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Guide for Air Supply Strategies in Spaces

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Chapter 1. Introduction

Heating, Ventilation, and Air-Conditioning (HVAC) systems are designed to maintain indoor air quality and thermal conditions in a space that supports conditions that are acceptable for human occupants, processes housed in the space, and building sustainability. Proper selection and sizing of HVAC systems require knowledge of contaminant, moisture, heating, and cooling loads and an understating of the mechanisms and characteristics of air and contaminant movement in ventilated and air-conditioned spaces.

The airflow pattern, the scale of air currents in the room, and temperature and contaminant distribution within the room depend on air supply methods and the type of air diffusers used, the types of sources, and the energy introduced by each source, as well as the configuration and dimensions of the room. The energy that propels the predominant turbulent flow that is created by each source transfers into variable air pulses that convert large air "eddies" into smaller eddies. The energy is finally converted into heat. The kinetic energy of air leaving the room through exhaust openings can be neglected. Typically, exhaust openings are protected by a grill, which does not allow large or medium-size energy-containing eddies to pass through. Chapter 7.3 of IVDG (2000) gives more detailed information.

This document describes basic principles guiding the selection and design of air supply in mechanically ventilated commercial, residential, and industrial spaces. For information on how to design air distribution systems, see the relevant chapter of the ASHRAE Handbook of Applications (1995).

Chapter 2. Supply and Exhaust Ventilation Systems

Supply and exhaust ventilation systems can be characterized as "general" or "local." General ventilation systems (both supply and exhaust) can be mechanical, natural, or mixed (mechanical supply, natural exhaust, or vice versa). When there are no special demands for thermal comfort conditions in the building during the cooling or heating season, natural ventilation through the windows, doors, or fixed air vents can be used provided that contaminant concentration in the breathing zone is controlled.

General Supply Systems

Mechanical supply systems are normally used for contaminant dilution, moisture control, and thermal comfort in the occupied zone. They consist of inlet sections, filters, heating and/or cooling equipment, fans, ducts, and air diffusers for distributing air within the ventilated space.

General Exhaust Systems

General exhaust systems complement local exhaust systems by removing air contaminated by fumes, gases, or particles not captured by local exhausts. Such systems usually consist of inlets, ducts, an air cleaner, and a fan. The efficiency of the air cleaner must be sufficient to meet the regulations of environmental agencies and may be affected by location, the concentration of contaminants in the atmosphere, nature of contaminants, and height and velocity of the air discharge stack. In some cases, the air cleaners may be excluded from the general exhaust system and general exhaust can be provided by the roof fans.

Recirculation

When hazardous gases and particles are not present or have been removed, it may be possible to recirculate part of the air cleaned in the general exhaust system, which will reduce heating and cooling costs. In this case, a return duct is used to bring the ventilated space air to the supply unit, or the air is cleaned in free-hanging air filter units. Portable units are also used for spot general control and pass the cleaned air recirculated directly back into the area being served.

Airflow rates for general ventilation should not only remove the key contaminants in the occupied zone (not captured by the local ventilation system) but should also help maintain the heating and cooling demands of the space and pressure control in the space to minimize infiltration through the building envelope and internal walls.

Local supply systems. Air is supplied locally to serve occupied areas such as desks in offices, seats in cinemas, theaters, and other assembly spaces (Figure 2-1) or spaces close to a few permanent working places in industrial facilities (Figure 2-2). Conditioned air is supplied to the occupants' breathing zone to maintain comfortable conditions and/or reduce the concentration of pollutants in the confined area. In local ventilation systems, air is supplied either through nozzles or grilles (e.g., for spot cooling); through specially designed low velocity/low turbulence devices (Kristensson and Lindqvist 1993); through perforated panels suspended on the vertical duct drops and positioned close to the workplace (VDI 1994), or through a combination of vertical perforated supply air panels that create a "comfort oasis" around the work area and that act as screens to separate the work area from the rest of the environment (REPUS Undated, ABB 1990, AIR-IX 1987).
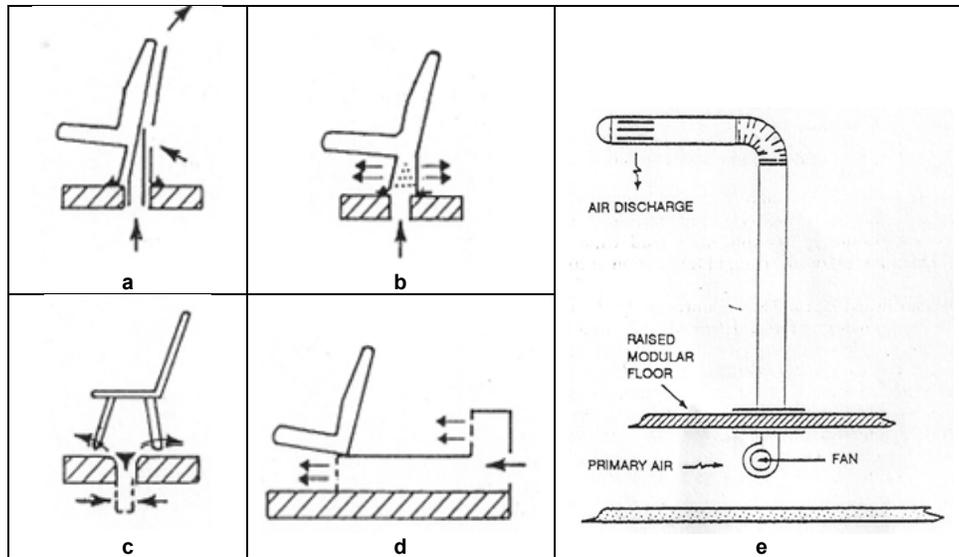
Figure 2-1. Local air supply: A-D in the auditorium and a theater, at the desk in the office.
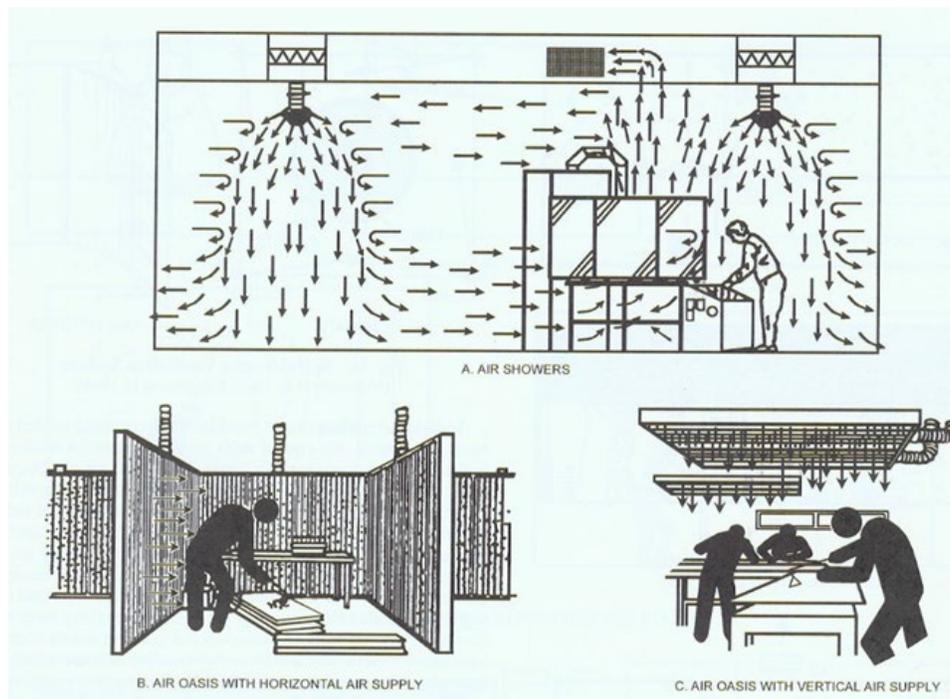
Figure 2-2. Local supply systems: (a) air showers, (b) air oasis with horizontal air supply, (c) air oasis with vertical air supply.

Figure 2-3. A bank of textile air dispersion systems using only the fabric porosity to create an air shower over a painting area.

Local exhaust systems. Local exhaust ventilation systems capture the contaminants and heat at or near their source. They can include the total process enclosure, a hood incorporated into process equipment, or fixtures close to the contaminant source; to overhead, downdraft, or side draft hoods; to flexible hoods; and to many other local exhaust designs (see Figure 2-3). Although local exhaust systems are not 100% effective in capturing heat and contaminants, they do significantly reduce the size of the general ventilation system.

