SECTION E

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Engineering Guide
Critical Environments

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

Introduction to Patient Care Areas

Each different room type in the hospital environment requires a unique and strategic approach to air distribution and temperature control. Past and ongoing industry research continues to impact the thinking of HVAC engineers involved in the design of health care facilities or the development of governing standards. This section will help explain some of the key North American codes and standards for health care facility design as they relate to each space type, as well as the logic behind these design guidelines. Following these design standards, air distribution strategies are presented for patient care areas, waiting rooms, operating rooms, and hospital pharmacies and labs. The many design examples included in this chapter should serve to further clarify the key points presented in each section.

The health care facility includes a number of different spaces, all with unique HVAC requirements. Well designed hospital HVAC systems should support asepsis control while also taking advantage of energy saving technologies and strategies. Understanding the needs and goals of each space as well as the regulations that govern their design are important first steps toward building a high performing facility.

The following chapter will discuss a number of key areas in the hospital and provide recommendation on available HVAC technology and how it can be applied. Examples throughout the chapter give more clarity on practical application of the concepts and standards presented.

Patient care areas in a health care facility are carefully controlled environments designed to provide contaminant control without sacrificing the comfort of patients or other occupants. Various hospital rooms fall into the 'Patient Care' category, including standard patient rooms, isolation rooms, and burn center intensive care units.These spaces are similar in that they are all used for patient treatment and/or recovery, however, each has unique design characteristics which clearly distinguish one type from another. With a different purpose for each room type, the standards and general HVAC considerations that dictate the design of the local air distribution systems are also different.

Standard Patient Rooms

The majority of patient care areas in a hospital are standard patient rooms. As such, the design of the air distribution systems in these rooms can greatly affect the performance and cost of operating the overall HVAC system. Standard patient rooms can be either single or multiple patient spaces and are typically dedicated to the care of individuals without serious health risk. These rooms will often have an exterior wall and window which can have a significant impact on room air patterns in certain climatic regions.

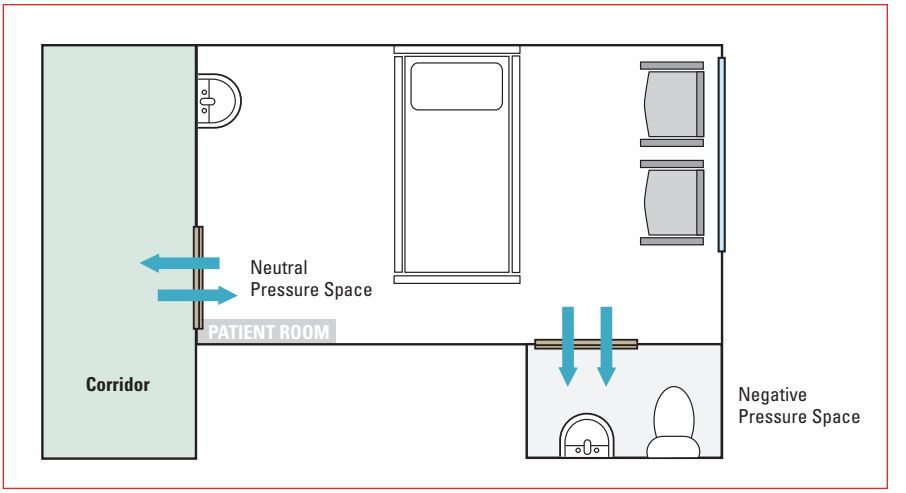

Figure 1: Typical patient room

The variability in occupancy levels, solar/conduction loads through exterior walls, and equipment loads will generally facilitate the need for reheat, VAV control, supplemental room heating or cooling, exterior shading, or any combination of these features. If insulating liners are used for supply terminal boxes, no fibers should be exposed to the air stream, with closed-cell foam liners the preferred option.Venturi valves are becoming more popular for return/exhaust applications in patient rooms as they can provide accurate flow control without the need for cross-flow sensors. Frequent linen changes in patient rooms can result in high levels of airborne lint which may interfere with the function of the cross-flow sensors typical to most terminal boxes. Another common design practice is to locate a radiant panel above the window on an exterior wall.The radiant panel serves to moderate the temperature around the exterior wall as to prevent unfavorable influence over the room air pattern.

ASHRAE suggests the use of either group A (horizontal throw) or group E (vertical throw) diffusers in standard patient rooms, however, the former is far more common for these applications. Use of displacement ventilation products in patient rooms is also gaining traction as research surrounding the contaminant control benefits of 'single-pass' DV air flow continues to justify support for the technology.

Supply and exhaust diffusers should be selected and located in such a way as to promote the movement of air from clean to less clean areas and also prevent uncomfortable drafts. Mixing type, group A diffusers will typically be located directly above the patient bed with exhaust outlets located near the door to the corridor or toilet room. Attention should be given to ceiling-mounted obstructions that may interfere with the discharge pattern of horizontal throw diffusers. Lift rails or curtains may redirect high velocity supply air directly at the patient if positioned too close to the diffuser. Exhaust outlets can be located at a low level, but eggcrate style ceiling returns are more common and preferred.

Isolation Rooms

Isolation rooms can be separated into two main categories: Airborne Infection Isolation (AII) and Protective Environment (PE) rooms. As the names would suggest, these rooms have different functions. All rooms are designed for patients with serious and contagious conditions (e.g. Tuberculosis), while PE rooms exist to protect patients with weakened immune systems or some form of impairment to their natural defense system. All rooms are designed with the primary purpose of protecting hospital occupants (other than the patient) from airborne infection, while PE rooms should protect the more vulnerable patients (e.g. bone marrow transplant patients) from airborne contaminants present in the hospital environment.

Airborne Infection Isolation (AII) Rooms

The general layout and some design conditions of an AII room will be similar to that of a standard patient room. The possibility of exterior walls and windows, an attached toilet room, and similar equipment loads are a few commonalities between these spaces. Yet the AII room serves a different purpose and consequently has a number of key differences with respect to standards and air distribution equipment.

The minimum total air-change rate for an AII room is typically 12 ach, but this may vary depending on the applicable code.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Introduction to Patient Care Areas

An Airborne Infection Isolation (AII) room must maintain a negative pressure differential relative to the hospital corridor. To achieve an acceptable pressure differential it is usually necessary to supply approximately 20% less air to the room than is exhausted. The actual air flow offset will need to be determined at the time of commissioning. Furthermore, all air exhausted from the AII room, or associated toilet room and anteroom, should be exhausted directly outdoors without mixing with the exhaust air from any other spaces.

HEPA filtration of the AII exhaust air is required in cases where the air is not discharged clear of building openings or mixes with exhaust air from non-AII spaces prior to being discharged outside.

Most standards do not require anterooms for AII rooms, however, most modern medical facilities utilize them. An AII anteroom should have a negative pressure differential relative to the corridor, and a positive pressure differential relative to the AII room. The minimum relative pressure differential between these spaces will depend on the jurisdiction and applicable standards. This pressurization scheme supports the movement of air and contaminants from clean (the corridor) to less clean (AII room) spaces, reducing the potential for further spread of infection. All rooms also require a permanently installed device to monitor the differential pressure between the room and adjacent areas (ASHRAE Standard 170-2008; CSA Z317.2-10). Most modern health care facilities achieve this level of monitoring by using a room pressure monitor with a built-in pressure sensor. Typically, the room pressure monitor is only used to detect and display differential pressure between the room and corridor, however, some models also include an integral controller for supply and exhaust valve control. Audible alarms are often used in conjunction with the visual signal displayed on the screen of the room pressure monitor.

The relatively high air-change rate requirements for AII rooms typically mean additional supply air is not necessary to meet cooling loads; however, VAV control is still preferable in AII rooms for a number of reasons:

- Adjustment of supply/exhaust air flow to maintain room pressure differential.
- Airborne infection isolation rooms should include provisions for standard patient care when AII precautions are not required (ASHRAE Standard 170-2008).
- The number of air-changes may be reduced when the room is unoccupied provided that pressure relationships with adjacent spaces are maintained at the reduced air-change rate (ASHRAE Standard 170-2008; CSA Z317.2-10).

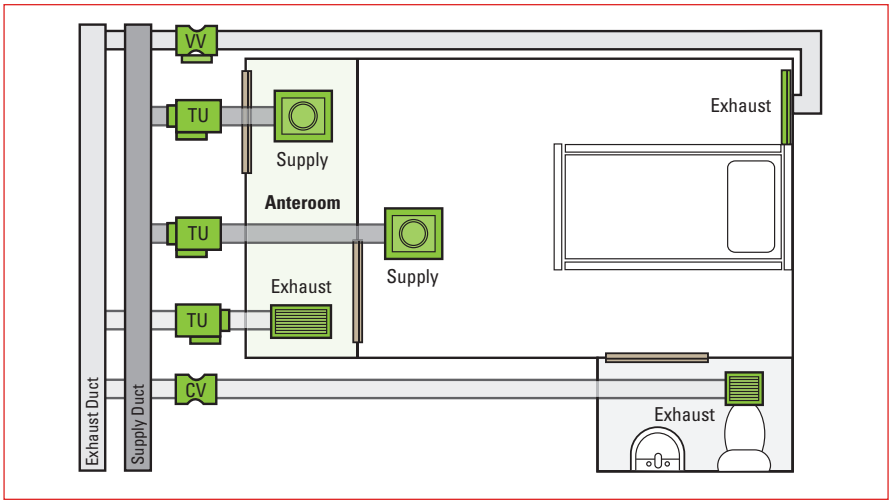

Figure 2: Airborne infection isolation (AII) venturi valve setup

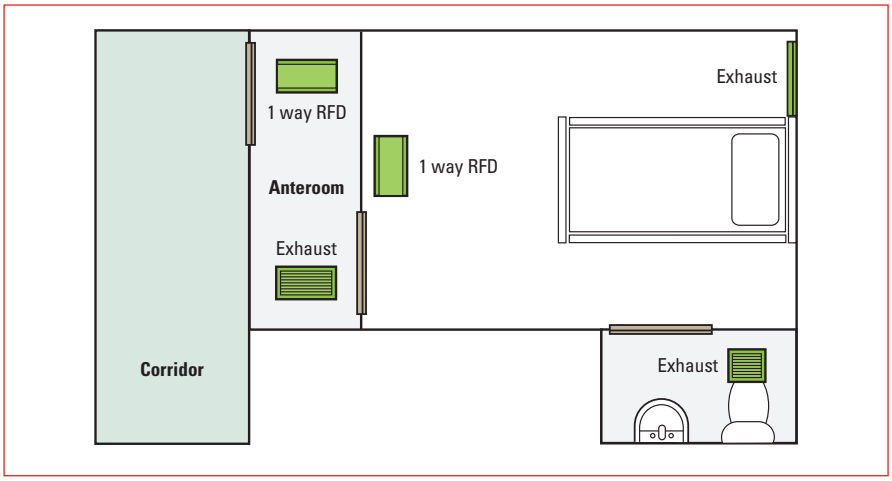

Figure 3: Airborne infection isolation (AII) supply and exhaust outlets

A typical VAV configuration for an AII room is shown in **Figure 2**. Venturi valve exhaust units are sometimes preferred in areas where airborne lint might be a concern.

The current ASHRAE Standard 170 suggests that group A (horizontal throw - mixing diffuser) or group E (vertical throw) diffusers are suitable for AII room applications. The CSA (2010) recommends non-aspirating diffusers (i.e. laminar flow diffusers). One challenge is supplying enough air while also minimizing airborne particle entrainment. To address this challenge, radial flow diffusers are a good alternative. Radial flow diffusers are high capacity outlets with short throws and one or two way discharge patterns. The units are also available with adjustable pattern controllers, but designers should exercise caution when selecting this option, particularly if pattern controllers can be adjusted without removing the face. Cleaning staff may inadvertently move pattern controllers, compromising the safety of healthy room occupants.

The position of the radial flow diffuser is equally as important as the selection of the diffuser itself. The diffuser should be installed to promote the movement of air toward the patient and away from the room entrance. With this in mind, the supply diffuser should be located near the room door. The exhaust grille, should be ceiling-mounted directly above the patient bed (ASHRAE Standard 170-2008) or wall-mounted at a low level near the head of the patient bed (CSA Z317.2-10) (ASHRAE Standard 170-2008), unless it can be demonstrated that such positions are impractical. The exhaust grilles should be sized to achieve a core velocity of approximately 500 fpm in order to realize a desirable noise and pressure drop level.

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Protective Environment (PE) Rooms

PE rooms, while also considered isolation rooms, are essentially used for the opposite purpose to that of an Airborne Infection Isolation (AII) room. Patients in these rooms have a high susceptibility to infection and need greater protection than the average hospital occupant to avoid further health complications. These patients include, but aren't limited to, burn patients, bone marrow or organ transplant patients, leukemia, and AIDS patients.

Most codes require a minimum total air-change rate of 12 ach for a PE room. The exhaust rate should be approximately 20% less than the supply to achieve a suitable positive pressure differential relative to the corridor. The actual exhaust air flow rate will need to be determined at the time of commissioning. Air from a PE room does not need to be exhausted directly to the outdoors as it should with an AII room, however, recirculating room HVAC units or finned tube elements are not permitted by most codes. Some standards will allow recirculation devices equipped with HEPA filters (ASHRAE Standard 170-2008).

A PE room does not require an anteroom unless the PE room also requires airborne infection isolation control (see next section). Standard PE rooms should exhibit a positive pressure differential relative to the corridor, toilet room, or any other adjacent spaces to prevent the infiltration of contaminants. PE rooms also require a permanently installed device to constantly monitor and provide a visual output of the differential pressure between the room and adjacent areas.

PE rooms can be used for standard patient care, but air flow parameters cannot be reversed for the purposes of switching between PE and AII room function.

Current standards suggest group E, non-aspirating supply diffusers, should be used for PE room applications. This would include laminar flow diffusers with average air velocities just below the diffuser face of 30 fpm. The same challenge exists in PE rooms with supplying high volumes of air while also minimizing airborne particle entrainment. Radial flow diffusers are better suited to address this issue than laminar flow diffusers. Radial flow diffusers are not yet categorized by ASHRAE standards, but there is strong precedent for their use in such critical applications and they typically require less ceiling space for equivalent air flow rates. Both radial and laminar flow diffusers are available with integral HEPA filters.

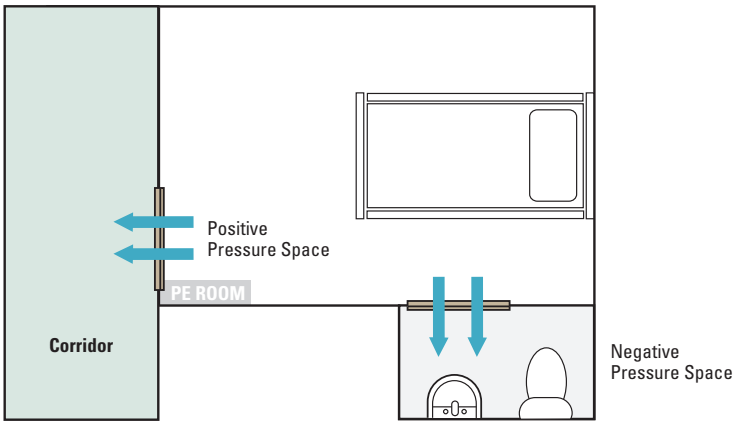

Figure 4: PE room air flow direction

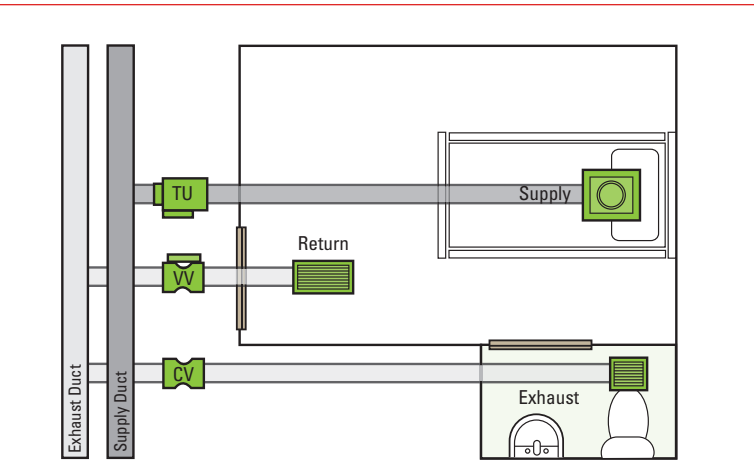

Figure 5: PE room venturi valve setup

Standards require the supply diffuser to be located directly above the patient bed with the exhaust outlet positioned near the door to the corridor or anteroom. The intent of this relative outlet location is to establish a vertical downward wash of clean air through the breathing zone of the patient before the air passes through the rest of the room. The style of exhaust grille will depend on the installation location. Louvered bar grilles are typically used for low-level applications and eggcrate for ceiling applications. Either grille type should be sized based on a core velocity of 500 fpm for optimal pressure drop and noise levels.

Introduction to Patient Care Areas

Airborne Infection Isolation (AII) in Protective Environment Rooms

In situations where a patient is both contagious and highly susceptible to further infection, the PE room design is modified to include airborne infection isolation.

