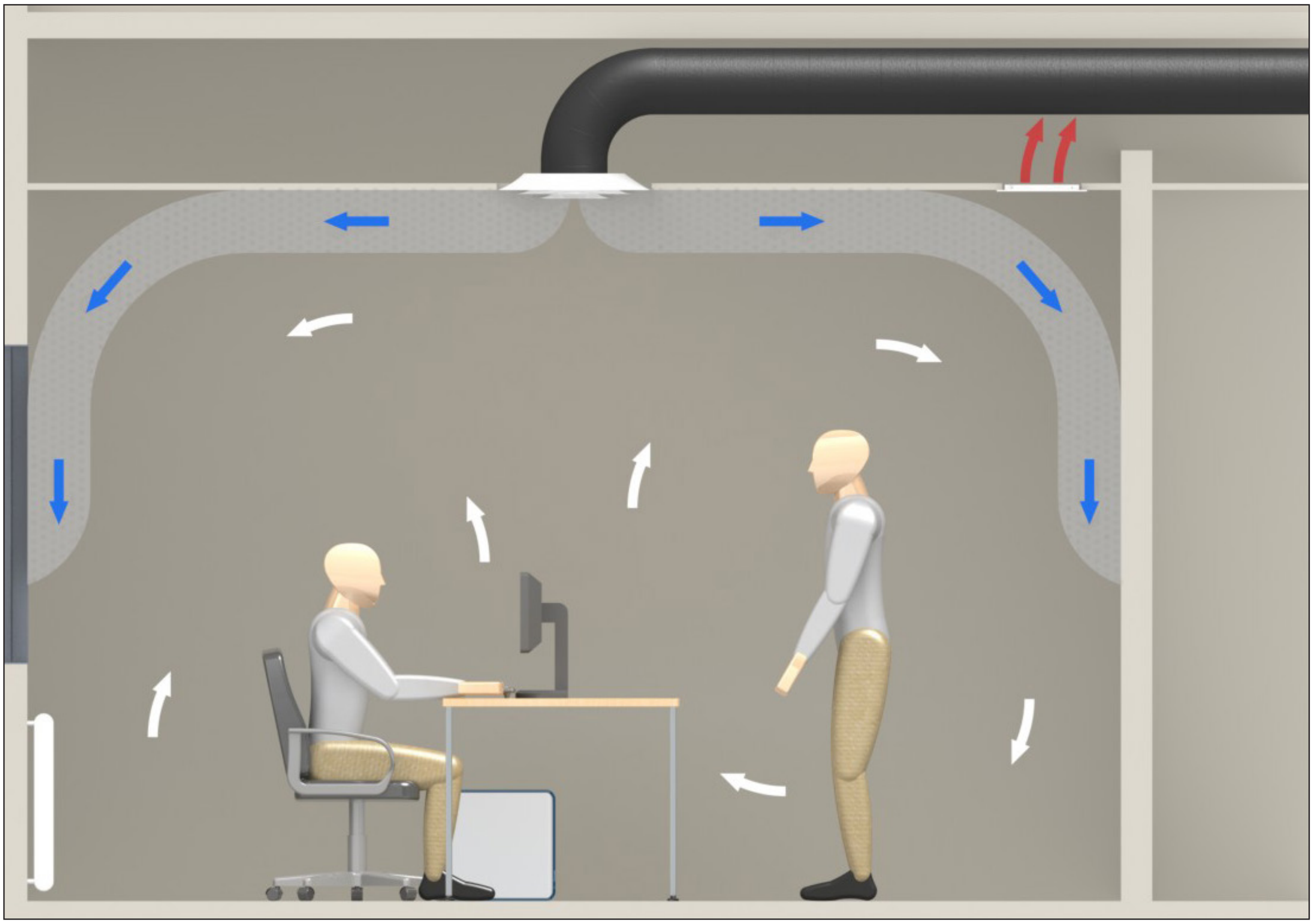

Figure 1. Fully mixed room air distribution

Buildings come in all shapes and sizes and are designed for any number of purposes. In order to create healthy and productive environments, air distribution systems must be selected that best meet the goals of designers. There are a wide range of choices available, but often one system can be identified as the best solution in terms of cost, comfort and energy. The purpose of this guide is to explain how displacement ventilation works, describe recommended applications and provide engineering guidance to the system designer.

INTRODUCTION TO DISPLACEMENT VENTILATION

In order to understand the advantages and limitations of displacement ventilation, it's important to understand the differences between conventional mixed air distribution and fully-stratified air distribution.

In fully mixed air distribution systems (see figure 1), cool or warm supply air is delivered at relatively high velocity from ceiling-mounted diffusers. When ceiling diffusers are properly selected and positioned, this high velocity air doesn't result in occupant discomfort because it is delivered outside the occupied zone. The purpose of the high velocity supply is to create low velocity room air motion through entrainment. Ideally, this air motion will thoroughly mix the supply air with the room air resulting in uniform temperatures and contaminant levels throughout the occupied zone. Internal heat loads and contaminants are eventually picked up and carried away by the return air.

In fully-stratified air distribution systems (see figure 2), cool supply air is typically delivered at reduced velocity from low sidewall diffusers. The supply air is always cooler than the room air, so it quickly drops to the floor and moves slowly across the room. When this slow moving air mass encounters a heat load, it rises and carries the heat and pollutants towards the ceiling. A layer of warm air forms above the occupied zone due to natural buoyancy. Internal heat loads and contaminants are carried away by the return air.