Chapter 3. Factors Influencing Airflow in Ventilated Spaces

Air and contaminant movement in ventilated spaces is affected by such different external and internal forces as:

- Supply air jets forced into the room by mechanical systems
- Free convective currents generated by air heating or cooling by surfaces (e.g., people, process equipment, external walls)
- Airflow in the vicinity of local exhausts (hoods) or general exhaust (due to the negative pressure in the duct produced by mechanical systems)
- Airflow, forced through intended or unintended openings in the building envelope, which depends on the pressure difference across the opening resulting from the wind pressure on the building envelope, the temperature difference between the indoor and outdoor air, and an imbalance in the mechanical exhaust ventilation performance versus the mechanical air supply (positive or negative pressure building)
- Air currents produced by moving people or process equipment (e.g., high-speed rotating machines such as pulverizes, high-speed belt material transfer systems, falling granular materials, and escaping compressed air from pneumatic tools).

Airflow in rooms dominated by supply jets. In rooms where energy is introduced primarily by air jets, air distribution methods are referred to as "mixing types." In a perfect mixing-type air distribution, the airflow pattern and air velocity at any point in the room are governed by supply jet momentum. However, if the air supply and air exhaust openings are located close to each other, a large proportion of supply air is extracted from the room without passing the occupied zone (Figure 3-1). Such a "short-circuiting situation results from a poor design and leads to an undesirable airflow pattern.
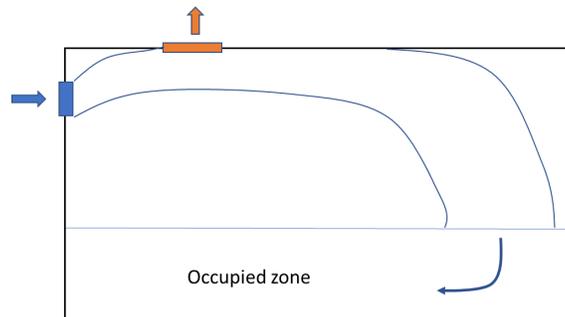
Figure 3-1. Short-circuiting of airflow in a ventilated room.

Mixing-type air distribution methods typically provide ventilation to the occupied zone using jets to intercept its upper boundary (Figure 3-2) Also, the occupied zone can be ventilated by the reverse flow produced as the supply jet degrades above the occupied zone level (Figure 3-3). Mixing-type air distribution methods include air supply with jets projected vertically downward (Figure 3-2a,b,c), inclined jets (Figure 3-4), jets directed vertically upward, and horizontal jets along room surfaces. In the latter case, the jet reaches the opposite wall/ceiling and follows room surfaces until it reaches the occupied zone (Figure 3-2d,e).

A. With radial jets attached to the ceiling   B. With downward projected conical jets   C. With downward projected compact jets

D. Through wall-mounted grille and linear diffusers;
occupied zone ventilated by the jet directly

E. Through wall-mounted grille; occupied zone ventilated
by jet and reverse flow
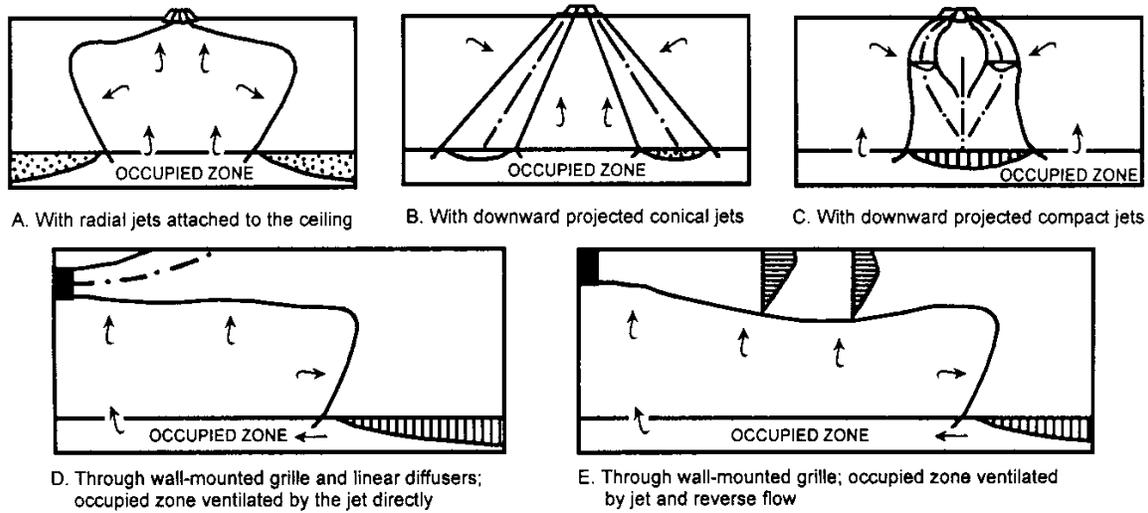
Figure 3-2. Examples of mixing-type air supply methods with the occupied zones ventilated directly by the jet.
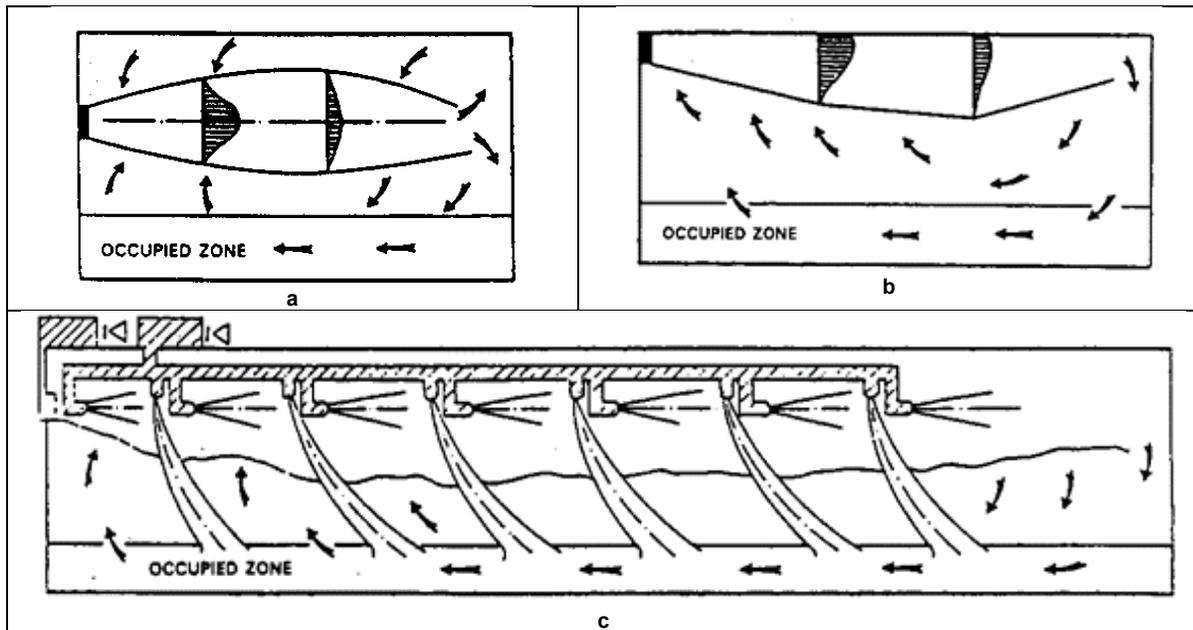
Figure 3-3. Concentrated air supply with the occupied zone ventilation by reverse flow: (a) non-attached horizontally projected jets, (b) horizontally projected jets attached to the ceiling; (c) horizontally projected jets supported by horizontal and vertical directing jets, e.g., a Dirivent system (ASHRAE 1999).
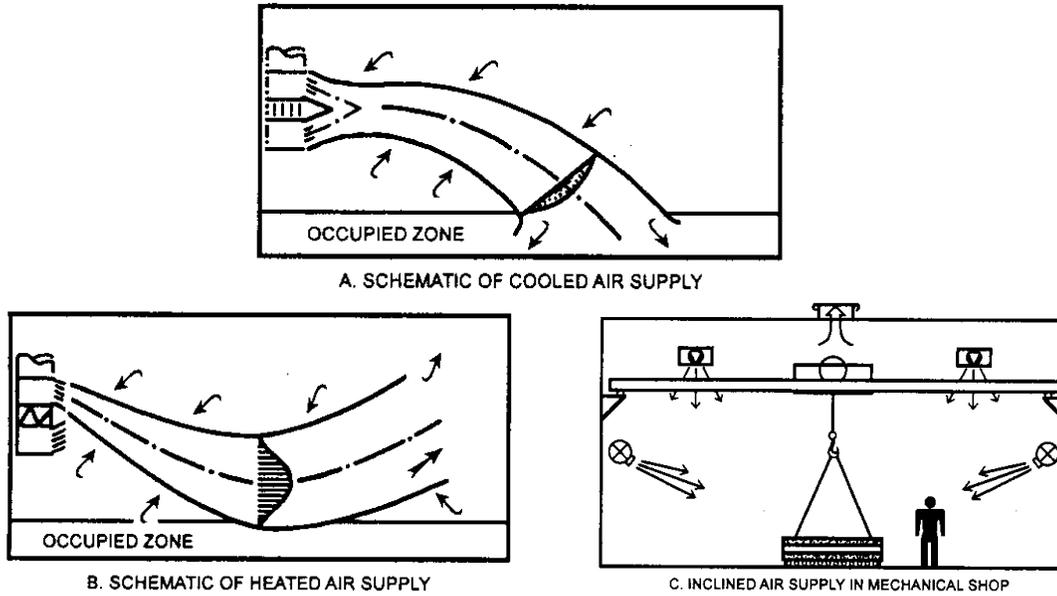
Figure 3-4. Examples of air supply in large industrial and commercial spaces with inclined jets (Zhivov 1992, AIR-IX 1987.