An anteroom is required for these spaces. Pressure in the anteroom relative to the PE room will generally be positive, allowing staff to gown and mask inside the anteroom with less serious risk. Also, positive pressure in the anteroom will reduce the probability of airborne contaminants from the corridor entering the space occupied by the patient.

Unlike the standard PE room, a PE room with AII precautions should be under negative pressure relative to adjacent spaces (other than the toilet room) to prevent exfiltration of contaminants into other occupied zones. Supply and return outlet selection and location should match standard PE room applications.

Burn Center Intensive Care Units (ICU)

Burn center ICUs have a number of unique challenges. These spaces are generally kept at higher relative humidity levels to prevent excess moisture loss from wounds and the associated complications. Draft represents another major consideration. In any application, steps are taken to reduce the possibility and inconvenience of draft in an occupied space, but draft in a burn center can result in severe pain for patients and must therefore be more carefully avoided. The last significant comfort related design criterion is the need for rapid temperature adjustment around the patient bed. During a wound dressing change it is preferable to raise the temperature around the patient from 10 °F to 15 °F. This reduces the ΔT between the air and wound temperature, thus creating a more comfortable condition for the patient.

A burn center ICU will typically have a positive differential pressure relative to adjacent spaces but this is not a requirement in all jurisdictions. Recirculating room HVAC units are not permitted but air filtration requirements are equivalent to that of a standard patient room.

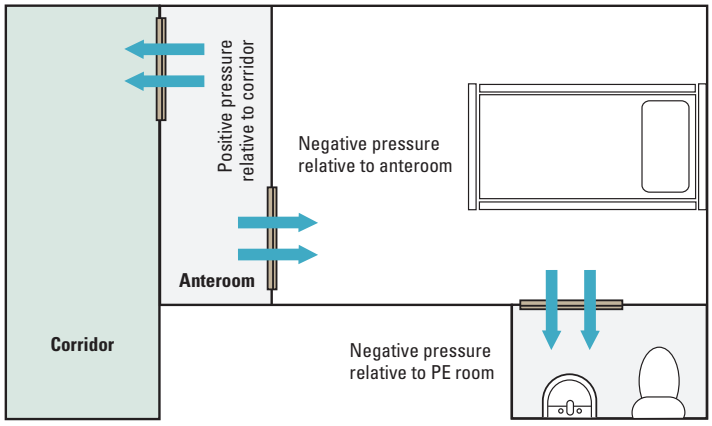

Figure 6: PE room with airborne infection isolation (AII)

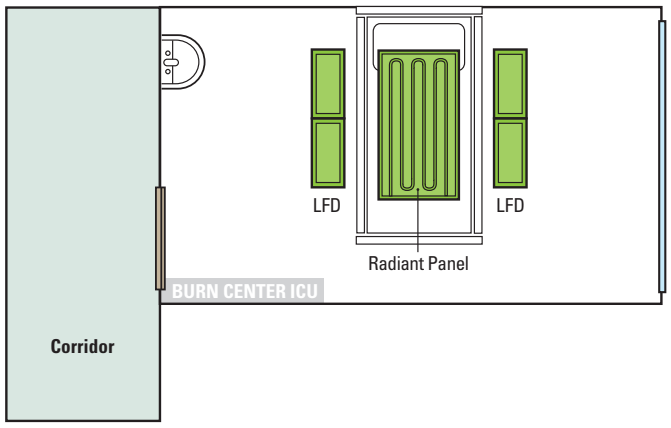

Figure 7: Burn center ICU equipment layout

Group E, non-aspirating diffusers (i.e. laminar flow diffusers) with HEPA filtration should be used in burn center ICU patient rooms for their low discharge velocity and entrainment characteristics. Buoyancy effects can be an issue when heating with laminar flow diffusers, which can result in the clean air not reaching the patient. To address this situation, radiant panels are often used to provide heat near the patient bed without significantly adjusting the supply air temperature. Exhaust outlets will typically be louvered bar grilles located at a low level near the room door to minimize mixing in the occupied space.

Waiting and Examination Rooms

Waiting and examination rooms involve a number of unknowns. In most cases, the patients in these rooms have yet to be diagnosed and the contagion risk must still be determined. The goal of the ventilation system in these spaces should be to provide a comfortable environment while also reducing the probability that infection will spread between occupants.

A waiting room is generally a patient holding area for one or more examination rooms. Waiting rooms are sized to allow for at least two chairs for each associated examination or treatment room. Depending on the size of the facility, waiting rooms can be quite large with a wide range of potential occupancy levels. Single bed examination rooms should have a minimum clear floor area of 120 ft² (FGI, 2010). For both spaces, ASHRAE recommends the use of either group A (horizontal throw) or group E (vertical throw) supply air outlets. Of the two, group A outlets are the most typical for these spaces.

One of the more common group A diffusers used in waiting rooms and examination rooms is the square plaque diffuser (**Figure 10**). These diffusers produce a 360° pattern and rapidly mix supply air with room air providing a high level of comfort and reduced risk of draft in the occupied zone. The flat face and removable center plaque make it easy to clean and sterilize.

Another group A outlet suitable for these applications, especially larger waiting rooms, is the louvered face directional ceiling diffuser (**Figure 11**). Similar to the square plaque diffuser, this outlet has a removable core and provides thorough mixing of room air. Louvered face type diffusers have long throws, and are further differentiated by the variety of available core styles and discharge patterns to suit different room layouts.

Other alternatives include displacement ventilation (DV) or active chilled beams (ACB). Although less common, these technologies are gaining traction in the North American market. In the case of ACBs, some codes allow the air volume induced through the beam to be considered part of the total room air-change rate. With typical induction ratios near 5:1, ACBs are often capable of satisfying total air-change requirements while only supplying the minimum outdoor air volume requirement directly to the diffuser.

When locating supply air outlets in the waiting room, care must be taken to ensure there are no uncomfortable drafts in the occupied zone. Diffuser selection and location should be such that the velocity of air inside the occupied zone does not exceed 50 fpm.

Emergency or chest X-ray (respiratory) waiting rooms should have a negative pressure differential relative to adjacent spaces; most codes also require total room air-change rates of 12 ach. Examination rooms will most often be maintained at a neutral pressure relative to adjacent spaces, but minimum air-change rates will vary significantly from one region to another.

Exhaust air outlets in these spaces are typically located in the ceiling, away from the supply outlets, with an eggcrate type grille representing a common selection. The size of the exhaust grilles should be based on a core velocity of approximately 500 fpm. Selection at 500 fpm will generally result in an acceptable pressure drop and noise level.

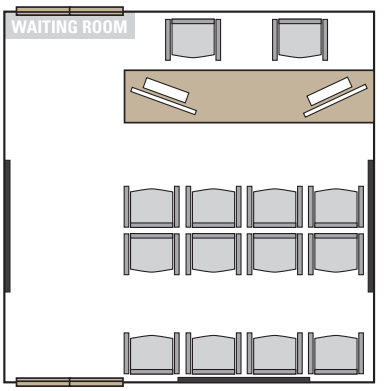

Figure 8: Waiting room

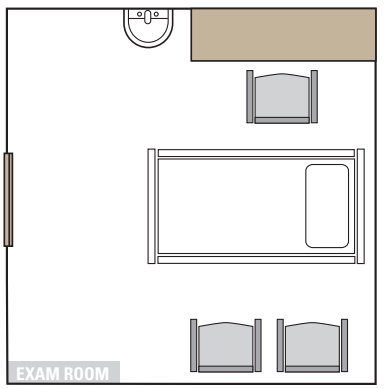

Figure 9: Single bed examination room

Figure 10: Square plaque diffuser

Figure 11: Louvered face supply diffuser

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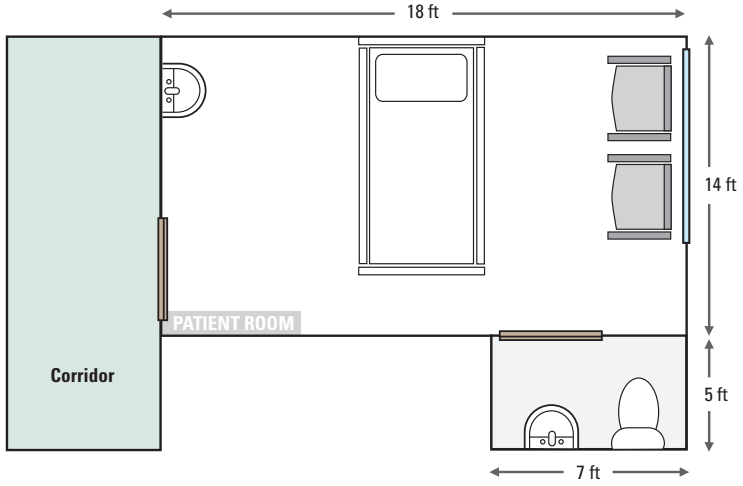
Example 1 - Patient Room with Mixing Diffusers

The patient room used for the following example is a single bed space designed for a maximum of five occupants (patient, medical staff and visitors). The room includes overhead lighting, one television and medical monitoring equipment. The control temperature for the space is 75 °F. The patient room is 14 ft x 18 ft with an attached 5 ft x 7 ft toilet room. Both spaces have a common 9 ft ceiling height. There is one exterior wall and window.



Patient Room Design Criteria (ASHRAE Standard 170-2008):

- 6 ach minimum (total, based on supply volume)
- Neutral room pressure relative to corridor

Toilet Room Design Criteria (ASHRAE Standard 170-2008):

- 10 ach minimum (total, exhaust only)
- Negative room pressure relative to patient room

Space Considerations

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

Supply air temperature is 55 °F

Specific heat of dry air, c_p, is 0.24 Btu/lb°F

Density of dry air, ρ, is 0.075 lb/ft^3

Patient Room Loads		
Heat Source	**Design Conditions (Btu/h)**	**Typical Daytime Conditions (Btu/h)**
Patient	160	160
Medical Staff/Visitors (4x)	1000	250
Television	500	500
Medical Equipment	500	250
Overhead Lighting	900	500
Solar/Conduction (ext. wall)	2750	1400
Total	5810	3060

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Example 1 - Patient Room with Mixing Diffusers

Calculating the supply air flow rate to satisfy the design load:

$$Q_{design} = \frac{q_{design}}{\rho c_p \Delta T} = \frac{5810 \text{ Btu/h}}{(0.075 \text{ lb/ft}^3)(0.24 \text{ Btu/lb°F})(75 \text{ °F} - 55 \text{ °F})(60 \text{ m/h})} = 269 \text{ cfm}$$

Calculating the supply air flow to satisfy the required air-change rate:

$$Q_{Supply} = \frac{(14 \text{ ft})(18 \text{ ft})(9 \text{ ft})(6 \text{ ach})}{60 \text{ m/h}} = 227 \text{ cfm}$$

Calculating the reheat threshold:

$$q_{design} = Q_{Supply} \rho c_p \Delta T$$

$$q_{design} = (227 \text{ cfm})(0.075 \text{ lb/ft}^3)(0.24 \text{ Btu/lb°F})(75 \text{ °F} - 55 \text{ °F})(60 \text{ m/h})$$

$$q_{design} = 4903 \text{ Btu/h}$$

The required air flow rate to satisfy the maximum cooling load (269 cfm) is greater than the minimum total air-change requirement based on code (227 cfm). As such, the supply air flow will range between 227 cfm and 269 cfm during occupied periods and reheat will be necessary to prevent overcooling whenever room loads are below 4903 Btu/h.

The typical daytime cooling load in this patient room is well below the design load for the space. Calculating the typical supply air temperature:

$$T_{Supply} = T_{sp} - \left(\frac{q_{design}}{\rho c_p Q_{Supply}} \right)$$

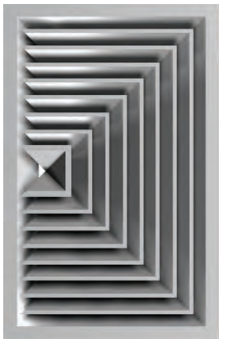
$$T_{Supplytyp} = 75 \text{ °F} - \left(\frac{3060 \text{ Btu/h}}{(0.075 \text{ lb/ft}^3)(0.24 \text{ Btu/lb°F})(227 \text{cfm})(60 \text{ m/h})} \right) \approx 63 \text{ °F}$$

Supply Air Outlet Sizing & Selection

Standard patient rooms are typically occupied by patients without highly contagious conditions or abnormal susceptibility to infection. As such, control of airborne contamination in these spaces is of somewhat less priority than it would be in isolation or operating rooms, allowing horizontal throw (mixing) diffusers to be a safe and practical alternative.

A number of different mixing diffusers can be used in standard patient rooms. As with any diffuser selection, the primary considerations for a patient room would be throw, noise, pressure drop and architectural appeal. Acceptable throw, noise and pressure drop levels can often be achieved with proper diffuser sizing; whereas architectural appeal is primarily based on designer preference and industry norms. Another consideration that applies to this example has to do with the asymmetry of the room. Since the floor area of this room is not square, a modular style diffuser with flexible discharge pattern options is most suitable. Louvered face directional diffusers offer this flexibility and are popular for patient rooms.

4 Way Louvered Diffuser



3 Way Louvered Diffuser

The placement of a louvered face directional diffuser will almost entirely dictate the style selected. A common location for this type of diffuser in a patient room would be directly above the patient bed near the head wall. This placement requires the selection of a 3 way discharge diffuser, as shown in the image above.

To reduce the chance of draft in this patient room, the 50 fpm terminal velocity of the discharge air should not enter the occupied zone (i.e. the discharge supply air should decelerate to 50 fpm or less before entering the occupied zone). Unlike in a waiting room, there will typically not be occupants in the 1 ft perimeter volume of the patient room.

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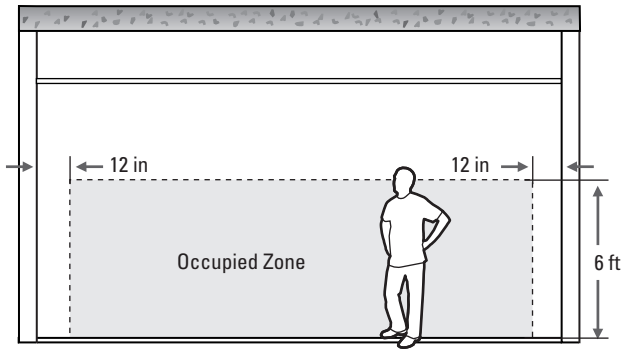
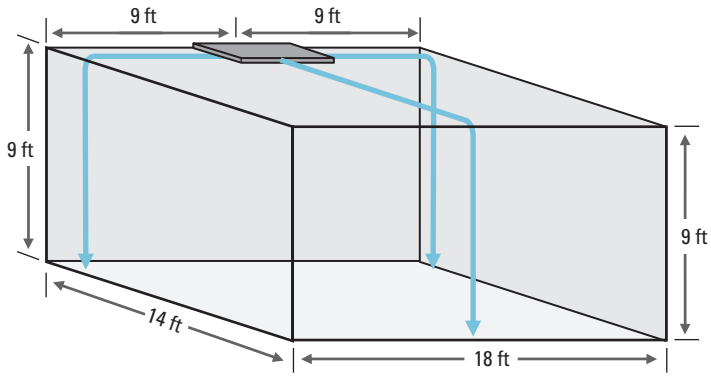
Example 1 - Patient Room with Mixing Diffusers

With this in mind, the occupied zone for this example will be the volume of the room from the finished floor to 6 ft above the floor, excluding the 1 ft perimeter volume along each wall. Based on this definition of occupied zone, the required throw to a 50 fpm terminal velocity can be approximated by adding the distance from the diffuser to the wall and the distance from the ceiling to the finished floor.

Given the dimensions of this patient room (18 ft x 14 ft x 9 ft) and the placement of the supply diffuser within the room (above the patient bed, against the head wall), the target throw characteristics from the diffuser would be 18 ft out either side of the diffuser and 23 ft in the direction of the foot wall.

Noise and total pressure drop are the other selection parameters that should be considered when choosing an air outlet. The target noise range for a patient room is between NC 25 and NC 35 (AHRI Standard 885-2008). Since cataloged diffuser NC values typically account for only the diffuser generated noise and not other duct related noise, the conservative approach is to select an air outlet near the low end of the target range. Please refer to the Acoustical Considerations in Health Care Spaces of the Price Engineer's Handbook for more detail on noise calculations.





With the diffuser requirements known, an appropriate air outlet can be selected from catalog data:

Performance Data - 3 Way Louvered Diffuser - Rectangular Neck							
Duct size, in.	**Neck Velocity, fpm**	**300**		**400**		**500**	
	Velocity Pressure, in. w.g.	**0.006**		**0.01**		**0.016**	
	Total Pressure, in. w.g.	**0.036**		**0.065**		**0.099**	
9 x 9 Duct Area 0.56 ft²	**Total cfm**	**169**		**225**		**282**	
	NC	**-**		**-**		**21**	
		A	**B**	**A**	**B**	**A**	**B**
	cfm/side	42	63	56	84	71	106
	throw, ft	6-9-16	7-11-18	8-12-18	10-15-21	10-15-21	12-16-23

From the performance data for the Price AMD with a 9 in. x 9 in. duct size (shown above), the 3A outlet will provide a 3 way throw combination close to the target for this example. From the catalog data, the 50 fpm terminal velocity throw is 18 ft out each side and 21 ft out the end of the diffuser (based on 225 cfm). When cooling loads demand air-change rates in excess of the minimum required by code (227 cfm), the throw is still close to the desired range. This diffuser selection also provides a noise level below the target of NC 25 as well as a relatively low pressure drop.