The main differences between these systems are:

Mixed air distribution

- Suitable for both heating and cooling with a supply temperature range generally between 55 to 90°F
- Conditioned air is discharged into the unoccupied zone at relatively high velocities
- Minimal temperature variations throughout the space
- Uniform contaminant concentrations throughout the space

Fully-stratified air distribution

- Suitable for cooling only with a supply temperature generally ranging from 62 to 70°F

- Conditioned air is discharged directly to the occupied zone at low velocity
- Sensible heat gains that emanate away from the conditioned air layer escape due to natural buoyancy
- Heat and respiratory pollutants rise into the upper unoccupied zone
- Occupied zone CO_2 concentrations are significantly reduced

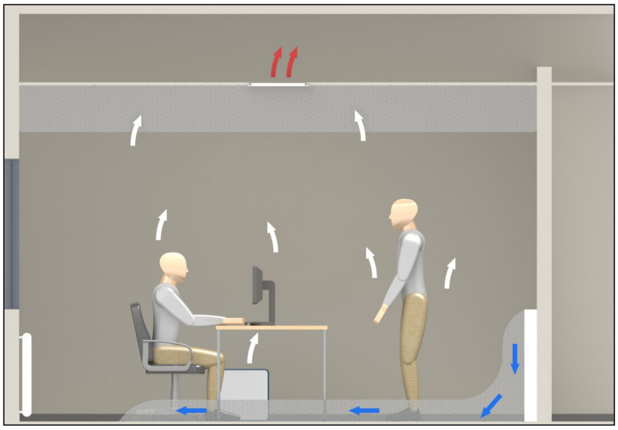

Figure 2. Stratified room air distribution

Figure 3 illustrates vertical temperature gradients that are representative to the two air distribution systems. Note that there is very little difference between the floor, control level and ceiling temperatures in the mixed air system while the displacement system floor temperature is lower than that at the control level and the ceiling (return) air temperature is typically several degrees higher. This is attributable to the buoyancy driven transport of sensible heat gains directly to the upper level of the space that occurs in thermal displacement systems. As those do not affect the thermal comfort of the occupants, they need not be factored into the space sensible cooling airflow calculation.

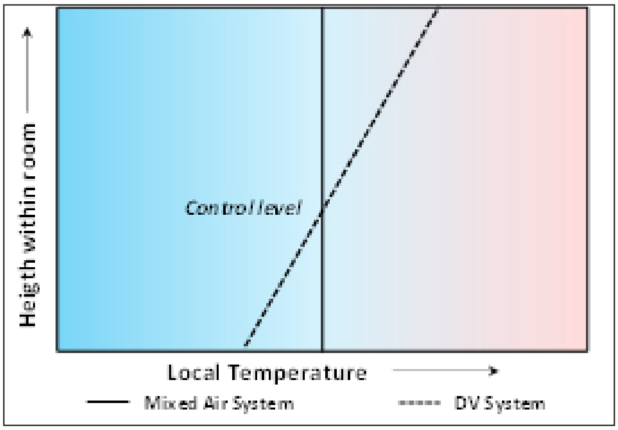

Figure 3. Vertical temperature gradients

AIR CHANGE EFFECTIVENESS

Figure 4 compares vertical CO_2 level concentrations that are representative of the two systems. Note that, while the mixed system concentration levels vary little over the room, those of the displacement system are 20 to 40% lower in the occupied portion of the space and higher in the upper levels of the space where the return inlet is located.

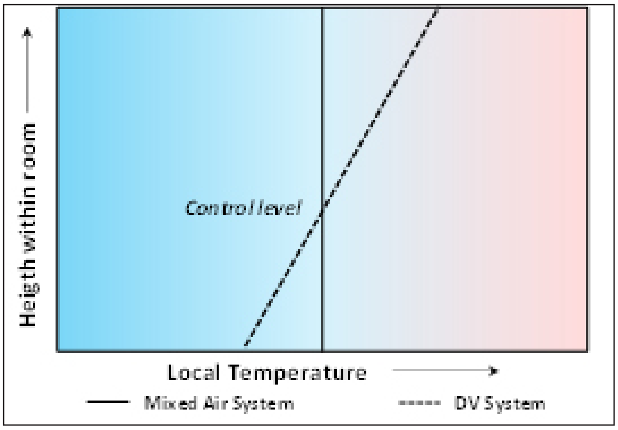

Figure 4. Vertical CO_2 gradient comparison

ASHRAE Standard 62.1-2016 'Ventilation for Acceptable Indoor Air Quality' Table 6-2 assigns a zone air distribution effectiveness value (E_z) of 1.0 for conventional mixed air systems and 1.2 for fully-stratified systems. This means that fully-stratified systems are 20% more effective than the best mixed air systems and can provide the same level of ventilation with an inversely proportional outdoor air volume flow rate.

TYPICAL APPLICATIONS

Ideal applications for displacement ventilation often involve large open spaces with tall ceilings. These include but are not limited to:

- Theaters and performance halls
- Meeting rooms and lecture halls
- Restaurants and cafeterias
- Hotel lobbies and atriums
- Shopping malls
- Gymnasiums
- Casinos
- Museums and exhibit halls
- Classrooms
- Airport terminals and train stations

Displacement ventilation can be a very effective strategy for removing contaminants from room air, because fully-stratified systems take advantage of the fact that airborne pollutants are generally lighter than air. The thermal buoyancy of tobacco smoke and human respiration allow these pollutants to escape directly to the upper regions of the space, carrying their contaminants and odors with them.

A couple of important considerations should be noted regarding the use of displacement ventilation systems:

- Displacement ventilation is not recommended for spaces where hazardous chemical spills could occur. In the event of a spill, a displacement ventilation system is likely to cause noxious fumes to be drawn from the floor and brought up to the breathing level, thereby increasing the possible hazard to occupants
- In rare situations where contaminants are heavier than air, accommodations should be made to allow some portion of the

room air to be extracted at a lower level

BENEFITS AND LIMITATIONS

Typical benefits of displacement ventilation include:

- Improved removal of airborne contaminants
- Reduced energy requirements to cool occupied spaces in mild climates
- Reduced ventilation air requirement due to increased air distribution effectiveness
- Very low diffuser noise levels

Although displacement ventilation is well-suited for a wide variety of applications, the following spaces may be better served by mixed air systems:

- Spaces with ceiling heights lower than 9 feet
- Spaces furnished with cubicles or other partitions
- Applications involving contaminants that are heavier and/or colder than room air in the occupied zone

OUTLET CHARACTERISTICS

Displacement ventilation requires outlets that supply air at extremely low velocities, (typically 50-70 fpm). These outlets are commonly located low on a sidewall or at the base of a column. The low average face velocity results in rather large diffuser panels. Since the outlets are located adjacent to the occupied zone and within easy reach of room occupants, they have the following special requirements:

- Should be elevated above the floor to prevent damage from cleaning equipment
- Construction and finishes must be rugged enough to prevent damage through occupant contact
- Should provide a concealed and tamperproof means for air pattern adjustment
- Face panels must be removable for cleaning and adjustment of air pattern controllers.