If the combination of room sizes (height, length, and width) allows such an airflow pattern, this room is considered to be "short" Figure 3-5). The room in which air jet dissipates before it reaches the opposite wall is considered to be "long."
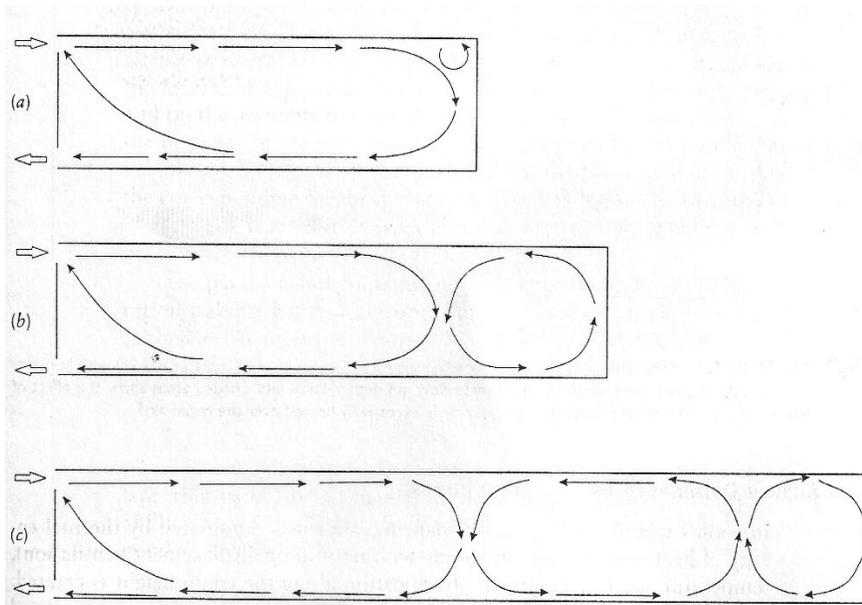
Figure 3-5. Flow pattern in rooms with horizontal air supply along the ceiling surface: (a) primary airflow in a short room, (b) primary and secondary airflows; (c) primary, secondary, and tertiary eddies in long rooms (Muller 1977).

In such rooms, the occupied zone is ventilated by reverse flow, and secondary and tertiary vortices.

Buoyant forces, e.g., when supply air is heated, can significantly affect the airflow pattern created by supply jets (Figure 3-6). Applying proper design principles prevents warm air from rising to the upper zone of the room without heating the occupied zone.
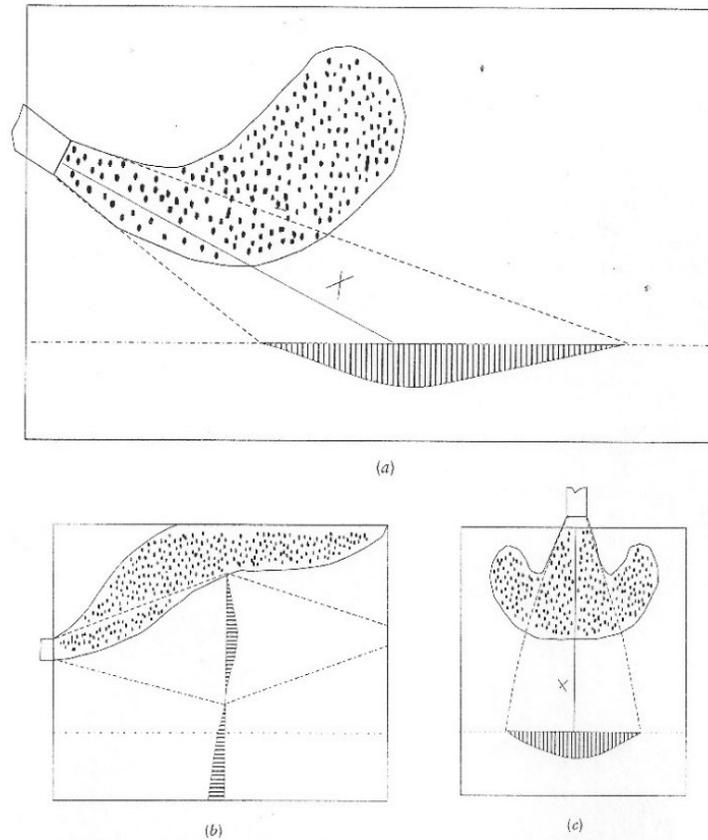
Figure 3-6. Effect of buoyancy forces on the trajectory of heated jet: (a) inclined jet, (b) horizontal jet, (c) vertical jet (IVDG 2001, Ch. 7)

Airflow dominated by thermal plumes (Zhivov et al. 2000), In rooms where air and contaminant movement is dominated by the thermal energy of heat sources (e.g., in rooms with natural or displacement ventilation systems), temperature and contaminant stratification along the room height is created. Air supply and exhaust in such rooms are designed not to disturb the natural pattern of air movement created by heat sources: cooled air enters the room in the lower zone close to the floor level and is exhausted from the upper zone. Since warm air is relatively buoyant, cold air spreads along the floor and floods the lower zone of the room. The air close to the heat source is heated and rises upward as a convective airstream (Figure 3-7). In the upper zone, this stream spreads along the ceiling. The lower part of the convective stream induces the colder air of the lower zone of the room, and the upper part of the convective airstream induces the heated air of the upper zone of the room. The height of the lower zone depends on the air volume discharged into the occupied zone and on the amount of convective heat discharged by the sources. In the presence of the temperature gradient, the convective plume may reach the height where the temperature difference in the plume and in the ambient air at the corresponding height disappears. This can happen with convective plumes above weak heat sources (e.g., above cigarette, point welding, person's body) in the presence of stronger heat sources (Figure 3-8). Temperature and contaminant gradients along the room height and separation stability between the upper and lower zones (Figure 3-9) are influenced by the turbulent exchange between these zones (Shilkrot and Zhivov 1996).
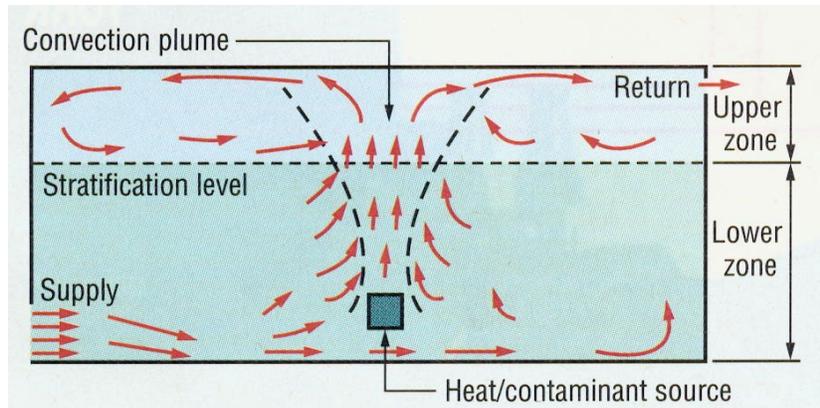
Figure 3-7. Schematics of airflow in rooms with displacement ventilation (Zhivov et al. 2000).