In rooms with multi-level ceilings, privacy curtains or other ceiling mounted obstructions, it should be verified that air flow will not be prematurely redirected toward the occupied zone.

Example 1 - Patient Room with Mixing Diffusers

Return Grille Sizing & Selection

Patient rooms are typically kept at a neutral or slight positive pressure relative to the corridor. For this example a neutral pressure differential relative to the corridor will be used, and the exhaust air flow will match that of the supply. The toilet room will have a negative pressure differential relative to the patient room, with an exhaust rate of 10 ach and no dedicated supply air outlet.

The first step in the process of selecting an appropriate return grille is to choose the location. Return grille location will dictate the type of grille selected and will affect room air flow patterns.

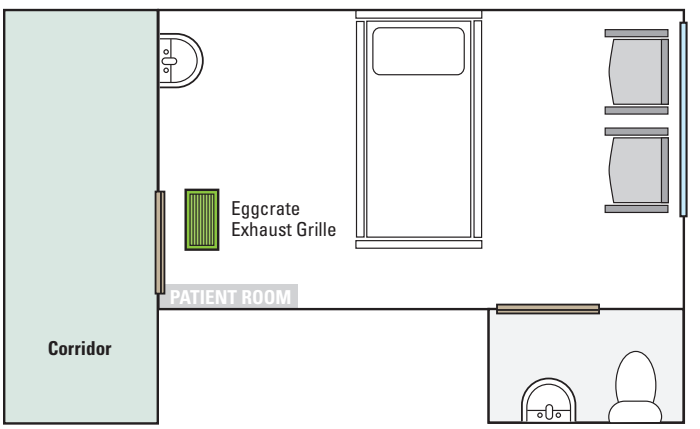
Since the supply diffuser selected for this example is a horizontal throw air outlet, the room air will be completely mixed. When this is the case, it is generally most practical to have a ceiling mounted return grille that will not be inadvertently blocked by furniture or other obstructions. Partially blocked returns will lead to higher noise and pressure drop levels than planned for during design. The return grille should also be located away from the supply diffuser to minimize the tendency for fresh supply air to be exhausted before conditioning the space. With the supply diffuser above the head of the patient bed, a suitable and common placement for the return grille is near the patient room door. A ceiling mounted, eggcrate grille is an economical selection that generally has attractive noise and pressure drop characteristics.

Since this patient room will be kept at neutral pressure relative to the corridor, the same volume of air will be exhausted from the total space as is supplied. Therefore, the total exhaust air flow from the patient room will range between 227 cfm and 269 cfm, depending on room loads. The toilet room will have a constant volume exhaust rate of 10 ach which must be accounted for in the total exhaust rate from the patient room itself.

$$Q_{Exhaust,Toilet} = \frac{(5\text{ ft})(7\text{ ft})(9\text{ ft})(10\text{ ach})}{60\,\text{m/h}} = 53\text{ cfm}$$

The exhaust rate inside the patient room (excluding the toilet room exhaust rate) will therefore range between 174 cfm and 216 cfm. It is important that the toilet room exhaust is accounted for in this calculation. If the toilet room exhaust rate was ignored, the net effect would be a negative pressure differential in the patient room relative to the corridor and an increase of infiltration air into the space.

When selecting a return grille from catalog data there will be an inverse relationship between the grille size and the noise and pressure drop generated by that grille. The return grille should be large enough that pressure drop and noise are at reasonable levels.



Suitable eggcrate grilles can be selected using catalog data:

Performance Data - 80 Core Eggcrate Exhaust Grille

Core Area, ft²	Nominal Size	Neck Velocity, fpm	600	700	800	1000
		Velocity Pressure, in. w.g.	.022	.031	.040	.062
		Negative Ps, in. w.g.	.047	.066	.085	.132
0.18	8 x 4	cfm	108	126	144	180
	6 x 6	NC	-	-	-	22
0.22	10 x 4	cfm	132	154	176	220
	7 x 6	NC	-	-	-	23
0.26	12 x 4	cfm	156	182	208	260
	8 x 6	NC	-	-	15	24
0.30	14 x 4	cfm	180	210	240	300
		NC	-	-	16	24

A number of options exist for the return grille selection. From the table above, any return grille with a core area of 0.22, 0.26, or 0.30 ft² would be appropriate for the exhaust outlet near the patient room door. All provide relatively low noise and pressure drop levels at the exhaust flow rates calculated in this example. At a constant flow rate of 53 cfm, the toilet room exhaust flow rate is easily satisfied with a 0.15 ft² core area eggcrate return grille. Positioning of the exhaust grille in the toilet room ceiling will have little impact on its effectiveness.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Example 2 - Energy Savings Displacement Ventilation vs. Overhead Mixing

Displacement Ventilation (DV) Design Procedure

DV in Patient Rooms

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 RP-949 (Chen et al., 1999) and the procedure outlined in the ASHRAE Design Guide, and incorporate the requirements of ASHRAE Standard 170-2008. Note that local codes may have different requirements.

For a complete explanation and derivation of the assumptions and equations used to develop this procedure, please refer to ASHRAE Standard 129-1997. The design procedure applies to typical North American health care spaces, such as patient and waiting rooms. 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 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 K/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:

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

Step 3: Determine the Minimum Flow Rate

Ventilation of health care spaces is typically regulated by code. ASHRAE (2008) and CSA (2010) define an air-change rate which is used to determine the minimum air flow rate:

$$Q_{oz} = \text{ach}\,\frac{V_{room}}{60\,\text{min/h}}$$

Step 4: Determine Supply Air Flow Rate

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:

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

Step 5: Determine Supply Air Temperature

The supply air temperature can be determined from the ASHRAE Design Guide equations and simplified to:

$$t_s = t_{sp} - \Delta t_{hf} - \frac{AQ_t}{2.456Q_s^2 + 1.08AQ_s}$$

Step 6: Determine Exhaust Air Temperature

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

$$t_e = t_s + \frac{q_t}{1.08(Q_s)}$$

Step 7: Selection of Diffusers

The goal is to maximize comfort in the space and minimize the quantity of diffusers. At a maximum, ASHRAE suggests 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.

Example 2 - Energy Savings
Displacement Ventilation vs. Overhead Mixing

Three patient rooms supplied by the same AHU all have exterior walls and windows facing different directions. The cooling load in each room varies significantly due to the large differences in solar heat gain. The geometry of the three rooms is identical, with a floor area of 250 ft² and 9 ft ceilings. The control temperature for each room is 75 °F.

Space Considerations

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

- The specific heat of dry air, cp = 0.24 Btu/lb°F
- Density of dry air, ρ = 0.075 lb/ft³
- The instantaneous cooling loads in each room are broken down as follows:

Design Considerations	Room #1	Room #2	Room #3
Occupants & Equipment q_{oe}	1300 Btu/h	1000 Btu/h	1100 Btu/h
Overhead Lighting q_l	400 Btu/h	700 Btu/h	700 Btu/h
Exterior Load q_{ex}	2700 Btu/h	700 Btu/h	1400 Btu/h
Total q_T	**4400 Btu/h**	**2400 Btu/h**	**3200 Btu/h**

Mixing Ventilation Calculations
Assumptions:

- Supply air temperature from AHU after dehumidification is 55 °F

Calculating the supply air flow rate to meet code (ASHRAE Standard 170-2008):

$$Q_{Minimum} = \frac{(250\,\text{ft}^2)(9\,\text{ft})(6\,\text{ach})}{60\,\text{min/h}} = 225\,\text{cfm}$$

Calculating the supply air temperature to each room:

$$t_{Supply} = t_{sp} - \left(\frac{q}{Q_{Supply}\,\rho\,c_p}\right)$$

$$t_{Supply\,(Room\,\#1)} = 75\,°\text{F} - \left(\frac{4400}{(225\,\text{cfm})(0.075\,\text{lb/ft}^3)(0.24\,\text{Btu/lb°F})(60\,\text{min/h})}\right) = 57\,°\text{F}$$

Calculating the instantaneous reheat requirements:

To warm the supply air from 55 °F to 57 °F:

$$q_{reheat} = Q_{Supply}\,\rho\,c_p\,\Delta t$$

$$q_{reheat\,(Room\,\#1)} = (225\,\text{cfm})(0.075\,\text{lb/ft}^3)(0.24\,\text{Btu/lb°F})(60\,\text{min/h})(57\,°\text{F} - 55\,°\text{F}) = 486\,\text{Btu/h}$$

Similarly,

$t_{supply(room\#2)}$	65 °F
$t_{supply(room\#3)}$	62 °F
$q_{reheat(room\#2)}$	2430 Btu/h
$q_{reheat(room\#3)}$	1701 Btu/h

Example 2 - Energy Savings
Displacement Ventilation vs. Overhead Mixing

Displacement Ventilation Calculations

Assumptions:

- Temperature difference from head to feet for sedentary occupants (Δt_{hf}) is 3.6 °F
- Supply air temperature from AHU after dehumidification and energy recovery is 65 °F

Calculating the displacement air flow requirement for each room:

Using the procedure described in example 16.1 of the Price Engineer's HVAC Handbook:

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

$$Q_{cooling\,(Room\,\#1)} = \frac{0.295(1300\ \text{Btu/h}) + 0.132(400\ \text{Btu/h}) + 0.185(2700\ \text{Btu/h})}{(0.075\ \text{lb/ft}^3)(0.24\ \text{Btu/lb°F})(3.6\,°\text{F})(60\ \text{min/h})} = 240\ \text{cfm}$$

Proposed changes to ASHRAE Standard 170-2008 would see the minimum total air change rate for single bed patient rooms applied to the occupied zone (0 - 6 ft) only when low-level displacement ventilation is used.

Calculating the supply air flow rate to meet code assuming six air changes in the occupied zone:

$$Q_{Minimum} = \frac{(250\,\text{ft}^2)(6\,\text{ft})(6\,\text{ach})}{60\ \text{min/h}} = 150\ \text{cfm}$$

Calculating the supply air temperature to each room:

$$t_{Supply} = t_{sp} - \Delta t_{hf} - \left(\frac{q_{total}\,A}{2.456\,Q_{Supply}^{\,2} + 1.08\,A\,Q_{Supply}}\right)$$

$$t_{Supply\,(Room\,\#1)} = 75\,°\text{F} - 3.6\,°\text{F} - \left(\frac{(4400\ \text{Btu/h})(250\ \text{ft}^2)}{2.456\,(240\ \text{cfm})^2 + 1.08\,(250\ \text{ft}^2)(240\ \text{cfm})}\right) = 66\,°\text{F}$$

Calculating the instantaneous reheat requirements for the room:

To warm the supply air from 65 °F to 66 °F:

$$q_{reheat\,(Room\,\#1)} = (240\,\text{cfm})(0.075\,\text{lb/ft}^3)(0.24\,\text{Btu/lb°F})(60\,\text{min/h})(66\,°\text{F} - 65\,°\text{F}) = 260\ \text{Btu/h}$$

Similarly,

$Q_{cooling(room\#2)}$	133 cfm
$Q_{cooling(room\#3)}$	174 cfm
$t_{supply(room\#2)}$	65 °F
$t_{supply(room\#3)}$	62 °F
$q_{reheat(room\#2)}$	0 Btu/h
$q_{reheat(room\#3)}$	0 Btu/h

Example 2 - Energy Savings
Displacement Ventilation vs. Overhead Mixing

Note the significant reduction in reheat relative to the mixing ventilation system. The above calculations for reheat with a displacement ventilation system are based on recent changes to the definition of minimum total air change rate in ASHRAE Standard 170-2008 for single bed patient rooms.

Recalculating the supply air temperature assuming no change to the definition of minimum total air change rate for single bed patient rooms (i.e. current ASHRAE Standard 170-2008 requirements and Q_{min} = 225 cfm):

$$t_{Supply} = t_{sp} - \Delta t_{hf} - \left(\frac{q_{total}\, A}{2.456\, Q_{Supply}{}^2 + 1.08\, A\, Q_{Supply}} \right)$$

$$t_{Supply\,(Room\,\#1)} = 75\,°\text{F} - 3.6\,°\text{F} - \left(\frac{(4400\ \text{Btu/h})(250\ \text{ft}^2)}{2.456\,(240\ \text{cfm})^2 + 1.08\,(250\ \text{ft}^2)(240\ \text{cfm})} \right) = 66\,°\text{F}$$

$$q_{reheat\,(Room\,\#1)} = (240\,\text{cfm})(0.075\,\text{lb/ft}^3)(0.24\,\text{Btu/lb°F})(60\,\text{min/h})(66\,°\text{F} - 65°\text{F}) = 260\,\text{Btu/h}$$

Similarly,

$t_{supply(room\#2)}$	68 °F
$t_{supply(room\#3)}$	67 °F
$q_{reheat(room\#2)}$	729 Btu/h
$q_{reheat(room\#3)}$	486 Btu/h

System Type	Room #1	Room #2	Room #3	Total Energy used for Reheat
Mixing (6 ach min.)	486 Btu/h	2430 Btu/h	1701 Btu/h	4617 Btu/h
Displacement (4 ach min.)	260 Btu/h	0	0	260 Btu/h
Displacement (6 ach min.)	260 Btu/h	729 Btu/h	486 Btu/h	1475 Btu/h

The narrow range of supply air temperatures combined with the superior energy recovery potential of displacement ventilation systems allows for the dramatic reheat savings demonstrated by this example.

Research Highlight 1 - Displacement Ventilation in Patient Rooms

Guity, Gulick & Marmion (2009) conducted research to answer a number of practical questions related to the use of displacement ventilation (DV) in single bed patient rooms. The research compared low sidewall DV at 4 air changes per hour (ach) with overhead ventilation (OHV) at 6 ach (current accepted minimum) in order to identify the impact of the following conditions and design parameters:

- Hot summer conditions (including solar gain)
- Supplemental cooling
- Cold winter conditions
- Supplemental heating
- Impact of air diffuser and grille locations
- Movement
- Coughing
- Metrics

Two main sets of metrics were chosen to compare performance; the first to evaluate thermal comfort and the other to evaluate ventilation effectiveness/contaminant removal.

Predicted percentage dissatisfied (PPD) was the metric used to compare performance from a thermal comfort perspective. PPD was empirically derived from human responses to test conditions where individuals reported their level of comfort. The metric was then built into the numerical model to describe a set of room conditions in terms of the percentage of occupants who are likely to be dissatisfied with those conditions.

Four indices were used to evaluate the results from a contaminant removal perspective; they fall into two main categories, as follows:

Ventilation Effectiveness (VE), or the ability of a ventilation system to remove internally generated pollutants from a building, zone, or space. Ventilation effectiveness (as defined by ASHRAE Standard 62.1-2007) is calculated for the caregiver, visitor, and whole room.

Air-change Effectiveness (ACE), or the ability of a ventilation system to distribute ventilation air to a building, zone, or space. ACE is defined by ASHRAE Standard 129-1997.

Results

The test results showed that DV at 4 ach performed equally or better than OHV at 6 ach for thermal comfort, ventilation effectiveness, and contaminant concentration. The figures below show the extent to which contaminants spread in the DV and OHV scenarios. The DV air flow pattern results in a more contained contaminant concentration at a high level.

Case	Description	Weather	Load	Air Changes/Hour	Supply Temp., °F [°C]
1	DV Limit w/Solar Load	Summer, 105 °F	Standard	4	Adjustable, 60 °F Min
2	Overhead	Winter, -10 °F	Reduced	6	Adjustable, 105 °F Max
3	DV w/Radiant Heating Panel	Winter, -10 °F	Reduced	4	67.1 °F
4	DV w/Baseboard Heater	Winter, -10 °F	Reduced	4	67.1 °F
5	DV w/Radiant Cooling Panel	Summer, 105 °F	Standard	4	67.1 °F
6	Overhead	Summer, 105 °F	Standard	6	Adjustable, 55 °F Min
7	DV w/Solar Load	Summer, 105 °F	Standard	4	60 °F
8	DV w/High Supply Temp.	Winter, -10 °F	Reduced	4	87 °F

Standard Load = Load from patient, caregiver, TV, and equipment.
Reduced Load = Load from patient and caregiver only.

Research Highlight 1 - Displacement Ventilation in Patient Rooms

Case	PPD Patient Area Visitor Area	VE - Caregiver 3.6 ft 5.6 ft	VE - Visitor 3.6 ft 5.6 ft	VE - Whole Room 3.6 ft 5.6 ft	ACE 3.6 ft 5.6 ft
1	8.61 8.91	1.22 1.05	1.45 1.38	1.31 1.18	0.95 0.91
2	7.06 21.10	0.99 0.87	1.12 1.09	1.04 1.00	0.84 0.82
3	5.89 9.84	1.67 1.48	1.39 1.37	1.71 1.63	1.60 1.18
4	7.05 8.37	3.24 2.58	2.53 2.37	3.07 2.67	0.85 0.71
5	5.90 6.64	0.97 0.80	1.19 1.19	1.01 0.90	0.92 0.91
6	5.75 14.99	1.03 0.94	1.09 1.08	1.06 1.03	0.75 0.74
7	12.53 11.57	1.65 1.35	1.91 1.60	1.77 1.53	1.08 1.28
8	8.17 8.74	0.80 0.76	0.94 0.94	0.81 0.79	0.79 0.80

Discussion

The research concluded that DV at 4 ach performed equally or better than OHV at 6 ach for thermal comfort, ventilation effectiveness and contaminant concentration, and that the performance of DV is dependent on several integrated elements of the air delivery and room exhaust air system design.

