver air along the floor in a thin layer typically less than 8 in. [0.20 m] in height. The supply air spreads across the floor in a similar manner to water flowing out of a tap, filling the entire space. If obstructions such as furniture or partitions are encountered, the air will flow around and beyond the obstruction, as illustrated in **Figure 6**. Even rooms with irregular geometries, as illustrated in **Figure 7** can be uniformly supplied with air.

When the cool air meets a heat source such as a person or piece of equipment, a portion of the conditioned air is captured by the thermal plume of the heat source, while the remainder of air continues further into the room.

When designing the system to deal with the cooling demand of the space, the penetration depth of a displacement diffuser can be 26-30 ft [8-9 m] or more from the face of the diffuser. For rooms exceeding 30 ft [9 m] in length or width, diffusers on several walls are suggested to promote even air distribution.

Diffuser Air Flow Pattern

In order to avoid draft and minimize induction of room air, it is essential for the displacement diffuser to uniformly deliver the supply air across the entire diffuser face at low velocity. This requires an internal equalization baffle in combination with a low free area face. Yuan, Chen & Glicksman, (1999) recommended 40 fpm [0.2 m/s] in order to maintain acceptable comfort.

A displacement diffuser supplying cool air will result in an air pattern (typically 5 - 10 °F [2 - 5 °C] cooler than the room set-point), resembling **Figure 8**. Due to the density of the cool supply air, it falls towards the floor a short distance from the diffuser face and continues along the floor at a depth of approximately 4-8 in. [0.1-0.2 m].

When supply air is isothermal (supply air is the same or less than 5 °F [2.5 °C] warmer than the room set-point), the flow will be distributed horizontally into the space, as shown in **Figure 9**.

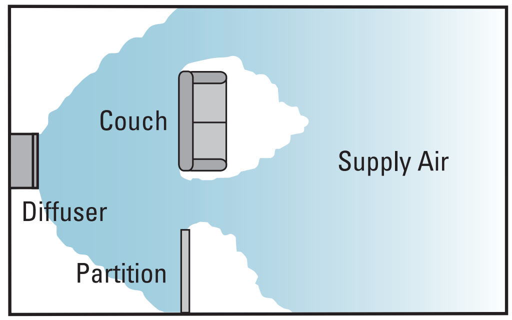


Figure 6: Obstruction

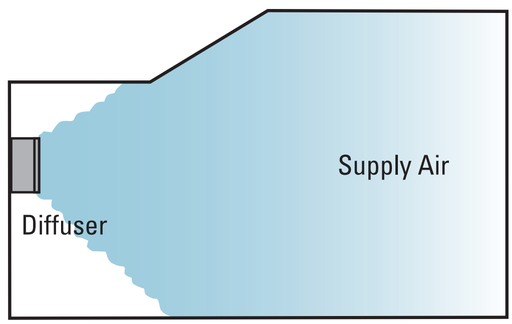


Figure 7: Irregular room geometry

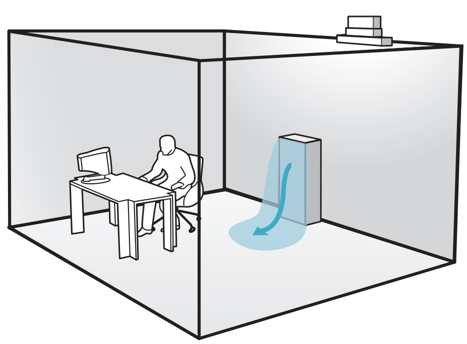


Figure 8: Cooling air flow pattern

Displacement Ventilation Characteristics

Contaminant Distribution

Contaminant distribution is influenced by several factors such as supply air method, contaminant source type, location within the space, heat sources, and space height.

Displacement ventilation improves occupant air quality by reducing the contaminants in the lower portion of the room. The general upward motion of air causes contaminants to concentrate within the upper zone **(Figure 10).**

With mixing ventilation, contaminants are diluted with supply air and are distributed evenly throughout the space. The figure represents contamination distribution in a room supplied with mixing and displacement ventilation for a typical case where the contaminant source is warm (a person, for example).

With displacement ventilation, because the upward convection around a person brings clean air from lower level to the breathing zone, the air in the breathing zone is cleaner than the room air at the same height. Contaminants that are heavier than air need to be extracted at a lower level through a second return if they present a safety concern.

Temperature Distribution

Controlling stratification in the occupied zone is critical to maintaining occupant comfort. ASHRAE Standard 55 requires that the temperature difference between the head and foot level not to exceed 5.4 °F [3 °C] for a standing person and 3.6 °F [2 °C] for a seated person.

ASHRAE (Chen et al., 1999) has determined a method for calculating the head-to-foot temperature stratification of a displacement system based on supply air volume and load distribution. This relationship was used to develop a design procedure for displacement ventilation systems. Using this design procedure, an acceptable room temperature stratification level can be achieved.

For commercial displacement ventilation systems, supply air temperatures ranging from 63 - 68 °F [17 - 20 °C] can be expected. As well, the temperature difference between return and supply in a stratified system will generally be between 13 - 20 °F [7 - 10 °C].

CONTROL TIP

Temperature stratification above the occupied zone is not a concern as long as the ceiling is over 8 ft [2.4 m]. To ensure stratification control, returns must extract from within 1ft [0.33 m] of the maximum ceiling height.

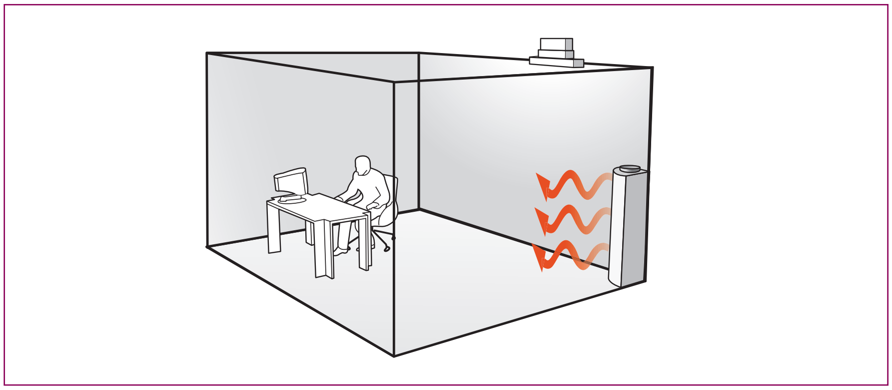


Figure 9: Isothermal air flow pattern

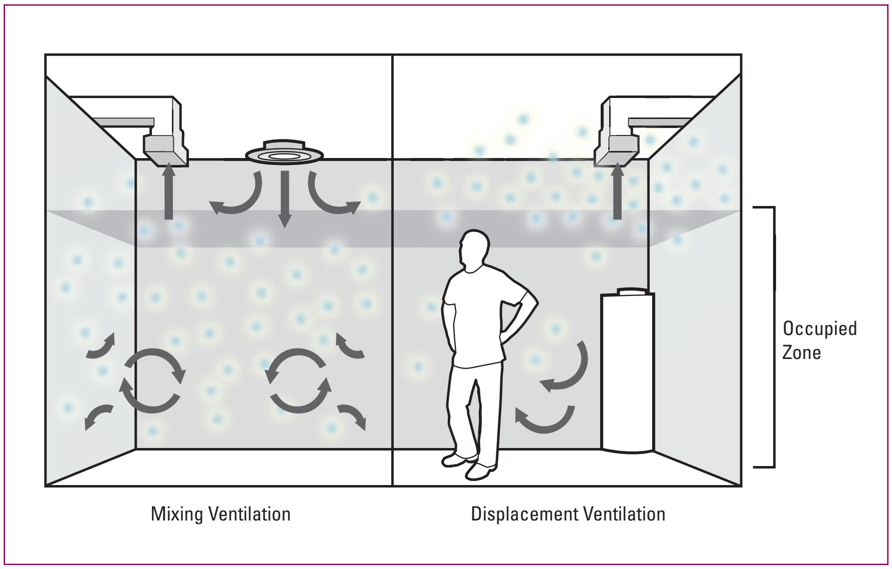


Figure 10: Contaminant distribution

Location of Returns

Returns should be located as high as possible in the space to remove as much of the stratified zone as possible, ideally at ceiling height. If the return is located below the ceiling, the air above the return may not be exhausted properly from the space. If the exhaust is located lower than 7 ft [2 m], some polluted/hot air may remain within the occupied zone. For lower ceilings it is best to place the return above the heat source in the space. In all cases, distributing the returns evenly thoughout the zone will promote even air movement in the room.

Heating with Displacement Diffusers

As previously discussed, displacement ventilation relies on buoyancy, or more specifically, the thermal plumes that surround heat sources, to drive the air movement through the space. These plumes pull the supply air toward occupants, equipment and the façade, as well as any other heat source that requires conditioning. This is all made possible by the pooling of fresh cool supply air at the floor level, which can be used to supply the plumes with cool, fresh air.

When heating is required, the warmer (and relatively buoyant) supply air may not have enough forward momentum from the diffuser to overcome the effects of buoyancy. This may result in the warm supply air rising to the ceiling and being exhausted or returned, potentially bypassing the occupied zone. The warmer the supply air, the higher the risk of 'short circuiting,' which can result in poor thermal comfort and ventilation effectiveness, as shown in **Figures 11** and **Figure 12**.

In practice, for climates with significant heating loads, diffusers with heat-cool changeover or integrated heat should be used. Alternatively, an auxiliary heating system such a radiant panels could be used. For milder climates, it may be possible to use the DV system at slightly elevated temperatures. Experience has demonstrated that reasonable performance may be achieved using up to 5 °F [3 °C] heating air.

Diffusers with Integrated Heat

Displacement diffusers with integrated heat feature a cooling section as well as a heater. In the case of the perimeter diffuser, the bottom section is a low velocity displacement outlet, used to manage the cooling load and provide ventilation air during heating periods. The upper section includes a heater in the enclosure that is to be cycled or modulated as required. The diffuser, shown in **Figure 13**, is designed to look and function like the perimeter radiation systems common to many commercial buildings. The fintube or electric coil in the heating section manages the skin load in the same manner as a typical baseboard heater. The cooling section below continues to supply ventilation air to the building occupants, typically at isothermal or slightly cooling temperatures. This type of outlet is shown in **Figures 14**.

In this configuration, the convective forces from the heating element are not substantial enough to draw the supply air into the front intake opening for the heater, so potential short circuiting of the conditioned supply air is minimized. These diffusers are ideal for use in perimeter offices, classrooms, and commercial spaces with large windows.

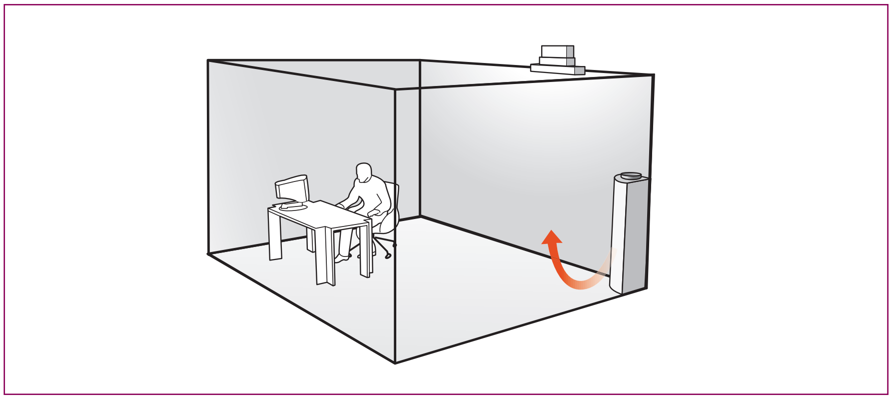


Figure 11: Heating air supply

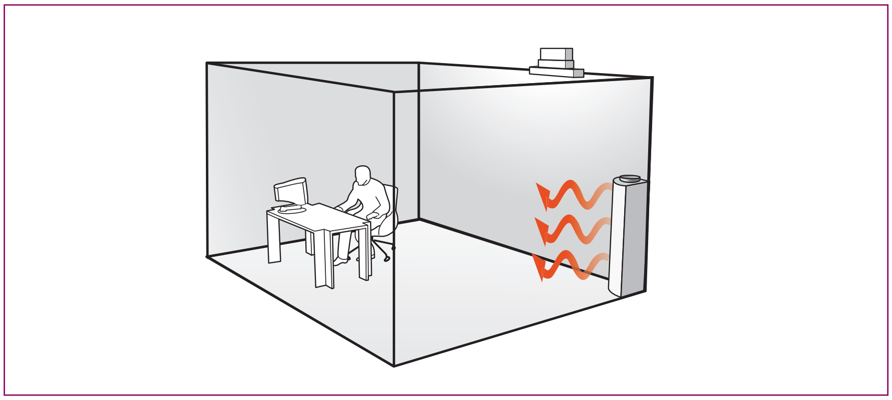


Figure 12: Isothermal air supply

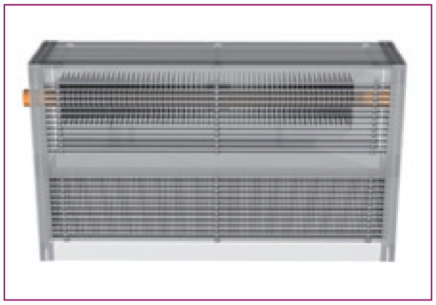


Figure 13: Perimeter diffuser with integrated heater

Figure 14: DV diffuser with integrated heat installed along the perimeter

Heating with Displacement Diffusers

Other types of outlets with heating functions include those that can change their discharge pattern to optimize the room air flow depending on the supply air temperature. These diffusers provide a typical displacement pattern in cooling mode, but can also switch over to a mixing pattern in heating mode. These diffusers incorporate a slot diffuser or a linear bar grille section in order to increase the discharge velocity of the air when mixing is required, as shown in **Figure 17** and **Figure 18**. The changeover is actuated automatically through a signal from either the building control system or a duct temperature sensor. When the supply air is warm, it is diverted into the secondary plenum and through the heating diffuser. This allows a single duct and a single diffuser to provide both heating and cooling with no manual changeover or secondary heating systems.

Other Heating Options

There are cases where it is preferred to separate the heating and cooling functions, particularly in cold climates where the skin load can be significant. Examples of this type of system include the following:

Fan Coils

Fan coils may be incorporated into a displacement system as an alternative heating source, as long as the fan coil is located outside the occupied zone and is used to treat perimeter walls and glass without mixing the occupied zone. For more information on fan coils please refer to Chapter 13—Introduction to Fan Coils and Blower Coils of the Price Engineer's Handbook.

Hydronic Systems

Utilizing a hydronic system in conjunction with a displacement system has numerous benefits. In addition to supplying heat to the zone, hydronic systems can be used to compensate for the sensible cooling demand and provide excellent comfort conditions to a space. There are several methods for supplying hydronic heat: perimeter radiation, radiant flooring, radiant panels **(Figure 19)**, and chilled sails **(Figure 20)**. More information on this option can be found in the following section, as well as in Chapter 18—Introduction to Radiant Heating and Cooling of the Price Engineer's Handbook.

Figure 15: Ceiling mounted DV diffuser with heat-cool changeover

Figure 16: DF1W-HC, displacement diffusers with heat-cool changeover

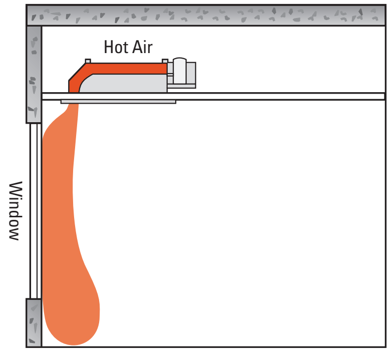


Figure 17: Air diverted through the heating plenum

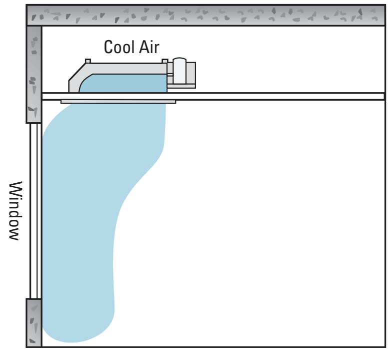


Figure 18: Air diverted through the cooling plenum

Figure 19: Radiant panel

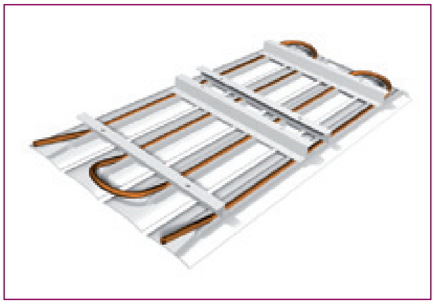


Figure 20: Chilled sail

Diffuser Types

Diffuser Types

A wide variety of displacement air diffuser types are available to suit the location restrictions and décor of a particular room or space. In some cases the diffusers are custom fabricated to meet an area's unique architectural design.

There are several categories of displacement diffusers:

- Free standing diffusers that mount on the floor, in most cases against a wall
- Wall diffusers that integrate into the wall or millwork
- Floor diffusers that install into the floor
- Ceiling diffusers that install in a ceiling
- Industrial diffusers which are designed to withstand harsher environments.

Free Standing Diffusers

Rectangular units are typically placed against a wall or partition or in a corner, but may also be located against pillars or, in some instances, stand in the middle of the room. They are available with rectangular or round faces in order to provide an aesthetic to compliment the room design. Depending on the design, these diffusers provide a 1 way, 3 way, or radial pattern, as shown in **Figures 21** to **Figure 24** for rectangular and round faced diffusers and various configurations.

The configuration of displacement diffusers are typically driven as much by architectural considerations as by performance characteristics. It is for this reason that there is such a large variety of displacement ventilation products in various shapes and sizes.

One common type of these diffusers are the corner outlets. These are specifically designed to fit into a 90° corner in a room and are available in flat or rounded faces, depending on the desired look. These diffusers are ideal for applications where wall space may be limited and corners are available for use.