Figure 3-8. Textile Air Dispersion system dispersing displacement supply air at very low velocities using fabric porosity (courtesy of DuctSox Co.).
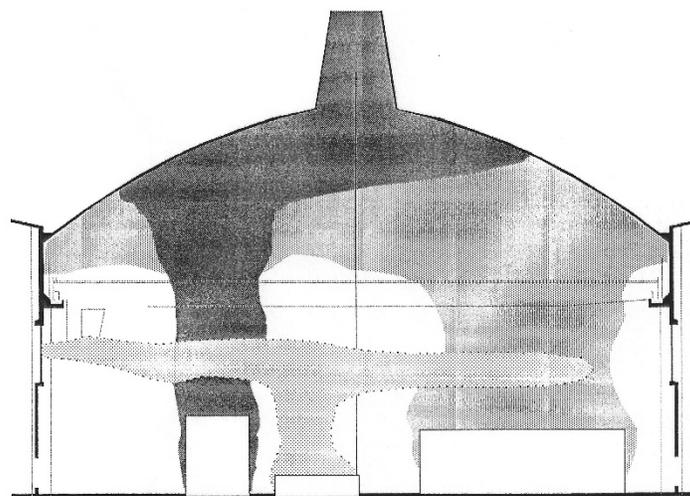
Figure 3-9. Stratification in rooms with several heat sources of different strengths (Skistad 1999).

Active displacement ventilation (Lycke 1999, Zhivov et al. 2000). Air with a temperature lower than the desired room air temperature in the occupied zone is supplied through air diffusers located above the occupied zone. Supply air velocity is lower than that of a mixing-type air supply, but higher than that of a thermal displacement ventilation. In the system shown in Figure 3-10d, air supplied through ducts with specially punched nozzles suppresses polluted air of the occupied zone and creates an overlying air cushion that displaces the contaminated air toward floor-level exhausts. Another type of such system (Figure 3-10a) supplies cooled air with low momentum through diffusers installed at a height of about 10 ft (3 m). Since warm air is relatively buoyant, cold air flows toward the occupied zone with some entrainment of the ambient air, spreads along the floor and floods the lower zone of the room. Air heated by the internal heat sources rises and is exhausted from the upper zone. Special controlled air diffusers allow for an active displacement air supply in the cooling mode, and a mixing-type air supply with a downward-projected or inclined jets in the heating mode (Figure 3-10b). This air supply method is predominantly used for ventilation of industrial spaces.
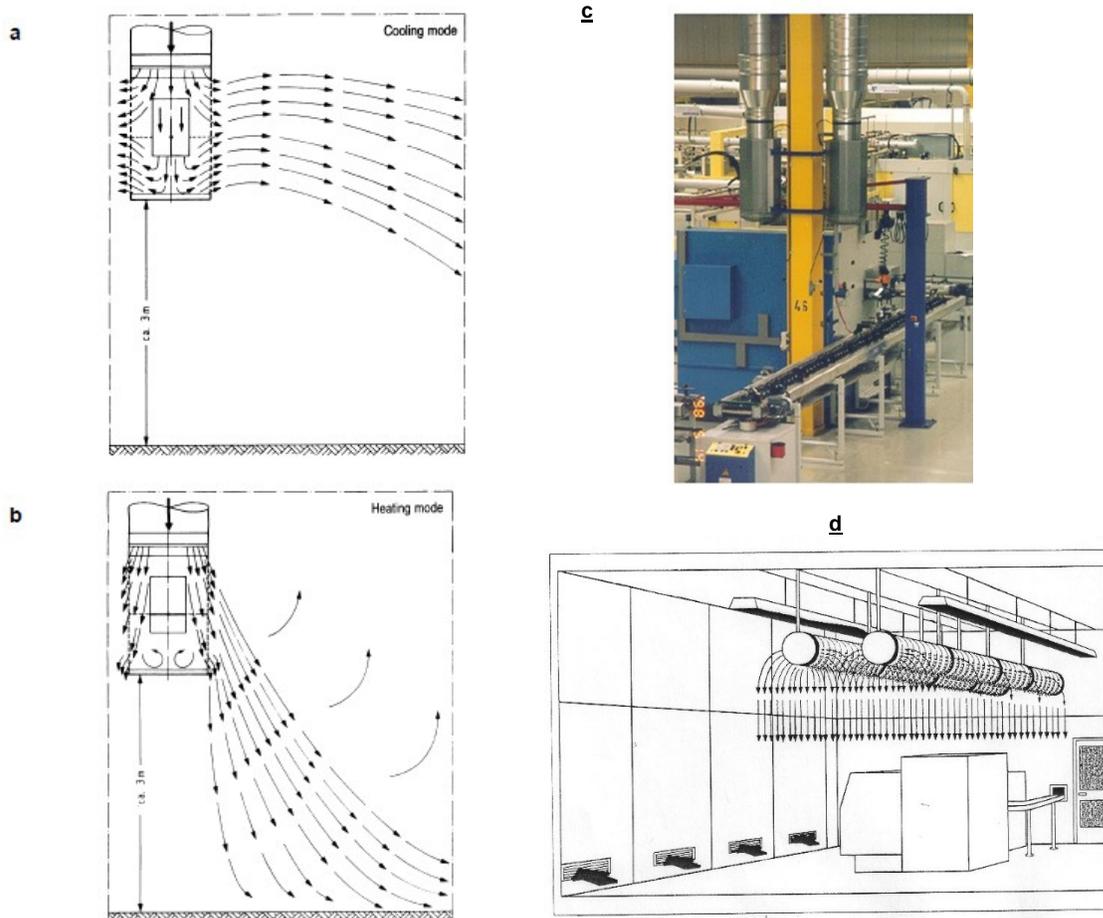
Figure 3-10. Schematics of active displacement ventilation air supply, through adjustable diffusers: (a) cooling mode: (b) heating mode, (c) example of installation in the machining shop; through perforated ducts in the mechanical shop. (Used with permission from KRANTZ-TKT [a,b] and ABB [c]).

Figure 3-11. Textile Air Dispersion system mounted at less than half of ceiling height and with air dispersion orifices for active displacement ventilation (Courtesy of DuctSox Co.).

Unidirectional flow. To create unidirectional low-turbulent airflow, the air is supplied at a low velocity; supply diffusers and exhaust openings have large surfaces (e.g., filter mats, perforated panels). Airflow can be either vertical, where air is supplied from the ceiling and exhausted from the floor or vice versa (Figure 3-11), or horizontal, where air is supplied through one wall and exhausted through returns located on the opposite wall. The outlets are uniformly distributed over the ceiling, floor, or wall to provide low turbulence plug-type flow across the entire room. This type of system is mainly used for ventilating clean rooms, in which the main objective is to remove contaminant particles within the room, or in halls with high heat and/or contaminant loads and a high air change rate.
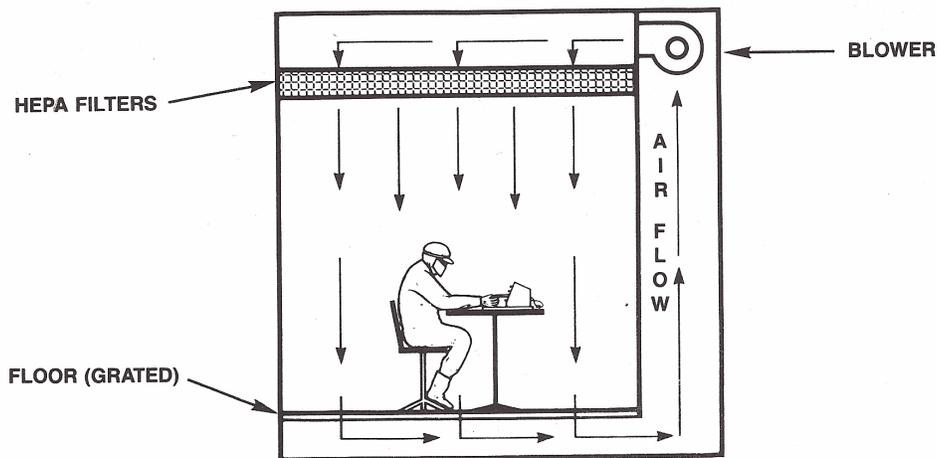
Figure 3-12. Unidirectional flow in a "clean room" (Cleanroom Guide).