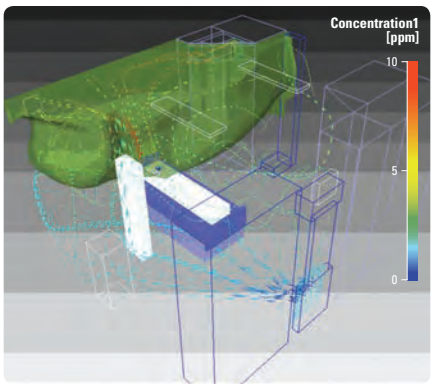
Impact of Hot Summer Conditions and Supplemental Cooling

Increases in direct solar gains had a significant impact on both thermal comfort and ventilation effectiveness. Thermal comfort proved to be the limiting factor as ventilation effectiveness requirements were always satisfied if thermal comfort was maintained.

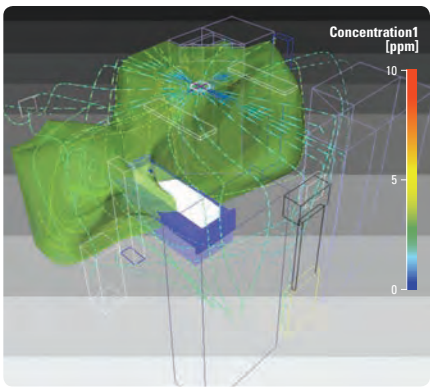
Lowering the DV supply air temperature to 60 °F (case 7) was sufficient to maintain thermal comfort, but too low to achieve air temperature stratification requirements (per ASHRAE Standard 55-2004). Supplying 67 °F air in combination with a supplemental cooling strategy (case 5) allowed both requirements of the standard to be satisfied simultaneously.

Room thermal gains and losses must be controlled if the performance of the DV system is to be maintained:

- Facades should be designed to minimize thermal gains and losses in order to prevent warm and cold surfaces—warm surfaces may cause fresh DV air to prematurely rise, while cold surfaces may impede stratification in areas of the room without warm objects (i.e. equipment or occupants) to drive air motion.
- Solar shading devices should be installed to minimize/eliminate direct solar gains. Floor surfaces warmed by direct solar gains can act as thermal hot spots, causing most of the displacement supply air to short circuit at the breathing level.
- Lighting and medical equipment loads should be minimized to keep cooling loads at acceptable levels and allow more DV air flow to be driven by occupants.



Displacement Ventilation at 4 ach



Overhead Ventilation at 6 ach

Impact of Cold Winter Conditions and Supplemental Heating Modes

Due to the same buoyancy principles that allow DV to work in cooling mode, supply air that is warmer than the room air will immediately rise as it enters the space (case 8). The warmer supply air cannot drop toward the floor to form stratified layers. The lower ventilation effectiveness and longer mean age of air in the occupied region indicate a reduction in performance of the DV system in terms of indoor air quality. Using either baseboard heating or ceiling mounted radiant panels while supplying air at 67 °F improved ventilation effectiveness and allowed thermal comfort to be maintained (case 3 and 4).

Impact of Movement on Displacement Ventilation

The study demonstrated that moving objects can carry contaminants in their wake. The movements can cause swings in the contaminant concentration in the breathing level of sitting and standing positions for 10 to 90 seconds. Since the variation lasted for 90 seconds or less, it would not likely change the exposure risk for occupants.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Research Highlight 1 - Displacement Ventilation in Patient Rooms

In most cases, the DV system with 4 ach provided better air quality than the OHV system with 6 ach. Around the bed region or close to the contaminant source, the DV system had a higher contaminant concentration than the OHV system. However, in most parts of the room, the concentration was lower with the DV system.

Impact of Coughing on Displacement Ventilation

Around the bed, where a caregiver is most likely to stand, the DV case showed a higher contaminant concentration for a short period of time after a coughing incident; while at the end of the simulation period (5 minutes), the concentration was lower than observed with the OHV system. The reason for this result is because displacement ventilation 'confines' or 'contains' the concentration around the patient, while OHV, given its mixing nature and higher air-change rate, is able to better dilute the contaminant levels around the bed. Consistent with the above discussion, DV delivers a better concentration in the visitor area, as it is further away from the patient. When looking at the room as a whole, the results achieved by the two ventilation systems in terms of response to coughing were comparable.

Impact of Air Diffuser and Grille Locations

The placement of the DV supply air diffuser is not critical, but should be coordinated with the room design. The diffuser should be located at a low level in a location that will not be blocked with solid furniture such as a storage cabinet. DV supply air is not discharged in a particular direction or pattern and will navigate around obstructions, unless they are completely blocking the outlet.

Related research by Yin et al., (2009) demonstrated that the toilet room transfer grille should be located at a high level. Performance of the system can be negatively affected if DV supply air is allowed to short circuit through an undercut or low-level transfer grille to the toilet room. This problem is the result of a negative pressure differential in the toilet room relative to the patient room itself. A low-level transfer point to the toilet room will pull a disproportionate amount of air into the toilet room when the toilet room door is closed. The negative pressure differential is effectively lost with the toilet room door open and will not have a significant impact on system performance.

Hospital Operating Rooms

Operating rooms are among the most unique spaces in any hospital. The patients who occupy operating rooms typically undergo invasive procedures that will expose internal tissue to room air. It is not uncommon for these patients to already have weakened immune defenses, and the physical interference with their organs and systems (skin, blood flow, body temperature, etc.) can make them even more susceptible to infection. The air distribution system in an operating room (OR) can either reduce or promote the frequency of surgical site infections, depending on the design.

Mixing type air distribution systems are not suitable for hospital operating rooms. In addition to uniform temperature distribution from floor to ceiling, a well-designed mixing system will produce an even distribution of contaminants in the air, increasing the risk of infection during surgical procedures.

In an OR, control of airborne contaminants and comfort are both major considerations. The three primary sources of airborne particulates are ventilation, infiltration and occupants. The particulate level of ventilation air is controlled using high efficiency filters, while space contamination through infiltration is minimized by maintaining a positive pressure differential between the OR and adjacent areas of the hospital. Consequently, these means of space contamination represent less of a concern than the presence of the patient and surgical team.

The largest source of airborne contamination in most modern operating rooms (and most challenging to control) is the surgical team and patient. Scrubbing and gowning tactics used by surgical teams help to minimize the amount of airborne particles released during a procedure, but they do not eliminate them completely. Also, with operating rooms maintaining a positive pressure differential with respect to adjacent areas, there will inherently be recirculating air (and contaminants) inside the room at all times. The goal is to control and isolate these contaminants in such a way as to minimize their time in the surgical zone. The OR air distribution system is the means by which this source of contamination is controlled, and it involves three main components. The first is dilution. Diluting airborne contaminants to an acceptable level has led to air supply exchange rates much in excess of those typically required for thermal control. These increased air exchange rates can lead to thermal discomfort due to drafts and the air distribution system must therefore be capable of introducing a large volume of supply air without compromising comfort in the occupied zone. The second and third requirements of the air distribution system are to remove particulates from the surgical zone and to reduce or eliminate the tendency for those particulates to reenter the clean air stream over the patient. An OR environment should be comfortable for occupants without contributing to the risk for surgical wound infection. Achieving this goal from an air distribution perspective involves control of a number of factors.

Design Considerations

The face velocity of non-aspirating diffusers over the surgical table should not exceed 35 fpm (Memarzadeh & Manning, 2002) as to avoid high velocity air near the patient. High velocity air in the surgical zone can have a number of negative consequences:

1. Elevated rate of skin particle erosion off surgical team members (Cook & Int-Hout, 2009).
2. Overcooling of the patient, resulting in hypothermic complications (Kurz, Sessler & Lenhardt, 1996).
3. Uncomfortable drafts.
4. Entrainment of contaminated air.

Operating rooms typically require a positive pressure differential relative to corridors and other adjacent spaces. This is achieved by supplying more air to the room than is exhausted. The actual air flow offset between supply and exhaust is dependent on the target differential pressure and leakage from the room envelope and cannot be determined in advance of room commissioning. However, a 20% offset is typically required to maintain a reasonable pressure differential.

All operating rooms should have individual temperature control and a device to monitor differential pressure between the room and adjacent spaces. Each category of OR will have different equipment needs (thermal loads), as well as different condition requirements with respect to air patterns and temperature.

Cooling Loads

In most cases, the total air-change rate required by code will be sufficient to meet OR cooling loads at the supply air temperature range required by the surgical team. However, for some procedures, including cardiac or transplant surgery, it is necessary improved thermal control, increased cooling capacity, or both.

Hospital Operating Rooms

Occupancy loads present an issue similar to lighting loads. Operating rooms will often have periods of high population density at certain times during a procedure, and the HVAC system should be capable of handling these peak loads with the ability to adjust for reduced occupancy levels.

Equipment loads often account for the majority of heat generation in the OR. Cooling load requirements should typically be based on the equipment manufacturers' Btu ratings, however, caution should be taken when using these ratings in calculations. For example, a blood pressure meter will have relatively constant power consumption whereas the peak draw of an X-ray machine will only occur during X-ray exposure lasting a fraction of a second.

Types Of Operating Room Air Distribution Systems

There are two ventilation systems commonly accepted for use in hospital operating rooms today: laminar diffuser systems and air curtain systems. Both systems have been widely used in all types of operating rooms and are described in more detail here.

Laminar Diffuser Systems

Laminar diffuser systems were developed to control airborne contamination in operating rooms by providing a downward wash of clean supply air at relatively low velocity.

The most effective laminar diffuser systems would see the entire ceiling filled with laminar flow diffusers and all air exhausting through floor grilles. By covering the entire ceiling with diffusers, room conditions would be close to isothermal, reducing the chance of supply air acceleration due to temperature gradients. The practice of covering the entire ceiling in diffusers is not only impractical for an operating room, but the supply air volumes would be well in excess of code requirements.

Reducing the size of the laminar diffuser array opens space for other ceiling-mounted equipment (lights, booms, gas columns, etc.). At minimum, laminar flow diffusers should cover 70% of the ceiling area directly above the area defined by the surgical table and a 12 in. offset (ASHRAE Standard 170-2008). This minimum requirement will usually not satisfy minimum room air-change requirements and additional supply diffusers beyond the primary diffuser array area are most often necessary. This minimum requirement is not common to all jurisdictions and should be verified prior to design.

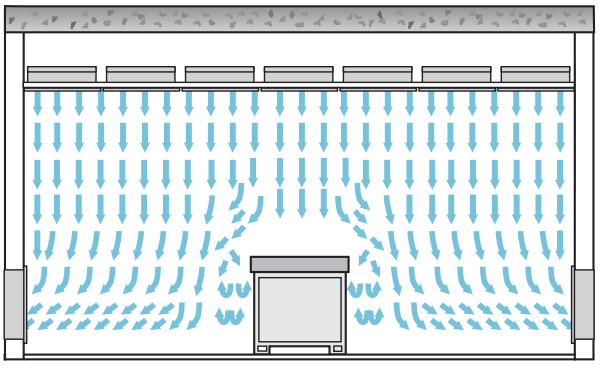


Figure 12: Laminar flow system with full ceiling coverage

Figure 13: Laminar flow diffuser

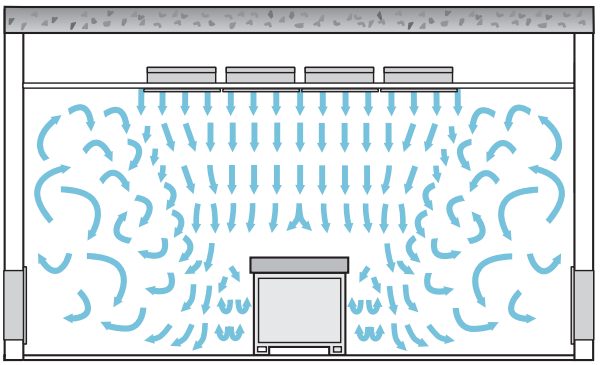


Figure 14: Laminar flow air pattern

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Hospital Operating Rooms

Although laminar flow diffusers are considered by many to be non-aspirating air outlets, some entrainment of room air still occurs within the first 3 to 6 in. below the diffuser face. The holes in the perforated face act as individual air jets, causing air to accelerate as it passes through the smaller free area. This is why many codes refer to the "average velocity" below the face of the diffuser. The average velocity near the face of a laminar flow diffuser is based on the air flow rate per nominal face area, not actual air velocity. Once through the perforated face, the air jets will expand, coalesce and decelerate. By the time the air mass is more than 6 in. from the diffuser face, the air velocity profile will be more consistent and the actual velocity of the supply air will be much closer to the average velocity. When the actual air velocity is in the 25 to 35 cfm/ft² range there is minimal entrainment of room air.

The supply air temperature in most operating rooms is 5 °F to 10 °F below the room set-point. The cooler supply will have a greater density than the surrounding room air, and will therefore have the tendency to accelerate toward the surgical table. The warmer room air will also transfer heat to the boundary layer of the laminar air flow, causing it to become more buoyant. The result of this thermal interaction causes the air in the center of the supply air column to accelerate toward the surgical zone at a higher rate than the air around its perimeter (i.e. boundary layer). The relatively high velocity will pull the column boundary inward, creating a tapered column of air. This tapering will occur under cooling conditions, regardless of the number of diffusers in the array. Depending on the magnitude of this effect, the surgical team may not get washed by the clean supply air, leading to discomfort and contamination issues. Extending the laminar diffuser array beyond the footprint of the surgical table will typically address this issue.

Air filtration in a laminar diffuser system can be accomplished in one of two ways. Typical practice is to use HEPA filters in either an upstream filter bank or directly in the laminar flow diffusers themselves. When multiple operating spaces are supplied through a common system, it is often most economical to use the HEPA filter bank approach. Additionally, with the HEPA filters located in a bank upstream of the operating room, filter service and maintenance can be performed without entering the sterile environment of the operating room.

Supply diffusers with integral room-side replaceable HEPA filters offer ease of accessibility through the diffuser face for filter service and replacement, but they must be accessed from inside the sterile operation room. With this arrangement it may be necessary for the operating room to be re-sterilized each time the filters are accessed. In some applications, a shut-off damper is installed in the branch duct feeding the laminar diffuser. This allows for the air supply to one diffuser to be shut off during filter change-out while maintaining the supply of air through the remaining laminar devices.

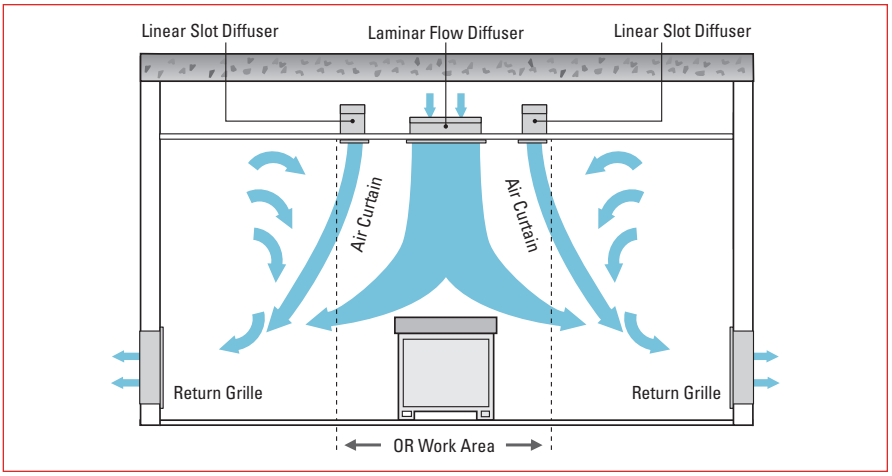


Figure 15: Air curtain systems air pattern

Figure 16: Air curtain systems

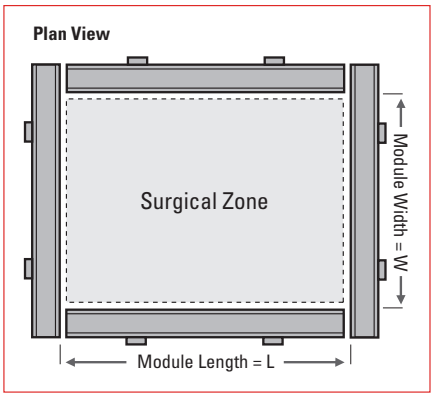


Figure 17: Air curtain systems

Air Curtain Systems

An air curtain system combines a laminar diffuser array above the surgical table with a four-sided linear slot diffuser system. The function of the linear slot diffusers is to create a barrier of air between recirculating contaminants in the perimeter of the room and the surgical zone. An air curtain system typically use less ceiling space (particularly above the surgical zone) to introduce the same volume of air into the operating room.