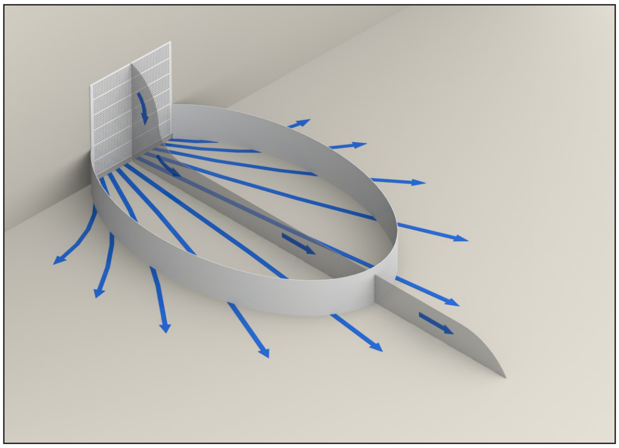

Figure 5. Diffuser adjacent zone

THE ADJACENT ZONE

The area immediately adjacent to a displacement ventilation outlet is known as the adjacent zone as illustrated in figure 5. This is any area in the occupied zone where local air velocities regularly exceed 50 fpm at a height 1" above the floor. Although this clear zone can often be in an aisle or corridor without creating potential comfort problems, location of stationary occupants within the outlet's adjacent zone is not advised. Cool air that drops from a sidewall diffuser and travels across the floor can easily be sensed by the stationary occupants at their ankle level.

It is important to note that all Titus displacement ventilation diffusers are supplied with adjustable air pattern controllers. The ability to adjust the shape of the air pattern and the resultant adjacent zone can be of great benefit when dealing with furniture, occupants and obstructions especially in smaller spaces.

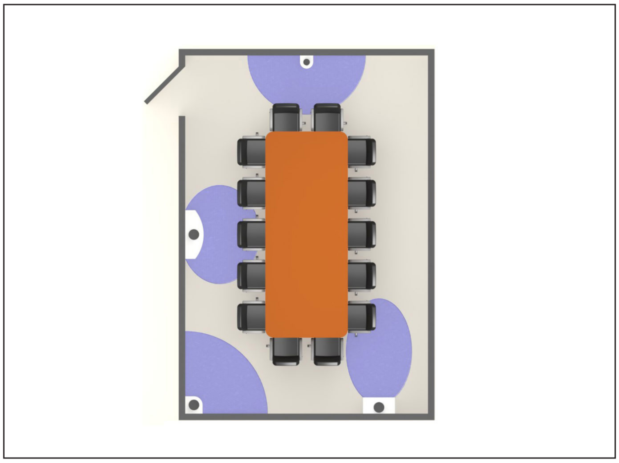

Figure 6. Standard air patterns

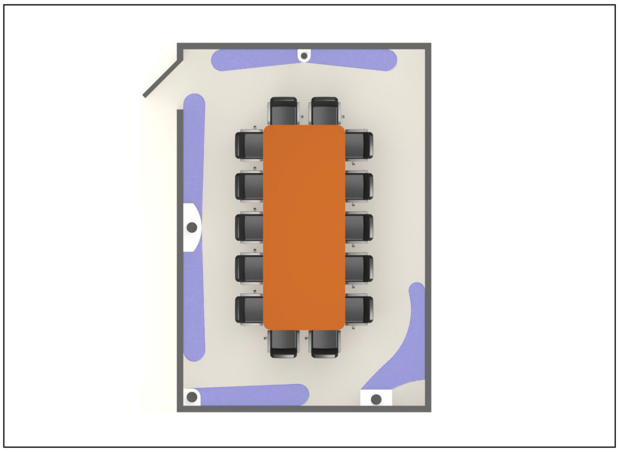

Figure 7. Adjusted air patterns

OUTLET CHOICES

Displacement ventilation diffusers are available in a wide range of styles and sizes. Unlike conventional ceiling diffusers, the size and placement of displacement ventilation diffusers require early coordination with architectural professionals for successful project integration. Generally speaking, displacement diffusers can be ducted from above, below or plenum-supplied.

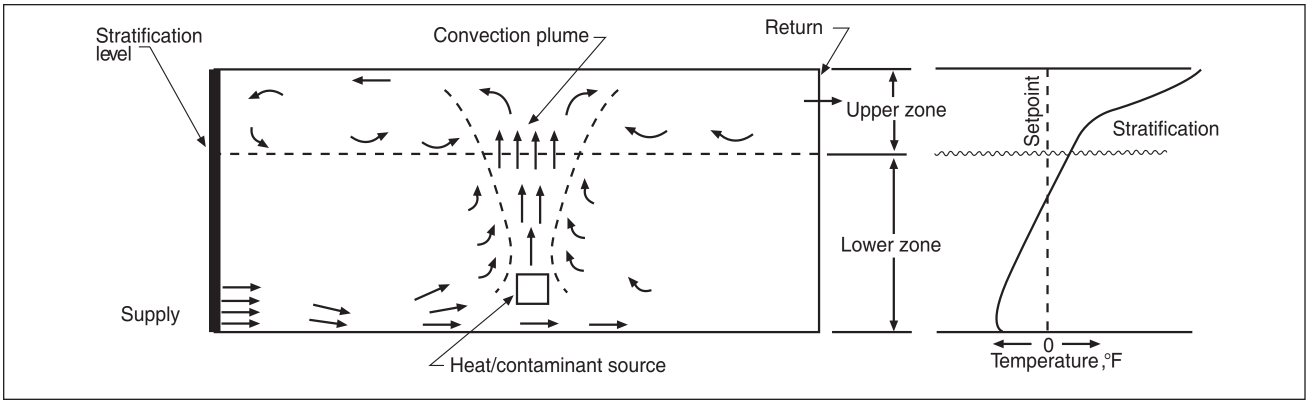

Figure 8. Convective heat sources

All Titus displacement diffusers include:

- Adjustable air pattern controllers
- Air balancing tap
- Removable face plate
- All metal construction (galvanized steel and aluminum)
- Standard #26 white powdercoat finish
- Optional telescoping duct cover (not applicable to DVR1)
- Optional 2-3/4 or 4 inch mounting base (not applicable to DVR1)

CONVECTIVE HEAT FLOWS

As shown in figure 8, the free flow of convective heat is essential in establishing a fully-stratified system. As heat moves from warmer surfaces to the cooler surrounding air, the buoyancy of the air increases and the heat rises to create stratification in the occupied zone. This upward air motion driven by convection also results in room air entrainment that increases the volumetric flow rate of the heat plume as it rises. Although radiant heat sources do not directly affect these convective heat plumes, they may increase plume formation by increasing surface temperatures of heat sources.

The characteristics of individual convective heat plumes may be influenced by each of the following:

- Heat source shape and size
- Heat source intensity
- Air motion surrounding heat source
- Vertical temperature gradient in the space
- Convective heat plumes will continue to rise until they encounter a horizontal obstruction or reach a stratification level of equal temperature.

SPACE TEMPERATURE GRADIENTS AND AIRFLOW RATES

Displacement diffusers supply conditioned air at higher cooling temperatures (typically 62 to 70°F) and lower discharge velocities (less than 70 fpm) than diffusers in mixed air systems. Since the supply air is always cooler than the room air, it can be said to cascade from the diffuser face to the floor. The negative buoyancy of the cooler air and the momentum of the supply air stream cause it to spread along the floor level. When it encounters a convective heat plume, the conditioned air is drawn upward to replace the warm air that is displaced by the convective plume. As the supply air warms, its buoyancy increases and transports it to the upper mixed zone near the ceiling. The distance from the floor to the upper mixed zone is known as the stratification height. Since the design goal of a displacement ventilation system is to create temperature stratification throughout the occupied zone, it is critical that the stratification height exceeds the height of the occupied zone.

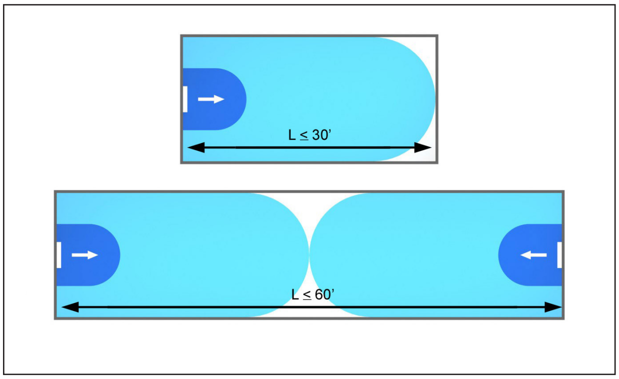

Figure 9. Coverage of displacement outlets

AIR PATTERN PROJECTION

Although displacement ventilation is typically supplied from a low sidewall, the resulting room pattern is very different from a conventional sidewall grille. Because the supply air is cooler than the room air and is discharging at low velocity, it immediately drops to the floor. The air moves across the floor in a thin layer typically no more than 6-8 inches deep.

This air pattern tends to stretch out and cover the entire room, even if the room shape is irregular. Obstructions such as partitions or furniture resting directly on the floor may result in coverage gaps, but the air pattern will quickly rejoin itself much like fluid passing around an object. Displacement diffusers can typically provide coverage into a room that is up to six times the length of the adjacent zone. Internal heat load concentrations can help to extend the projection of a displacement system by drawing the air across the room. Large rooms can be supplied from the side walls so long as the distance from the diffuser face to the most distant heat source is no more than 30 feet. When room dimensions exceed 30 feet in both length and width, it is best to place displacement diffusers on more than one wall. By placing diffusers on opposing walls, rooms up to 60 feet can be supplied from side walls. Another solution for large rooms is to place 360-degree (column type) diffusers within the space.

SYSTEM DESIGN CONSIDERATIONS

A successful displacement ventilation design should provide a supply airflow rate to meet the thermal gradient profile of an occupied space in accordance with ASHRAE comfort guidelines. ASHRAE Standard 55-2017 'Thermal Environmental Conditions for Human Occupancy' recommends that vertical temperature differential (see figure 10) between a seated occupant's ankle and head regions (roughly 4 to 43 in) should be no more than 5.4°F to deliver acceptable comfort to 95% or more of the occupants. For a stationary standing person same guideline a maximum 7.8°F differential is recommended.

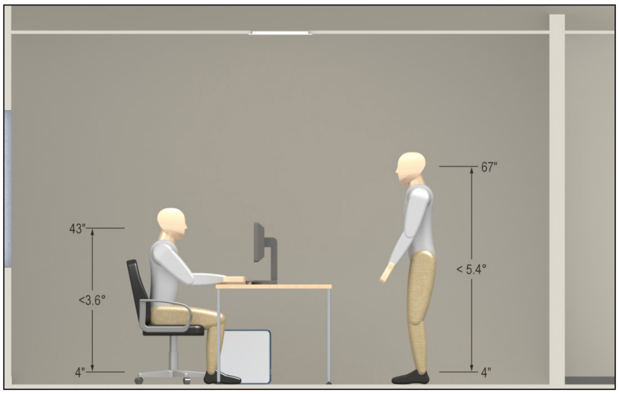

Figure 10. Vertical temperature differentials

SUPPLY AIR CONNECTIONS

Displacement ventilation diffusers are usually supplied by ductwork and they can be supplied from either above or below.