Underfloor air distribution systems (UFAD). This air distribution method is based on the same principles as displacement ventilation (Figure 3-12). Both systems have higher ventilation performance than the mixing one in both cooling and heating modes (Lee et al. 2009). The difference is that, instead of using vertical air diffusers installed at or slightly above the floor level, the air is discharged into the occupied zone from air diffusers installed in the raised floor providing mixing only within the occupied zone. The system is designed to maintain stratification above the occupied zone. This system originally designed for IT data centers is currently widely used in office buildings, command centers, schools, museums, and similar applications. According to ASHRAE (ASHRAE 2015),

a major concern with UFAD is the perimeter zone that has widely varying loads between summer and winter conditions, especially in buildings with large glass exterior elements. Thermostatically controlled fan coils beneath the floor can be a cost-effective solution to this problem (Figure 3-13). Also, most UFAD diffusers can be adjusted automatically by the space thermostat or manually by the occupant.

(a)   **(b)**
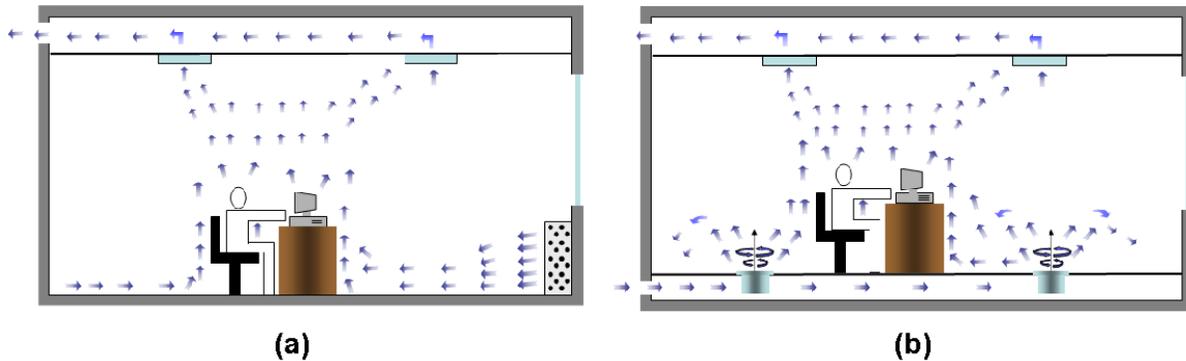
Figure 3-13. Displacement (a) vs. underfloor (b) air supply (Lee et al. 2009).

Spiral vortex flow. Spiral vortex air distribution can be used to localize air contaminants in certain room areas and to evacuate the polluted air from those areas. A spiral vortex in a space can be formed by supplying air through the vertical supply ducts located along a closed contour (preferably along the walls) to generate a vertical vortex. An exhaust outlet can be located in the ceiling near the center of the rotational flow. Such a combination of air supply and exhaust systems allows contaminants to be concentrated in the vortex core and transported to the exhaust outlet along the core axis (Figure 3-14). Low pressure in the vortex core collects contaminants and prevents their diffusion into the clean space (Kuzmina et al. 1986; Nagasawa et al. 1990).
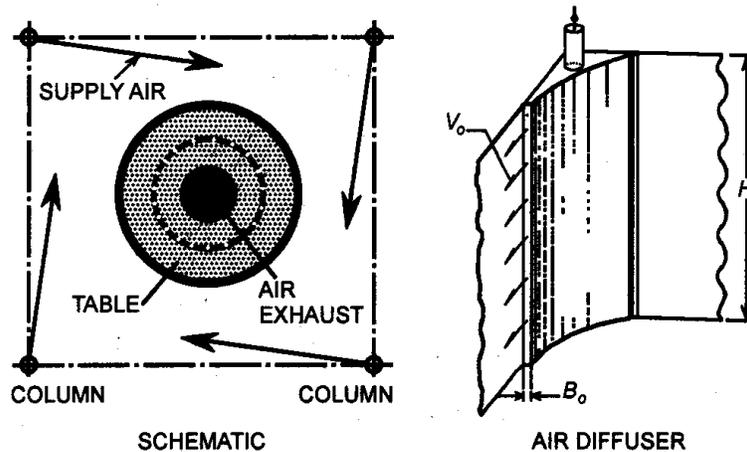
Figure 3-14. Spiral vortex ventilation system (Kuzmina et al. 1986; Nagasawa et al. 1990).

- **The airflow created by exhaust performance.** The effect of exhaust performance on room air movement is limited compared to the effect produced by air jets. The distance from the opening to the point where air velocity drops to 10% of the initial velocity value (Figure 3-15) is approximately equal to one characteristic size of the exhaust opening (D for the round duct) and 60 characteristic sizes for the supply outlet (60D for the round nozzle).
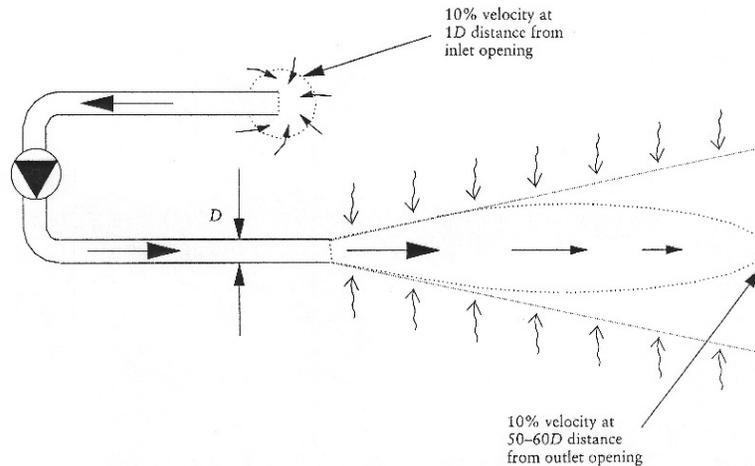
Figure 3-15. Supply air jet through is much longer than the capture length of a sink (IVDG 2001, Ch. 7).

Local exhausts do not affect the airflow pattern in the ventilated space. They are designed to capture air pollutants and heat at the source, and thus their location near the source and the exhausted airflow rate should ensure sufficient capture velocity. General exhausts typically do not prevent contaminants and heat from mixing with the room air so that, theoretically, it does not matter where the exhaust openings are located (Figure 3-16).
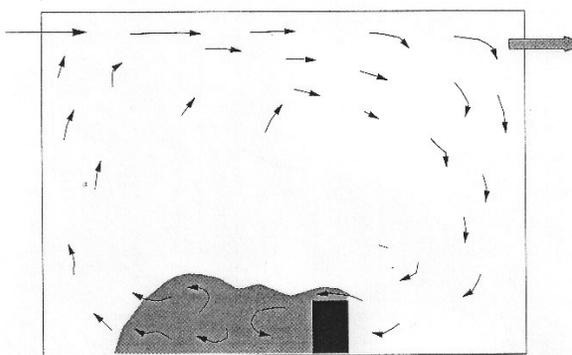
Figure 3-16. Exhaust from the room with a perfect mixing. The location of the exhaust opening is not important (IVDG 2001, Ch. 7).

In practice, the air seldom mixes as completely as in theory. The pollutants spread within the room air, but there may be areas where contamination is higher than the average for the room. This must be considered when determining the location of the exhaust opening(s) (IVDG 2001,

In rooms with ventilation systems creating temperature and/or contaminant stratification, one might consider placing an exhaust opening at, or close to, the equilibrium height for the main contaminant (Figure 3-17). In addition, there should be a general exhaust either at the ceiling level or at floor level, depending on the buoyancy of contaminants. When it is not certain if contaminants have negative or positive buoyancy, one should place exhaust openings both under the ceiling and at the floor level. For more information about general exhaust, and location selection see IVDG (2001).