The air curtain is created using specially designed linear slot diffusers on each of the four sides around the surgical table. The linear slot diffusers are installed in the ceiling with a minimum 3 ft offset between the surgical table and the inside of the linear slot diffusers. This gives the surgical staff space to move around the table without entering the protective, air curtain field. The linear slot diffusers typically feature fixed or adjustable deflection blades to discharge supply air at an angle of between 5° and 15° from vertical, away from the table.

The linear slot diffusers which form the air curtain typically have two slots, creating a thick, uniform curtain of air around the surgical area. The air curtain presents a barrier of high velocity clean air between the laminar flow diffusers and any particulates which may be recirculating in the room, particularly near the ceiling level where laminar diffuser are most likely to entrain room air. The air curtain entrains room air and any particulates in its outer boundary layer, carrying them down and away from the surgical area toward the low-level exhaust grilles. The air curtain should be sized to deliver between 25 and 45 cfm/ft.

Hospital Operating Rooms

At flow rates below 25 cfm/ft, the air curtain may not properly isolate the interior laminar flow diffusers, increasing the possibility of surgical zone contamination due to entrainment of the recirculating room air. In contrast, air flow exceeding 45 cfm/ft will increase the potential for re-entrainment of particulates and debris that may have settled on the floor.

The purpose of an air curtain system goes beyond simply creating a barrier between the surgical zone and the perimeter area of the operating room. The air curtain also serves to control air velocity at the operating table level. This system characteristic is of utmost importance given the existing research that suggests high velocity air at the surgical table can increase the risk of surgical site infection.

As the relatively lower velocity air exits the laminar diffusers above the patient, the higher velocity air from the linear slot diffusers will induce (pull) the laminar flow outward. The laminar flow will expand to fill the zone enclosed by the air curtain, mitigating the tendency for the cold air mass to accelerate as described earlier. The net result is the ability to maintain air velocities at the operating table close to, or even slower than, those at the diffuser face.

Air curtain systems allow slightly less flexibility with regard to filter locations compared with all laminar systems. The linear slot diffuser plenums, which make up the four-sided air curtain, are too narrow to effectively incorporate an integral high efficiency filter without resulting in significant and undesirable pressure drop. The laminar flow diffusers above the surgical table may still include integral filters, but all air supplied to the linear slot diffusers must be filtered upstream of the system.

No official guideline exists to define the division of total supply air between the linear slot diffusers and laminar flow diffusers of an air curtain system. However a common method is to supply 60% to 75% of the total supply air through the linear slot diffusers with the remaining volume supplied through the laminar diffusers. Since the surgical zone served by the laminar flow diffusers is typically less than 25% of the total operating room area, the net result is a higher air-change rate within the air curtain than the room average. The result is faster dilution and removal of particulates at the surgical table.

Selection of the air curtain system must take into account the standard air distribution design parameters such as sound, pressure drop and comfort, plus the additional issue of particulate control.

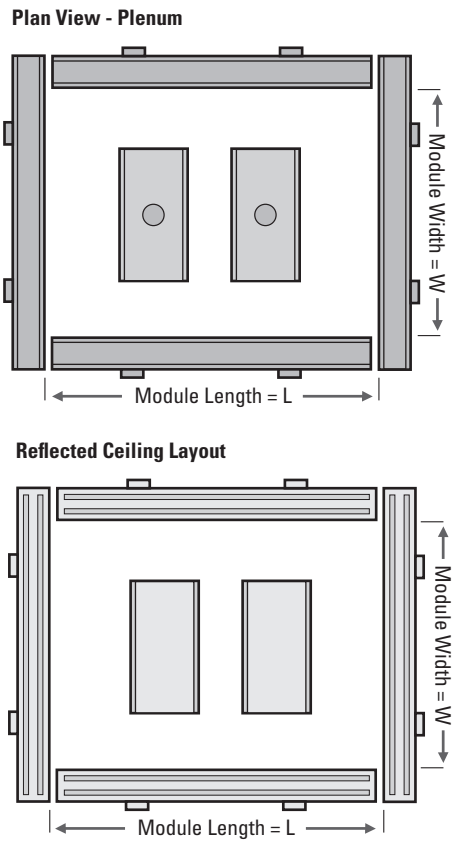


Figure 18: Modular air curtain room-side and plan view

The module size of the air curtain is governed by four factors:

Size of the area to be contained; recommended practice is to take the size of the surgical table plus a 3 ft perimeter work area for the surgical team.

The total air volume required by the room to achieve the desired number of air-changes. It may be necessary in the case of very large operating rooms to use longer linear slot diffusers (i.e. larger module size) than is dictated by the surgical work area in order to meet the air-change requirements of the operating room.

Allowances and concessions may have to be made for other ceiling-mounted equipment such as I.V. tracks, surgical lights, gas columns, general lighting, etc.

The shape of the operating room may restrict the space in one direction or the other. It is recommended to keep at least 3 ft between the linear slot diffusers and the OR walls.

Air Curtain Types

There are two types of air curtain systems commonly used today: the modular and the continuous plenum. The selection of one over the other is typically associated with the available space for facilitating duct connections.

Modular Plenum Air Curtain

The modular plenum air curtain system has four independent plenums. Since the plenums are independent, each one must be ducted separately. The quantity and/or size of the inlets are dependent on the length of the linear slot. For slot lengths of up to 120 in., a single inlet located close to the center of the plenum is sufficient to provide equal air flow along the entire length of the slot. For lengths greater than 120 in., multiple inlets are usually required. Multiple inlets should be equally spaced along the entire diffuser length to support air flow equalization along the full slot length. Inlets are typically rectangular and sized for an inlet velocity of approximately 500 fpm.

Hospital Operating Rooms

The modular plenum style offers the following advantages:

The corners of the modular plenums overlap, resulting in an almost continuous air curtain with only small gaps at the corners. This minimizes the potential for particulates to migrate inside the air curtain and into the laminar air flowing over the patient.

Fewer field connections. Since each diffuser section is independent there is no need to connect plenums with corner elbows.

With inlet connections on all sides of the air system, it is easier to achieve a uniform air distribution along the entire length of the linear slot diffusers, resulting in a more effective barrier.

The perceived disadvantage to the modular system is the quantity of inlets. With each diffuser section separately ducted, ceiling space can become congested.

Continuous Plenum Air Curtain

The continuous plenum air curtain system has one common ring plenum with each side connected by flanged elbows above the ceiling. Since all plenum sections are connected, it is possible to use fewer inlets, however, it is still beneficial to have one inlet on each side of the air curtain to help equalize air flow. The minimum number of inlets recommended for this system is two, with the inlets located as far apart as possible to support effective equalization around the entire air curtain. As with the modular systems, these inlets should be sized for an inlet velocity of approximately 500 fpm.

Advantages of the continuous systems include:

- Fewer inlets, resulting in less ductwork and connections.

Perceived disadvantages include:

- With fewer inlets and the same air volume, ductwork to each inlet will be larger.
- Continuous plenum air curtains have four elbows which require field connection and sealing.
- The active slots do not overlap in the corners, resulting in larger gaps in the air barrier than with modular systems.

Return Grilles

Ideally, there should be four low-level return grilles, centered in each wall, or mounted in each corner of the room (Memarzadeh & Manning, 2002). Since space is usually at a premium in operating rooms, there is not always sufficient room to include four returns. In this case, the next best option is to use two return grilles located as far apart as possible.

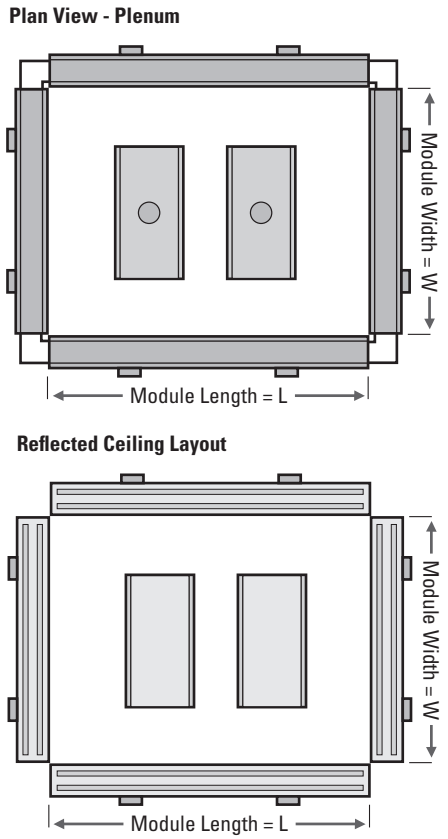


Figure 19: Continuous air curtain room-side and plan view

Return grilles are typically located at a low level, with the bottom of the grille installed approximately 8 in. above the floor (ASHRAE Standard 170-2008). This location is beneficial for ease of cleaning as well as for removal of heavier-than-air gases, a category which includes most medical gases (i.e. CO_2, N_2O, O_2).

Return grilles are most often constructed of stainless steel, typically for strength and durability properties as opposed to the need for corrosion resistance. Since these grilles are typically located at a low level, the potential is high for impact and damage by cleaning staff or mobile equipment.

Return grilles are sized to return less air than is supplied in order to maintain a positive pressure in the operating room. Once the total return air flow has been determined, the air flow per grille should be calculated with grilles sized based on a core velocity of approximately 500 fpm. Selecting returns at 500 fpm will provide desirable noise and pressure drop levels.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Example 3 - Operating Room Air Distribution

The operating room (OR) used in the following example includes an anesthesia machine, two LCD monitors, two surgical lights and overhead lighting. The OR was designed for a maximum of seven occupants (patient, surgical team and support staff) and has one 2 ft x 6.5 ft surgical table in the center of the room. Control temperature for the space is 68 °F and the room dimensions are 22 ft x 22 ft with a 12 ft ceiling height.

Operating Room Design Criteria (ASHRAE Standard 170-2008)

- 20 ach minimum (total, based on supply volume)
- Positive room pressure relative to corridor
- Target HVAC system noise range 25 to 35 NC (AHRI Standard 885-2008)

Space Considerations

- Some of the assumptions made for this space are as follows:
- Supply air temperature is 55 °F
- Specific heat of dry air (c_p) is 0.24 Btu/lb°F
- Density of dry air, ρ is 0.075 lb/ft^3

Operating Room Loads	
Heat Source	**Design Conditions (Btu/h)**
Patient	160
Surgical Team (4)	1200
Support Staff (2)	600
Anesthesia equipment	900
LCD monitors	850
Surgical lights	1500
Overhead lighting	2400
Total	7610

Air Flow Rate Calculations:

Calculating the supply air flow rate to satisfy the design load:

$$Q_{cooling} = \frac{q_{design}}{\rho c_p \Delta T} = \frac{7610 \text{ Btu/h}}{(0.075 \text{ lb/ft}^3)(0.24 \text{ Btu/lb°F})(68 \text{ °F} - 55 \text{ °F})(60 \text{ m/h})} = 542 \text{ cfm}$$

For rooms with a positive pressure differential relative to adjacent zones, the minimum air-change rate is based on the supply air volume. Calculating the supply air flow to satisfy the required air-change rate:

$$Q_{Supply} = \frac{(22 \text{ ft})(22 \text{ ft})(12 \text{ ft})(20 \text{ ach})}{60 \text{ m/h}} = 1936 \text{ cfm}$$

Less air must be exhausted from the operating room than is supplied in order to maintain the required positive differential pressure. The actual air flow offset will need to be determined at the time of commissioning, but a 20% offset will be used for this example. Calculating the exhaust air flow rate to provide the desired positive differential pressurization:

$$Q_{exhaust} = (1936 \text{ cfm})(1 - 0.2) = 1549 \text{ cfm}$$

The minimum total supply air flow rate for this OR is 20 ach or 1936 cfm (ASHRAE Standard 170-2008). The air flow rate required to satisfy the design load conditions is only 542 cfm, well below the ASHRAE specified minimum. As a result, there will be constant volume supply and exhaust air flow to and from this OR and reheat will be required at all times to prevent overcooling.

Calculating the minimum supply air temperature:

$$T_{Supply} = T_{SP} - \left(\frac{q_{design}}{\rho c_p Q_{Supply}} \right)$$

$$T_{Supply} = 68 \text{ °F} - \left(\frac{7610 \text{ Btu/h}}{(0.075 \text{ lb/ft}^3)(0.24 \text{ Btu/lb°F})(1936 \text{ cfm})(60 \text{ m/h})} \right) = 64 \text{ °F}$$

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Example 3 - Operating Room Air Distribution

The difference between the supply air temperature and the room temperature (ΔT) must be monitored to ensure there is no significant acceleration of supply air directly over the patient and surgical team. As ΔT values increase, so do the chances of acceleration of laminar air flow over the patient. Air-change rates can be reduced when the room is unoccupied, however, pressure relationships should be maintained.

Choosing a System

The two widely accepted air distribution systems for operating rooms are the laminar diffuser system and the air curtain system. No scientific, comparative performance test results exist for which each of these systems was included. As such, there are conflicting opinions on which system is more effective in controlling airborne contamination.

Standards and guidelines that govern the design of operating room air distribution systems vary from one to the next in terms of requirements (i.e. air-change rates, diffuser quantity/type/location, etc.). ASHRAE Standard 170 is currently the primary document used for OR air distribution design in the US and many countries abroad. For this reason, it will be used to dictate the design of the OR air distribution systems detailed in this example.

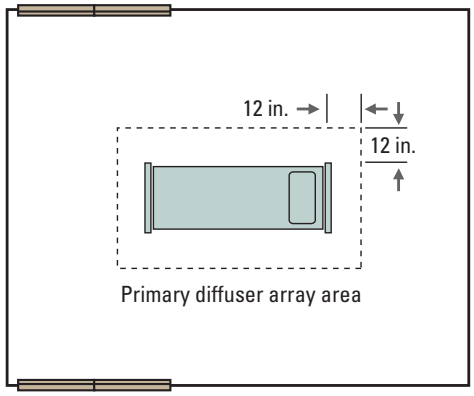
The key details from ASHRAE Standard 170-2008 that relate to the design of the OR air distribution system are summarized below:

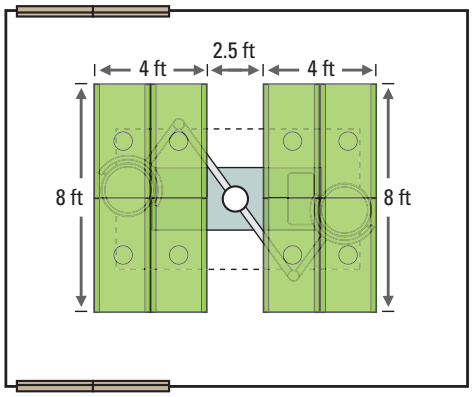
- Room should be maintained at a positive pressure relative to all adjoining spaces.
- The primary diffuser array above the operating table should satisfy the following conditions:
  - Diffusers should be 'non-aspirating' with unidirectional, downward air flow at an average velocity of 25 to 35 cfm/ft² at the diffuser face (i.e. laminar flow diffusers).
  - Cover 70% of the ceiling area directly above the area defined by the surgical table plus a 12 in. offset on all sides.
- Diffusers outside the primary diffuser array area should discharge air vertically downward.
- OR should have at least two low-level (8 in. above floor) sidewall return or exhaust grilles spaced as far apart as possible.

This example will detail how each system should be designed based on this criteria.

Laminar Diffuser Array

Laminar diffuser array systems use laminar flow diffusers concentrated above the surgical table to deliver 100% of the supply air to the operating room. ASHRAE defines the primary diffuser array area as the ceiling area directly above the operating table plus a 12 in. offset on all sides of the table. Up to 30% of this primary diffuser array area may be used for non-diffuser equipment (i.e. lights, booms, etc.).





30% of primary diffuser array area used for non-diffuser application

The average discharge velocity from the laminar flow diffusers should be between 25 and 35 cfm/ft² (Memarzadeh & Manning, 2002). For this example, a target of 30 cfm/ft² will be used.

Calculating the required laminar flow diffuser face area:

$$\frac{1936 \text{ cfm}}{30 \text{ cfm/ft}^2} \approx 64 \text{ ft}^2$$

There are a wide range of laminar flow diffuser sizes available, as well as custom sizes. A combination of diffusers must be arrayed above the surgical table, ensuring no more than 30% of the primary diffuser array area is used for non-diffuser equipment.

There is no 'correct' way to arrange the laminar diffusers and the placement is often partially dictated by the location of other ceiling mounted equipment. To reduce the acceleration potential of supply air over the surgical zone, it is preferable to have gaps between diffusers when practical. Spacing diffusers is often a challenge with laminar diffuser systems due to the large total diffuser face area. The diffuser face area requirement for this example can be satisfied with eight 24 in. x 48 in. laminar flow diffusers arranged around the surgical light boom, as in the image below.

The inlet neck selection has negligible impact on the performance of these diffusers. At 30 cfm/ft², each laminar flow diffuser will supply approximately 242 cfm and either a 10 in. or 12 in. inlet neck diameter will result in suitable pressure drop levels and noise levels below our target range of 25 to 35 NC.