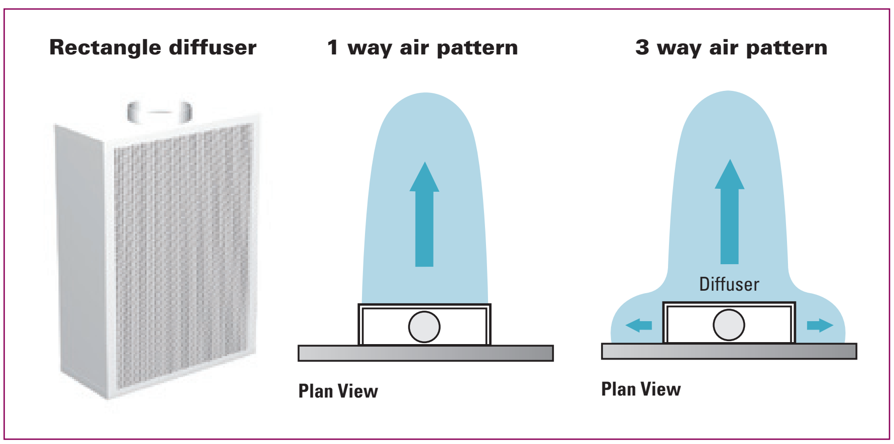


Figure 21: Rectangle diffusers

Figure 22: DF1L installed against wall

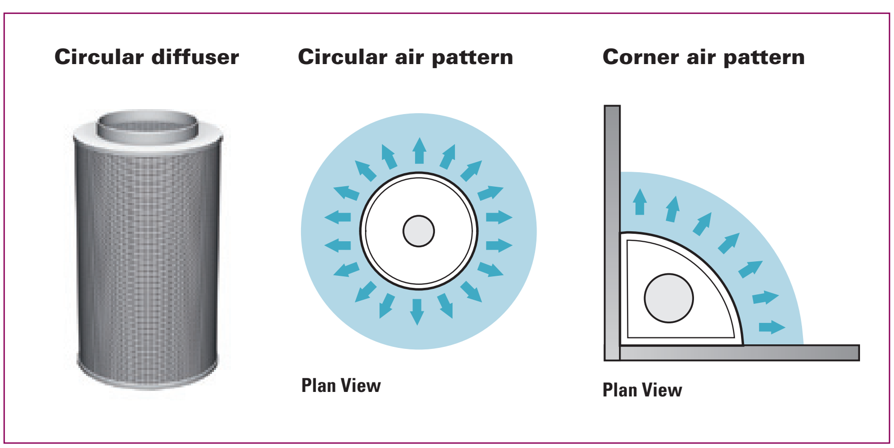


Figure 23: Circular diffusers

Figure 24: DR180 installed in free space

Diffuser Types

Wall Mounted Diffusers

Wall mounted displacement diffusers are designed to be integrated into the wall **(Figure 25)**. The most common wall integrated diffuser features a narrow plenum and rectangular inlet to accommodate duct connection in a standard 4 in. [100 mm] studded wall. A recessed diffuser is another example of a wall mounted diffuser **(Figure 27)**. It has no plenum or inlet, and is designed for plenum fed applications, as might be found when mounted in a stair riser, wall or cabinet. Another type of wall mounted diffuser features a linear grille, as shown in **Figure 28**. These diffusers are typically installed on the perimeter and are available with an integrated heating element.

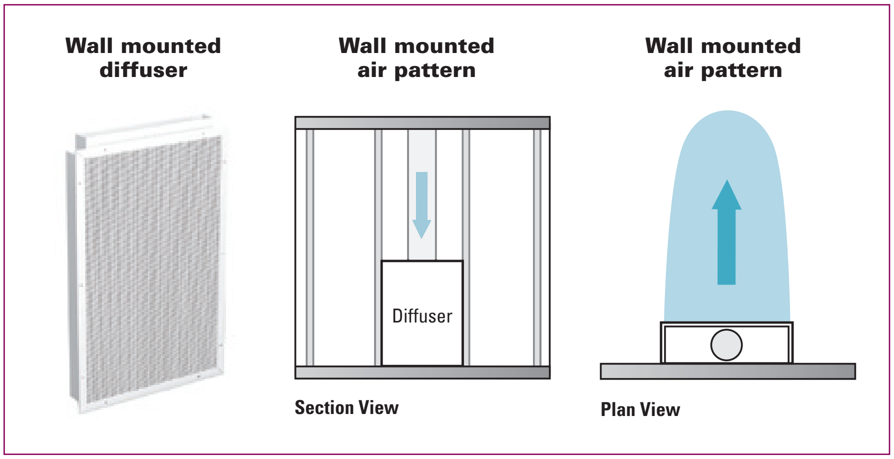


Figure 25: Wall mounted diffusers

Figure 26: DF1W installed in-wall

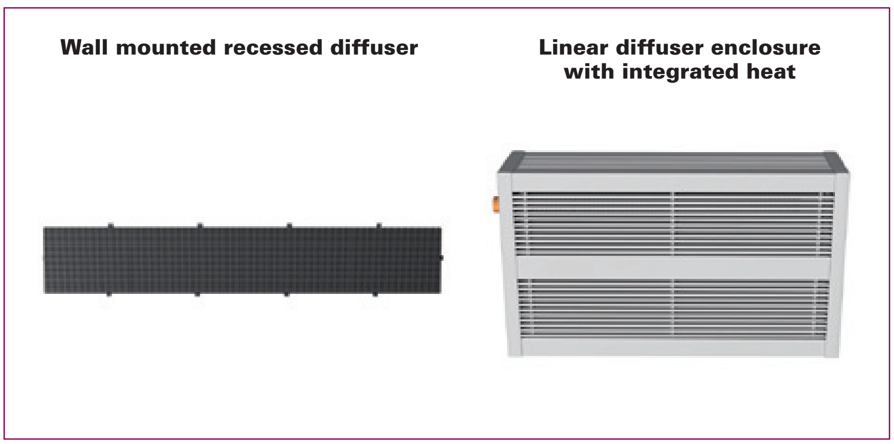


Figure 27: Wall mounted diffusers

Figure 28: DLE-H installed against wall

Diffuser Types

Floor Diffusers

Displacement diffusers are available for integration with a raised floor air distribution system. **Figure 29** to **Figure 33** are some types of displacement floor diffusers available. These diffusers produce a low velocity radial pattern across the floor.

Displacement floor grilles can also be fan assisted **(Figure 31)** when additional air volumes are required and a fan terminal is not economical.

In some instances, such as in a highly aesthetic area or along a perimeter, a continuous grille is preferred. In these situations the linear version of the displacement floor grille is a good choice **(Figure 32)**.

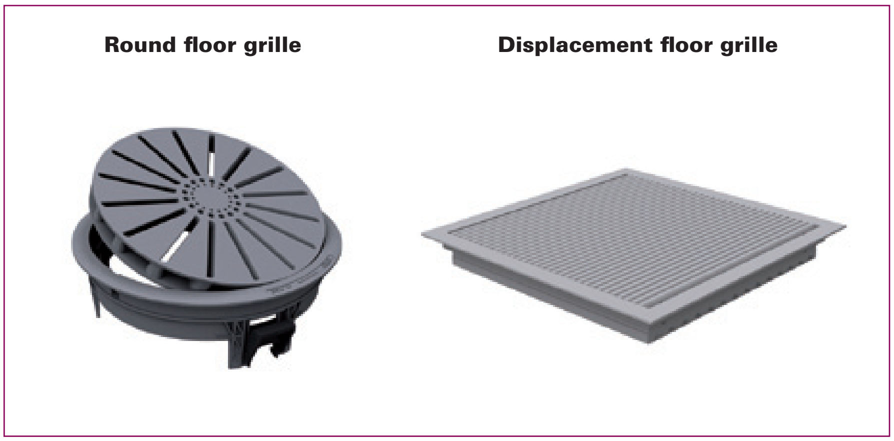


Figure 29: Floor Diffusers

Figure 30: RFDD installed in floor

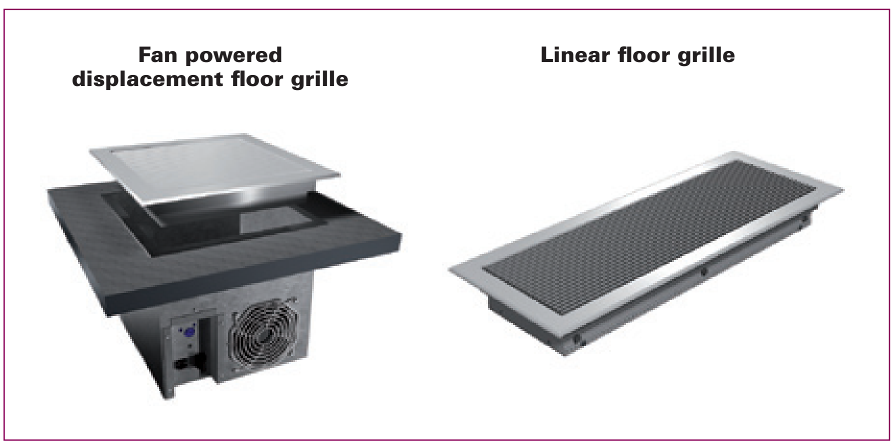


Figure 31: Floor Diffusers

Figure 32: DFGL installed continusly along a perimeter

Diffuser Types

Ceiling Diffusers

Displacement ventilation diffusers may also be located outside of the occupied zone, which is a good location for a diffuser in a space that either does not have a lot of wall space or where space is at a premium, such as in a private office. These diffusers can either fit in to a T-bar ceiling system or can be mounted directly to ductwork. Some of these products are also available with heat-cool changeover options (**Figure 32** and **Figure 34**)

Due to the supply air falling through the warmer air above, there will be some amount of heat gain of the supply air before it reaches the floor. There is also the potential for some entrainment of pollutants that are collected in the upper zone. While the amount of heat gained and pollutants entrained is small, it is often desired to minimize this as much as possible. It is therefore common to locate the supply outlets near a wall to take advantage of the Coanda effect, wherein the supply air will travel down the wall to the occupied zone. This reduces the size of the area where the supply air interacts with the stagnant air, and thereby the heating effect.

If the face velocity of the ceiling mounted displacement diffuser is within the rage recommended for those located in the occupied zone, the velocity of the air falling past occupants should remain low. To ensure that this does not pose a risk of draft, it is good practice to place diffusers that cannot be located near a wall above corridors or office pathways (**Figure 36** and **Figure 37**)

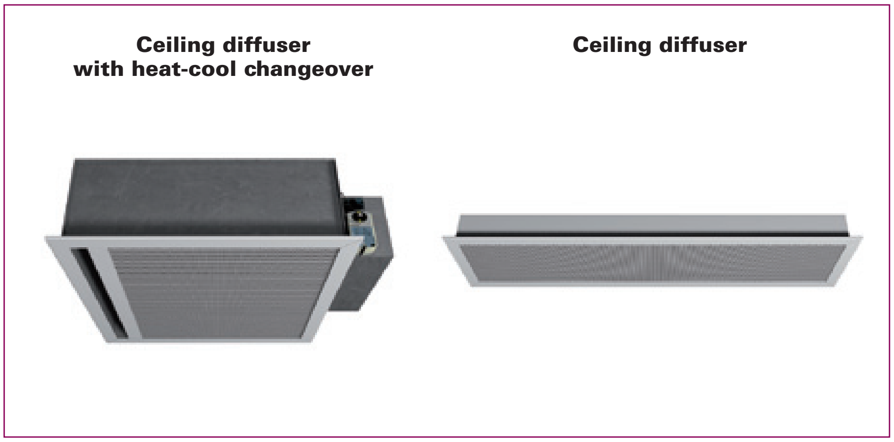


Figure 33: Ceiling Diffusers

Figure 34: DF1W-HC installed in wall

Industrial Diffusers

For the industrial environment, diffusers must be able to withstand impact from moving equipment or be mounted above the working space and designed to supply air deep into the space. Flat industrial displacement diffusers are intended to be placed on the industrial floor space and provide supply air. The robust design allows this diffuser to withstand the impact forces common to the industrial sector. Industrial diffusers are designed to be mounted above the occupied zone, and have integrated heating and cooling supply air modes (**Figure 35** and **Figure 36)**.

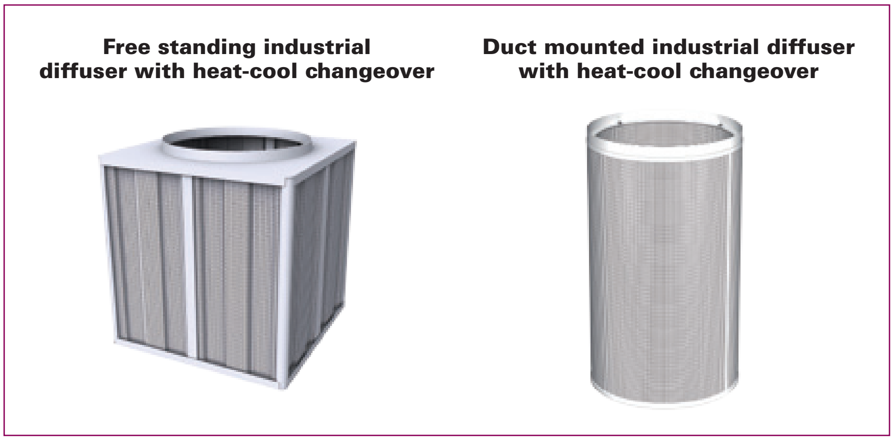


Figure 35: Industrial Diffusers

Figure 36: DFXi installed in floor

Design Procedure - Displacement Ventilation

ASHRAE

The following step by step design procedure is offered as a simplified approach to determine the ventilation rate and supply air temperature for typical displacement ventilation applications. The procedures presented are based on the findings of ASHRAE Research Project-949 (Chen, Glicksman, Yuan, Hu, & Yang, 1999) and the procedure outlined by Chen & Glicksman (2003).

The design procedure applies to typical North American office spaces and classrooms. These procedures should be used with care when applied to large spaces such as theaters or atria; a computational fluid dynamic analysis (CFD) of large spaces is recommended to optimize the air supply volume.

Only the sensible loads should be used for the preceding calculations. These calculations are only for determining the air flow requirements to maintain the set-point in the space; the total building load remains the same as with a mixing system.

Step 1: Determine the Summer Cooling Load

Use a cooling load program or the ASHRAE manual method to determine the design cooling load of the space in the summer. If possible, assume a 1.1 °F/ft [2 °C/m] vertical temperature gradient in the space for the computer simulation as the room air temperature is not uniform with displacement ventilation. Itemize the cooling load into the following categories:

- The occupants, desk lamps and equipment, q_{oe} (Btu/h [W])
- The overhead lighting, q_l (Btu/h [W])
- The heat conduction through the room envelope and transmitted solar radiation, q_{ex} (Btu/h [W])

Step 2: Determine the Cooling Load Ventilation Flow Rate, Q_{DV}:

The flow rate required for summer cooling, using standard air, is:

IP
$$Q_{DV} = \frac{0.295\, q_{oe} + 0.132\, q_l + 0.185\, q_{ex}}{60 \rho c_p \Delta t_{hf}} \quad \text{J1}$$

SI
$$Q_{DV} = \frac{0.295\, q_{oe} + 0.132\, q_l + 0.185\, q_{ex}}{\rho c_p \Delta t_{hf}} \quad \text{J1}$$

Where:

Q_{DV} = air required to satisfy the sensible cooling load in a DV system, cfm [L/s]
ρ = air density, lb/ft^3 [kg/m^3]
c_p = specific heat of the air at constant pressure, Btu/lb°F [kJ/kgK]
t_{hf} = temperature difference from head to foot level, °F [°C]

Step 3: Determine Flow Rate of Fresh Air, Q_{oz}:

ASHRAE Standard 62.1-2004 Ventilation Rate Procedure includes default values for ventilation effectiveness. From ASHRAE Standard 62.1-2004: equation 6-1 is used to determine the breathing zone outdoor air flow Vbz and equation 6-2 is used to determine the zone outdoor air flow Q_{oz}.

$$Q_{oz} = \frac{R_p P_z + R_A A_z}{E_z} \quad \text{J2}$$

where:

Q_{oz} = the required volume of outdoor air, as determined from ASHRAE Standard 62.1-2004, based on room application. Note that local codes may not allow the discount for the ventilation effectiveness, or may have stricter requirements.
R_p = outdoor air flow rate required per person, as determined from Table 6-1 in ASHRAE 62.1-2004, cfm/person [L/s person]
R_A = outdoor air flow rate required per unit area, as determined from Table 6-1 in ASHRAE 62.1, cfm/ft^2 [L/sm^2]
P_z = zone population: the largest number of people expected to occupy the zone during typical usage, persons
A_z = zone floor area, ft^2 [m^2]
E_z = the ventilation effectiveness of the air distribution system in the zone

Design Procedure - Displacement Ventilation

Step 4: Determine Supply Air Flow Rate, Q_s:

Choose the greater of the required flow rate for summer cooling and the required ventilation rate as the design flow rate of the supply air:

$$Q_s = \max\left[Q_{DV}, Q_{oz}\right] \quad \text{J3}$$

Step 5: Determine Supply Air Temperature, T_s:

The supply air temperature can be determined from equations and simplified to:

IP

$$t_s = t_{sp} - \Delta t_{hf} - \frac{A_z q_t}{2.456 Q_s^2 + 1.08 A Q_s} \quad \text{J4}$$

SI

$$t_s = t_{sp} - \Delta t_{hf} - \frac{A_z q_t}{0.584 Q_s^2 + 1.208 A Q_s} \quad \text{J4}$$

Step 6: Determine Exhaust Air Temperature

The exhaust air temperature can be determined by the following method:

IP

$$t_e = t_s + \frac{q_t}{1.08\left(Q_s\right)} \quad \text{J5}$$

SI

$$t_e = t_s + \frac{q_t}{1.208\left(Q_s\right)} \quad \text{J5}$$

Step 7: Evaluate Calculated Supply Temperature.

Since displacement ventilation provides the cool conditioned air along the floor level, a minimum supply air temperature of 63 °F should be observed to ensure the floor level does not become excessively cool. Occasionally the supply temperature calculated in Step 5 above will end up below 63 °F, in which case the following steps should be taken to rebalance the cooling airflow with a minimum supply temperature of 63 °F or higher.

Step 8: Rebalance Supply Air Volume (As required)

Using a derivation of equation 15.25, the supply air volume will be recalculated with the new supply air temperature, using the previous inputs and the calculated exhaust air temperature.

IP

$$Q_{DV} = \frac{q_t}{1.08\,(t_e - t_s)} \quad \text{J6}$$

SI

$$Q_{DV} = \frac{q_T}{1.208\,(t_e - t_{SS})} \quad \text{J6}$$

Step 9: Selection of Diffusers

The goal is to maximize comfort in the space and minimize the quantity of diffusers. At a maximum, Chen & Glicksman (2003) suggest a 40 fpm face velocity, but this value may increase or decrease depending on the space and comfort requirements. A CFD simulation can validate the design and is recommended for larger spaces.