- Optional telescoping duct covers are available to hide otherwise visible supply ductwork for a clean finished appearance
- Optional mounting bases (2-3/4 or 4 inch) are recommended to prevent possible damage due to traffic and floor cleaning equipment. These mounting bases are also recommended to simplify installation when air will be supplied from below.

It is also possible to supply displacement ventilation diffusers from a pressurized plenum.

ACOUSTICAL PERFORMANCE

Like any building space, those supplied by displacement ventilation create an acoustical environment with contributions from air handlers, terminal units, diffusers and structure-borne sound. Properly sized and selected displacement ventilation diffusers are rarely the cause of noise complaints because they operate at low pressure and low velocity and therefore do not generate audible noise. The catalog sound performance rating of a displacement diffuser is usually expressed in terms of a noise criteria (NC) level based upon a typical space with room absorption of 10 dB in each octave band per ASHRAE Standard 70-2006 (Appendix D). While this typical space effect has been used for many years to estimate the sound level of a diffuser serving a small office, this certainly isn't the typical environment in which displacement ventilation is employed. Since we are often dealing with much larger spaces and taller ceilings, different methods must be employed to better estimate sound levels. A space effect for each octave band can be calculated based upon the size of the room and the distance between the source and observer using the following equation per AHRI Standard 885-2008:

$$\text{Space Effect} = 25 - 10 \log(\text{ft}) - 5 \log(\text{ft}^3) - 3 \log(\text{Hz})$$

Where:

- ft = Distance between the source and observer
- ft^3 = Room volume
- Hz = Octave band center frequency

When considering sound contributions from multiple diffusers, we can logarithmically add or multiply, but this is typically unnecessary. In large spaces, diffusers are rarely close enough together to contribute to the overall room sound level. As a general rule for smaller spaces, it is advisable to select diffusers for an NC level that is 10 points lower than the desired room sound level. This has the effect of masking the sound contribution of the diffusers in the background sound level. For larger spaces, the NC level of the diffuser is less critical because the room effect is so much greater.

AIRFLOW RATE CALCULATIONS

The design air volume supplied by a displacement ventilation system must be capable of meeting both the cooling and minimum ventilation requirements for a given space. It is also recommended that the space latent cooling requirements be considered when calculating the required supply airflow rate. The applicable space ventilation airflow rate (CFM_V) can be determined by procedures in ASHRAE Standard 62.1-2016 or by other overriding ventilation needs. The airflow rates necessary to offset the sensible and latent heat gains respectively can be calculated by using the following equations:

$$CFM_{SENS} = ESHG / (60\rho c_P \times \Delta T_{SE})$$

$$CFM_{LAT} = TLHG / (1076 \times 60\rho \times \Delta W_{SE})$$

where:

ESHG = Effective space sensible heat gain, Btu/h
TLHG = Total space latent heat gain, Btu/h
= Density of the room air, lbm/ft^3
c_P = Specific heat of air, Btu/lbm-°F
ΔTSE = ΔT between supply and return air, °F
ΔWSE = Humidity ratio differential between supply and return air, lbm/lb-DA

The supply airflow rate must then be the greatest of the three airflow rates determined above.

There are two sources for methods of airflow calculation in displacement ventilation systems, an ASHRAE method and that which is published by the

REHVA (Federation of European Heating, Ventilating and Air Conditioning Associations) Guidebook for Displacement Ventilation. The examples that are presented here will use and compare the results of both methods.

ASHRAE AIRFLOW CALCULATION METHOD

ASHRAE's publication 'System Performance Evaluation and Design Guidelines for Displacement Ventilation' (2003) presents a procedure for calculating displacement system supply airflow rates. The procedure involves applying factors to each of the sensible heat gains listed above in order to arrive at an effective sensible heat gain (ESHG) for the space.

Step 1: Establish the total space sensible heat gain (TSHG)

In order to determine the cooling design air volume, the type, location and magnitude of all space sensible heat gains must be identified. These loads can be classified as:

- Heat generated by occupants, Q_O
- Heat generated by equipment in the space, Q_E
- Heat generated by overhead lighting, Q_L
- Heat from the exterior wall and window surfaces including transmitted solar radiation, Q_{EX}

Step 2: Determine the effective sensible heat gain (ESHG)

The procedure establishes the following derating factors for each category of sensible heat gains listed above.

- Occupant heat gains: $ESHG_O = Q_O \times 0.295$
- Equipment heat gains: $ESHG_E = Q_E \times 0.295$
- Lighting heat gains: $ESHG = Q_L \times 0.132$
- External heat gains, $ESHG_{EX} = Q_{EX} \times 0.185$

The individual ESHG's are then summed to arrive at a total ESHG for the space. The required space sensible cooling airflow rate can then be estimated by the following equation:

$$CFM_{SENS} = \sum ESHG\text{'s} / (60\rho cP \times \Delta T_{hf})$$

where, ΔT_{hf} is the maximum allowable head to ankle ΔT

At standard density ($\rho = 0.075$ lb/ft³), the equation becomes:

$$CFM_{SENS} = \sum ESHG\text{'s} / (1.08 \times \Delta T_{hf})$$

Step 3: Compare CFM_{SENS} to CFM_{LAT} and CFM_V

Compare the three airflow calculations. Note that the ventilation component (CFM_V) should be corrected to CFM_{VC} according to the zone ventilation effectiveness factor (E_Z) assigned by ASHRAE Standard 62.1:

$$CFM_{VC} = CFM_V / E_Z$$

The greatest of the three will be the required space airflow rate (CFM_{SA}).