Example 3 - Operating Room Air Distribution

The following performance table is for a typical laminar flow diffuser (Price LFD):

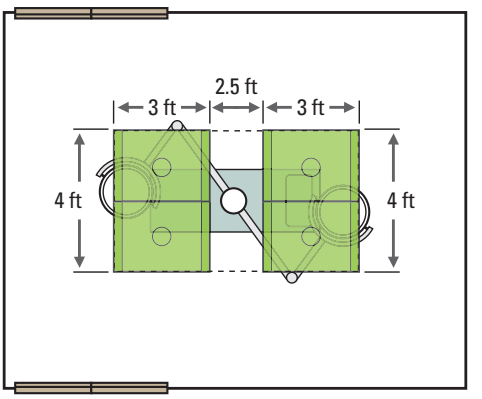
Performance Data - Laminar Flow Diffuser			
24 in. x 48 in. Panel - 10 in. Round Inlet			
cfm	**P_s , in. w.g.**	**P_t , in. w.g.**	**NC**
160	0.01	0.02	-
240	0.02	0.04	-
320	0.04	0.06	-
400	0.07	0.10	17
480	0.09	0.14	23
Performance Data - Laminar Flow Diffuser			
24 in. x 48 in. Panel - 12 in. Round Inlet			
cfm	**P_s , in. w.g.**	**P_t , in. w.g.**	**NC**
160	-	0.01	-
240	0.02	0.03	-
320	0.04	0.05	-
400	0.06	0.07	-
480	0.08	0.11	16

Air Curtain System

An air curtain system consists of a primary laminar diffuser array above the operating table and a four-sided linear slot diffuser system surrounding the surgical zone. The linear slot diffusers must be specifically engineered for OR applications, as conventional slot diffusers are not manufactured to accommodate thorough sterilization, nor is the air pattern they produce appropriate for protection of the surgical zone.

The laminar flow diffusers are located directly above the surgical table, while the linear slot diffusers surround the operating table and a perimeter work area for the surgical team. It is also recommended that the linear slot diffusers are installed at least 1 ft from the laminar flow diffusers to reduce the potential for entrainment of laminar air flow into the higher velocity discharge of the linear slots. The target supply rate for a linear slot diffuser system should be anywhere between 25 and 45 cfm/ft. Flow rates below this range will not create an effective curtain, while higher flow rates may stir particles that are settled on the floor. Typically 60 to 75% of the total supply air volume will be through the linear slot diffusers with the balance supplied through the laminar diffuser array.

The first step in the design of the air curtain system is to establish the size and layout of the primary laminar diffuser array to ensure ASHRAE requirements are satisfied. The recommended practice is to keep the size of the primary diffuser array to the minimum ASHRAE requirement and deliver the balance of the supply air through the linear slot diffusers. Following this process will keep more of the prime ceiling area above the surgical zone available for non-diffuser equipment while also minimizing the potential for acceleration of supply air over the patient and surgical team.



30% of primary diffuser array area used for non-diffuser application

Determine the Size and Quantity of Laminar Flow Diffusers

Based on the assumption that the operating table is 2 ft x 6.5 ft, the required diffuser face area for the primary diffuser array should be approximately 24 ft².

$$\text{Operating table size} = 2 \text{ ft} \times 6.5 \text{ ft}$$

$$\text{Primary diffuser array area} = [(2 \text{ ft} + 2 \text{ ft}) \times (6.5 \text{ ft} + 2 \text{ ft})] \times (1 - 0.3) = 23.8 \text{ ft}^2 \approx 24 \text{ ft}^2$$

At the recommended 30 cfm/ft² average discharge velocity through the diffuser face area, the total supply air volume through the primary diffuser array will be 720 cfm. The selection and arrangement of laminar flow diffusers for the primary diffuser array will depend in part on the location of other ceiling mounted equipment. Since the surgical light boom is mounted in the center of the room above the operating table, the primary diffuser array must be positioned around this obstruction.

Four 24 in. x 36 in. laminar flow diffusers positioned as shown below will provide sufficient coverage of the primary diffuser array area without interfering with the surgical light boom.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Example 3 - Operating Room Air Distribution

Determine the Size of the Linear Slot Air Curtain

The linear slot diffuser layout is defined in different ways by different manufacturers. The inside dimensions of the linear slot diffuser layout are often used to describe the system. As a starting point, the linear slot diffusers will be sized to enclose the operating table plus a 3 ft perimeter. This 3 ft perimeter will allow the surgical team to move around the patient and work in different positions without passing through or standing directly beneath the air curtain. It should also be verified that the linear slot diffusers are at least 12 in. from any laminar flow diffuser as noted earlier in this example.

Based on the 2 ft x 6.5 ft surgical table, the inside dimensions of the linear slot diffuser system will be 8 ft x 12.5 ft. The ceiling plan should be referenced to ensure the linear slot diffusers do not interfere with any other ceiling mounted equipment. It should also be noted that the 3 ft perimeter work space around the operating table is a general guideline. Depending on the type of operating room, the associated number and position of surgeons, and space restrictions, the perimeter may be more or less than this 3 ft value. The linear slot diffuser system should be adjusted accordingly.

There are three options to consider if interference exists between the linear slot diffusers and other equipment, in order of preference:

Reposition the other ceiling mounted equipment – typically not done

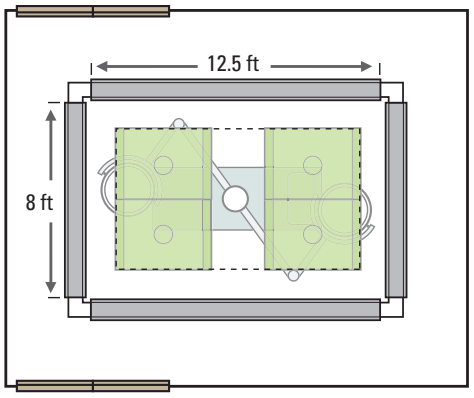
Increase the size of the air curtain until interference with other equipment no longer exists

Split the air curtain around the other ceiling mounted equipment

The third option is the least desirable as it will create a gap in the air curtain where airborne contaminants could enter the surgical zone. For this example it is assumed that there is no interference between the linear slot diffuser system and other ceiling mounted equipment. The inside dimensions of the linear slot diffuser system will therefore remain 8 ft x 12.5 ft.

The total linear slot length can be approximated by adding these two dimensions and doubling that value. The actual linear slot length will depend on the width of the linear slot face (will depend on manufacturer) and the type of system used (i.e. modular or continuous). The approximate total linear slot length will be:

$$(2)(8\text{ ft} + 12.5\text{ ft}) = 41\text{ ft}$$



Performance Data - HORD - Linear Slot Diffuser

	cfm/ft	25	30	35	40	45
7 ft - 6 ft	NC	-	-	-	19	24
	Throw*	0-1-4	1-2-6	1-2-6	1-3-7	2-4-7
	P_s , in. w.g.	0.048	0.070	0.054	0.070	0.088
8 ft - 6 ft	NC	-	-	16	21	25
	Throw*	1-1-5	1-2-6	1-2-7	1-3-7	2-4-8
	P_s , in. w.g.	0.051	0.074	0.061	0.080	0.101
10 ft	NC	-	-	18	23	28
	Throw*	1-1-6	1-2-7	1-3-7	2-4-8	2-5-8
	P_s , in. w.g.	0.049	0.071	0.065	0.084	0.107
12 ft	NC	-	15	21	26	31
	Throw*	1-2-7	1-2-8	1-3-8	2-4-9	2-5-9
	P_s , in. w.g.	0.066	0.095	0.066	0.086	0.109
14 ft	NC	-	18	23	28	33
	Throw*	1-2-7	1-3-8	2-4-9	2-5-10	3-6-10
	P_s , in. w.g.	0.069	0.099	0.076	0.099	0.126

*Throw values are given in feet to terminal velocities of 150-100-50 fpm based on 10 °F cooling

It should be verified that the air flow through the linear slot diffuser system falls within the 25 to 45 cfm/ft target range.

$$\frac{(1936\text{ cfm} - 720\text{ cfm})}{41\text{ ft}} = 29.7\text{ cfm/ft}$$

This value is well within the target range and acceptable for this operating room. The following table highlights the sound and pressure drop levels of the Price HORD linear slot diffusers. All values at the calculated flow rate of 30 cfm/ft are well below the recommended HVAC noise range of 25 to 35 NC for operating rooms.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

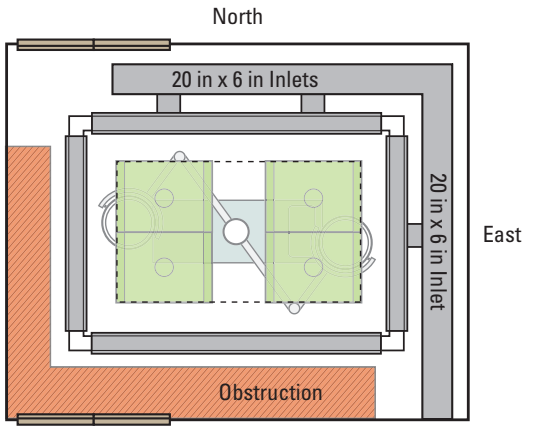
Example 3 - Operating Room Air Distribution

Size & Quantity of Inlets on the Air Curtain Plenum

The size, quantity and location of inlets on the air curtain plenum are all important considerations. Each of these variables will impact the distribution of supply air along the length of the linear slot diffusers. Effective designs will produce an equal distribution of supply air across the diffuser, creating a thick, uniform air curtain around the surgical zone.

For this example it will be assumed that the ceiling plenum is fairly congested and access to the linear slot diffuser sections is somewhat restricted. Further to this assumption, access to the linear slot diffusers is only possible from the north and east sides of the system. In situations where it is impractical to include duct connections on all four sides of the air curtain, a continuous plenum system is the appropriate selection.

The continuous plenum systems utilize corner sections to connect each of the four linear slot diffuser plenums, allowing air to pass between them. Air flow from a continuous plenum system can be equalized across all four sides of the linear slot diffuser without having primary air supplied directly to each plenum.The corners of a continuous plenum system are blanked off on the room side, whereas the modular systems feature linear slot diffusers that overlap in the corners.



Considerations for sizing and locating inlets on OR linear slot diffuser systems:

- Target inlet velocity should be 500 fpm
- Preferable to have at least one inlet for every 10 ft of linear slot length – modular systems (inlets required on all four sides of the system)
- Preferable to have at least one inlet for every 16 ft of linear slot length – continuous systems (inlets required on at least two sides of the system)

Calculating the required total inlet area for system based on 500 fpm air velocity:

$$\text{Total Inlet Area} = \frac{1936\ \text{cfm} - 720\ \text{cfm}}{500\ \text{fpm}} = 2.43\ \text{ft}^2 = 350\ \text{in}^2$$

Calculating the total inlet width based on a 6 in. inlet height:

$$\text{Total Inlet Width} = \frac{350\ \text{in}^2}{6\ \text{in}} \approx 58\ \text{in}$$

With a total linear slot length of 41 ft, three 20 in. x 6 in. inlets will work for this continuous plenum system. Two inlets on the north side plenum and one on the east side plenum represent the most suitable distribution of inlets given the restrictions of this example.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

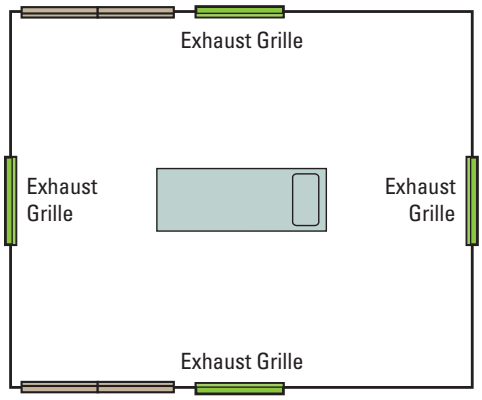
Example 3 - Operating Room Air Distribution

Return Grille Selection (Applies to all system types)

As previously calculated, 1549 cfm will be exhausted from this OR to provide the required positive pressure differential with adjoining spaces.

The return or exhaust air should be exhausted through at least two (ASHRAE Standard 170-2008), but preferably four low-level return grilles, spaced as far from each other as possible (i.e. one in each corner of the room). If we use four grilles for this example, each will need to handle 387 cfm of return or exhaust air. The grilles should be sized for a core velocity of approximately 500 fpm.

From this performance data, four exhaust grilles with a core area of 0.69 ft² each will be suitable. The exhaust grilles should be installed 8 in. above the floor at the midpoint of each wall or in the corners of the room.



Performance Data - Stainless Steel Louvered Face Return - 3/4 in. Blade Spacing

Core Area, ft²	Nominal Size	Neck Velocity, fpm	300	400	500	600	700
		Velocity Pressure, in. w.g.	.006	.010	.016	.022	.031
		Negative Ps , in. w.g.	.025	.044	.069	.100	.136
0.60	10 x 10	cfm	180	240	300	360	420
	12 x 8	NC	-	16	23	28	32
0.69	12 x 10	cfm	207	276	345	414	483
	14 x 8	NC	-	17	23	28	32
0.81	14 x 10	cfm	243	324	405	486	567
		NC	-	17	23	28	33

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Hospital Pharmacies

Unlike most health care applications where patient safety and infection control are the primary issues, the concern in the pharmacy is primarily particulate control. Pharmaceutical preparation in a health care facility presents multiple challenges with respect to air distribution and HVAC design. A number of considerations exist to ensure that air cleanliness is not compromised and medications remain contaminant free. Proper filtration, dilution and air flow patterns will reduce the probability of airborne particulates mixing with and contaminating product or process.

Standards & Guidelines

The ventilation requirements for hospital pharmacies are discussed in several standards and guidelines, many of which are adopted as local code requirements and by-laws. USP 797, a Guidebook to Pharmaceutical Compounding-Sterile Preparations, is one of the more widely referenced documents covering HVAC design in pharmaceutical manufacturing environments. The recommendations given in this section are based primarily on USP 797.

Hospital Pharmacies

USP 797 uses ISO classifications for identification of airborne particulate matter concentrations. **Table 1** defines the relevant classifications for pharmaceutical manufacturing spaces.

Before dealing with air patterns and/or particle dilution, general terms and layouts related to pharmaceutical compounding areas should be explained.

- **Compound Sterile Preparation (CSP)**
  A product (oral and intravenous medication, chemotherapy, etc.) that requires some type of manipulation prior to administration to patients.
- **Laminar Air Flow Workbench (LAFW)**
  Either a laminar flow clean bench or a bio safety cabinet.
- **Barrier Isolator (BI)**
  Is designed to provide clean makeup air to the workspace and a seamless barrier between the workspace and the operator.
- **Buffer Area/Room**
  Is a clean space where all LAFWs or BIs are located.
- **Anteroom**
  Is a clean space located between the entrance to the buffer room and other adjacent spaces, with a wall separating it from the buffer room.
- **Ante Space**
  Is a clean space in the pharmacy that is logistically but not physically separated from the buffer room.

Table 1: Classifications for pharmaceutical manufacturing spaces.

Class Name	Particle Count (> 0.5 µm)	
ISO Class	**Per m³**	**Per ft³**
3	35.2	1
4	352	10
5	3520	100
6	35,200	1000
7	352,000	10,000
8	3,520,000	100,000

- **Risk Level**
  (low, medium or high) relates to the risk of the product becoming contaminated, not the danger to the operator (refer to USP 797, 2008) for details on low, medium or high risk level classification).

To minimize the risk of contamination during manipulation of CSP, the work should be done in an ISO Class 5 environment. The most common types of equipment used to provide this environmental classification are LAFW, bio safety cabinets or barrier isolators. It is not necessary to use this equipment if the overall buffer area meets ISO Class 5 requirements, but achieving this classification in a large space is often prohibitively expensive. Regardless of the type of equipment used to create the ISO Class 5 environment, it should be located in a buffer area designed to maintain ISO Class 7 conditions. An anteroom or ante space maintained at ISO Class 8 conditions will also be required.

The ISO Class 5 environment is achieved in a self-contained system (i.e., LAFW) and does not require design guidance. This section will therefore focus on the buffer and anteroom of the pharmacy.

Low velocity, vertical discharge diffusers are recommended for pharmacy applications to prevent air turbulence and potential interference with the face velocity of LAFW or similar equipment. With the primary goal of particulate control in the pharmacy, all supply air should be HEPA-filtered. This can be done at an upstream filter bank or at the diffuser(s). Installing HEPA filters in a filter bank or at the air handler is beneficial from a maintenance perspective as the filters can be changed without exposing the room to contaminated filter media and potentially compromising the clean environment. If HEPA filters are installed upstream of the diffusers, there should be no duct or attenuator lining between the filter and the room (Rousseau & Bare, 2007). Good practice is to install MERV 7 (25%) filters upstream of any HEPA filters to increase the useful life of the more expensive HEPA filters.

In renovation work, where the pharmacy is tying into an existing system, there may be insufficient fan power to overcome the filter pressure drop and supply enough air to the space. In these situations, fan filter units with integral HEPA filters can provide the static boost necessary to meet the supply requirements. Also, fan filter units are available with electrically commutated motors which will compensate for the loading of the HEPA filter and maintain a constant air flow.

The total supply air volume to the buffer area should be no less than 30 ach. If the room has an ISO Class 5 recirculating device (i.e., recirculating LAFW or bio safety cabinet, etc.) then a minimum of 15 ach is adequate providing the combined air-change rate is not less than 30 ach (USP 797, 2008). This minimum air-change rate may not be sufficient to address cooling loads and satisfy comfort requirements in the space. Pharmacies typically have a high density of heat-generating equipment and it is important to determine the maximum cooling load for the space to ensure enough diffusers are available to meet this requirement.