REHVA

The Federation of European Heating and Air-conditioning Associations (REHVA) presents two procedures for the air volume calculation in DV systems. The first is based on thermal comfort, and calculates the air volume required to satisfy the loads in the space. As this is the most relevant to the current discussion, it will be presented here. The second is based on air quality, and predicts the contaminant distribution in a room. For areas where AQ is critical, this method (Skistad, 2002) could be used as a check after the thermal comfort procedure is used.

Design Procedure - Displacement Ventilation

The thermal comfort procedure offered by REHVA is based on the assumption that half of the temperature gain in the room is realized between the supply and the floor level. The REHVA procedure is not dissimilar to the ASHRAE methods, but it does have slightly different assumptions.

Step 1: Determine the Design Conditions and the Cooling Load
Determine total room load using a cooling load program or the ASHRAE manual method:

$$q_{total} = q_{oe} + q_l + q_{ex} \qquad \text{J7}$$

Step 2: Determine the Maximum Temperature Rise Through the Room
Using the design stratification, s, and the room height, h:

$$t_e - t_s = 2sh \qquad \text{J8}$$

Note: REHVA does not recommend that the maximum temperature rise exceed 18 °F [10 K] for standard commercial spaces with typical office ceiling heights (9-10 ft [2.75 – 3 m]), and recommends adjusting the value of s so that the evaluation of equation J7 is not greater than the limit.

Step 3: Calculate the Supply Air Temperature
The supply air temperature is calculated as:

$$t_s = t_{sp} - s\left(h + h_{sp}\right) \qquad \text{J9}$$

(for further explaination please refer to the Price Engineer's Handbook, Chapter 15 - Introduction to Displacement Ventilation)

Step 4: Determine Supply Air Flow Rate
Since:

IP
$$q = 60\rho c_p Q \Delta t \qquad \text{J10}$$

SI
$$q = \rho c_p Q \Delta t \qquad \text{J10}$$

and

IP
$$Q_{DV} = \frac{q_t}{60\rho c_p\left(t_e - t_s\right)} = \frac{1}{2}\frac{q_t}{60\rho c_p sh} \qquad \text{J11}$$

SI
$$Q_{DV} = \frac{q_t}{\rho c_p\left(t_e - t_s\right)} = \frac{1}{2}\frac{q_t}{\rho c_p sh} \qquad \text{J11}$$

As with the ASHRAE method, the supply air rate should satisfy both equation J7 and local code requirements:

$$Q_s = \max\left[Q_{DV}, Q_{oz}\right] \qquad \text{J3}$$

Step 5: Re-evaluate the Comfort Conditions
Calculate the temperature at the floor and ensure that the stratification limit has not been exceeded:

IP
$$t_{af} = t_s + \frac{1}{2}\frac{q_t}{60\rho c_p Q_s} \qquad \text{J12}$$

SI
$$t_{af} = t_s + \frac{1}{2}\frac{q_t}{\rho c_p Q_s} \qquad \text{J12}$$

$$s = \frac{t_e - t_s}{2h} \qquad \text{J13}$$

Diffuser Selection, Location and Layout

Component Selection

The aim of diffuser selection is to choose an air outlet that will perform well (the function will vary by application) and not cause discomfort. These goals do not change significantly when designing a displacement ventilation system. In fact, the procedure for selecting DV outlets is often significantly easier than with mixing diffusers. The room air flow characteristics of a DV system are such that they make concepts of throw, spread and drop meaningless. As discussed, it is the heat sources in the room that drive the air diffusion, not the momentum from the air outlet. In addition, the noise generated from displacement diffusers is so low that NC is often not a factor when choosing a product.

As a result, the primary factor when selecting a displacement diffuser is thermal comfort. As discussed in the thermal comfort section, the primary factors affecting thermal comfort with a displacement ventilation system are stratification and draft. The procedure for determining the air volume already accounts for stratification, and so the primary selection criteria with displacement diffusers is draft.

A common industry metric that evaluates the performance of DV outlets is the adjacent zone (AZ). The AZ defines the region around the outlet where the velocity is 40 fpm [0.2 m/s], 1 or 2 in. [25 or 50 mm] above the floor. Unfortunately, the measurement heights and lack of temperature influence does not correspond well with North American comfort standards. It it is, therefore, difficult to get a sense on how this data translates to the thermal comfort of occupants.

A more appropriate metric is the draft ratio from ASHRAE Standard 55-2004 and ISO 7730-2005. As discussed in Chapter 4—Introduction to Indoor Environmental Quality of the Price Engineer's Hadbook, DR identifies the percentage of people dissatisfied based on a combination of temperature, velocity and turbulence intensity. Performance data presented in this form gives a real idea of how these outlets will impact the thermal comfort in the zone, which is the primary concern for DV diffuser selection and layout. In both ASHRAE Standard 55-2004 and ISO 7730-2005, the range of acceptable DR is between 0 and 20.

Figure 37 and **Figure 38** show the DR performance of a floor mounted displacement diffuser at 5 °F [2.8 °C] and 10 °F [5.5 °C] cooling.

The discomfort due to draft, DR, is highest in areas closest to the diffuser, where the lowest temperatures and often the highest velocity exist. Moving away from the outlet, the velocity decreases and the supply air temperature increases due to entrainment of room air and convective heat transfer from surrounding surfaces while the draft is decreased. Because of this, higher supply air temperatures and lower diffuser face velocities are desirable in order to minimize draft. The face velocity will have an impact on the size of the selected diffuser. For example, 200 cfm out of a diffuser with a face velocity of 40 fpm will require 5 ft^2 of diffuser face, whereas a face velocity of 50 fpm only requires 4 ft2 of diffuser face.

ASHRAE (Chen & Glicksman, 2003) recommends a maximum face velocity of 40 fpm [0.2 m/s] for regularly occupied commercial spaces. In practice this number seems to be a good compromise between draft risk and diffuser size. Depending on the application, some adjustment to the face velocity is possible. If people are seated adjacent to the outlet, such as in a theater with diffusers in risers behind the seats, a lower face velocity is preferred. In transient spaces such as lobbies or airports, the engineer may be able to select higher face velocities, perhaps as high as 50 -55 fpm [0.25 – 0.275 m/s]. In areas where draft is less of a concern, such as in a machine shop, significantly higher face velocities may be appropriate. It is not unusual to have a face velocity as high as 100 fpm [0.5 m/s] in industrial spaces.

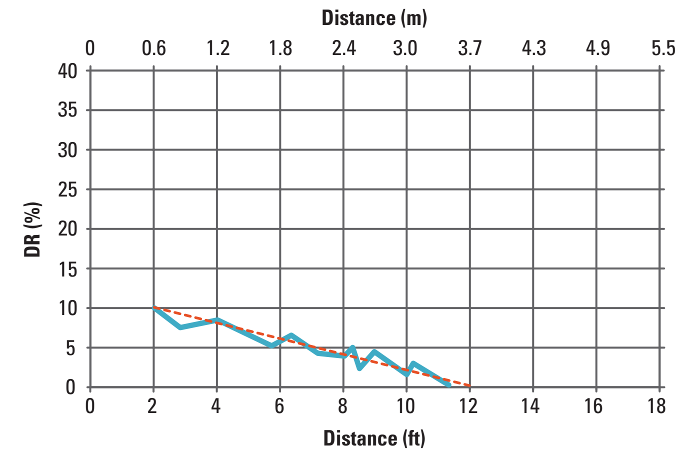


Figure 37: DR vs. distance, 40 fpm [0.2 m/s] face velocity, supply air 5 oF [2.7 °C] below troom

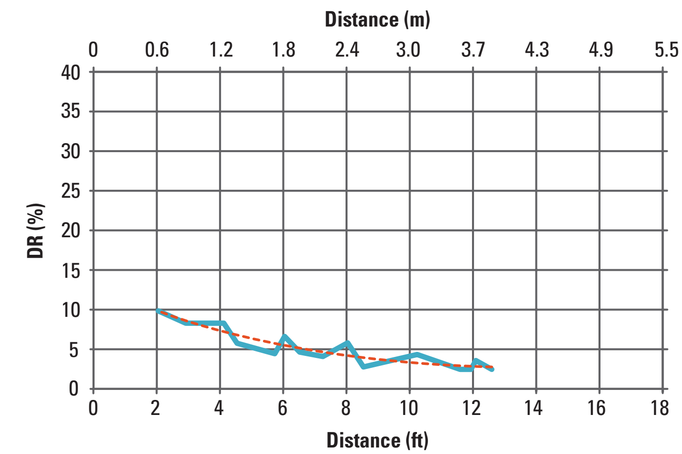


Figure 38: DR vs. distance, 40 fpm [0.2 m/s] face velocity, supply air 10 °F [2.7 °C] below troom

Diffuser Selection, Location and Layout

These typical face velocities are general rules of thumb and have been determined to be acceptable for most spaces. A more accurate method for determining an appropriate diffuser face velocity is to evaluate the draft, or DR, around the diffuser. The diffuser face velocity directly impacts the air velocity at the floor level, thereby affecting the draft. Laboratory testing can provide relationships between the diffuser face velocity and supply air temperature to the DR in a room. Other factors that will affect comfort with a given velocity and temperature include:

- Occupant's metabolic rate
- Occupant's clothing level
- Occupant density
- Room loads

Due to these complex relationships, CFD modeling or selection software may be used to refine the diffuser size and face velocity based on a specific application.

The selection of a specific diffuser is often as much of an architectural choice as an engineering one. For the most part, various diffuser types all supply air in a similar pattern; the type of diffuser will therefore not have a large impact on the room air dynamics. The type of diffuser is selected in order suit the architectural requirements, integrate into millwork, incorporate heat, or promote an even air distribution in the space.

Once the air volume has been determined, the next step is to select the product. Some of the products feature options that are required for the application, such as a diffuser with integrated heat or a diffuser that is face adjustable for providing personal control. If the type of diffuser is already determined due to room constraints or application (floor mounted diffusers for an open office, for example), then the next step is to divide the room air volume by the face velocity:

$$A_{face} = \frac{Q_t}{V_{face}}$$

Knowing the face area, the designer can then determine either the size or number of diffusers required. Product performance pages include information about the diffuser face size to facilitate this selection, as shown in **Table 2**.

In general, selecting taller diffusers will cause a larger area with a DR of 20 or above due to the cool supply air having a longer path to the floor. This air gains momentum whenever travelling in the direction of the buoyancy forces (vertically), which will lead to higher velocity along the floor adjacent to the outlet. Diffusers that are wider do not have a significant impact on the local DR due to the constant diffuser height. As a result, it is often preferred to add face area by adding length or width to the diffuser before adding height if more diffuser capacity is required.

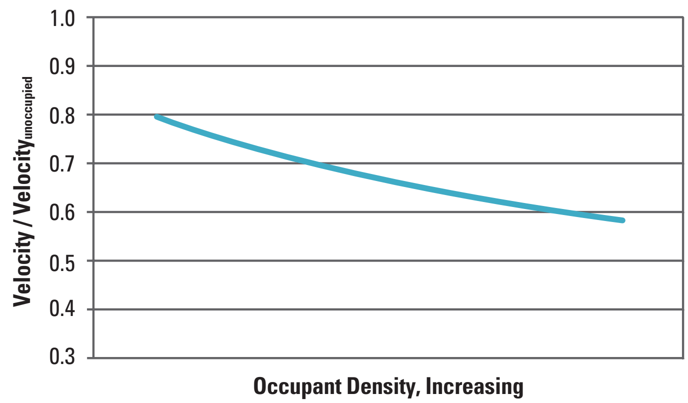


Figure 39: Reduction in air velocity due to occupant density

Unit Size, in. [Face Area. ft²] W x H x D	Inset Size in.	Face Velocity fpm	Air Flow cfm	Total Pressure in. w.g.	Static Pressure in. w.g.	Noise Criteria NC
24x24x13 [3.6]	8	20	72	---	---	---
	8	30	108	0.02	---	---
	8	40	144	0.03	---	---
	8	50	180	0.05	0.03	---
48x24x13 [7.4]	8	20	148	0.02	---	---
	8	30	222	0.04	---	---
	8	40	295	0.07	0.02	---
	8	50	369	0.11	0.04	20
60x24x13 [9.3]	8	20	186	0.02	---	---
	8	30	278	0.05	---	---
	8	40	371	0.09	---	17
	8	50	464	0.14	0.03	25
	10	30	278	0.03	---	---
	10	40	371	0.06	0.03	---
48x36x16 [11.3]	10	20	226	0.02	---	---
	10	30	338	0.04	---	---
	10	40	451	0.08	0.03	16
	10	50	564	0.12	0.05	23
	12	30	338	0.03	---	---

Table 2: DF1 Series - performance data in IP units

For more information on additional imperial and metric sizes please visit priceindustries.com or contact your local Price representative.

Diffuser Selection, Location and Layout

Diffuser Layout

By primarily considering comfort and trusting that the heat sources will effectively distribute the air through the room, the concepts of coverage or colliding air streams are not really applicable. Instead, there are several rules of thumb that can help the designer to lay the diffusers out in the room. ASHRAE (Chen & Glicksman, 2003) recommends the following:

- There should not be large obstacles near the diffusers.
- The diffusers should be placed on the walls opposite the exterior walls/windows.
- The diffusers can be placed in the center of a room around a column, for example.
- More diffusers should be placed in the spaces with higher cooling load.

Even though the heat sources move the supply air throughout the zone, it is good practice to supply the air in such a way that promotes even air distribution. Using multiple smaller diffusers instead of a single large one can help supply the air to all corners of the room while also improving the thermal comfort.

Some rules of thumb that may be used are:

- For rooms with dimensions larger than 30 ft [9 m], consider using multiple outlets, evenly spaced or mounting the diffuser on two opposite walls or corners.
- For large open spaces, such as a casino or exhibition hall where there is limited wall space against which to locate diffusers, supply outlets should be located in the middle of the zone.
- When ducting from below a diffuser, it is important to supply the diffuser with a base for easy connection to the diffuser.
- When mounting displacement diffusers along walls, it is important to provide support in order to hold the weight of the outlets.
- In installations where the ductwork is supplied from above the diffuser and needs to be hidden, the use of a duct cover will properly conceal the ductwork. If a perforated cover is preferred, the ductwork should be painted to conceal it completely.

PRODUCT TIP

To conceal ductwork located between the ceiling and the floor mounted diffuser, a duct covering may be used. These covers are designed to match the look of the diffuser for a consistent architectural finish.

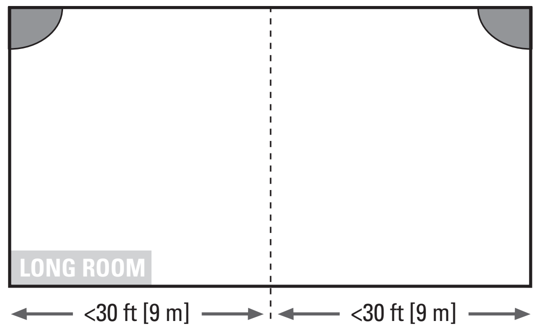


Figure 40: Long rooms

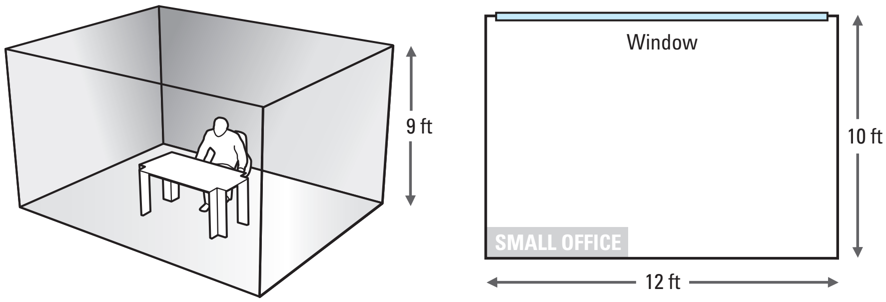
There are some additional factors to keep in mind for ceiling mounted diffusers, as well as sloped floor applications. The heavier supply air will make its way down to the lowest level in the space. For ceiling mounted diffusers, this would typically mean the floor. Care must be taken in these instances not to locate the diffuser directly above seated occupants due to the risk of draft caused by the air passing through the zone. The ideal location for ceiling mounted diffusers in an office environment is against an interior wall, which will ensure that the chances of someone being seated below the outlet is low. It will also make use of the Coanda effect (refer to Chapter 2—Introduction to Fluid Mechanics of the Price Engineer's Handbook for more information) to pull the supply air against the wall, reducing the amount of mixing that occurs between the supply air and the warm polluted air trapped at the ceiling, and thereby reducing the potential reduction in air quality from entrainment of pollutants from the return air.

In the case of a sloped floor, such as in a theater, the heavier air will make its way down to the lowest point, potentially reducing the air temperature in that zone and starving the higher elevations.

Location of Return Grilles

The only requirement for return grilles is that they are placed at the highest point in the space. If not all of the return air can be drawn from this location, placement of returns below the highest point is acceptable, as long as there is some relief at the peak to remove any trapped air or moisture. The model type does not change from mixing systems.

Example 1 — Office Design (IP)



Space Design

The owner of an office building is renovating and would like to consider using displacement ventilation in the office areas. This example examines a small office in this space. The office is a north facing room, used primarily during the hours from 8:00 am to 5:00 pm. The space is designed for 2 occupants, a computer with LCD monitor, T8 florescent lighting, and has a control temperature of 72 °F. The room is 10 ft wide, 12 ft long, and 9 ft from floor to ceiling. The owner expressed interest in supplying the office spaces with wall mounted displacement diffusers or corner displacement diffuser as space is limited.

Design Considerations	
Occupants	2
Set-Point	72 °F
Floor Area	120 ft^2
Exterior Wall	90 ft^2
Volume	1080 ft^3
q_{oz}	800 Btu/h
q_l	825 Btu/h
q_{ex}	450 Btu/h
q_T	2075 Btu/h

Space Considerations

One of the primary considerations when using a DV system is comfort. As previously discussed, ASHRAE Standard 55-2010 stipulates the maximum combination of velocity and temperature in the occupied zone, PPD due to draft, as well as the stratification in the space. According to ASHRAE Standard 55-2010, the recommended stratification limit between head and foot is 5.4 °F.