When people or contaminating processes are in a semi-enclosed part of the room, the exhausts should be located so that a displacement flow is created through the passage into the semi-enclosure. With this arrangement, there may be no need for a door in the semi-enclosure, which may be very impractical.
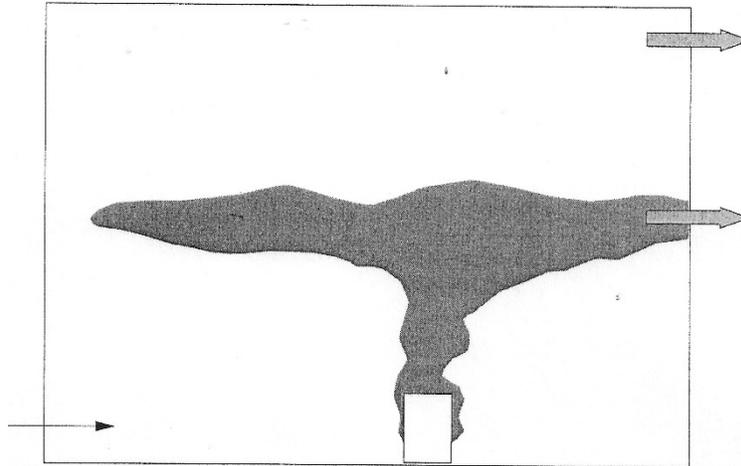
Figure 3-17. Thermal stratification and contaminant distribution should be considered in selecting the location of the exhaust opening (IVDG 2001, Ch. 7).

Chapter 4. Criteria of Air Distribution Performance Evaluation

The objective of air diffusion using ventilating and air-conditioning systems is to create the proper combination of air quality, temperature, humidity, and air movement in the occupied zone of the ventilated and conditioned room. In most cases, the occupied zone is considered to be the volume from the floor to 6ft (1.8m) above the floor level. To obtain comfort conditions within this zone, standard limits have been established as acceptable effective draft temperature, which combines the effects of air temperature, air motion, and relative humidity on a human body. Variation from accepted standards causes occupant discomfort. Lack of uniform conditions within the space or excessive fluctuation of conditions in the same part of the space also produces discomfort. Such discomfort can arise because of (1) excessive room air temperature variations (horizontally, vertically, or both), (2) excessive air motion (draft), (3) failure to deliver or distribute air according to the load requirements at different locations, (4) or overly rapid fluctuation of room temperature.

To define the difference in effective temperature, θ, between any point in the occupied zone and control conditions, Koestel and Tuve (1955) used the following equation proposed by Rydberg and Norback (1949) and modified by H. Straub in the discussion of the paper by Koestel and Tuve.

$$\Theta = t_x - t_r - 0.07 (V_x - 30), \tag{1}$$

where:

t_x = Local airstream dry-bulb temperature. °F
t_r = average room dry-bulb temperature, °F
V_x = local airstream velocity, fpm.

Equation 1 accounts for the feeling of coolness produced by air motion and is used to establish the neutral line in Figure 3-1. In summer, the local air stream temperature, t_x, is below the control temperature. Hence, both temperature and velocity terms are negative when velocity, V_x, is greater than 30 fpm (9.1 m/min), they both add to the feeling of coolness. If, in winter, t_x, is above the control temperature, any velocity above 30 fpm (9.1 m/min) subtracts from the feeling of warmth produced by t_x.

Air Diffusion Performance Index (ADPI). A high percentage of people are comfortable in sedentary (office) occupation, where the effective draft temperature, Θ, as defined in Equation (1), is between -3 and 2 °F (-18 to 17 °C) and the air velocity is below 70 fpm (21.3 m/min). If several measurements of air velocity and air temperature were made throughout the occupied zone of an office, ADPI would be defined as the percentage of locations where measurements that meet the previous specifications on effective draft temperature and air velocity were taken. If the ADPI is maximum (approaching 100%), the most desirable conditions are achieved (Miller and Nevins 1974). The ADPI is based only on air velocity and effective draft temperature and is not directly related to the level of dry-bulb temperature or relative humidity. These and similar effects, such as mean radiant temperature, must be accounted for separately according to ASHRAE Standard 55 (2020) recommendations. Also, ADPI was designed as a measure of cooling mode performance; however, due to low air speeds present in a heating mode, ADPI is not a good means of evaluation in a heating mode. Heating conditions can be evaluated using ASHRAE Standard 55 guidelines.

A probabilistic approach may be applied to the occupied zone comfort and contaminant distribution. Air velocity and temperature have been measured in realistic rooms and in room ventilation simulators by numerous researchers. Jackman (1970, 1971, 1973) studied air movement with side-wall-mounted grilles, with circular and linear diffusers. For all these methods of air supply, he found the correlation between momentum, $\mathbf{J_o}$, of supply air and average velocity $\mathbf{V_{o.z.}}$, fpm, in the occupied zone. Fissore et al. (1991) studied air velocity distribution in a space ventilated by slot-type diffusers and found that the velocity distribution matches well with the Gaussian assumption:

$$\sigma_v = 0.3 V_{o.z.} \tag{2}$$

with the variation coefficient

$$K_v = \sigma_v / V_{o.z.} = 0.0053 \qquad (3)$$

where σ_v is the standard deviation of velocity, fpm.

Grimitlin et al. (1986) used both scaled physical models and full-scale field facilities to study the distribution of velocities and temperatures in the occupied zone with air supplied into rooms through grilles and ceiling-mounted diffusers. For different air supply methods, they found the ranges for room areas ventilated by one air jet that provide the most uniform distribution of air velocities and temperatures throughout the occupied zone. Researchers suggested that the velocities and temperatures in the occupied zone can be described by evaluating the air jet's maximum velocity V_x and temperature difference Δt_x, at the point within the room where the jet enters the occupied zone. In the case of velocities, the deviation created from the maximum value can be described as:

$$V_{o.z.} = V_x - 2\sigma_v \qquad (4)$$

The minimum air velocity in the occupied zone, V_{min} can be obtained from:

$$V_{min} = V_x - 4\sigma_v \qquad (5)$$

In this way, the resulting range of velocities in the occupied zone is based on the maximum velocity in the jet entering the occupied zone determined from the air-jet theory. Using two or four times the experimental deviation will provide confidence that 95% of the average occupied zone velocities will be within the predicted range.

A temperature equation for the occupied zone is organized the same way. Grimitlyn et al. (1986) recommended using four times the experimental deviation of the temperature, σ_t. This results in a 95% confidence interval for both maximum max $(t_{o.z})$ and minimum min $(t_{o.z})$ air temperatures, °F, in the occupied zone:

$$\max (t_{o.z}) - \min (t_{o.z}) = 4\sigma_v \qquad (6)$$

Through this technique, the designer can determine if velocity and temperature are within an acceptable range. If they are not, a different diffuser velocity and air temperature, and /or diffuser type will need to be selected.

Obstructions influence the uniformity of velocity and temperature distributions and will be accounted for by corresponding values of deviation variables. This approach allows designing air distribution based on a combination of air-jet theory and mathematical (statistical) relationships of velocities and temperatures in the occupied zone.

Ventilation effectiveness. Ventilation effectiveness describes an air distribution system's ability to remove internally generated pollutants from the ventilated space and to remove excessive heat or to compensate for heat losses. Ventilation effectiveness E_v can be defined as (ASHRAE 62.1 2019)

$$E_v = (C_{ext} - C_o.)/(C_{o.z.} - C_o.) \qquad (7)$$

where:

E_v = ventilation effectiveness
C_{exh} = contaminant concentration at the exhaust, lb/ft^3
C_o = contaminant concentration at the supply, lb/ft^3
$C_{o.z.}$ = average contaminant concentration in the occupied zone, lb/ft^3

It can be used to calculate the minimum airflow rate required for contaminant control in ventilated space:

$$Qo = Qexh + \frac{G(1-\varepsilon) - Qexh\ (Co.z. - Co.)}{Ev\ (Co.z. - Co.)} \qquad (8)$$

where:

Q_0 = air supply rate; cfm
Q_{exh} = local ventilation exhaust rate, c
G = rate of pollutant generation, lb/min
ε = local ventilation capturing effectiveness (unitless)
$C_{o.z.}$ = desired concentration of fume, gas, particulates in the occupied zone air, lb/ft^3: $C_{o.z.} <$ TLV$^{®}$
TLV$^{®}$ = Threshold Limit Value of the contaminating fume, gas, or particulates, lb/ft^3
C_o = concentration of fume, gas, and particulates in the supply air, lb/ft^3

Table 4-1 lists default values of E_v for the air distribution configurations described in the table. The reference E_v value of 1 is typical of ideal mixing in the zone. The strategy of removing contaminants or displacing contaminants from the breathing zone may result in an effective E_v value greater than 1, which is typical of stratified or localized systems.

Table 4-1. Ventilation effectiveness EV (ASHRAE 2019).