The number of supply diffusers in a particular area of the pharmacy will depend on the particulate concentration classification of that space. The supply outlets should be located to ensure they do not interfere with the flow of air into the equipment generating the ISO Class 5 environments. Supply air velocity should not exceed 50 fpm at the face of the LAFW (ASHRAE, 2007) to prevent interference with face velocity and performance of the equipment.

Low sidewall grilles are recommended for exhausting air from ceiling-mounted diffusers. This will help create a general downward flow of filtered air from ceiling to floor, displacing any particulates down and away from the people and the processes. Low-level exhaust grilles are typically the louvered face style with the blades oriented at a downward angle to reduce visibility of the ductwork. Grille construction will usually be stainless steel for appearance, durability and cleaning purposes. To facilitate cleaning, these exhaust grilles are often specified with quick-release fasteners for easy removal.

To help control and maintain the particle count density in the various areas of the pharmacy, a positive pressure differential should exist to maintain air flow direction from the buffer area to the anteroom and from the anteroom to the corridor.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Laboratories

There are a number of different laboratory types that exist from animal research to electronic cleanrooms, each having different HVAC design requirements.The primary goal of air control in labs is to ensure occupant safety without compromising the process or accuracy of experiments and research conducted in the facility. As with any air distribution system, occupant comfort is also a critical design consideration.

The design of a laboratory HVAC system should start with an assessment of what is driving ventilation rates for the room.

- Load-driven spaces most often include high equipment loads and/or external walls and windows, with occupants generating a smaller percentage of the total load. Supply air volumes for these spaces are determined by cooling or heating requirements typically resulting in the supply of more ventilation air than would otherwise be required to meet minimum air-change rates.
- Labs with multiple fume hoods or containment cabinets are often exhaust-driven spaces, with primary supply air volumes based on the rate at which air is exhausted using this equipment. Supply air requirements for this category will fluctuate depending on how many of these exhausted safety enclosures are in use at any time. Fume hood and/or room pressurization controls are often used in combination with lab exhaust valves or venturi valves to maintain proper fume hood function and room pressure conditions.
- The minimum air-change rate is sometimes the factor dictating the room ventilation rate, particularly in microelectronic lab applications where air cleanliness requirements are so high. Different systems, diffusers and configurations are suited to each type of lab so the distinction should be made early in the design process.

Laboratory Types and Space Considerations

Most laboratory spaces require minimum ventilation rates of 6 to 12 ach, with some exceptionally critical spaces demanding even more. Each type of lab will present different design considerations.

Laboratories and the associated design considerations can be divided into the following general categories (ASHRAE, 2007):

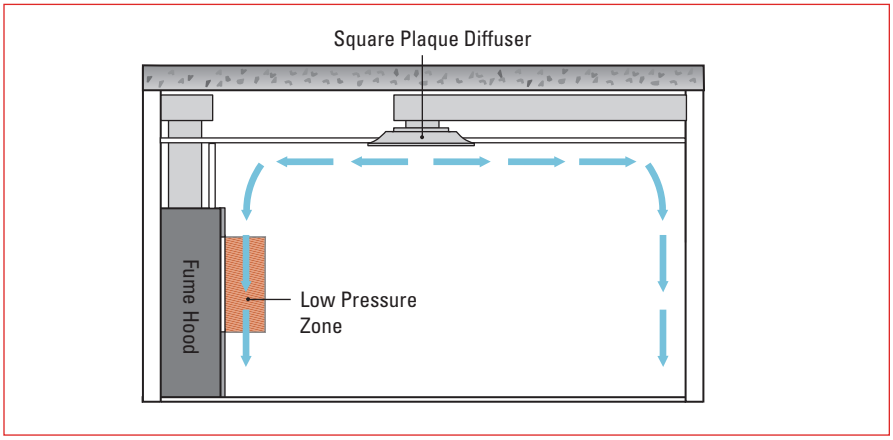


Figure 20: Low pressure zone

Figure 21: Laminar flow diffuser

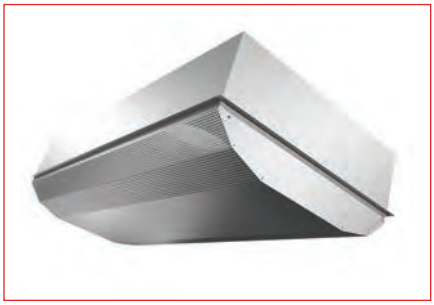


Figure 22: Radial flow diffuser

Biological Laboratories

- Contain biologically active material
- Chemical fume hoods and biological safety cabinets
- Negative room pressurization

Chemical Laboratories

- Synthesis and analytical functions
- Includes materials and electronic sciences
- Multiple fume hoods
- Negative room pressurization

Animal Laboratories

- Manipulation, surgical modification and observation of laboratory animals
- Animal holding rooms
- Negative room pressurization

Physical Laboratories

- Spaces associated with physics
- High precision analytical equipment
- High air-change rates
- Typically positive room pressurization

Fume Hood Face Velocity

Room air flow patterns can impact fume hood face velocity if supply or return outlets are improperly selected or positioned.To ensure proper containment, a laboratory fume hood should maintain a consistent face velocity. Often this face velocity will be 100 fpm, however many facilities are being designed with low-flow hoods which require face velocities as low as 50 to 60 fpm. If fume hood face velocities are too low or inconsistent, air within the hood is more likely to escape into the general lab area.The combination of these disturbances is sometimes referred to as 'challenge velocity' (McIntosh, Dorgan & Dorgan 2001). As a rule, supply air velocity should not exceed 50 fpm in the occupied zone (6 ft above the floor) to protect against unwanted disturbances of hood face velocity as well as avoid draft. However, it is highly recommended that challenge velocity not exceed 20% of the design face velocity for a given hood. Therefore, supply air velocities around the hood opening must be less than 20% of the hood face velocity in order to meet challenge velocity criteria.

Laboratories

Diffuser Layout & Selection

Unlike typical office spaces where diffusers are spaced to promote even mixing and a uniform temperature profile throughout the room, laboratories need to supply large volumes of air at low velocities to make up for exhaust air volumes without interfering with fume hood function. Conventional mixing diffusers installed near fume hoods can represent a threat to fume hood containment. High velocity discharge air from horizontal throw diffusers will travel along the ceiling and down the face of a fume hood, creating a potential low pressure zone near the fume hood opening (**Figure 20**).

This situation will often result in harmful gases being pulled into the room or directly into the face of a lab technician. As such, laminar flow diffusers (**Figure 24**) or high-capacity radial flow diffusers (**Figure 25**) are preferable for lab applications.

Close attention must be given to the throw characteristics of selected diffusers to ensure both the horizontal and vertical velocity profiles do not exceed 50 fpm inside the occupied zone. The higher throws of most ceiling diffusers are more likely to result in air velocities greater than 50 fpm in the occupied zone, resulting in potential drafts or fume hood interference. As a general rule, diffusers with a high velocity discharge and a long throw should be located as far from fume hoods as practically possible. Aside from the concern surrounding fume hood face velocity, positioning high throw diffusers too close to fume hoods may result in 'short circuiting' of the supply air into the fume hood before conditioning the room.

In the case of laminar or radial flow diffusers, air is supplied to the space in a more controlled manner with lower velocities and shorter throws. Both diffuser types should be installed no closer than 3 ft from the face of a fume hood. Furthermore, radial flow diffusers should be installed with the radial discharge perpendicular to any fume hood in close proximity (3 to 12 ft).

Figure 23: Bio safety level 4 laboratory with radial flow diffusers

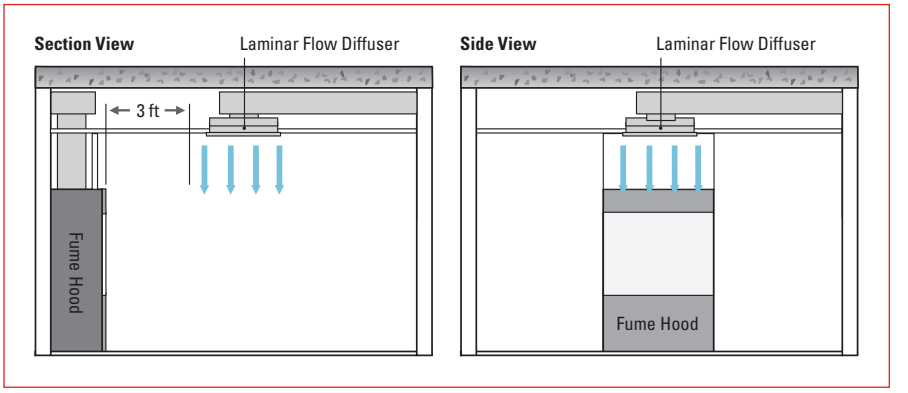


Figure 24: Laminar flow diffuser installed near fume hood

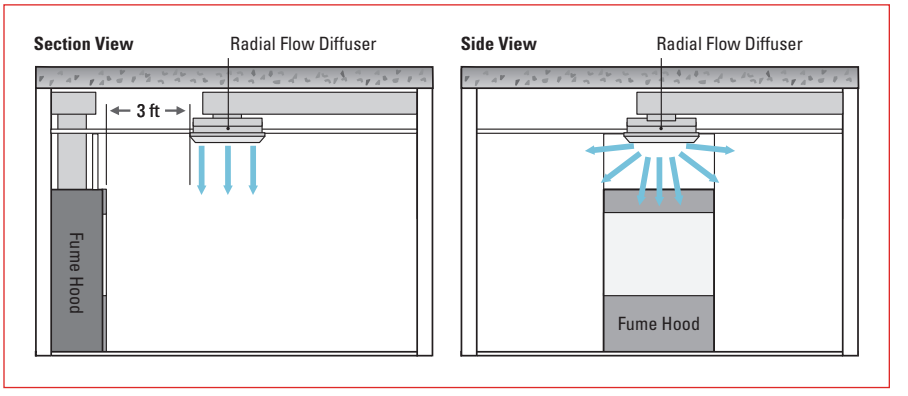


Figure 25: Radial flow diffuser installed near fume hood

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Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Laboratories

Exhaust Air Systems

The exhaust air system must remove air from the various containment devices and from the laboratory itself. The exhaust system must be controlled and operate in conjunction with the supply system to maintain the required room pressurization.

In a typical laboratory, seldom are all the exhaust devices in operation at the same time. Energy conservation and lower equipment capacities may be possible if the system can be sized for the maximum expected usage. This can be determined through consultations with the owners and research staff.

There are two typical options for exhausting the air from a laboratory. Fume hoods and containment cabinets can either be supplied with individual exhaust fans or have their exhausts manifolded to a central exhaust fan.

Central Exhaust System (Manifold)

An exhaust system where the exhaust from each fume hood and containment cabinet is ducted to a common exhaust manifold can be either a pressure-dependant or pressure-independent system.

The pressure-dependent system can only be a constant volume system. The exhaust duct from each exhaust device will have a manually adjusted damper for balancing of the system. One draw back to this type of system arises when an additional exhaust hood or device must be added to the system. The entire system must be rebalanced, and the capacity of the exhaust fan may need to be adjusted.

The pressure-independent system can be constant volume, variable volume or a combination of the two. Two (2) advantages this system has over the pressure-dependent system is the flexibility to add additional exhaust devices without rebalancing the entire system, and variable volume control of the exhaust devices.

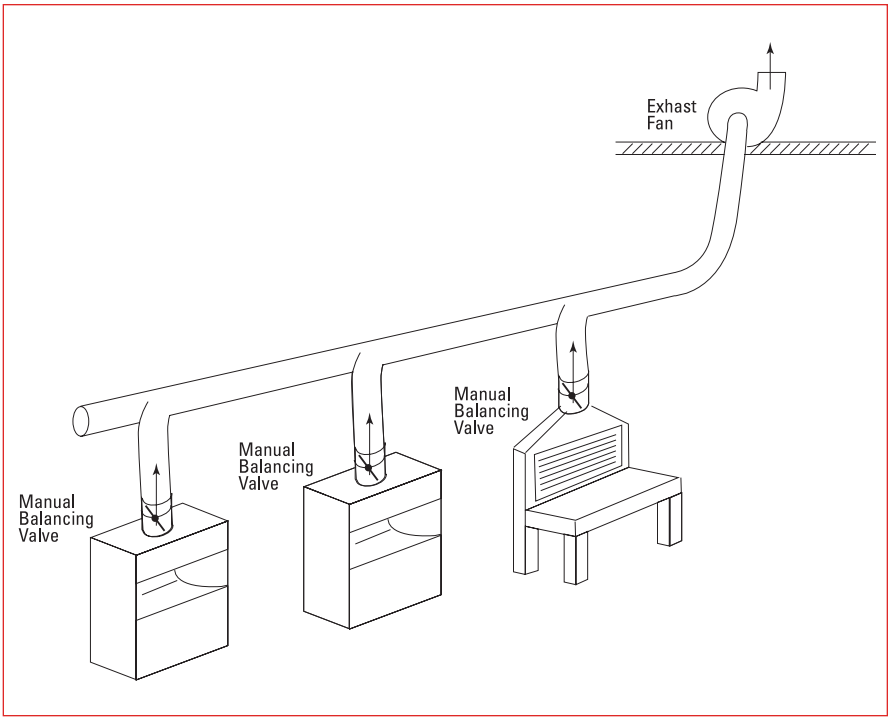

Figure 26: Pressure-dependant system

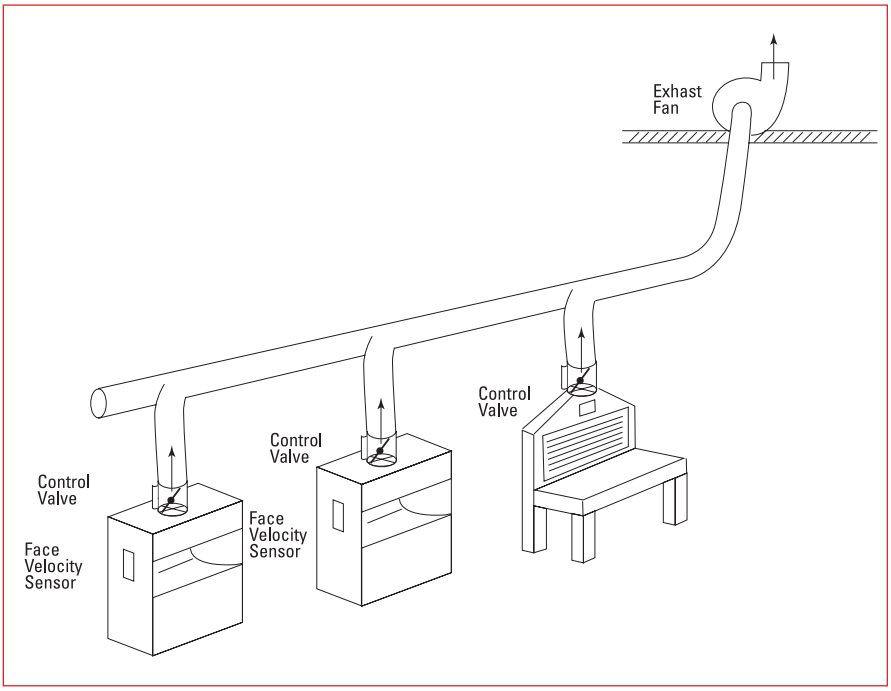

Figure 27: Pressure-independant system

Laboratories

Individually Ducted Exhaust Systems

As the name implies, an individually ducted exhaust system has a separate duct, exhaust fan and discharge stack for each exhaust device. This type of exhaust system allows for the installation of special exhaust filtration or treatment systems for specific applications, customized ductwork and exhaust fan corrosion control when required, provision for selected emergency power back up, and simpler initial balancing. Maintaining the correct flow at each exhaust fan requires periodic maintenance and balancing, as well as consideration of the flow rates with the fume hood sash in different positions.

Exhaust Control Valves

With most laboratories utilizing manifolded exhaust systems, it is important to use pressure-independent exhaust valves which will compensate for system pressure changes. These pressure changes could be the result of many factors including opening/closing of fume hood sash doors as well as adjustments to control valves modulating the laboratory general exhaust. Control valves that are not pressure-independent will allow air flows to fluctuate slightly in response to system pressure changes, leading to a potentially unsafe variance in volumetric offset and room pressurization in the laboratory.

Venturi valves offer this critical pressure independence by incorporating an internal plunger assembly which includes a carefully engineered spring. As duct static pressure fluctuates, this spring will expand or compress to facilitate changes to the relative plunger position within the venturi valve body, keeping the air flow constant through the valve. VAV venturi valves will usually incorporate a rotary potentiometer to measure the position of the shaft controlling the plunger position. Based on factory characterization of the venturi valve and the pressure-independent plunger assembly, air flow through the valve is known based on the output from this potentiometer.

Depending on the type of laboratory, the control valves can be exposed to fairly corrosive and/or contaminated environments. As a result, laboratory control valves used for fume hood type exhaust applications are typically constructed of either heresite coated aluminum or stainless steel to withstand corrosion and chemical attack.