The assumptions made for the space are as follows:

- Load per person is 250 Btu/h
- Lighting load in the space is 6.82 Btu/hft^2
- Computer load is 308 Btu/h (CPU and LCD Monitor)
- Conduction through the window and wall is 5 Btu/hft^2
- The specific heat and density of the air for this example will be 0.24 Btu/lb°F and 0.075 lb/ft^3 respectively.

Example 1 — Office Design (IP)

Using the ASHRAE Procedure (Chen & Glicksman, 2003)

The loads are broken down as follows:

q_{oz} = (2 People × 250 Btu/h) + 300 Btu/h = 800 Btu/h

q_l = 120 ft² × 6.87 Btu/hft² = 825 Btu/h

q_{ex} = 90 ft² × 5 Btu/hft² = 450 Btu/h

q_T = 2075 Btu/h

Total cooling load for this space (q_T) is 2075 Btu/h, and 17.33 Btu/hft².

ASHRAE Standard 62-2004 requires 0.06 cfm/ft² outdoor air flow rate per unit area, R_a, and 5 cfm/person outdoor air flow rate per person, R_p, be delivered to the space for moderately active office work applications. For displacement ventilation, ventilation effectiveness or zone air distribution effectiveness (E_z) is assumed to be 1.2 (Table 6-2, ASHRAE Standard 62-2004).

Step 1: Determine the air flow rate to meet the cooling load.

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

$$Q_{DV} = \frac{0.295(800) + 0.132(825) + 0.185(450)}{60(.24)(.075)5.4}$$

$$Q_{DV} = 73\text{ cfm}$$

Step 2: Determine fresh air flow rate.

$$Q_{oz} = \frac{R_p P_z + R_A A_z}{E_z}$$

$$Q_{oz} = \frac{(5)2 + (.06)120}{1.2} = 14\text{ cfm}$$

Note: Some local codes may not allow the discount for q_{oe}, or may have stricter requirements, and they should be used instead of this calculation.

The total supply air volume for cooling is then the maximum value between Q_{DV} and Q_{oz}.

$$Q_s = \max\left[Q_{DV}, Q_{oz}\right] = 73\text{ cfm}$$

Step 3: Calculate the supply air temperature.

$$t_s = t_{sp} - \Delta t_{hf} - \frac{A_z q_T}{2.456 Q_s^2 + 1.08 A Q_s} = 72 - 5.4 - \frac{(120)(2075)}{2.456(73)^2 + 1.08(120)(73)} = 55.6\text{ °F}$$

Step 4: Determine the return air temperature.

$$t_e = t_s + \frac{q_T}{1.08 Q_s} = 55.6\text{ °F} + \frac{(2075)}{1.08(73)} = 81.8\text{ °F}$$

Example 1 — Office Design (IP)

Step 5: Adjust for new supply temperature

The supply temperature should be 10 °F less than tsp or 63 °F, whichever is higher.

$$t_s = 63\,°\mathrm{F}$$

$$Q_{DV} = \frac{q_t}{60\rho c_p (t_e - t_s)} = \frac{2075}{60(0.075)(0.24)(82-63)}$$

$$Q_{DV} = 101 \text{ cfm}$$

Using the REHVA Thermal Comfort Procedure (Skistad, 2002)

Step 1: Determine the design conditions and the cooling load.

$$q_t = 2075\,\mathrm{Btu/h}$$

Step 2: Determine the maximum temperature rise through the room.

$$t_e - t_s = 2sh = 2\left(1.1\frac{°\mathrm{F}}{\mathrm{ft}}\right)(9) = 19.8\,°\mathrm{F}$$

This is greater than the 18 °F maximum temperature rise recommended by REHVA, adjusting accordingly:

$$t_e - t_s = 2s'h = 18\,°\mathrm{F}$$

$$s' = \frac{18\,°\mathrm{F}}{2(9\,\mathrm{ft})} = 1\frac{°\mathrm{F}}{\mathrm{ft}}$$

Step 3: Calculate the supply air temperature.

$$t_s = t_{sp} - s\left(h + h_{sp}\right)$$

$$t_s = 72\,°\mathrm{F} - (1)\left(9\,\mathrm{ft} + 42\,\mathrm{in}\frac{1\,\mathrm{ft}}{12\,\mathrm{in}}\right) = 59.5\,°\mathrm{F}$$

Step 4: Determine supply air flow rate.

$$Q_{DV} = \frac{1}{2}\frac{q_t}{60\rho c_p s'h}$$

$$Q_{DV} = \frac{1}{2}\frac{2075\,\mathrm{Btu/h}}{60\left(0.24\,\mathrm{lb/ft^3}\right)\left(0.075\,\mathrm{Btu/lb_m\,°F}\right)\left(1\,°\mathrm{F/ft}\right)\left(9\,\mathrm{ft}\right)} = 107\,\mathrm{cfm}$$

As with the ASHRAE method, the supply air rate should satisfy both equation J10 and local code requirements:

$$Q_s = \max\left[Q_{DV}, Q_{oz}\right] = 107\,\mathrm{cfm}$$

Example 1 — Office Design (IP)

Step 5: Re-evaluate the comfort conditions.

Calculate the temperature at the floor and ensure that the stratification limit has not been exceeded:

$$t_{af} = 59.5\ °\text{F} + \frac{1}{2}\frac{2075\ \text{Btu/h}}{1.08\left(107\ \text{cfm}\right)} = 68.5\ °\text{F}$$

This corresponds to a 3.5 °F temperature differential between head and foot, which is acceptable according to ASHRAE Standard 55-2004.

It is important to note that the supply air temperature in this example is below what is generally recommended for DV applications. If this example was redone using a cooling differential, t_{sp}-t_s = 10 °F, the following values would be obtained:

$$t_s = 62\ °\text{F}$$

$$t_s = t_{sp} - s''\left(h + h_{sp}\right)$$

$$s'' = \frac{t_{sp} - t_s}{\left(h + h_{sp}\right)} = \frac{10\ °\text{F}}{\left(9\ \text{ft} + 42\ \text{in}\dfrac{1\ \text{ft}}{12\ \text{in}}\right)} = 0.8\frac{°\text{F}}{\text{ft}}$$

$$Q_{DV} = \frac{1}{2}\frac{q_t}{60\rho c_p s'' h} = 133\ \text{cfm}$$

$$t_{af} = 62\ °\text{F} + \frac{1}{2}\frac{2075\ \text{Btu/h}}{1.08\left(133\ \text{cfm}\right)} = 69\ °\text{F}$$

$$t_e = t_s + 2s''h = 69\ °\text{F} + 2\left(0.8\frac{°\text{F}}{\text{ft}}\right)\left(9\ \text{ft}\right) = 83.4\ °\text{F}$$

Comparing all solutions in this example:

Value	ASHRAE	REHVA	REHVA (Adjusted Ts)
t_s	62.7 °F	59.5 °F	62 °F
Q_s	110 cfm	107 cfm	133 cfm
t_e	81.8 °F	77.5 °F	83.4 °F
θ_f	0.32	0.5	0.5
t_{hf}	3.6 °F	3.5 °F	3 °F

In the table we notice that the supply air temperature from the ASHRAE procedure is higher than that of REHVA, with the exception of the iterative procedure. This can be largely attributed to the way each method predicts the room loads' contribution to the heat gain in the occupied zone, as is also shown in the variance in the value of θ_f between the cases.

For additional REHVA sample calculations please refer to the Price Engineer's Handbook.

Example 1 — Office Design (IP)

Selection of Diffusers

For this application we are limited to wall mounted or corner diffusers at the request of the owner. Traditional displacement diffusers are limited to 40 fpm face velocity in standard commercial applications in order to meet comfort criteria. With a supply air rate of 101 cfm and a face velocity of 40 fpm, 2.53 ft^2 of diffuser face area is required. For the Price DF1W, DF1R or DF1C, a 24 in. x 18 in. diffuser will provide a face area of 3 ft^2. A Price DR90 unit that is 30 in. tall and has an 18 in. diameter will provide a face area of 2.94 ft^2.

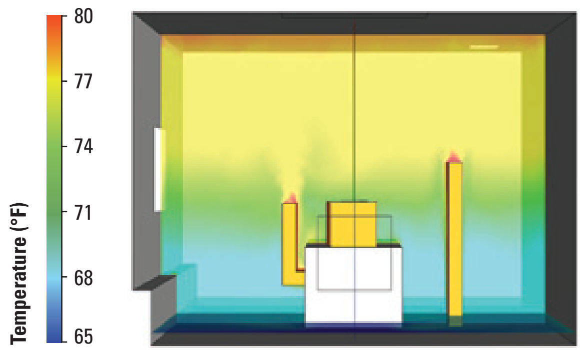
Layout of the Office

The corner diffusers could be placed in any of the corners to supply this room, as long as the occupant is comfortable. The wall diffusers can be placed on any of the walls in the room.

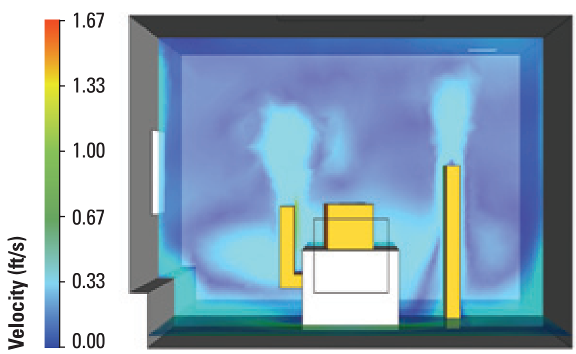
Flow Visualization

A CFD analysis was run for this example using the conditions, calculated air flow, and supply air temperature for the small office with the Price DF1W to give a visual representation of the temperature distribution, air movement, and draft temperatures.

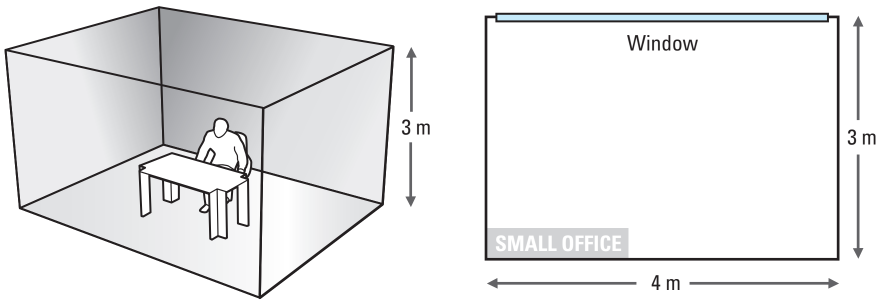
The CFD plot of air temperature is shown below. The DF1W produces the predicted temperature stratification in the space. Also visible are the heat plumes off the occupants and computer. The seated occupant experiences ambient air temperatures from 69 °F to 72 °F, and the standing occupant 69 °F to 75 °F. Both are within the thermal stratification comfort conditions set by ASHRAE (Chen & Glicksman, 2003).



The velocity profile illustrates the slow moving air throughout the space. The images also show the plumes off the occupants and computer, as well as the general shape of the air pattern leaving the diffuser.



Example 1 — Office Design (SI)



Space Design

The owner of an office building is renovating and would like to consider using displacement ventilation in the office areas. This example examines a small office in this space. The office is a north facing room, used primarily during the hours from 8:00 am 5:00 pm. The space is designed for 2 occupants, a computer with LCD monitor, T8 florescent lighting, and has a control temperature of 22 °C. The room is 3 m wide, 4 m long, and 3 m from floor to ceiling. The owner expressed interest in supplying the office spaces with wall mounted displacement diffusers or corner displacement diffuser as space is limited.

Design Considerations	
Occupants	2
Set-Point	22 °C
Floor Area	12 m^2
Exterior Wall	9 m^2
Volume	36 m^3
q_{oz}	210 W
q_l	300 W
q_{ex}	135 W
q_T	645 W

Space Considerations

One of the primary considerations when using a DV system is comfort. As previously discussed, ASHRAE Standard 55-2010 stipulates the maximum combination of velocity and temperature in the occupied zone, PPD due to draft, as well as the stratification in the space. The stratification according to ASHRAE Standard 55-2010 and ASHRAE Research Project-949 (Chen et al., 1999) is 3 °C.

The assumptions made for the space are as follows:

- Load per person is 75 W
- Lighting load in the space 25 W/m^2
- Computer load is 60 W (CPU and LCD Monitor)
- Conduction through the window and wall is 15 W/m^2
- The specific heat and density of the air for this example will be 1.007 kJ/(kgK) and 1.2 kg/m^3 respectively.

Example 1 — Office Design (SI)

Using the ASHRAE Procedure (Chen & Glicksman, 2003)

The loads are broken down as follows:

q_{oz} = (2 People × 75 W) + 60 W = 210 W

q_l = 12 m2 × 25 W/m2 = 300 W

q_{ex} = 9 m² × 15 W/m² = 135 W

q_T = 645 W

Total cooling load for this space (q_T) is 645 W, and approximately 54 W/m².

ASHRAE Standard 62-2004 requires 0.3 L/s m² outdoor air flow rate per unit area, R_a, and 2.5 L/s per person outdoor air flow rate per person, R_p, be delivered to the space for moderately active office work applications. For displacement ventilation, ventilation effectiveness or zone air distribution effectiveness (E_z) is assumed to be 1.2 (Table 6-2, ASHRAE Standard 62-2004).

Step 1: Determine the air flow rate to meet the cooling load.

$$Q_{DV} = \frac{0.295 q_{oe} + 0.132\, q_l + 0.185\, q_{ex}}{\rho c_p \Delta\, t_{hf}}$$

$$Q_{DV} = \frac{0.295(210) + 0.132(300) + 0.185(135)}{(1.2)(1.007)(3)}$$

$$Q_{DV} = 34.9 \text{ L/s}$$

Step 2: Determine fresh air flow rate.

$$Q_{oz} = \frac{R_p P_z + R_A A_z}{E_z}$$

$$Q_{oz} = \frac{(2.5)2 + (0.3)(12)}{1.2} = 7.2 \text{ L/s}$$

Note: Some local codes may not allow the discount for q_{oe}, or may have stricter requirements, and they should be used instead of this calculation.

The total supply air volume for cooling is then the maximum value between Q_{DV} and Q_{oz}.

$$Q_s = \max\left[\, Q_{DV}, Q_{oz} \right] = 34.9 \text{ L/s}$$

Step 3: Calculate the supply air temperature.

$$t_s = t_{sp} - \Delta t_{hf} - \frac{A_z q_T}{0.584 Q_s^2 + 1.208 A Q_s} = 22 - 3 - \frac{(12)(645)}{0.584(34.9)^2 + 1.208(12)(34.9)} = 12.6\ °\text{C}$$

Step 4: Determine the return air temperature.

$$t_e = t_s + \frac{q_T}{1.208 Q_s} = 12.6\ °\text{C} + \frac{645}{1.208(34.9)} = 27.9 °\text{C}$$

Example 1 — Office Design (SI)

Step 5: Adjust for new supply temperature.

The supply temperature should be 5.5 °C less than tsp or 17 °C, whichever is higher.

$$t_s = 17\ ^\circ\mathrm{C}$$

$$Q_{DV} = \frac{q_T}{\rho\, c_p\,(t_e - t_{SS})} = \frac{645}{(1.21)(1.006)(27.9 - 17)}$$

$$Q_{DV} = 48.6\ \mathrm{L/s}$$

Using the REHVA Thermal Comfort Procedure (Skistad, 2002)

Step 1: Determine the design conditions and the cooling load.

$$q_t = 645\ \mathrm{W}$$

Step 2: Determine the maximum temperature rise through the room.

$$t_e - t_s = 2sh = 2\left(2\frac{\mathrm{K}}{\mathrm{m}}\right)(3\,\mathrm{m}) = 12\ \mathrm{K} = 12\ ^\circ\mathrm{C}$$

This is greater than the 10 °C maximum temperature rise recommended by REHVA, adjusting accordingly:

$$t_e - t_s = 2s'h = 10\ ^\circ\mathrm{C}$$

$$s' = \frac{10\ ^\circ\mathrm{C}}{2(3\ \mathrm{m})} = 1.67\frac{^\circ\mathrm{C}}{\mathrm{m}}$$

Step 3: Calculate the supply air temperature.

$$t_s = t_{sp} - s\left(h + h_{sp}\right)$$

$$t_s = 22\ ^\circ\mathrm{C} - (1.67)(3\ \mathrm{m} + 1.1\ \mathrm{m}) = 15.2\ ^\circ\mathrm{C}$$

Step 4: Determine supply air flow rate.

$$Q_{DV} = \frac{1}{2}\frac{q_t}{\rho c_p s'h}$$

$$Q_{DV} = \frac{1}{2}\frac{645\ \mathrm{W}}{(1.2\ \mathrm{kg/m^3})(1.007\ \mathrm{kJ/kg\,K})(1.67\ ^\circ\mathrm{C/m})(3\ \mathrm{m})} = 53\ \mathrm{L/s}$$

As with the ASHRAE method, the supply air rate should satisfy both equation J10 and local code requirements:

$$Q_s = \max\left[Q_{DV}, Q_{oz}\right] = 53\ \mathrm{L/s}$$

Example 1 — Office Design (SI)

Step 5: Re-evaluate the comfort conditions.

Calculate the temperature at the floor and ensure that the stratification limit has not been exceeded:

$$t_{af} = 15.2\ °\text{C} + \frac{1}{2}\,\frac{645\ \text{W}}{1.2\,(1.007)(53)} = 20.2\ °\text{C}$$

This corresponds to a 1.8 °C temperature differential between head and foot, which is acceptable according to ASHRAE Standard 55-2004.

It is important to note that the supply air temperature in this example is below what is generally recommended for DV applications. If this example was redone using a cooling differential, t_{sp}-t_s = 5.5 °C, the following values would be obtained:

$$t_s = 16.5\ °\text{C}$$

$$t_s = t_{sp} - s''(h + h_{sp})$$

$$s'' = \frac{t_{sp} - t_s}{(h + h_{sp})} = \frac{5.5\ °\text{C}}{(3\ \text{m} + 1.1\ \text{m})} = 1.3\frac{°\text{C}}{\text{m}}$$

$$Q_{DV} = \frac{1}{2}\,\frac{q_t}{\rho c_p s'' h} = 68.5\ \text{L/s}$$

$$t_{af} = 16.5\ °\text{C} + \frac{1}{2}\,\frac{645\ \text{W}}{1.2\,(1.007)(68.5)} = 20.4\ °\text{C}$$

$$t_e = t_s + 2s''h = 16.5\ °\text{C} + 2\left(1.3\frac{°\text{C}}{\text{m}}\right)(3\ \text{m}) = 24.3\ °\text{C}$$

Comparing all solutions in this example:

Value	ASHRAE	REHVA	REHVA (Adjusted Ts)
t_s	16.7 °C	15.2 °C	16.5 °C
Q_s	52 L/s	53 L/s	68.5 L/s
t_e	25.6 °C	25.2 °C	24.3 °C
θ_f	0.32	0.5	0.5
t_{hf}	2 °C	2 °C	1.43 °C

In the table we notice that the supply air temperature from the ASHRAE procedure is higher than that of REHVA, with the exception of the iterative procedure. This can be largely attributed to the way each method predicts the room loads' contribution to the heat gain in the occupied zone, as is also shown in the variance in the value of θ_f between the cases.