Air Distribution Configuration	E_z
Well-Mixed Air Distribution Systems	
Ceiling supply of cool air	1.0
Ceiling supply of warm air and floor return	1.0
Ceiling supply of warm air 15 °F (8 °C) or more above space temperature and ceiling return	0.8
Ceiling supply of warm air less than 15 °F (18 °C) above average space temperature where the supply air-jet velocity is less than 150 fpm (0.8 m/s) within 4.5 ft (1.4 m) of the floor and ceiling return	0.8
Ceiling supply of warm air less than 15 °F (8 °C) above average space temperature where the supply air-jet velocity is equal to or greater than 150 fpm (0.8 m/s) within 4.5 ft (1.4 m) of the floor and ceiling return	1.0
Floor supply of warm air and floor return	1.0
Floor supply of warm air and ceiling return	0.7
Makeup supply outlet located more than half the length of the space from the exhaust, return, or both	0.8
Makeup supply outlet located less than half the length of the space from the exhaust, return, or both	0.5
Stratified Air Distribution Systems (Section 6.2.1.2.1)	
Floor supply of cool air where the vertical throw is greater than or equal to 60 fpm (0.25 m/s) at a height of 4.5 ft (1.4 m) above the floor and ceiling return at a height less than or equal to 18 ft (5.5 m) above the floor	1.05
Floor supply of cool air where the venial throw is less than or equal to 60 fpm (0.25 m/s) at a height of 4.5 ft (1.4 m) above the floor and ceiling return at a height less than or equal to 18 ft (5.5 m) above the floor	1.2
Floor supply of cool as where the vertical throw is less than or equal to 60 fpm (0.25 m/s) at a height of 4.5 ft (1.4 m) above the floor and ceiling return at a height greater than 18 ft (5.5 m) above the floor	1.5
Personalized Ventilation Systems (Section 6.2.1.2.2)	
Personalized air at a height of 4.5 ft (1.4 m) above the floor combined with ceiling supply of cool air and ceiling return	1.40
Personalized air at a height of 4.5 ft (1.4 m) above the floor combined with ceiling supply of warm air and ceiling return	1.40
Personalized air at a height of 4.5 ft (1.4 m) above the floor combined with a stratified air distribution system with nonaspirating floor supply devices and ceiling return	1.20
Personalized air at a height of 4.5 ft (1.4 m) above the floor combined with a stratified air distribution system with aspirating floor supply devices and ailing return	1.50

<u>Heat supply/removal efficiency.</u> This criterion is similar to ventilation effectiveness and describes an air distribution system's ability to remove excessive heat or to compensate for heat losses. ASHRAE (1995) defines heat removal efficiency, K_t, as

$$K_t = (t_{exh} - t_o)/(t_{o.z.} - t_o) \tag{9}$$

where:

t_{exh} = air temperature at the exhaust, °F
t_o = supply air temperature, °F
$t_{o.z.}$ = average occupied zone air temperature., °F

Heat supply/removal efficiency, which can be used to calculate the minimum airflow rate required for air heating or for assimilating surplus heat, can be obtained from the following equation based on the heat balance in the premises:

$$Q = Q_{exh} + \frac{W - \rho C_p Q_{exh}(t_{o.z.} - t_o)}{C_p K_t (t_{o.z.} - t_o)} \tag{10}$$

where:

W = surplus heat generated in the room, Btu/min
ρ = air density (lb/ft^3)
C_p = specific heat (Btu/lb °F)
$t_{o.z.}, t_o$ = air temperature in the occupied zone and supplied air accordingly, °F

Table 4-2 provides default values of K_t for the air distribution configurations described in the table.

**Table 4-2. Heat and moisture removal efficiency coefficients for mechanically
ventilated spaces with insignificant heat loads.**

Air Supply Method*	Heat/moisture removal coefficients, K_t/K_w			
	Air Change per Hour (ACH)			
	3	5	10	>15
Mixing air supply (ASHRAE 1995).				
Concentrated air jets	0.95/1.1	1.0/1.05	1.0/1.0	1.0/1.0
Concentrated air jets with vertical and/or horizontal directing jets	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0
Inclined air jets from a height of				
Greater, than 13 ft (4 m)	1.15/1.4	1.1/1.2	1.0/1.1	1.0/1.0
Less, than 13 ft (4 m)	1.0/1.2	1..0/1.1	1.0/1.05	1.0/1.0
Through ceiling-mounted air diffusers with				
Radial/linear attached to the ceiling jets	0.95/1.1	1.0/1.05	1.0/1.0	1.0/1.0
Conical/compact jets	1.05/1.1	1.0/1.05	1.0/1.0	1.0/1.0
Air supply with thermal stratification (VGAI 2000)				

Thermal displacement
    Heating mode
    Cooling mode
	$K_t = 1.0$
	$K_t = 1.0\text{-}1.4$			

Active displacement	$K_t = 1.2\text{-}1.8$			
Unidirectional Flow	$K_t > 2$			
*Note: Insignificant heat load is defined as below 2.2 Btu per hour per ft^3.				

The reference K_t value of 1 is typical of ideal mixing in the zone. The strategy of creating temperature stratification along the space height may result in an effective K_t value greater than 1, which is typical for thermal displacement or localized systems. Providing space air heating by supplying warm air under the ceiling with not sufficient air velocity will result in K_t value smaller than 1.

Water vapor removal efficiency. Water vapor removal efficiency, K_w, was defined (ASHRAE 1995) as

$$K_t = (W_{exh} - W_o)/(W_{o.z.} - W_o) \hspace{4cm} (11)$$

where:

    K_w =   ventilation effectiveness
    W_{exh} =   humidity ratio at the exhaust, lb water per lb or dry air
    W_o =   humidity ratio in supply air, lb water per lb or dry air
    $W_{o.z.}$ =   average occupied zone air humidity ratio. lb water per lb or dry air.

Table 4-2 lists the default values of K_w.

Chapter 5. Methods of Air Distribution Design

Consulting engineers use a variety of methods to design room air diffusion and to select and size air diffusers. The following approaches are most common (Zhivov 1992):

- Perfect mixing is often assumed, meaning air distribution is uniform within a room. This approach is acceptable for many situations. However, there is no assurance that the clean air actually reaches the occupants nor is there a method of predicting thermal and air quality parameters as they vary throughout the room.
- A design method can employ empirical relations determined through research and testing. The ADPI is one example of this technique. Air velocity and effective draft temperature (a combination of local temperature differences from the room air average) are measured and converted into ADPI. ADPI applies only to cooling conditions. There are many more of these types of indices outlined for the design and analysis of occupied regions, each with their limitations.
- Computational fluid dynamics (CFD) codes predict temperature, velocity, and contaminant distributions. Boundary descriptions complexities and computational requirements limit general acceptance in the design community, although some specialty system designs have benefited from CFD simulations. It is especially difficult to adequately describe the boundary conditions. CFD has great potential but has not evolved into a design tool except for specialty and high-value applications. The use of CFD codes for practical three-dimensional computation requires expertise, experience, and computational expertise not always common to typical designers.
- Air-jet theory is a well-practiced technique for analyzing and designing human environments. To use air-jet theory, the room is divided into a jet region and mixing zones (as opposed to the assumption used for "perfect mixing" design). Design methods based on air-jet theory allow for the prediction of extreme values of air velocities and air temperatures in the occupied zone. Current air-jet theory techniques account for the effect of buoyancy, confinement, and jet interaction.
- An analytical model may be used for passive thermal displacement ventilation design. This approach is based on information on airflows created by supply devices, convective plumes above heat sources, and ventilation effectiveness of displacement ventilation.

This document focuses primarily on two latter methods of air distribution design.

Chapter 6. Elements of Room Air Distribution

Air Supply with Air Jets

<u>Classification</u>**.** As a rule, air supplied to rooms through various types of outlets (e.g., grille-like, ceiling diffusers, perforated panels) is distributed by turbulent air jets. These air jets are the primary factors affecting room air motion. For more information on the relationship between the air jet and the occupied zone air motion, see IVDG (2000). If air is not obstructed by walls, ceiling, or internal obstructions, it is considered a ***free jet***. If the air jet is attached to a surface, it is an ***attached jet***.

Characteristics of the air jet in the room might be influenced by reverse flows created by the same jet entraining the ambient air. This air jet is called a ***confined jet.*** If the temperature of the supplied air is equal to the temperature of the ambient room air, the air jet is called an ***isothermal jet.*** A jet with an initial temperature different from the temperature of the ambient air is called a ***non-isothermal jet.*** The air temperature differential between supplied and ambient room air generates thermal forces in jets, affecting (1) the trajectory of the jet, (2) the location at which the jet attaches and separates from the ceiling/floor, and (3) the throw of the jet. The significance of such effects depends on the ratio between thermal buoyancy and inertial forces (characterized by the Archimedes number Ar.)