Step 4: Determine the supply air temperature

The minimum supply air temperature (T_S) required to maintain the prescribed ΔT_{hf} value can be determined as:

$$T_S = T_h - \Delta T_{hf} - A \times TSHG / (2.456 \times CFM_S^2 + 1.08 \times A \times CFM_{SA})$$

where,

T_H = Desired temperature at the occupant head level, °F
A = Floor area of the space, ft²

Step 5: Determine the exhaust air temperature (T_E)

The exhaust air temperature T_E can be estimated as:

$$T_E = T_S + TSHG / (1.08 \times CFM_{SA})$$

REHVA AIRFLOW CALCULATION METHOD

The REHVA method is based on applying "typical" space temperature gradient relationships to determine acceptable supply and return temperatures, then using the resultant temperature difference between those to estimate the airflow rate required to accomplish the space sensible cooling. The procedures do not involve factoring of the individual heat gains but do assume that the heat gains are "typical" in both relative magnitude and location.

REHVA suggests that the space under consideration be classified according to its ceiling height and proposes separate methodologies for room heights less than 4.5 m (around 14 feet) and those with higher ceilings. These two methods are referred to as the "50/50 rule" and the "1/3, 2/3 rule" respectively.

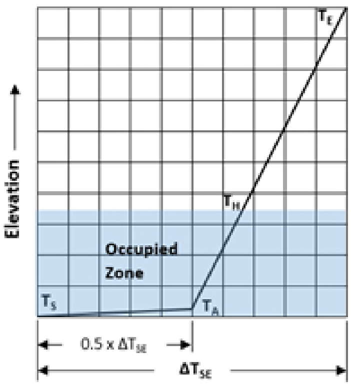

Figure 11. REHVA "50/50" rule

Figure 11 illustrates REHVA's 50/50 rule. It is intended for use with ceiling heights between 9 and 14 feet and assumes that 50% of the differential (ΔT_{SE}) between the supply (T_S) and return (T_E) temperature is dissipated between the time the air leaves the diffuser and contacts the ankle (T_A) of an occupant outside the outlet's adjacent zone. It also models the vertical temperature gradient as a straight line from ankle level to the overhead return.

For seated occupants, the head level is considered to be 3.6 feet above the floor, thus the occupied zone height (H_{OZ}) is 3.6 feet. REHVA also suggests that the head to foot temperature gradient not exceed 1°F per foot, thus occupied zone temperature differential (ΔT_{OZ}) should not exceed 3.6°F for seated occupants. The room vertical temperature gradient (ΔT_{AE}) can be determined by the following equation where HROOM is the ceiling height:

$$\Delta T_{AE} = \Delta T_{OZ} / (H_{OZ}/H_{ROOM})$$

Applying REHVA's 50/50 rule, the maximum supply to return temperature differential (ΔT_{SE}) is twice that of ΔT_{AE}. ΔT_{SE}, as well as the appropriate supply and return temperatures can now be calculated.

$$\Delta TSE = 2 \times \Delta T_{OZ} / (H_{OZ}/H_{ROOM})$$

$$T_S = (T_H - \Delta T_{OZ}) - (0.5 \times \Delta T_{SE})$$

$$T_E = T_S + \Delta T_{SE}$$

When ceiling heights exceed 14 feet, REHVA suggests that 1/3 of the overall space temperature differential (ΔT_{SE}) is dissipated before the air reaches the occupant ankle level and the space temperature gradient comprises the other 2/3. Figure 12 illustrates REHVA's 33 /67 rule applied to a 20 foot ceiling.

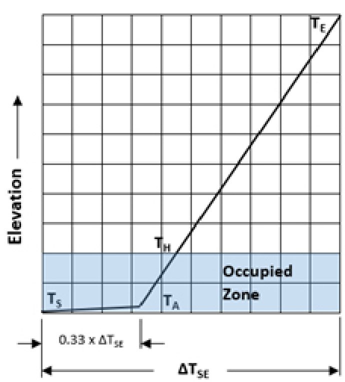

Figure 12. REHVA "33/67" rule

Note that the occupied zone now represents a much smaller percentage of the room volume, in this case only 17.5%. The 3.6°F occupied zone temperature differential (ΔT_{OZ}) now only represents 11.7% of the supply to return temperature differential ΔT_{SE}. The supply to return temperature differential ΔT_{SE} and the supply (T_S) and return (T_E) temperatures can be calculated as:

$$\Delta T_{SE} = 1.5 \times \Delta T_{OZ} / (H_{OZ}/H_{ROOM})$$

$$T_S = (T_H - \Delta T_{OZ}) - (0.33 \times \Delta T_{SE})$$

$$T_E = T_S + \Delta T_{SE}$$

For applications that involve locating stationary occupants in close proximity to the supply outlets, it is recommended that supply (T_S) temperatures below 65°F not be employed.

The following examples compare the airflow requirement results from each source.

Example 1:

A 250 ft² conference room with 10 foot ceilings is to be cooled by a displacement system. The room is designed for 10 occupants and has the following sensible cooling loads:

Occupants: Q_O = 2,500 Btu/h
Equipment: Q_E = 825 Btu/h
Lights: Q_L = 1,275 Btu/h
External: Q_{EX} = 4,800 Btu/h

Airflow rates for latent cooling (CFM_{LAT}) and ventilation (CFM_{VC}) are 210 and 150 CFM, respectively. The temperature at the 3.6 foot level is to be maintained at 72°F.

Calculate the airflow requirement for sensible cooling using the ASHRAE and the REHVA methods.