For additional REHVA sample calculations please refer to the Price Engineer's Handbook.

Example 1 — Office Design (SI)

Selection of Diffusers

For this application we are limited to wall mounted or corner diffusers at the request of the owner. Traditional displacement diffusers are limited to 0.2 m/s face velocity in standard commercial applications in order to meet comfort criteria. With a supply air rate of 49 L/s and a face velocity of 0.2 m/s, 0.245 m^2 of diffuser face area is required. For the Price DF1W, DF1R or DF1C, a 600 mm x 450 mm diffuser will provide a face area of 0.27 m^2. A Price DR90 unit with an 450 mm diameter and 750 mm tall will provide a face area of 0.265 m^2.

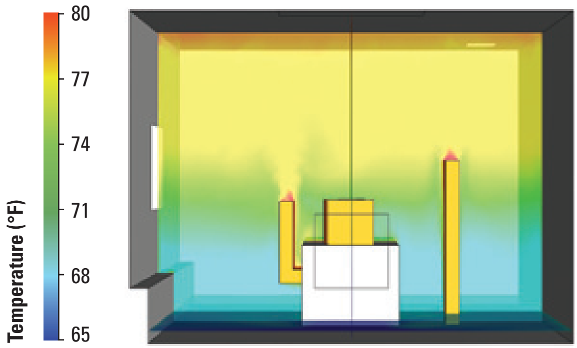
Layout of the Office

The corner diffusers could be placed in any of the corners to supply this room, as long as the occupant is comfortable. The wall diffusers can be placed on any of the walls in the room.

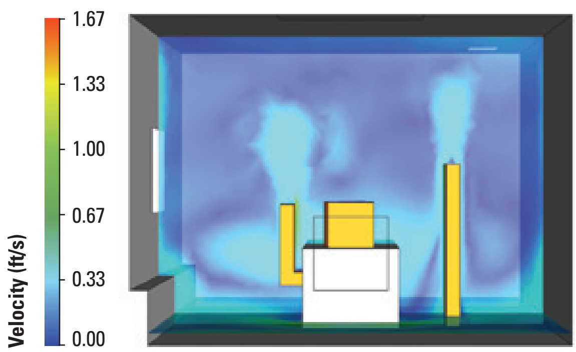
Flow Visualization

A CFD analysis was run for this example using the conditions, calculated air flow, and supply air temperature for the small office with the Price DF1W to give a visual representation of the temperature distribution, air movement, and draft temperatures in the space.

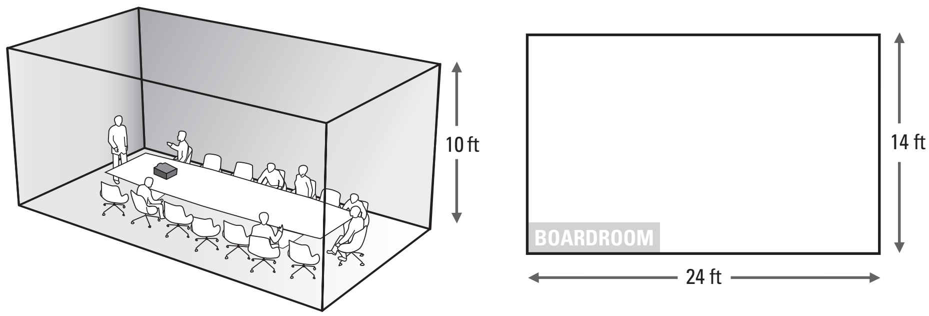
The CFD plot of air temperature is shown below. The DF1W produces the predicted temperature stratification in the space. Also visible are the heat plumes off the occupants and computer. The seated occupant experiences ambient air temperatures from 20.6 °C to 22.2 °C, and the standing occupant 20.6 °C to 23.9 °C. Both are within the thermal stratification comfort conditions set by ASHRAE (Chen & Glicksman, 2003).



The velocity profile illustrates the slow moving air throughout the space. The images also show the plumes off the occupants and computer, as well as the general shape of the air pattern leaving the diffuser.



Example 2 — Boardroom Design A (IP)



Space Design

The owner of a new office building wants to use a displacement ventilation system for all occupied spaces. This example examines a private boardroom that is located in the center of the building without any exterior surfaces. The space is designed for 8 occupants, a computer with LCD monitor, a projector, T8 florescent lighting, and has a control temperature of 72 °F. The room is 24 ft wide, 14 ft long, and 10 ft from floor to ceiling. There is a large whiteboard at the west side of the room and cabinets along the south and east sides of the room. The owner and architect want the displacement diffusers in the space to fit seamlessly into the room.

Design Considerations	
Occupants	8
Set-Point	72 °F
Floor Area	336 ft²
Volume	3360 ft²
q_{oz}	2496 Btu/h
q_l	2292 Btu/h
q_{ex}	0 Btu/h
q_T	4788 Btu/h

Space Considerations

Some of the assumptions made for the space are as follows:

- The head to foot gradient recommended by ASHRAE (Chen & Glicksman, 2003) is 5.4 °F for seated occupants.
- Load per person is 250 Btu/h
- Lighting load in the space is 6.82 Btu/hft²
- Computer and LCD load is 308 Btu/h
- Projector load is 188 Btu/h
- The specific heat and density of the air for this example will be 0.24 Btu/lb °F and 0.075 lb/ft³ respectively.

The loads are broken down as follows:

$$q_{oz} = (8 \text{ People} \times 250 \text{ Btu/h}) + 308 \text{ Btu/h} + 188 \text{ Btu/h} = 2492 \text{ Btu/h}$$

$$q_l = 336 \text{ ft}^2 \times 6.82 \text{ Btu/h/ft}^2 = 2296 \text{ Btu/h}$$

$$q_{ex} = 0 \text{ Btu/h}$$

$$q_T = 4788 \text{ Btu/h}$$

Total cooling load for this space (q_T) is 4788 Btu/h, and approximately 14.5 Btu/hft².

ASHRAE Standard 62-2004 requires 0.06 cfm/ft² outdoor air flow rate per unit area, R_a, and 5 cfm/person outdoor air flow rate per person, R_p, be delivered to the space for moderately active office work applications. For displacement ventilation, ventilation effectiveness or zone air distribution effectiveness (E_z) is assumed to be 1.2 (Table 6-2, ASHRAE Standard 62-2004).

Example 2 — Boardroom Design A (IP)

Determine the air flow rate to meet the cooling load.

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

$$Q_{DV} = \frac{0.295(2496) + 0.132(2292) + 0.185(0)}{60\,(.24)(.075)\,5.4}$$

$$Q_{DV} = 178 \text{ cfm}$$

Determine the fresh air flow rate.

$$Q_{oz} = \frac{R_p P_z + R_A A_z}{E_z}$$

$$Q_{oz} = \frac{(5)8 + (.06)336}{1.2}$$

$$Q_{oz} = 50 \text{ cfm}$$

Note: Some local codes may not allow the discount for Q_{oz}, or may have stricter requirements, and they should be used instead of this calculation.

The total supply air volume for cooling is then the maximum value between Q_{DV} and Q_{oz}.

$$Q_s = \max\left[\,Q_{DV}, Q_{oe}\right] = 178 \text{ cfm}$$

Calculate the supply air temperature.

$$t_s = t_{sp} - 5.4 - \left(\frac{q_t}{1.08\,Q_s}\right)\left(\frac{A}{2.27376\,Q_s + 1.08(A)}\right)$$

$$t_s = 72 - 5.4 - \left(\frac{4788}{1.08(267)}\right)\left(\frac{336}{2.27376(178) + 1.08(336)}\right)$$

$$t_s = 59\ °\text{F}$$

Determine the return air temperature.

$$t_e = t_s + \frac{q_t}{1.08(Q_s)} = 59 + \frac{4788}{1.08(178)} = 84°\text{F}$$

Adjust for new supply temperature

The supply temperature should be 10 °F less than tsp or 63 °F, whichever is higher.

$$t_s = 63°\text{F}$$

$$Q_{DV} = \frac{q_t}{60\,\rho\,c_p(t_e - t_s)} = \frac{4788}{60(0.075)(0.24)(82-63)}$$

$$Q_{DV} = 211 \text{ cfm}$$

Example 2 — Boardroom Design A (IP)

Selection of Diffusers

For this application we have three goals set by the owner:

1. Quiet operation
2. Thermal comfort to the space
3. Hidden diffusers

Inherently, displacement ventilation diffusers are quiet, but care has to be taken to limit the sound generated from the HVAC air supply. Price recommends limiting the duct velocity to 1200 fpm in order to minimize noise from ductwork. For thermal comfort, a face velocity of 40 fpm is required. At 211 cfm a diffuser face area of 5.275 ft^2 would be required.

There are two options to make these diffusers as unobtrusive as possible: mount them in the wall or as part of the furniture.

Layout of the Boardroom

For a concealed look, the Price DF1R displacement diffuser could be installed at the base of the cabinets or in the wall under the whiteboard in a pressurized plenum. Two diffusers at 48 in. x 8 in. will be able to meet the 40 fpm requirement. The diffusers can be placed on any of the walls in the room, but it is essential to ensure that sedentary occupants will be located a comfortable distance from the diffuser.

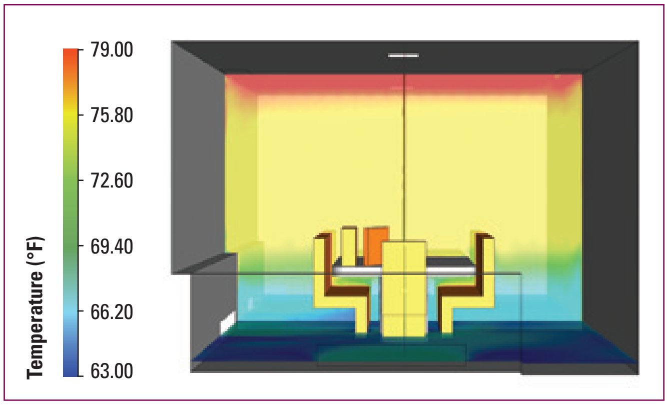
Flow Visualization

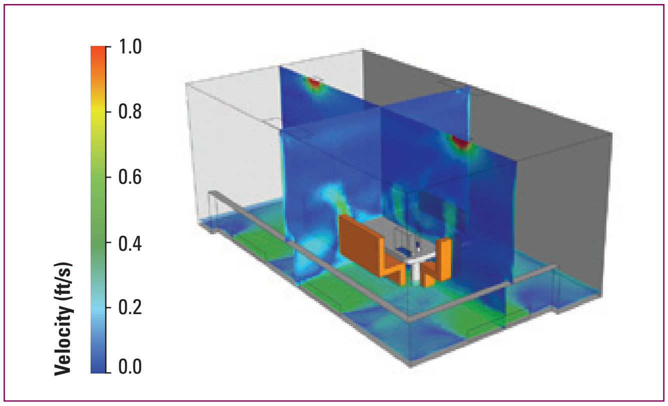
A CFD analysis was run for this example using the conditions, calculated air flow, and supply air temperature for the boardroom to give a visual representation of the temperature distribution, air movement, and draft temperatures in the space.

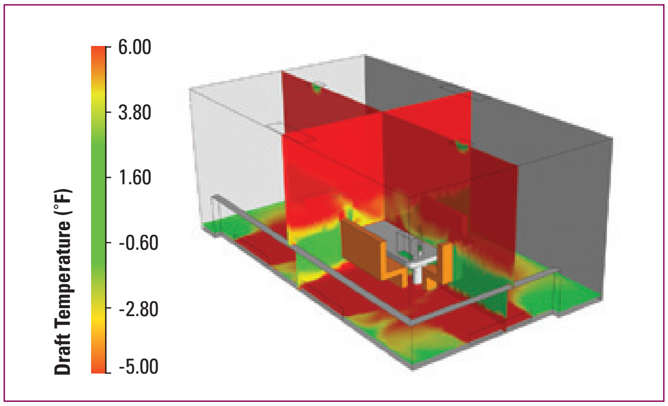
The image on the following page shows the temperature profiles across the space and a reasonable temperature stratification is predicted. Also visible are the heat plumes off the occupants and computer. The seated occupant experiences ambient air temperatures from 69 °F to 72 °F, and the standing occupant 69 °F to 75 °F. Both are within the thermal stratification comfort conditions set by ASHRAE (Chen & Glicksman, 2003).

The image below depicts the velocity profile. The images show the plumes off the occupants and computer, as well as the general shape of the air pattern leaving the diffuser and slowly entering the zone.

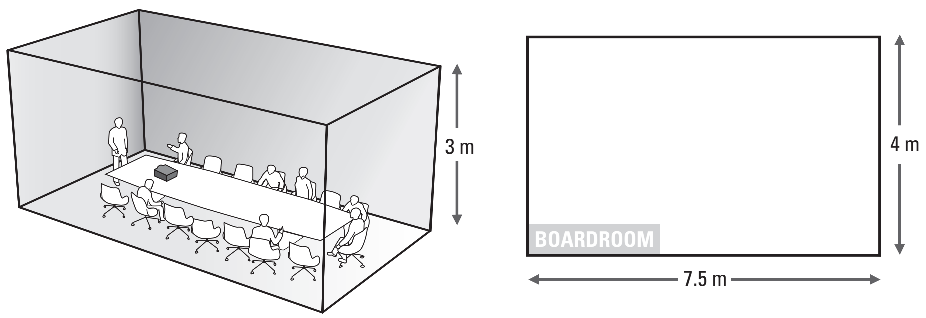
The image blow depicts the draft temperature for the space. The range in which people will feel the most comfortable is indicated in green. The DF1R diffusers produce a thermally comfortable space.







Example 2 — Boardroom Design A (SI)



Space Design

The owner of a new office building wants to use a displacement ventilation system for all occupied spaces. This example examines a private boardroom that is located in the center of the building without any exterior surfaces. The space is designed for 8 occupants, a computer with LCD monitor, a projector, T8 florescent lighting, and has a control temperature of 22 °C. The room is 7.5 m wide, 4 m long, and 3 m from floor to ceiling. There is a large whiteboard at the west side of the room and cabinets along the south and east sides of the room. The owner and architect want the displacement diffusers in the space to fit seamlessly into the room.

Design Considerations	
Occupants	8
Set-Point	22 °C
Floor Area	30 m²
Volume	90 m³
q_{oz}	700 W
q_l	750 W
q_{ex}	0 W
q_T	1450 W

Space Considerations

Some of the assumptions made for the space are as follows:

- The head to foot gradient recommended by ASHRAE (Chen & Glicksman, 2003) is 2 °C from head to foot for seated occupants.
- Load per person is 75 W
- Lighting load in the space is 25 W/m²
- Computer and LCD load is 60 W
- Projector load is 40 W
- The specific heat and density of the air for this example will be 1.007 kJ/(kgK) and 1.2 kg/m² respectively. The loads are broken down as follows:

$$q_{oz} = (8 \text{ People} \times 275 \text{ W}) + 60 \text{ W} + 640 \text{ W} = 700 \text{ W}$$

$$q_l = 30 \text{ m}^2 \times 25 \text{ W/m}^2 = 750 \text{ W}$$

$$q_{ex} = 0 \text{ W}$$

$$q_T = 1450 \text{ W}$$

Total cooling load for this space (q_T) is 1450 W, and approximately 48 W/m².

ASHRAE Standard 62-2004 requires 0.3 L/s m² outdoor air flow rate per unit area, R_a, and 2.5 L/s per person outdoor air flow rate per person, R_p, be delivered to the space for moderately active office work applications. For displacement ventilation, ventilation effectiveness or zone air distribution effectiveness (E_z) is assumed to be 1.2 (Table 6-2, ASHRAE Standard 62-2004).

Example 2 — Boardroom Design A (SI)

Determine the air flow rate to meet the cooling load.

$$Q_{DV} = \frac{0.295\,q_{oe} + 0.132\,q_l + 0.185\,q_{ex}}{\rho c_p \Delta t_{hf}}$$

$$Q_{DV} = \frac{0.295(700) + 0.132(750) + 0.185(0)}{(1.2)(1.007)(3)}$$

$$Q_{DV} = 84.3 \text{ L/s}$$

Determine the fresh air flow rate.

$$Q_{oz} = \frac{R_p P_z + R_A A_z}{E_z}$$

$$Q_{oz} = \frac{(2.5)8 + (0.3)30}{1.2}$$

$$Q_{oz} = 24 \text{ L/s}$$

Note: Some local codes may not allow the discount for Q_{oz}, or may have stricter requirements, and they should be used instead of this calculation.

The total supply air volume for cooling is then the maximum value between Q_{DV} and Q_{oz}.

$$Q_s = \max\left[Q_{DV}, Q_{oe}\right] = 84.3 \text{ L/s}$$

Calculate the supply air temperature.

$$t_s = t_{sp} - \Delta t_{hf} - \frac{Aq_t}{0.584 Q_s^2 + 1.208 A Q_s}$$

$$t_s = 22 - 3 - \frac{30(1450)}{0.584(84.3)^2 + 1.208(30)(84.3)} = 13.0\,°\text{C}$$

Determine the return air temperature.

$$t_e = t_s + \frac{q_t}{1.208(Q_s)}$$

$$t_e = 13 + \frac{1450}{1.208(84.3)} = 27.2\,°\text{C}$$

Adjust for new supply temperature.
The supply temperature should be 5.5 °C less than tsp or 17 °C, whichever is higher.

$$t_s = 17\,°\text{C}$$

$$Q_{DV} = \frac{q_T}{\rho c_p (t_e - t_{SS})} = \frac{1450}{(1.21)(1.006)(27.2 - 17)}$$

$$Q_{DV} = 116.8 \text{ L/s}$$

Example 2 — Boardroom Design A (SI)

Selection of Diffusers

For this application we have three goals set by the owner:

1. Quiet operation
2. Thermal comfort to the space
3. Hidden diffusers

Inherently, displacement ventilation diffusers are quiet, but care has to be taken to limit the sound generated from the HVAC air supply. Price recommends limiting the duct velocity to 5 m/s to minimize noise from ductwork. For thermal comfort, a face velocity of 0.2 m/s fpm is required. At 117 L/s, a diffuser face area of 0.584 m2 would be required.