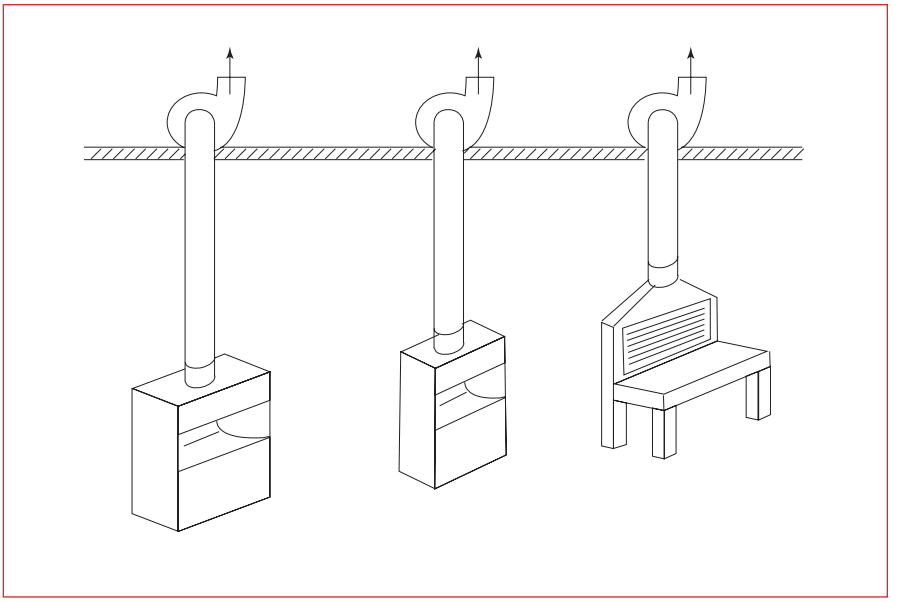

Figure 28: Individually ducted system

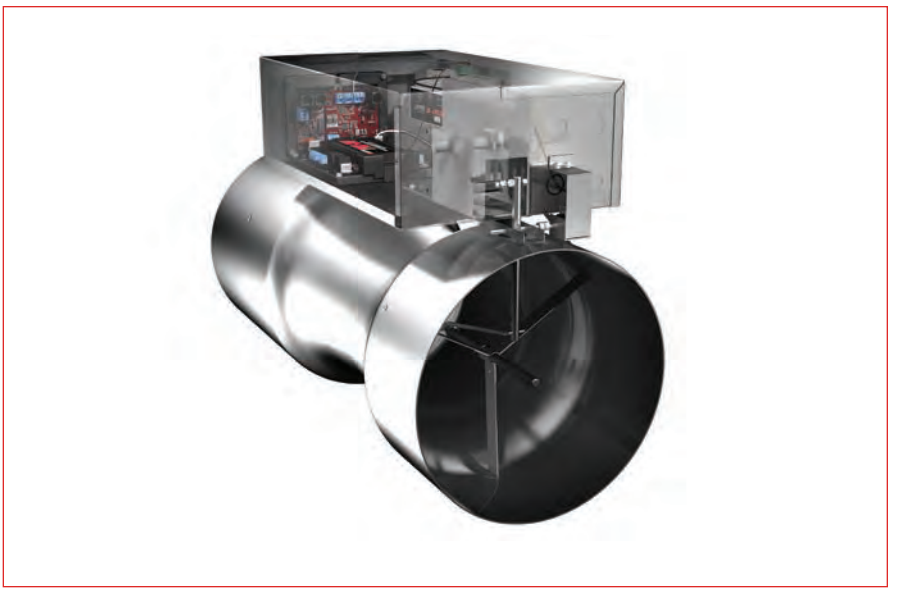

Figure 29: Venturi valves

Cleanrooms

Cleanroom performance depends on the quality of control of particulate concentrations and dispersion, temperature, humidity, vibration, noise, air flow pattern and construction. The goal is to control these parameters while maintaining reasonable installation and operation costs.

In a typical clean space/cleanroom, there are two (2) main sources of particulates, external and internal sources.

External sources consist of the following:

- Outside make-up air introduced into the room. This is typically the largest source of external particulates.
- Infiltration through doors, windows and other penetration through the cleanroom barriers.

External sources of particulates are controlled primarily through:

- Make-up air filtration.
- Room pressurization.
- Sealing of all penetrations into the space.

Internal sources consist of the following:

- People in the clean area. People are potentially the largest source of internally generated particulates.
- Cleanroom surface shedding.
- Process equipment.
- Material ingress.
- Manufacturing processes.

Internal sources of particulates are controlled by:

- Air flow designed to continually wash occupants with clean air;
- New cleanroom garments;
- Proper gowning procedures.

Externally generated particulates are prevented from entering the clean space through the use of proper air filtration. Either HEPA or ULPA filters are used to filter the outside make-up air. HEPA filters have 99.97% to 99.997% removal efficiency on 0.3µ particles, while ULPA filters have 99.9997% removal efficiency on 0.12µ particles. The decision regarding which filter to use is based on the Class of cleanroom desired **(refer to Table 2)**.

The configuration of the air flow pattern in the cleanroom is the first step in good cleanroom design. Air turbulence in the space can cause particulates which have settled onto the floor and work surfaces to become re-entrained in the air. Air turbulence is greatly influenced by the configuration of air supply and return grilles, people traffic and process equipment layout.

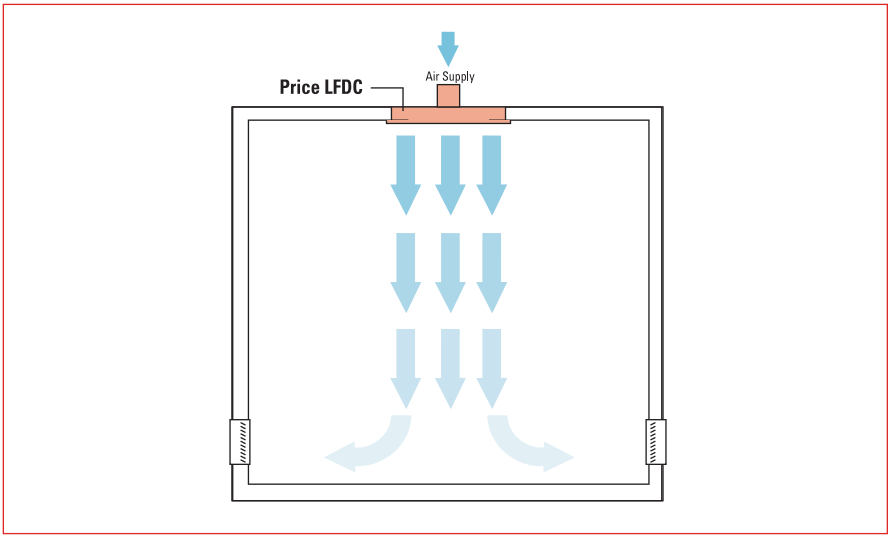

Figure 30: Unidirectional air flow

There are generally two (2) air pattern configurations used in cleanroom design, unidirectional and non-unidirectional. **(refer to Table 2)**. Unidirectional air flow is not truly laminar, but is characterized by air flowing in a single pass in a single direction through a cleanroom or clean zone with generally parallel streamlines. A state of constant velocity is approximated with unidirectional air flow. Non-unidirectional air flow is any air flow which does not meet the definition of unidirectional air flow.

Unidirectional Air Flow

With unidirectional air flow, air pattern control and turbulence reduction are optimized. Clean make-up air is typically introduced at the ceiling and returned through a raised floor or at the base of the sidewalls. Horizontal flow cleanrooms use a similar approach, but with a supply wall on one side and a return wall on the other.

Typically a downflow cleanroom consists of HEPA filtered units mounted in the ceiling. As the class of the cleanroom gets lower, more of the ceiling consists of HEPA units, until, at Class 5, the entire ceiling will require HEPA filtration **(refer to Table 3)**. The flow of air in a downflow cleanroom bathes the room in a downward flow of clean air. Contamination generated in the room is generally swept down and out through the return. With the downflow cleanroom, sealing of the ceiling system and HEPA filters assists with the desired air flow pattern by removing a potential leak point at the level where the clean air is introduced.

The horizontal flow cleanroom uses the same filtration air flow technique as the downflow, except the air flows across the room from the supply wall to the return wall. One major limitation of the horizontal flow design is that the downstream contamination in the direction of air flow increases.

Cleanrooms

Non-Unidirectional Air Flow

With non-unidirectional air flow there can be numerous variations based on the location of supply air inlets and return air outlets and air filter locations.

Air is typically supplied into the space by one of two methods. The first uses supply diffusers and HEPA filters. The HEPA filter may be integral to the supply diffuser or it may be located upstream in the ductwork or air handler. The second method has the supply air pre-filtered upstream of the cleanroom and introduced into the space through HEPA filtered work stations. Non-unidirectional air flow may provide satisfactory control for cleanliness levels of Class 6 to Class 9.

Non-unidirectional air flow can result in turbulent air flow in the space. This turbulent flow enhances the mixing of low and high particle concentrations, producing a homogenous particle concentration acceptable to the process.

Air Velocity

There are no current standards, which specify air velocities in a cleanroom or clean space though the 90 fpm velocity specified in an earlier standard (Federal Standard 209B) is widely accepted in the cleanroom industry.

Ceiling Construction

The ceiling of the cleanroom is another potential location for contaminants to enter the clean zone. Pressurization of the cleanroom helps to prevent this, however this can lead to contaminants from the processes in the cleanroom being forced out into the area surrounding the cleanroom. To reduce the chance of this happening, the cleanroom ceiling is sealed. The type of seal is determined by the cleanliness class of the cleanroom. For Class 6 and higher (less clean), the ceiling grid can be gasketed aluminum T-bar with a 1 in. face tee. Refer to the Price Unitee CR section in this brochure. A Class 5 cleanroom should have a gasketed aluminum T-bar grid with 2 in. face tees and Class 4 and cleaner should have a modular /T-bar ceiling grid with a gel seal **(refer to table 5)**.

The gasketed T-bar system has an integral vinyl, or similar material, gasket. The gasket is compressed between the base of the tee and the ceiling panel or diffuser. Hold-down clips are used to maintain the compression on all non-access related panels.

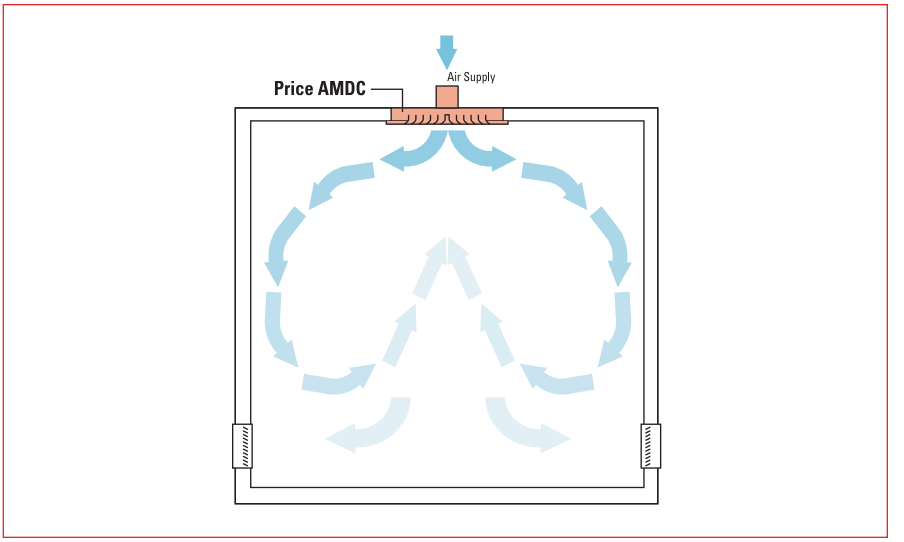

Figure 31: Non-unidirectional air flow

The gel grid T-bar system has a groove running the full length of both sides of the tee. The groove is filled with a suitable sealing gel. This type of ceiling is typically used in cleanrooms where 100% of the ceiling consists of filters or fan filter units. The filters and/or fan filter units have a knife edge around the perimeter which goes into the gel forming a seal.

The type of ceiling panels used in a cleanroom ceiling also depend on the cleanliness class of the space. Class 6 and above (less clean) can have cleanroom approved, vinyl covered panels or blank aluminum panels while Class 5 and cleaner can only have blank aluminum panels.

Ceiling grid support is determined by cleanliness class as well. Class 5 and cleaner should have all-thread rod with strut and turnbuckles while Class 6 and above should have 12 ga hanger wires to the grid and 10 ga hanger wires to the filters. The hanger wires should be installed at the grid intersections.

Air Return

The air return system is another critical component of the cleanroom air distribution system. The amount of air being returned will depend on the process taking place within the clean space. The process will dictate whether positive, negative or neutral pressurization is required. For a space requiring positive pressurization, the return air volume is typically 15% less than the total supply air volume. Negative pressurization will typically have 15% more return air volume than supply. With neutral pressurization, the supply and return air volumes are equal.

The location and type of air return is determined by the cleanliness class of the clean space. For Class 4 and cleaner, a raised floor return system is recommended. The raised floor should be perforated with floor grilles distributed to assist the laminar flow past processes and personnel in the space. The floor grilles may include optional balancing dampers. Low wall diffusers can be used in Class 5 and above (less clean) clean spaces. Filter grilles or balancing grilles may be used here. With Class 8 and above (less clean), either low wall or ceiling returns can be used, though the low wall is preferred. Ceiling returns may result in short circuiting of the air flow with little or no clean air reaching the occupied space.

The material of construction for the return grilles will be determined by the process taking place in the clean space, though stainless steel is used quite often for its appearance and cleanability. Due to its corrosion resistance, the use of stainless steel grilles also allows for processes to be changed periodically without changing grilles.

All Metric dimensions ( ) are soft conversion.
Imperial dimensions are converted to metric and rounded to the nearest millimeter.

Design Criteria For Cleanrooms

Table 2: Cleanroom design

Cleanliness Class	ISO Class	Air Flow Type[1]	Average Air Flow Velocity	Air Changes Per Hour*
M7 & M6.5 (Class 100,000)	8	Nonunidirectional, mixed	0.02-0.04 m/sec (4-8 fpm)	24-50
M6 & M5.5 (Class 10,000)	7	Nonunidirectional, mixed	0.04-0.09 m/sec (8-15 fpm)	50-100
M5 & M4.5 (Class 1,000)	6	Nonunidirectional, mixed	0.13-0.18 m/sec (25-35 fpm)	150-200
M4 & M3.5 (Class 100)	5	Unidirectional, Nonunidirectional, mixed	0.23-0.28 m/sec (40-60 fpm)	270-330
M3 & M2.5 (Class 10)	4	Unidirectional	0.3-0.36 m/sec (60-70 fpm)	350-425
M2 & M1.5 (Class 1)	3	Unidirectional	0.36-0.43 m/sec (70-85 fpm)	425-500
M1 & Cleaner	2 & cleaner	Unidirectional	0.43-0.51 m/sec (85-100 fpm)	500-600

[1]Type listed represents the more common air flow characteristics for cleanrooms of that class.
*Air changes per hour based on 10 foot ceiling height.

Table 3: Filter type & coverage

Cleanliness Class	ISO Class	Filter Type	L/s/M² (cfm/ft²)	Ceiling Filter Coverage
M7 & M6.5 (Class 100,000)	8	HEPA	51 (10)	10%
M6 & M5.5 (Class 10,000)	7	HEPA	102 (20)	20%
M5 & M4.5 (Class 1,000)	6	HEPA	203 (40)	25-40%
M4 & M3.5 (Class 100) Service corridor	5	HEPA	254 (50)	50%
M4 & M3.5 (Class 100) Gown room	5	HEPA	254 (50)	50%
M4 & M3.5 (Class 100)	5	HEPA	457 (90)	100%
M3 & M2.5 (Class 10)	4	ULPA	508 (100)	100%
M2 & M1.5 (Class 1)	3	ULPA	508 (100)	100%

Table 4: Filter efficiencies

M2.5 (Class 10)	ISO Class 4	99.9995% @ 0.12 micron
M3.5 (Class 100)	ISO Class 5	99.99% @ 0.3 micron
M4.5 (Class 1,000)	ISO Class 6	99.99% @ 0.3 micron
M5.5 (Class 10,000)	ISO Class 7	99.99% @ 0.3 micron
M6.5 (Class 100,000)	ISO Class 8	99.97% @ 0.3 micron

Table 5: Ceiling type

M2.5 (Class 10)	ISO Class 4	Gel grid
M3.5 (Class 100)	ISO Class 5	2" T-bar with gasket, extruded aluminum
M4.5 (Class 1,000)	ISO Class 6	1" T-bar with gasket, extruded aluminum
M5.5 (Class 10,000)	ISO Class 7	1" T-bar with gasket, extruded aluminum
M6.5 (Class 100,000)	ISO Class 8	Side access HEPA filter housings

Table 6: Air peturns

M2.5 (Class 10)	ISO Class 4	Raised floor
M3.5 (Class 100)	ISO Class 5	Low wall/perimeter/long axis
M4.5 (Class 1,000)	ISO Class 6	Low wall
M5.5 (Class 10,000)	ISO Class 7	Low wall
M6.5 (Class 100,000)	ISO Class 8	Low wall or ceiling

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