Solution:

ASHRAE calculation

Step 1: Establish the total sensible heat gains (TSHG):

$$TSHG = Q_O + Q_E + Q_L + Q_{EX} = 9{,}400 \text{ Btu/h}$$

Step 2: Establish the effective sensible heat gain (ESHG):

$$ESHG = 0.295Q_O + 0.295Q_E + 0.132Q_L + 0.185Q_{EX} = 2{,}037 \text{ Btu/h}$$

Step 3: Establish the airflow rate to provide sensible cooling:

$$CFM_{SENS} = ESHG / (1.08 \times \Delta T_{HA}) = 524 \text{ CFM}$$

Step 4: Is CFM_{SENS} greater than CFM_{LAT} and CFM_{VC} (Yes):

Step 5: Determine required supply air temperature (T_S):

$$T_S = T_H - \Delta T_{HA} - (A \times TSHG) / (2.456 \times CFM_{SA}{}^2 + 1.08A \times CFM_{SA})$$

$$T_S = 72 - 3.6 - (250 \times 9{,}400) / (2.456 \times 524^2 + 1.08 \times 250 \times 924)$$

$$T_S = 65.5°F$$

Step 6: Calculate resultant return air temperature (T_E):

$$T_E = T_S + TSHG / (1.08 \times CFM_{SA}) = 65.5 + 9{,}400 / (1.08 \times 524)$$

$T_E = 65.5 + 16.6 = 82.1°F$

REHVA 50/50 rule:

Step 1: Identify the total sensible heat gain (TSHG):

TSHG = 9,400 Btu/h

Step 2: Calculate ΔT_{OZ} as a percentage of the total space temperature differential ΔT_{SE}:

$\Delta T_{OZ} = \Delta T_{AE} * (H_{OZ} / H) = (0.5 \times \Delta T_{SE}) * (3.6 / 10) = 0.18\ \Delta T_{SE}$

Step 3: Determine temperature differential ΔTSE:

$\Delta T_{SE} = \Delta T_{OZ} / 0.18 = 3.6 / 0.18 = 20.0°F$

Step 4: Determine the airflow rate for sensible cooling (CFM_{SENS}):

$CFM_{SENS} = TSHG / (1.08 \times \Delta T_{SE}) = 435$ CFM

Step 5: Is CFMSENS greater than CFMLAT and CFMVC? (Yes)

Step 6: Determine the supply air temperature (TS):

$T_S = T_{ROOM} - \Delta T_{OZ} - (0.5 \times \Delta T_{SE}) = 58.4°F$

Step 7: Determine the return air temperature (TS):

$T_E = T_S + \Delta T_{SE} = 58.4 + 20.0 = 78.4°F$:

Example 2:

A 750 ft² classroom with 10 foot ceilings is to be cooled by a displacement system. The classroom is designed for 30 occupants and has the following sensible cooling loads:

Occupants: $Q_O = 7,500$ Btu/h
Equipment: $Q_E = 2,560$ Btu/h
Lights: $Q_L = 3,840$ Btu/h
External: $Q_{EX} = 6,750$ Btu/h

Airflow rates for latent cooling (CFM_{LAT}) and ventilation (CFM_{VC}) are 575 and 325 CFM, respectively. The temperature at the 3.6 foot level is to be maintained at 75°F.

Calculate the airflow requirement for sensible cooling using the ASHRAE and the REHVA methods.

Solution:

ASHRAE calculation:

Step 1: Establish the total sensible heat gains (TSHG):

$TSHG = Q_O + Q_E + Q_L + Q_{EX} = 20,650$ Btu/h

Step 2: Establish the effective sensible heat gain (ESHG):

$ESHG = 0.295Q_O + 0.295Q_E + 0.132Q_L + 0.185Q_{EX} = 4,723$ Btu/h

Step 3: Establish the airflow rate to provide sensible cooling:

$CFM_{SENS} = ESHG / (1.08 \times \Delta THA) = 1,215$ CFM

Step 4: Is CFM_{SENS} greater than CFM_{LAT} and CFM_{VC} (Yes):

Step 5: Determine required supply air temperature (T_S):

$T_S = T_H - \Delta T_{HA} - (A \times TSHG) / (2.456 \times CFM_{SA}^2 + 1.08A \times CFM_{SA})$

$T_S = 75 - 3.6 - (750 \times 20,650) / (2.456 \times 1,215\ 2 + 1.08 \times 750 \times 1,215)$

$T_S = 68.0°F$

Step 6: Calculate resultant return air temperature (T_E):

$T_E = T_S + TSHG / (1.08 \times CFM_{SA}) = 68.0 + 20,650 / (1.08 \times 1,215)$

$T_E = 68.0 + 15.7 = 83.7°F$

REHVA 50/50 rule:

Step 1: Identify the total sensible heat gain (TSHG):

TSHG = 20,650 Btu/h

Step 2: Calculate ΔT_{OZ} as a percentage of the total space temperature differential ΔT_{SE}:

$\Delta T_{OZ} = \Delta T_{AE} * (H_{OZ} / H) = (0.5 \times \Delta T_{SE}) * (3.6 / 10) = 0.18\ \Delta T_{SE}$

Step 3: Determine temperature differential ΔT_{SE}:

$\Delta T_{SE} = \Delta T_{OZ} / 0.18 = 3.6 / 0.18 = 20.0°F$

Step 4: Determine the airflow rate for sensible cooling (CFM_{SENS}):

$CFM_{SENS} = TSHG / (1.08 \times \Delta T_{SE}) = 956$ CFM

Step 5: Is CFM_{SENS} greater than CFM_{LAT} and CFM_{VC}? (Yes)

Step 6: Determine the supply air temperature (T_S):

$T_S = T_{ROOM} - \Delta T_{OZ} - (0.5 \times \Delta T_{SE}) = 61.4°F$

Step 7: Determine the return air temperature (T_S):

$T_E = T_S + \Delta T_{SE} = 61.4 + 20.0 = 81.4°F$

Example 3:

Calculate the sensible cooling airflow requirement.

For the classroom from Example 2 but with a 20 foot ceiling height, the sensible cooling loads remaining the same.

Occupants: Q_O = 7,500 Btu/h
Equipment: Q_E = 2,560 Btu/h
Lights: Q_L = 3,840 Btu/h
External: Q_{EX} = 6,750 Btu/h

Airflow rates for latent cooling (CFM_{LAT}) and ventilation (CFM_{VC}) are again 575 and 325 CFM, respectively. The temperature at the 3.6 foot level is to be maintained at 75°F.