There are two options to make these diffusers as unobtrusive as possible: mount them in the wall or as part of the furniture.

Layout of the Boardroom

For a concealed look, the DF1R displacement diffuser could be installed at the base of the cabinets or in the wall under the whiteboard in a pressurized plenum. Two diffusers at 1.5 m x 0.2 m will be able to meet the 0.2 m/s requirement. The diffusers can be placed on any of the walls in the room, but it is essential to ensure that sedentary occupants will be located a comfortable distance from the diffuser.

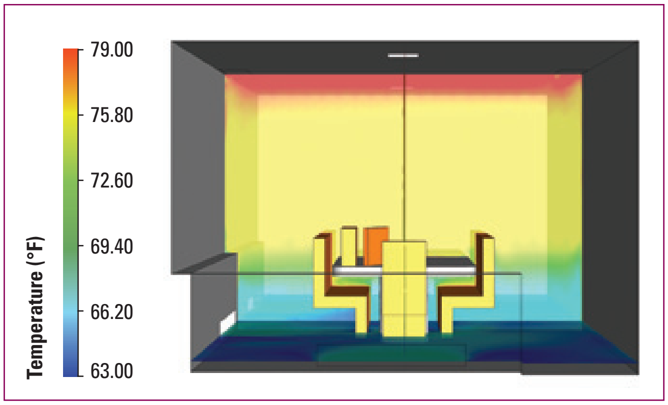
Flow Visualization

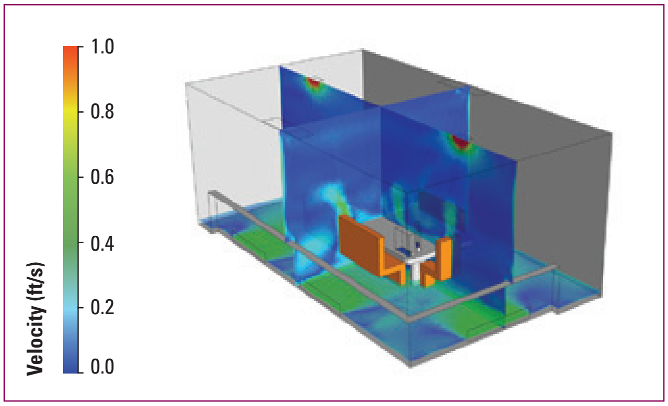
A CFD analysis was run for this example using the conditions, calculated air flow and supply air temperature for the boardroom to give a visual representation of the temperature distribution, air movement, and draft temperatures in the space.

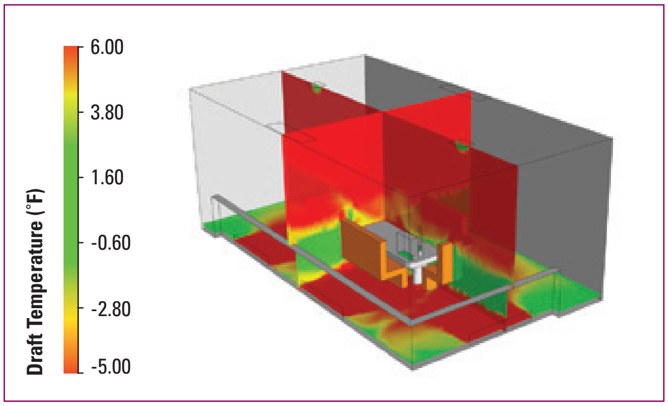
The image on the following page shows the temperature profiles across the space and a reasonable temperature stratification is predicted. Also visible are the heat plumes off the occupants and computer. The seated occupant experiences ambient air temperatures from 20.6 °C to 22.2 °C, and the standing occupant 20.6 °C to 23.9 °C. Both are within the thermal stratification comfort conditions set by ASHRAE (Chen & Glicksman, 2003).

The image below depicts the velocity profile. The images show the plumes off the occupants and computer, as well as the general shape of the air pattern leaving the diffuser and slowly entering the zone.

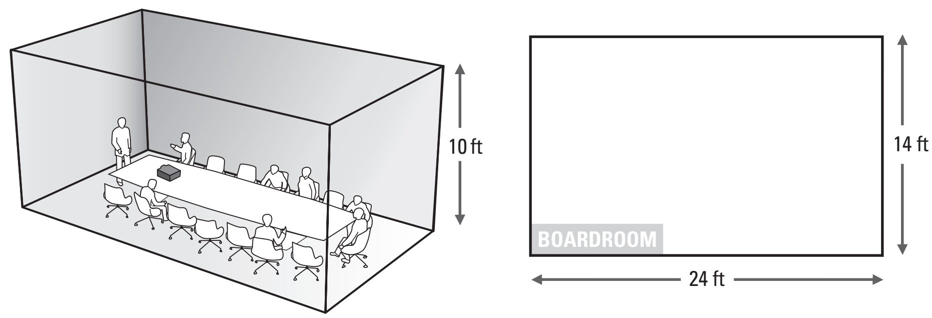
The image blow depicts the draft temperature for the space. The range in which people will feel the most comfortable is indicated in green. The DF1R diffusers produce a thermally comfortable space.







Example 3 — Boardroom Design B



Space Design

This boardroom design example is an extension to the Boardroom Design in Example 2 but adds an external load requirement for summer and winter. The exterior wall and window are located along a 14 ft long side of the room.

Design Considerations - Cooling	
Occupants	8
Set-Point	72 °F
Floor Area	336 ft²
Volume	3360 ft³
q_{oz}	2496 Btu/h
q_l	2292 Btu/h
q_{ex}	1200 Btu/h
q_T	5988 Btu/h

Design Considerations - Heating	
Occupants	8
Set-Point	72 °F
Floor Area	336 ft²
Volume	3360 ft³
q_{oz}	2496 Btu/h
q_l	2292 Btu/h
q_{ex}	-7665 Btu/h
q_T	-2877 Btu/h

Space Considerations

Some of the assumptions made for the space are as follows:

- Internal loading conditions identical to Example 2
- The head to foot gradient recommended by ASHRAE Research Project-949 (Chen et al., 1999) is 5.4 °F from head to foot for sedentary occupants

The assumptions made for the external load case are as follows:

- During summer, the external cooling load is 1200 Btu/h
- During winter, the heating load is 7665 Btu/h
- The building owners would like all ventilation, heating, and cooling to be ceiling installed

Cooling Calculations:

The loads are broken down as follows:

$$q_{oz} = (8 \text{ People} \times 250 \text{ Btu/h}) + 308 \text{ Btu/h} + 188 \text{ Btu/h} = 2496 \text{ Btu/h}$$

$$q_l = 336 \text{ ft}^2 \times 6.82 \text{ Btu/h/ft}^2 = 2292 \text{ Btu/h}$$

$$q_{ex} = 1200 \text{ Btu/h}$$

$$q_T = 5988 \text{ Btu/h}$$

Total cooling load for this space (q_T) is 5988 Btu/h, and approximately 17.8 Btu/h/ft².

For more information on additional imperial and metric sizes please visit priceindustries.com or contact your local Price representative.

Example 3 — Boardroom Design B

ASHRAE Standard 62-2004 requires 0.06 cfm/ft² outdoor air flow rate per unit area, R_a, and 5 cfm/person outdoor air flow rate per person, R_p, be delivered to the space for moderately active office work applications. For displacement ventilation, ventilation effectiveness or zone air distribution effectiveness (E_z) is assumed to be 1.2 (Table 6-2, ASHRAE Standard 62-2004). As indicated in ASHRAE Research Project-949 (Chen et al., 1999), the design vertical temperature difference is 3.6 °F for mainly sedentary occupants.

Determine the air flow rate to meet the cooling load.

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

$$Q_{DV} = \frac{0.295(2496) + 0.132(2292) + 0.185(1200)}{60(0.24)(0.075)(5.4)}$$

$$Q_{DV} = 216 \text{ cfm}$$

From example 2, the fresh air flow rate was determined to be:

$$Q_{oz} = 50 \text{ cfm}$$

The total supply air volume for cooling is then the maximum value between Q_{DV} and Q_{oz}:

$$Q_s = \max\left[Q_{DV}, Q_{oe}\right] = 216 \text{ cfm}$$

Using the procedure from Example 2:

$$t_s = 56.5\,°\text{F}$$

$$t_e = 82.5\,°\text{F}$$

Adjust for new supply temperature

The supply temperature should be 10 °F less than t_{sp} or 63 °F, whichever is higher:

$$t_s = 63\,°\text{F}$$

$$Q_{DV} = \frac{q_t}{60 \rho c_p (t_e - t_s)} = \frac{5988}{60(0.075)(0.24)(82-63)}$$

$$Q_{DV} = 292 \text{ cfm}$$

Heating Calculations

Determine the flow rate for the heating load.

The total heating load, q_T , heating, is 2877 Btu/h, and the heating supply air for the space is set at 90 °F:

$$\Delta t = t_{s,\, heating} - t_{sp}$$

$$\Delta t = 90\text{ °F} - 72\text{ °F}$$

$$\Delta t = 18\text{ °F}$$

Example 3 — Boardroom Design B

The air flow rate required to deliver the required heating load is determined using J9 and the table below:

IP (Btu/h)	
Air	$q = 1.08Q\Delta t,\ Q$ in cfm
Water	$q = 500Q\Delta t,\ Q$ in gpm

$$Q_h = \frac{q_{T,\,heating}}{1.080(\Delta t)}$$

$$Q_h = \frac{2877}{1.080(18)}$$

$$Q_h = 148 \text{ cfm}$$

Determine the fresh air flow rate.

Since displacement ventilation is not recommended for heating, the fresh air flow rate must be recalculated with a ventilation effectiveness appropriate for the ventilation method per ASHRAE Standard 62-2004, Table 6-2. For this example, the heating will be supplied and returned from the ceiling, and the value of E_z is 0.8.

$$Q_{oz} = \frac{R_p P_z + R_A A_z}{E_z}$$

$$Q_{oz} = \frac{5(8) + 0.06(336)}{0.8}$$

$$Q_{oz} = 76 \text{ cfm}$$

The total supply air volume for heating is then the maximum value between Q_{oz} and Q_h:

$$Q_s = \max\left[Q_{DV}, Q_{oz}\right] = 148 \text{ cfm}$$

Selection of Diffusers

For this application we have three goals set by the owner:

1. Heating and cooling modes available in one diffuser
2. Thermal comfort of the space
3. Diffusers must occupy minimal floor space and should be preferably mounted in the T-bar ceiling

The Price DF1L-HC is selected as the diffuser of choice. This diffuser fits into a standard T-bar ceiling and combines heating and displacement ventilation cooling into one unit. The DF1L-HC is best placed such that the slot is nearest the exterior wall so that the air pattern in heating will wash the window. When in cooling mode, the cool air will simply cascade out of the diffuser to the floor. Care should be taken to ensure that the occupants are not seated directly under the diffuser.

Price recommends limiting the duct velocity to 1200 fpm to minimize noise from the ductwork. And, to ensure thermal comfort of the occupants, the face velocity should be 40 fpm or less. At 292 cfm a diffuser face area of 7.3 ft^2 would be required for a 40 fpm face velocity.

For cooling, provide two 24 in. x 48 in. DF1L-HC diffusers providing 146 cfm of air is required. The performance data below is presented in DR%, which is the percent people dissatisfied due to draft per ASHRAE Standard 55-2010. For the 24 in. x 48 in. DF1L-HC selected, this means that at a distance of 2 ft from the diffuser, the DR% will be 20% for a ΔT = 10 °F. This performance data is useful in determining the required distance between sedentary occupants and the diffuser to maintain comfort.

Example 3 — Boardroom Design B

Performance Data - Price DF1L-HC Cooling									
Unit Size,in. [Face Area, ft²] L x W	Face Velocity, fpm	Air flow, cfm	Total Pressure, in. w.g.	Total Static, in. w.g.	Noise Criteria (NC)	Proximity to Outlet			
						ΔT = 5 °F Radius, ft		ΔT = 10 °F Radius, ft	
						DR%		DR%	
						15	20	15	20
24 x 24 [2.6]	20	51	-	-	-	-	-	2	-
	30	77	0.02	0.02	-	1	-	3	1
	40	102	0.04	0.04	-	1	-	3	1
	50	120	0.06	0.06	-	2	-	4	2
48 x 24 [5.5]	20	110	-	-	-	1	-	3	1
	30	165	0.02	0.02	-	2	-	4	2
	40	221	0.04	0.03	-	2	-	5	2
	50	276	0.06	0.05	17	3	1	5	3

For heating mode, the diffusers must supply 148 cfm total as calculated. Each DF1L-HC will supply 74 cfm during heating mode and will be located such that the heating slot is parallel and adjacent to the window producing a horizontal heating air pattern towards the window. The diffuser is located 1 ft from the window. To ensure proper heating distribution and comfort, the supply air in heating mode should achieve a terminal velocity of 50 fpm approximately 2/3 down the window. The boardroom in this example has a 7 ft window flush to the ceiling, so the heating air must reach approximately 4 1/2 ft down the window at a terminal velocity of 50 fpm.

Using the performance data table below for the 24 in. x 48 in. DF1L-HC, the flow rate of 74 cfm in heating falls between the cataloged points of 108 cfm and 162 cfm with 12 and 18 ft throws to 50 fpm respectively. Through linear interpolation we find the throw to 50 fpm at 74 cfm is approximately 7 ft. Adjusting the isothermal throw for an 18 °F heating differential the throw becomes 7 x 0.8 = 5.5 ft with the diffuser mounted 1 ft from the window the air will project across the ceiling and down the window to the floor, meeting our requirements.

$$\frac{108-54}{12-4} = \frac{108-74}{12-x}$$

$$x = 7\,\text{ft}$$

Performance Data - Price DF1L-HC Heating Horizontal Pattern							
	Neck Velocity, fpm	**50**	**100**	**150**	**200**	**250**	**300**
	Velocity Pressure	**0.000**	**0.001**	**0.001**	**0.002**	**0.004**	**0.006**
24 x 24 [2.6]	Total Pressure (in. w.g.)	0.01	0.03	0.07	0.13	0.20	0.29
	Flow Rate (cfm)	27	54	81	108	136	163
	NC	-	-	18	26	32	37
	Throw 150, 100, 50	0-0-1	0-1-4	1-3-5	2-4-7	3-4-9	4-5-11
24 x 48 [5.5]	Total Pressure (in. w.g.)	0.01	0.02	0.05	0.09	0.14	0.21
	Flow Rate (cfm)	54	108	162	217	271	325
	NC	-	-	22	30	36	41
	Throw 150, 100, 50	0-1-4	2-4-12	4-9-18	8-12-21	10-15-23	12-18-25

Typical Applications

Displacement Ventilation systems are very versatile, and can generally be used wherever traditional overhead systems can be used. Displacement applications can be divided into four key areas.

SCHOOLS

The primary market for displacement ventilation is schools. Indoor air quality, silent operation, and thermal comfort are all important design considerations for schools. As a result, many school districts are mandating the use of DV in schools.

The Collaborative for High Performance Schools (CHPS), is a national K-12 green school rating system in the United States whose goal is to improve student performance and educational experience by applying high performance building criteria to schools, as well as using the best possible building technology. CHPS recommends Displacement Ventilation as the preferred air distribution method and awards up to 4 points for the use of Displacement.

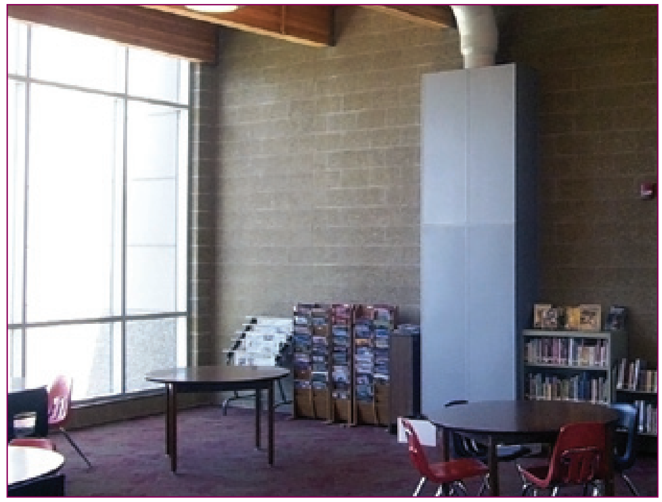
Products

Wall Mounted Diffuser Family – All diffusers from the Wall Mounted Diffuser Family are suitable for applications in schools. These diffusers feature robust construction and optional bases and duct covers which are popular with school applications. DF1C's and DR90's with bases and duct covers can be applied to the corners of classrooms, providing high quality air from a discrete location. DF1 and DF3's can be integrated into library book cases or trophy cases, and DR180's and DR180U's with bases and duct covers can be used as column like architectural features in hallways, cafeterias, change-rooms, and libraries.

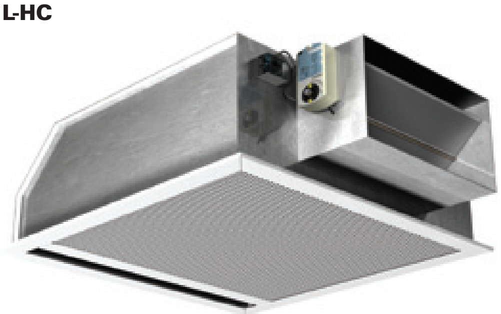
In Wall Diffuser Family - These diffusers provide high quality air and integrate into the wall space, leaving more free space in the room. The DF1R can provide perimeter cooling integrated under cabinetry or an enclosure under a bank of windows; it can also be used for an interior space such as a theatre or lecture hall. The DF1W integrates into plaster walls and is designed to fit between standard wall stud spaces; the DF1W is suitable for many areas of a school, from classrooms to offices to lecture theatres.

Linear Displacement Enclosure Family – Designed for perimeter applications, the DLE Series provides displacement ventilation with a linear bar grille appearance. The DLE can be used to provide cooling along the perimeter of classrooms or hallways. The DLE-H provides displacement ventilation cooling and either hydronic or electric heat in one unit, and is ideal for perimeter application in climates that require cooling and moderate heating.

Ceiling Mounted Diffuser Family – This diverse group provides displacement ventilation through ceiling diffusers, and includes, among the traditional benefits of displacement ventilation, a reduced amount of ductwork and minimal footprint. Both the DF1L and DF1L-HC integrate into standard suspended ceilings; the DF1L provides cooling only and is suitable for interior and perimeter use, while the DF1L-HC provides cooling and light heating from one unit for perimeter use. The DR90H is designed to be installed on plaster surfaces, and is available with duct covers for a continuous look. The DR360DH is typically applied to spaces with higher ceilings that require higher air flows, such as gymnasiums.



DF1L-HC



For more information on additional imperial and metric sizes please visit priceindustries.com or contact your local Price representative.

Typical Applications

LARGE PUBLIC SPACES

Large public spaces often have large, open areas, high ceilings, and varying occupancy levels. Since displacement systems only condition the first six feet of a space, there is potential for large energy savings in spaces with high ceilings.

Additionally, many large public spaces – such as theatres, airports, and places of worship – can have specific architectural designs that the air distribution must accommodate. Price has the custom design capability to provide diffusers that integrate seamlessly into such architectural features.

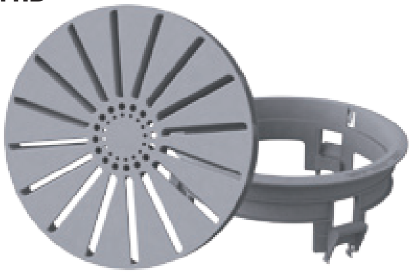
Products

In Wall Diffusers In Wall Diffuser Family - These diffusers provide high quality air and integrate into the wall space, leaving more free area in the room. The DF1R can provide perimeter cooling integrated under cabinetry or an enclosure under a bank of windows; it can also be used for an interior space such as a theatre, lecture hall, or lobby. The DF1W integrates into plaster walls and is designed to fit between standard wall stud spaces; the DF1W is suitable for many large public spaces, from lounges to airports to theatres. Both of these diffusers are highly customizable and are easily integrated into architectural features.

Free Standing Diffusers – Providing a unique architectural feature, Free Standing Diffusers, namely the DR360, is available with an optional base, can be highly customised, and can also be integrated into columns and with electrical supply. The DR360 can be applied to large public spaces such as lounges, casinos, airports, and lobbies.

Floor Mounted Diffusers - Installed in raised floors or concrete slabs, Floor Mounted Diffusers provide a subtle accent to a space. The RFDD is suitable for high churn, low wear spaces such as large offices, while the ARFDD lends itself to higher with higher foot traffic and loading, such as casinos. The DFG and DFGL offer floor supplied displacement ventilation with a linear bar grille appearance, and can be applied in areas such as lobbies, corridors, and waiting areas.

RFDD/ARFHD



Typical Applications

HEALTH CARE

DV has become an accepted technology in hospital patient rooms and can provide a cleaner, safer, environment for patients, health care providers, and visitors while providing energy savings through a lower air-change requirement. ASHRAE has passed Addendum G to Standard 170-2008 "Ventilation of Health Care Facilities," which officially recognizes the use of displacement ventilation in health care facilities and provides guidelines for its use.

Price has been instrumental in research for applying DV in health care in North America. Visit www.priceindustries.com/sustainable/research to access our research papers.

In-Wall Diffuser Family – The Puraflo is specifically designed for use in hospital patient rooms, featuring a standard removable face and optional tamper-proof fasteners. The DF1W features an optional stainless steel face and plenum and is ideal for use in MRI Rooms.

Ceiling Mounted Diffuser Family – the DF1L and the DR90H can be used in hallways, corridors, and areas such as nursing stations. The DF1L integrates into standard suspended ceilings, while the DR90H requires plaster walls and ceiling for mounting. Both diffusers are ceiling installed and maximize available floorspace.

INDUSTRIAL FACILITIES

Displacement Ventilation was originally introduced in Europe to manage the pollutants found in industrial facilities, and is gaining acceptance in this application in North America.

Products

Displacement systems can carry harmful contaminants such as welding and adhesive fumes (when they are lighter than air) up and out of the breathing zone, leaving employees with safe, healthy breathing air.

The DFXi – With a reinforced corrugated face, the DFXi is built to withstand tough industrial environments. Installed on the floor, the DFXi is applicable for use in warehouses, factories, welding shops, and school shops classes.

DR360DH – Part of the Ceiling Mounted Diffuser Family, the DR360DH can be applied to industrial settings. The DR360DH can supply large volumes of air and takes up minimal floorspace.

DR360DH

References

AHRI (2008). *AHRI standard 885-2008: Procedure for estimating occupied space sound levels in the application of air terminals and air outlets*. Arlington, VA: Air-Conditioning, Heating, and Refrigeration Institute.

ASHRAE (2002). *Standard 129-1997—Measuring air-change effectiveness.* Atlanta, GA: American Society for Heating, Refrigerating and Air Conditioning Engineers.

ASHRAE (2004a). *Standard 55-2004—Thermal environmental conditions for human occupancy.* Atlanta, GA: American Society for Heating, Refrigerating and Air Conditioning Engineers.

ASHRAE (2004b). *Standard 62.1-2004—Ventilation for acceptable indoor air quality.* Atlanta, GA: American Society for Heating, Refrigerating and Air Conditioning Engineers.

ASHRAE (2008). *Standard 170-2008—Ventilation of health care facilities.* Atlanta, GA: American Society for Heating, Refrigerating and Air Conditioning Engineers.

ASHRAE (2009a). *ASHRAE handbook--Fundamentals.* Atlanta, GA: American Society for Heating, Refrigerating and Air Conditioning Engineers.

ASHRAE (2009b). *Standard research project-1373—Air distribution effectiveness with stratified air distribution systems.* Atlanta, GA: American Society for Heating, Refrigerating and Air Conditioning Engineers.

ASHRAE (2010). *Addendum G to Standard 170-2008—Ventilation of health care facilities.* Atlanta, GA: American Society of Heating Refrigeration and Air-Conditioning Engineers.

Bakó-Biró, Z., Kochhar, N., Clements-Croome, D.J., Awbi, H.B., & Williams, M. (2007). Ventilation rates in schools and learning performance. Proceedings from *Clima 2007 WellBeing Indoors*. Helsinki.

Behne, M. (1999). Indoor air quality in rooms with cooled ceilings, mixing ventilation or rather displacement ventilation. *Energy and Buildings, 30*, 155-166.

Berube-Dufour, M., Derome, D., & Zmeureanu, R. (2008). In situ assessment of the air distribution in a room with a displacement ventilation system, the Auberge Saint-Antoine (field study report). Sustainable Buildings & Communities, Natural Resources Canada.

Breum, N. O., Helbo, F., & Laustsen, O. (1989). Dilution versus displacement ventilation—An intervention study. *Annuals of Occupational Hygiene, 33* (3), 321-329.

Breum N. O., & Orhede E. (1993). Proceedings from the *14th AIVC Conference: Energy Impact of Ventilation and Air Infiltration*. Copenhagen, Denmark.

Breum, N. O., & Oerhede, E. (1994). Dilution versus displacement ventilation—Environmental conditions in a garment sewing plant. *American Industrial Hygiene Association Journal, 55* (2), 140-148.

Breum, N. O. & Skotte, J. (1991). Displacement air flow in a printing plant measured with a rapid response tracer gas system. *Building Services Engineering Research and Technology, 12* (1), 39-43.

Breum, N. O. & Skotte, J. (1992). Displacement ventilation in industry—A design principle for improved air quality. *Building and Environment, 27* (4), 447-453.

Brohus, H. (1997). Personal exposure to contaminant sources in ventilated rooms. PhD thesis, Aalborg University, Aalborg, Denmark.

Brohus, H., & Nielsen P. V. (1996). Personal exposure in displacement ventilated rooms. *IndoorAir, 6* (3), 149 – 217.

Buckley, J., Schneider, M., & Shang, Y. (2004). *The effects of school facility quality On teacher retention in urban school districts*. Retrieved from: http://www.edfacilities.org/pubs/teacherretention.pdf

Chaudhury, H., Mahmood, A. & Valente, M. (2003). The use of single patient rooms vs. multiple occupancy rooms in acute care environments—A review and analysis of the literature. Unpublished Report: The Coalition for Health Environments Research.

References

Chen, Q. & Glicksman, L. (2003). *System performance evaluation and design guidelines for displacement ventilation*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-conditioning Engineers, Inc.

Chen, Q., Glicksman, L.R., Yuan, X., Hu, S., & Yang, X. (1999). Final report for *ASHRAE RP-949: Performance evaluation and development of design guidelines for displacement ventilation.* Cambridge, MA: Department of Architecture, Massachusetts Institute of Technology.

Cho, S., Im, P., & Haberl, J. (2005). Literature review of displacement ventilation. College Station, TX: Energy Systems Laboratory, Texas A&M University System.

Choi, N., Sagara, K., Yamanaka, T., Kotani, H., Suzuki, T., & Yamashita, T. (2008). Displacement ventilation system with radiation panel for sickroom—Influence of radiation panel on contaminant concentration profile. Proceedings from the *29th AIVC Conference: Advanced building ventilation and environmental technology for addressing climate change issues*. Kyoto, Japan.

Community Action to Fight Asthma (2007). Asthma and indoor air quality in schools. Retrieved from: http://www.rampasthma.org/uploads/resources/R1_Schools%20footnotes.1.pdf

CEC (2008). *Efficiency standards for residential and nonresidential buildings.* Sacramento, CA: California Energy Commission.

CHPS (2006). *Design best practices manual, volume II—Design for high performance schools*. San Francisco, CA: Collaborative for High Performance Schools.

CSA (2010). *CSA Z317.0-10—Special requirements for heating, ventilation, and air-conditioning (HVAC) systems in health care facilities*. Mississauga, ON: Canadian Standards Association.

Fitzner, K. (1996). Displacement ventilation and cooled ceilings—Results of laboratory test and practical installations. Proceedings from *Indoor Air '96*. Nagoya, Japan.

Fitzpatrick, M. (2009, January 17). Cut classroom noise so students can hear, experts urge. *Winnipeg Free Press*, A20.

Guity, A., Gulick, B., & Marmion, P. (2009). *Healthcare ventilation research collaborative: Displacement ventilation research* [PDF document]. Retrieved from: http://www.noharm.org/us_canada/reports/2009/dec/rep2009-12-01.php

Haverinen-Shaughnessy, U., Moschandreas, D., Nevalainen, A., & Shaughnessy, R.J. (2007). Indoor environmental quality in schools in relation to academic performance of students: Observations of potential contributors to poor IEQ. Proceedings from *Clima 2007 WellBeing Indoors*. Helsinki.

Healthy Schools Network (2007). *Guide to school design: Healthy and high performance schools* [PDF document]. Retrieved from: http://www.healthyschools.org/documents/School_Design_Guide.pdf

Heiselberg, P. & Sandberg, M. (1990). Convection from a Slender Cylinder in a Ventilated Room. Proceedings from *Roomvent '90*. Oslo, Norway.

Helenius, T., Seppanen, .O, Maranen, A., et al. (1987). Measurements of Air Exchange Efficiency and Ventilation Effectiveness. Proceedings from *Roomvent '87: Air Distribution in Ventilated Spaces*. Stockholm, Sweden.

Holmberg, S. & Li, Y. (1998a). Modeling of indoor environment—Particle dispersion and deposition. *Indoor Air*, 8, 113-122.

Holmberg, S. & Li, Y. (1998b). Non-passive particle dispersion in a displacementventilated room—A numerical study. Proceedings from *Roomvent'98: The 6th International Conference on Air Distribution in Rooms*. Stockholm, Sweden.

Holmberg, S. & Chen, Q. (2003). Air flow and particle control with different ventilation systems in a classroom. *Indoor Air*, 13, 200-204.

References

Husman, T. (2002). Indoor Air 2002: Respiratory infections among children inmoisture damaged schools. Kuopio, Finland: National Public Health Institute.

IARC (1997) [1990]. Volume 49: Chromium, nickel, and welding—Summary of data reported and evaluation. Retrieved from http://monographs.iarc.fr/ENG/Monographs/vol49/volume49.pdf.

ISO (2005). *International Standard ISO 7730-2005—Ergonomics of the thermal environment – Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria.* Geneva, Switzerland: International Standards Organization.

Jung, A. & Zeller, M. (1994). Analysis and Testing of Methods to Determine Indoor Air Quality and Air Change Effectiveness. Aachen, Germany: Rheinisch-Westfalische Technical University of Aachen.

Kobayashi, N. & Chen, Q. (2003). Floor-supply displacement ventilation in a small office. *Indoor and Built Environment, 12* (4), 281-292.

Lain, M. & Hensen, J. (2004). Combination of low energy and mechanical cooling technologies for buildings in Central Europe. Proceedings from the *5th International IIR Conference Compressors 2004.* Casta Papiernicka, Slovakia: International Institute of Refrigeration.

Lee, K.S., Jiang, Z., & Chen, Q. (2009). Air distribution effectiveness with stratified air distribution systems. *ASHRAE Transactions, 115* (2).

Lee, K.S., Zhang, T., Jiang, Z., & Chen, Q. (2009). Comparison of airflow and contaminant distributions in rooms with traditional displacement ventilation and under-floor air distribution systems. *ASHRAE Transactions, 115* (2).

Livchak, A. & Nall, D. (2001). Displacement ventilation – Application for hot and humid climate. Proceedings from *Clima 2000/Napoli 2001 World Congress*. Napoli, Italy.

Loftness, V. (2009). *Green schools: Attributes for health and learning* [PDF Document] Retrieved from: http://www.chps.net/content/315/NASSchoolsCHPSFINAL.pdf

Matsumoto, H. & Ohba, Y. (2004). The influence of a moving object on air distribution in displacement ventilated rooms. *Journal of Asian Architecture and Building Engineering, 3* (1), 71-75.

Mundt, E. (1995). Displacement ventilation systems convection flows and temperature gradients. *Building and Environment, 30* (1), 129-133.

National Research Council (2006). *Green schools—Attributes for health and learning*. Washington, DC: The National Academies Press.

Nielsen, P.V. (1981a). Contaminant distribution in industrial areas with forcedventilation and two-dimensional flow. Danfoss A/s. IRR-Joint Meeting, Commission E1. Essen, Germany.

Nielsen, P.V. (1981b). Ventilation of Working Areas. *SBI-Rapport 128*. Statens Byggefoskningsinstitut.

Nielsen, P.V., Hoff, L., & Pedersen, L.G. (1988). Displacement ventilation by different types of diffusers. Proceedings from the *9th AIVC Conference*. Ghent, Belgium: Warwick.

Niemela, R., Koskela, H., & Engstrom, K. (2001). Stratification of welding fumes and grinding particles in a large factory hall equipped with displacement ventilation. *Annuals of Occupational Hygiene, 45* (6), 467-471.

Niu, J. (1994). Modeling of cooled-ceiling air-conditioning systems. Ph.D. Dissertation, Delft University of Technology, The Netherlands.

Novoselac, A., Burley, B.J., & Srebric, J. (2005). Development of new and validation of existing convection correlations for rooms with displacement ventilation systems. *Energy and Buildings, 38*, 163-173.

Novoselac, A. & Srebric, J. (2002). A critical review on the performance and design of combined cooling ceiling and displacement ventilation systems. *Energy and Buildings, 34* (5), 497-509.

References

Oppl, L., (1969) Luftströmung in Gelüftenten Räumen, Ö1 – und Gasfeuerung, nr. 9, 1969, 908.

Price Industries (2011). *Price engineer's HVAC handbook—A comprehensive guide to HVAC fundamentals.* Winnipeg, MB: Price Industries Limited.

Review of final report: Assessment of displacement ventilation at Prairieview School, Vulcan, Alberta (2009).

Rimmer, J., Tully, B., Dyk, A., & Buck, M. (2010). Displacement ventilation as aviable air solution for hospital patient rooms. Proceedings from *Clima 2010*. Antalya, Turkey.

Rimmer, J., Tully, B., & Guity, A. (2010). A field study of the air change effectiveness of overhead air distribution and displacement ventilation in healthcare. Proceedings from *IAQ 2010*. Kuala Lampur, Malaysia.

Shaughnessy, R.J., Haverinen-Shaughnessy, U., Nevalainen, A., & Moschandreas, D. (2006). A preliminary study on the association between ventilation rates in classrooms and student performance. *Indoor Air, 16*, 465-468.

Shendell, D., Prill, R., Fisk, W., Apte, M., Blake, D., & Faulkner, D. (2004). Associations between classroom CO2 concentrations and student attendance in Washington and Idaho. *Indoor Air, 14*, 333-341.

Shield, B.M. & Dockrell, J.E. (2003). The effects of classroom noise on children's academic attainments. Proceedings from *Euronoise 2003*. Naples, Italy.

Seppanen, O.A., Fisk, W.J., Eto, J., & Grimsrud, D.T. (1989). Comparison of conventional mixing and displacement air-conditioning and ventilating systems in U.S. commercial buildings. *ASHRAE Transactions, 95* (2).

Skaret, E. (1985). Ventilation by displacement – Characterization and design application. Proceedings from *Ventilation '85*. Amsterdam: Elsevier Science Publishers.

Skistad, H. (1994). *Displacement ventilation.* Taunton, Somerset, England: Research Studies Press Ltd.

Skistad, H. (2002). *REHVA guidebook no. 1: Displacement ventilation in non-industrial premises*. Brussels, Belgium: Federation of European Heating and Air-conditioning Associations (REHVA).

Smedje, G & Norback, D. (2000). New ventilation systems at select schools in Sweden—Effects on asthma and exposure. *Archives of Environmental Health, 35*(1), 18-25.

Tan, H., Murata, T., Aoki, K., & Kurabuchi, T. (1998). Cooled ceiling/displacement ventilation hybrid air conditioning system—Design criteria. Proceedings from *Roomvent '98*. Stockholm, Sweden.

USGBC (2009). *LEED for schools rating system—New construction and major Renovations* [PDF document]. Retrieved from: http://www.usgbc.org/ShowFile.aspx?DocumentID=8872

Warila, D. T. (1994). Dehumidification without reheat using face and bypass dampers. Proceedings from the *9th Symposium on Improving Building Systems in Hot and Humid Climates*. Arlington, TX.