Solution:

Note that the ASHRAE calculation does not differentiate between ceiling heights and is only based on maintaining a specified occupied zone temperature differential for the given sensible load conditions, thus the results using the ASHRAE calculation are the same as those derived in Example 1:

$$TSHG = Q_O + Q_E + Q_L + Q_{EX} = 20{,}650 \text{ Btu/h}$$

$$ESHG = 0.295Q_O + 0.295Q_E + 0.132QL + 0.185QEX = 4{,}723 \text{ Btu/h}$$

$$CFM_{SENS} = ESHG / (1.08 \times \Delta THA) = 1{,}215 \text{ CFM}$$

$$T_S = 68.0°F \text{ and } T_E = 68.0 + 15.7 = 83.7°F$$

REHVA 33/67 rule:

Step 1: Determine the total sensible heat gain (TSHG):

$$TSHG = 20{,}650 \text{ Btu/h}$$

Step 2: Calculate ΔT_{OZ} as a percentage of the total space temperature differential ΔT_{SE}:

$$\Delta TOZ = \Delta TAE * (HOZ / H) = (0.67 \times \Delta TSE) * (3.6 / 20) = 0.12\ \Delta TSE$$

Step 3: Determine temperature differential ΔT_{SE}:

$$\Delta TSE = \Delta TOZ / 0.12 = 3.6 / 0.12 = 30.0°F$$

Step 4: Determine the airflow rate for sensible cooling (CFM_{SENS}):

$$CFM_{SENS} = TSHG / (1.08 \times \Delta T_{SE}) = 637 \text{ CFM}$$

Step 5: Is CFM_{SENS} greater than CFM_{LAT} and CFM_{VC}? (Yes)

Step 6: Determine the supply air temperature (T_S):

$$T_S = T_{ROOM} - \Delta T_{OZ} - (0.33 \times \Delta T_{SE}) = 61.5°F$$

Step 7: Determine the return air temperature (T_S):

$$T_E = T_S + \Delta T_{SE} = 61.5 + 30.0 = 91.5°F$$

Comparing the airflow calculation models

The preceding examples illustrate the following observations regarding the airflow rate calculation models:

1. There is a significant difference between the supply air temperatures suggested by the models. In all of the examples studied, the REHVA methods suggest supply air temperatures some 7°F cooler than those calculated by the ASHRAE method. These are often quite low especially for spaces where occupants must be located near the outlet and zone temperature gradient (ΔT_{OZ}) limits are 3.6°F or less.
2. There is a significant difference between the sensible cooling airflow rates generated by the models. In the examples with conventional (10 foot) ceiling heights the ASHRAE method airflow rates were 20 to 25% higher than those predicted by the REHVA 50/50 rule.
3. 3) The ASHRAE model does not differentiate between ceiling heights, basing its calculations solely upon the space sensible heat gains and the specified occupied zone temperature gradient (ΔT_{OZ}).
4. 4) Both models calculate the predicted sensible cooling airflow rate and a minimum supply air temperature that will limit the occupied zone temperature gradient (ΔT_{OZ}) to that specified when subjected to the specified space total sensible heat gain (TSHG). Since the TSHG for multiple spaces served by the same air distribution system will vary, so too will the space minimum supply air temperatures. Therefore, the system supply air temperature should be the highest of the spaces it serves and the other space airflow rates should be re-calculated accordingly.

Simplified calculation method

The chart on the following page introduces a simplified airflow calculation method that is based on the REHVA models. Table 1 recommends supply air temperatures and design supply to return temperature differentials for use according to the listed application.

The values in table 1 can be entered into the appropriate REHVA model to calculate the airflow rate for space sensible cooling. The recommended design ΔTSE values for ceiling heights up to 12 feet are intended for use with the REHVA 50/50 model while those for higher ceilings (indicated by shading in the table) apply to REHVA's 33/67 rule. The following example illustrates the use of the values in the table.

Example 4:

Repeat the classroom example (example 2) using the values from Table 1. Assume that the distance from the supply outlet to the closest stationary occupant is 5 feet.

Occupancy Type	Max. ΔT_{OZ} °F	Distance to outlet ft	Max. ΔT_{SH} °F	Design ΔT_{SH} / ΔT_{SE}					
				Ceiling height, ft.					
				9	12	15*	18*	21*	24*
Transient	7.2	Any	14	0.82	0.78	0.58	0.54	0.51	0.48
Stationary *(standing)*	5.4	10 or more	13	0.81	0.73	0.58	0.53	0.51	0.48
		6 to 9	12						
		Within 5	11						
Stationary *(seated)*	3.6	9 or more	13	0.70	0.65	0.49	0.46	0.44	0.43
		5 to 8	12						
		Within 4	10						

**Values for ceiling heights above 12 feet apply to REHVA 33/67 rule*

Table 1: Simplied airflow calculation

Solution:

Table 1 suggests a supply to room air temperature differential (ΔT_{SH}) of 12°F be used for the application where occupants are 5 feet from the supply outlet. The $\Delta T_{SH}/\Delta T_{SE}$ ratio for a 10 foot ceiling height of 10 feet can be estimated by interpolation, in this case 0.683. Since the maximum value of ΔT_{SH} is defined in the table as 11°F, the value of ΔT_{SE} can be calculated:

$$\Delta T_{SE} = \Delta T_{SH} / 0.683 = 12 / 0.683 = 17.6°F$$

Knowing ΔT_{SE}, the classroom airflow requirement can now be calculated:

$$CFM = TSHG / (1.08 \times \Delta T_{SE}) = 20{,}650 / (1.08 \times 17.6) = 1{,}088$$

Using the 50/50 rule, the occupied zone temperature gradient ΔT_{OZ} can then be calculated:

$$\Delta T_{OZ} = 0.5 \times \Delta T_{SE} \times (H_{OZ} / H) = (0.5 \times 17.6) \times (3.5 / 10) = 3.1°F$$

This is indeed within the recommended 3.6°F limit.