Yin, Y., Xu, W., Gupta, J.K., Guity, A., Marmion, P., Manning, A.,…Chen, Q. (2009). Experimental study on displacement and mixing ventilation systems for a patient ward. *HVAC&R Research, 15* (6), 1175-1191.

Yuan, X., Chen, Q., Glicksman, L.R. (1999). Performance evaluation and design guidelines for displacement ventilation. *ASHRAE Transactions, 105* (1), 298-309.

Zhang, H., Huizenga, C., Arens, E., & Yu, T. (2005). *Modeling thermal comfort in stratified environments.* Berkeley, CA: Center for the Built Environment, University of California.

Typical Applications

DESCRIPTION OF OPERATION

The PIC-MB controls package is an optional feature that can be supplied with a parallel flow terminal unit (FDV8 & FDVLP8). The controls package will deliver air at a constant pressure and temperature, both of which are field adjustable. This terminal unit and controls package is best suited for a mixing box application where it draws warm return air and mixes that with cool fresh air from a rooftop or air handling unit, and then supplies the resultant tempered air to one or more zones. This controls package is intended for displacement ventilation systems.

Regardless of whether the mixing box is used to supply one or more zones, each zone must have its own form of VAV control such as Price SDV8 terminal units with PIC controllers. The mixing box should not be used to supply air directly to grilles or diffusers because the total flow rate does not adjust based on an input such as a thermostat. Instead, downstream static pressure is held constant (default 0.3" W.C.), which makes this mixing box ideal for serving VAV zones with throttling dampers connected to displacement diffusers. The zone control is independent of the mixing box.

The PIC-MB monitors and controls both downstream duct static pressure and discharge air temperature (DAT) when installed on a parallel flow fan terminal unit (FDV8 or FDVLP8). The mixing box must be equipped with a modulating ECM motor and some type of modulating reheat (electric or hot water). This system will deliver air at a constant pressure and temperature. To understand how a mixing box works, it can be broken down into two parts: pressure control and DAT control.

The PIC-MB's pressure control loop regulates static pressure in the downstream duct work using a field installed static pressure sensor. The default downstream pressure set point is 0.3" in.w.g. and is field adjustable using the LCD setup tool or Linker2 service tool. The inlet damper of the mixing box is modulated in order to regulate the downstream static pressure. If the downstream static pressure reading is higher than the set point, the damper will close to allow less air through, thereby reducing the pressure. If the downstream static pressure reading is lower than the set point, the damper will open to allow more air through, thereby increasing the pressure. This constant monitoring and regulating of pressure happens on a slower time base than the DAT control, meaning it's slower to react than the fan and heater (DAT loop).

The PIC-MB's DAT loop regulates the air temperature at the discharge of the mixing box. Typical displacement ventilation systems require the discharge air temperature to be between 62 and 68 °F. The default discharge air temperature set point of the PIC-MB is 63 °F and is field adjustable using the LCD setup tool or Linker2 service tool.

The PIC-MB utilizes two stages of analog heat in order to maintain the DAT setpoint.

Upon detection of a discharge air temperature lower than the set point, the PIC-MB ramps up the ECM motor on the mixing box in order to draw more return air into the box. The process of adding return air that is assumed to be warmer than primary air effectively increases the discharge air temperature while consuming only a small amount of energy. If the fan reaches its maximum capacity (field adjustable) and DAT is still lower than the set point, the PIC-MB will utilize the mixing box's analog reheat and increase its capacity until the DAT set point set point is reached. This constant monitoring and regulating of the DAT happens on a faster time base than the pressure control, meaning it's quicker to react than the inlet damper (pressure loop).

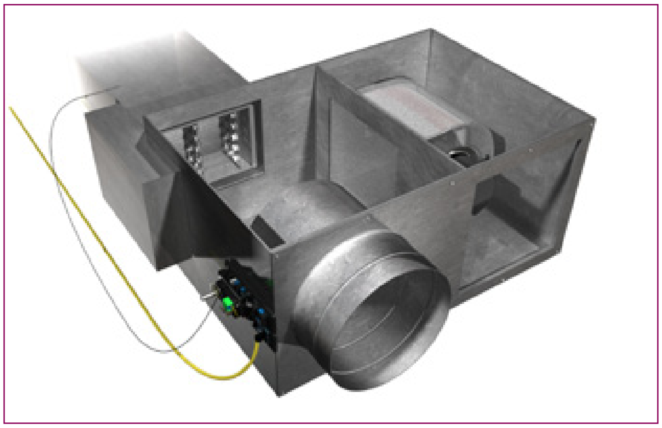


Figure 1: FDV8 Terminal Unit with PIC-MB Package

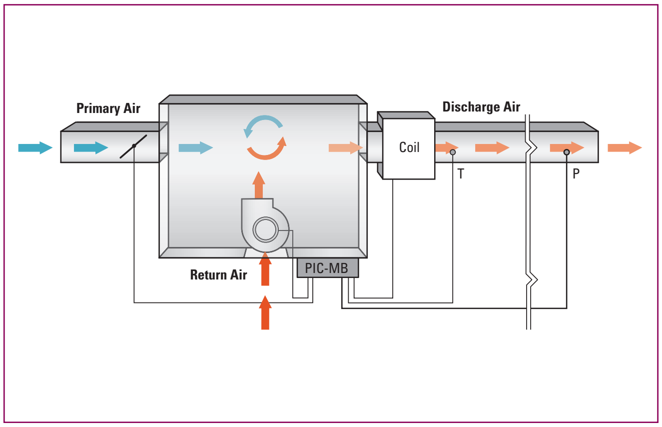


Figure 2: Mixing Box Application with Reheat Coil

Figure 3: Price Linker2 (Setup Tool) & Stand-alone LCD-SETUP

Typical Applications

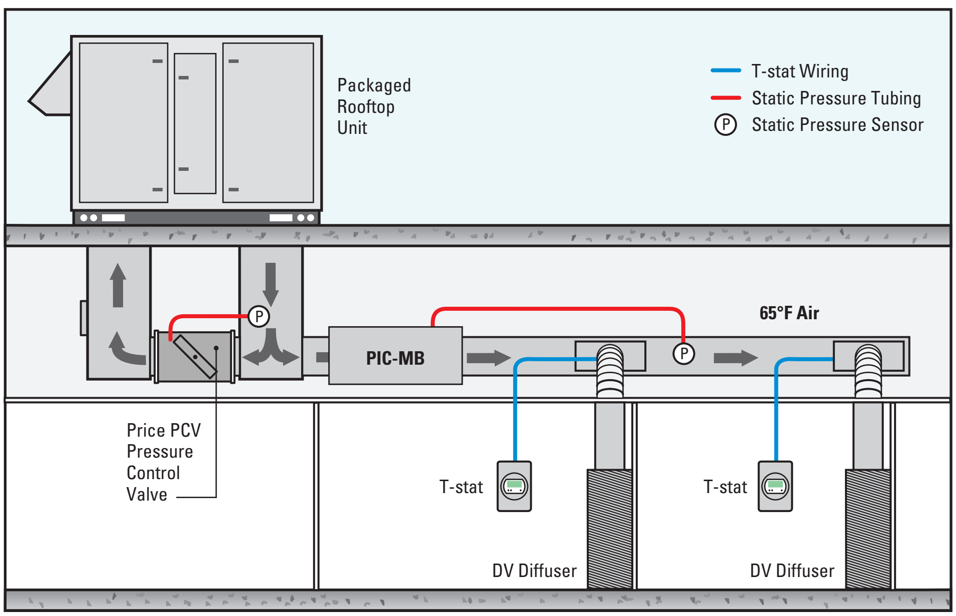
Applications

The PIC-MB is designed to be used with cool supply air and warm return air. It mixes these two air streams together to provide favourable conditions for displacement ventilation systems. The PIC-MB regulates discharge air temperature and pressure. It is typically installed in the following types of installations:

- In a building with a packaged rooftop unit and displacement ventilation diffusers. The packaged rooftop is incapable of directly delivering 65 °F air at a constant temperature and variable volume.
- In a larger building with an air handling unit that serves other zones. The other zones typically have lower air temperature requirements, for example typical VAV zones with Prodigy diffusers.

Packaged Rooftop Units with Displacement Ventilation

When using a packaged rooftop unit with a displacement ventilation (DV) system, there is a need to raise discharge air temperature to approximately 62-68°F. This can be done by mixing colder air from the rooftop unit with warmer return air from the ceiling return plenum or ductwork. The PIC-MB controls the mixing and provides a constant discharge air temperature to all the DV zones. Each zone may have its own form of VAV control to vary the amount of tempered air that goes to each diffuser, thereby controlling space temperature.



The rooftop unit will need to be equipped with either a variable speed fan or a bypass to ensure the rooftop can deliver a varying amount of conditioned air. Since most packaged rooftops are not sold with a variable speed fan, a bypass must be used in the system in order to maintain an acceptable static pressure in the ductwork. The PIC-MB will throttle the amount of incoming air from the rooftop, meaning some of the rooftop's air must be diverted elsewhere. Therefore it is paramount that some form of duct static pressure control is present in the system, upstream of the PIC-MB. Price recommends the installation of a pressure control valve (PCV) in bypass mode to maintain duct static pressure. The PCV will control static pressure in the ductwork and allow the PIC-MB to operate correctly.

To control the rooftop unit, Price recommends using the Price Rooftop Unit controller (PRTU). The rooftop unit controller can be networked to all the individual zones as well as to the pressure control valve. By polling all the zones for information about heating and cooling demands, the PRTU will activate the necessary stages of heating or cooling. Because the PIC-MB must receive cool air from the rooftop in order to function properly, the PRTU can be set up to favour cooling or even completely lock out heating stages. Furthermore, the PRTU can be calibrated to limit the off-coil temperature of the rooftop unit. Calibrating the limits of the rooftop unit discharge will ensure the PIC-MB receives cool air in the usable range and not frigid air that must be reheated significantly before it can be fed to displacement diffusers. This is advantageous by simultaneously increasing system stability and reducing overall operating costs.

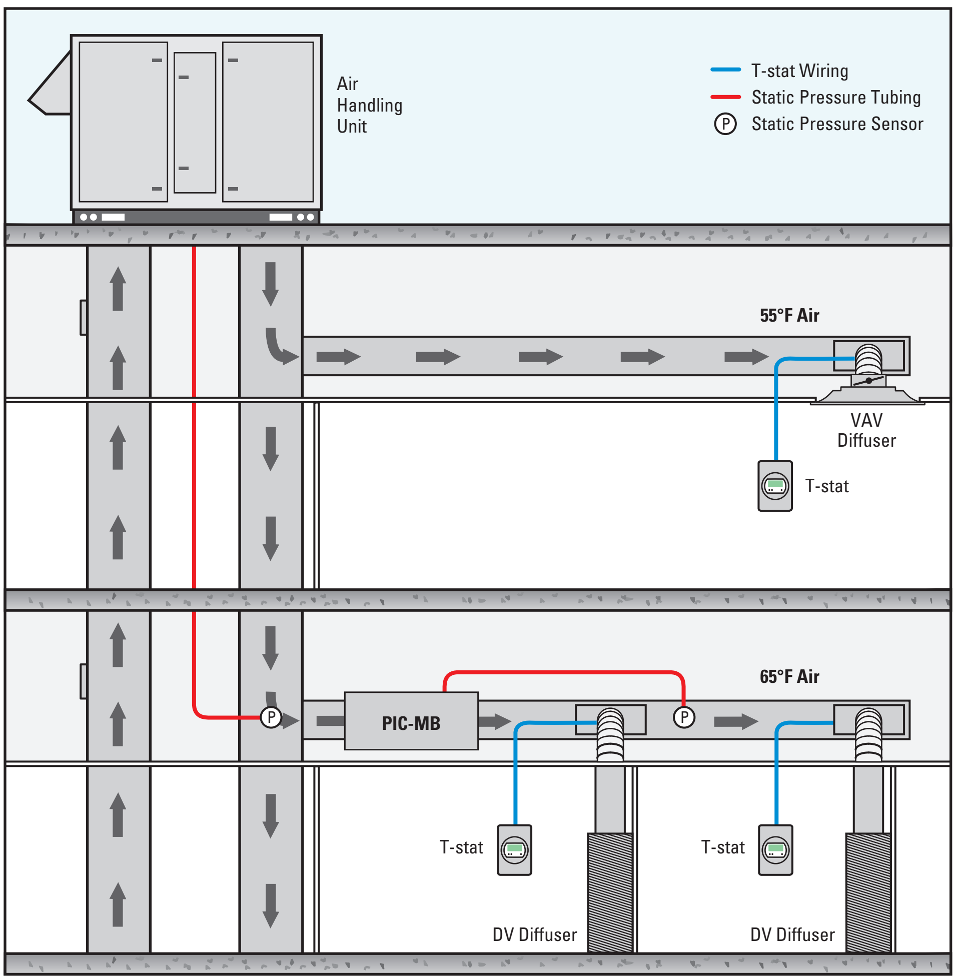
For more information on additional imperial and metric sizes please visit priceindustries.com or contact your local Price representative.

Typical Applications

Air Handling Units with Other Zones and Displacement Ventilation

Air handling units are generally versatile enough and have enough controllability to handle DV requirements without the need for the PIC-MB. An air handler with an energy recovery loop or face and bypass system can cool the incoming air enough to remove the humidity and then reheat the air back up to comfortable temperate air, around 65 °F. This is the ideal solution for larger buildings where DV is used exclusively with an air handling unit. Underfloor air distribution (UFAD) systems also use an elevated supply air temperature, so a building with separate UFAD and DV zones can generally be fed by the same air handling unit.

However, when one building has both DV and traditional overhead VAV systems supplied by the same air handling unit, a conflict of requirements occurs. The AHU is unable to supply both 55 °F air to the DV system and 65 °F air to the VAV system. The solution is to use the AHU for the VAV system and then use a PIC-MB to mix 55 °F air with local return air to deliver 65°F air to the DV zone(s). In this way, the AHU can simply run as it normally would, supplying 55°F air to all of the zones, but the DV diffusers would receive the correct higher temperature air due to the PIC-MB.



Typical Applications

PIC-MB Requirements

Typical set points for a PIC-MB are 63-68 °F for the DAT and 0.20-0.50 in.w.g. for the static pressure. In order for the PIC-MB to function properly, it can only add heat to the incoming air stream by adding return air and/or reheating the air if required. To avoid using reheat, it is recommended that the incoming air be between 55 and 60 °F, so that the PIC-MB can raise the DAT to a satisfactory level without excessive use of the analog reheat. This will ensure that the least amount of additional energy is used by the PIC-MB so as to keep operating costs down. The primary inlet static pressure should be equal to or only slightly higher than the desired target static pressure so that the FDV8 primary damper is able to maintain downstream pressure effectively. When a Price PCV is used, the PCV target static pressure can be adjusted to supply the correct primary pressure to the PIC-MB.

PIC-MB Sequences

The following diagram describes the sequence of operation of the PIC-MB.

Pressure Control:

On an increase in duct static pressure the contoller will close the inlet damper to decrease the amount of air delivered the downstream of the box. On a decrease in duct static pressure the controller will open the inlet damper to increase the amount of air delivered downstream of the box. Duct static pressure is held constant.

Upon detection of air handler shutdown (Zone duct pressure with VAV damper fully open), the controller/actuator will place the damper at the pre-selected setback position (default: 50% open)

Discharge Air Temperature (DAT) Control:

When the DAT falls below the set point, the fan will speed up to increase the amount of return air as a first stage of heat control. If the fan is at maximum speed and DAT is still below set point, a second stage of analog reheat will slow down to draw less return air.

Note:

Primary air must be cooler than the DAT set point because the controller can only add heat to the primary air.

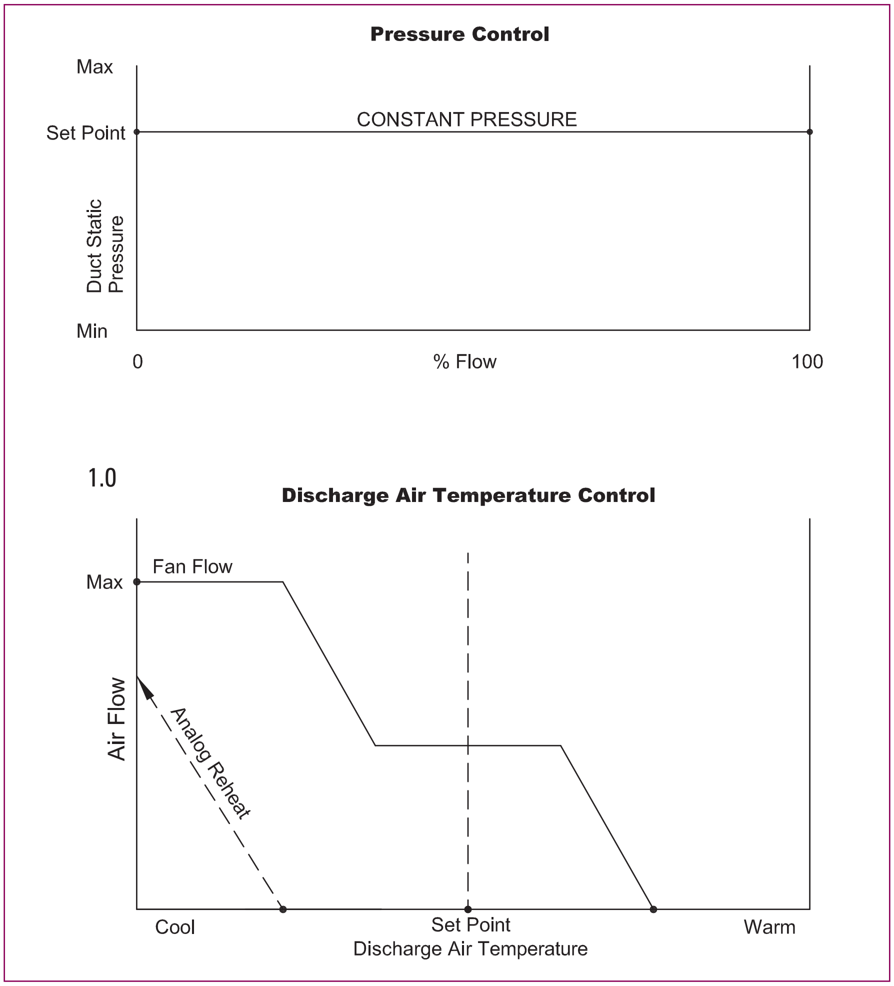


Figure 4: Pressure Control with Discharge Air Temperature Controls