Depending on the diffuser type, air jets can be classified as:

- Compact air jets, which are formed by cylindrical tubes, square or rectangular nozzles, with a small aspect ratio, unshaded or shaded by perforated panels, grilles, etc. The maximum velocity in the cross-section of the compact jet is on the axis.
- Linear air jets are formed by slots or rectangular openings with a large aspect ratio. The jet flows are approximately two-dimensional. Air velocity is symmetric in the plane at which air velocities in the cross-section are maximum. At some point from the diffuser, linear jets tend to transform into compact jets.
- Radial air jets are formed when the ceiling cylindrical air diffusers with flat disks or multi-diffusers direct the air horizontally in all directions.
- Conical air jets are formed by cone-type or regulated multi-diffuser ceiling-mounted air distribution devices that have an axis of symmetry. The vectors of air velocities are parallel to the conical surface (with an angle at the top of the cone equal to 120 degrees). Maximum velocities in cross-sections perpendicular to the axis occur in the conical surface.
- Incomplete radial jets are supplied through outlets with grilles having diverging vanes and have coerced angle of expansion. At some distance, this kind of jet tends to transform into a compact one.
- Swirling jets, which are supplied to the room through air diffusers with vortex-forming devices creating rotational motion, have tangential as well as radial velocity vectors. Depending on the type of air diffuser, swirling jets can be compact, conical, or radial (Figure 6-1).

Isothermal Free Jet

Different types of free jets and air diffusers can yield similarities in the resulting flows. Four major zones are recognized along a free jet. These zones, as described by Tuve (1953) may be roughly defined in terms of the maximum or center core velocity that exists at the jet cross-section being considered (Figure 6-2a).

Zone 1 is a short zone, extending about two to six diffuser diameters (for compact and radial jets) of slot width (for linear jets) from the diffuser face. In this zone, the centerline velocity of the jet remains nearly equal to the original supply velocity throughout its length.

Zone 2 is a transition zone, and its length depends on the diffuser type, diffuser's aspect ratio, initial airflow turbulence, and so forth. Within this zone, the maximum velocity may vary inversely with the square root of the distance from the outlet. For practical purposes, some researchers suggest using the simplified scheme of the jet with a transition cross-section.

Zone 3 is the zone of fully established turbulent flow. The length of this zone depends on the air jet shape, the type and size of the supply air diffuser, the initial velocity of the air jet, and the turbulence characteristics of the ambient air, and may be between 25 and 100 equivalent air diffuser diameters (width for slot-type air diffusers).
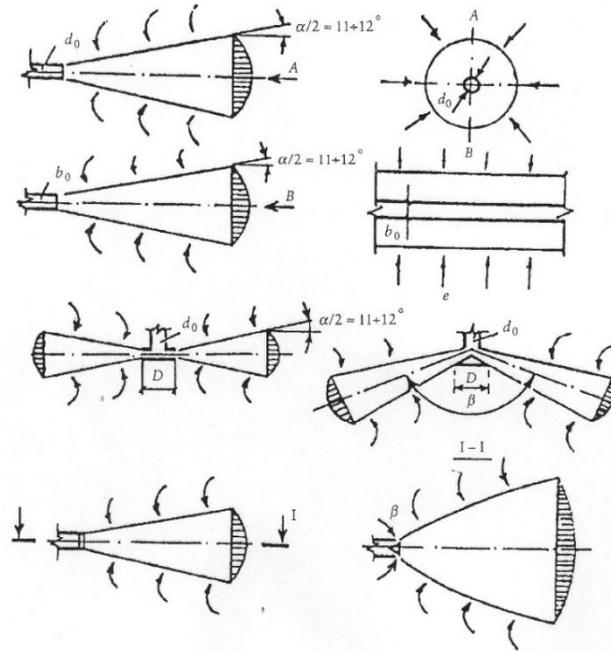
Figure 6-1. Types of diffuser jets: (a) compact, (b) linear, (c) radial, (d) incomplete radial, (e) conical (Grimitlyn 1994).
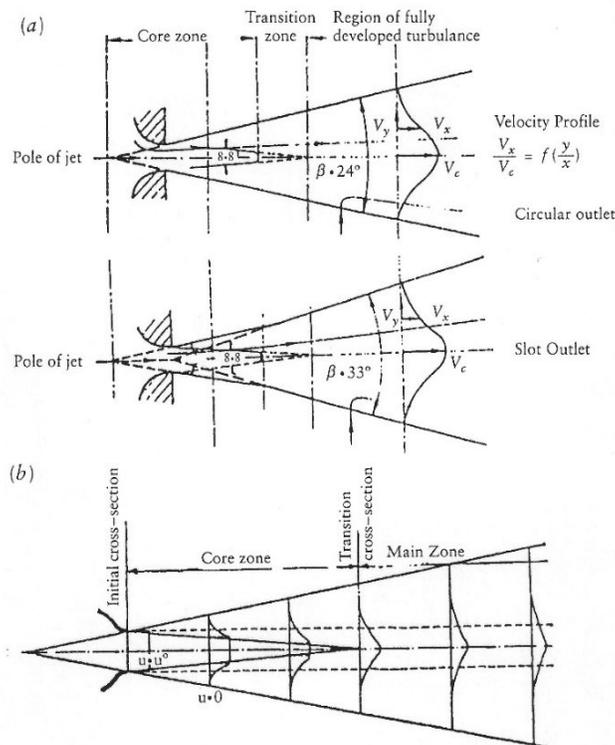
Figure 6-2. Turbulent jet: (a) schematic with four zones, (b) simplified jet schematic (IVDG 2001, Ch. 7).

Zone 4 is a terminal zone in which the residual velocity decays quickly into large-scale turbulence. Within a few diameters, the air velocity becomes less than 50 fpm (0.254 m/s). Though this zone has been studied by several researchers, its characteristics are still not well understood.

Zone 3 has major engineering importance since it is usually in this zone that the jet enters the occupied region. Therefore, further discussion will be primarily focused on this zone. For more information related to other jet zones, see the *ASHRAE Handbook of Fundamentals* (ASHRAE 2021a) and IVDG (2001).

In zone 3, the maximum or centerline velocity of the isothermal jet can be determined from the following equations:

$$V_x = K_1 V_0 A_o^{1/2}/X \text{ (for compact and radial jets)} \tag{12}$$

$$V_x = K_1 V_o (H_o/X)^{1/2} \text{ (for linear jet)} \tag{13}$$

where:

V_x = centerline velocity, fpm
K_1 = centerline velocity decay coefficient[*]
V_o = average initial velocity at discharge, fpm
A_o = free area, core area or neck area of air diffusor, ft^2
H_o = height of the slot, ft
X = distance from the face of the outlet, ft.

Jet throw. Diffuser jet through, L, is a parameter used in a diffuser sizing defined as the distance from the diffuser face to the jet cross-section where the centerline velocity equals a terminal velocity V_x (V_x is often assumed to be 50 fpm [15.2 m/min]). Therefore, the throw, L, can be determined from the velocity decay equation with the V_x equal to the terminal velocity:

$$L = K_1 (A_o)^{1/2} V_o / V_x \text{ (for compact and radial jets)} \tag{14}$$

$$L = H_o (K_1 V_o /V_x)^2 \text{ (for linear jets)} \tag{15}$$

Entrainment ratio. An air jet supplied into a ventilated space entrains ambient air resulting in increased airflow rate along its axial distance. The increase in the airflow moving across the jet's cross-section is characterized by the parameter called an ***entrainment ratio***. This jet characteristic depends on several parameters, including the type of air jet, velocity decay characteristic of air diffuser, and initial jet turbulence. The amount of air entrained by the jet affects the overall air movement in the space; this can also be used to evaluate mass transfer between different zones within the ventilated space. For a given size of diffuser opening, the entrainment ratio is proportional to the distance X along the jet axis from the air diffuser and can be calculated using the following equations: K_1

$$O_x/Q_o = = \frac{2}{K1} \frac{X}{\sqrt{Ao}} \text{ (for a compact jet)} \tag{16}$$

$$O_x/Q_o = \frac{\sqrt{2}}{K1} \left(\frac{X}{Ho}\right)^{1/2} \text{ (for a linear jet [per a unit of a slot length])} \tag{17}$$

$$O_x/Q_o = = \frac{\sqrt{2}}{K1} \frac{X}{\sqrt{Ao}} \text{ (for a radial jet)} \tag{18}$$

where:

Q_x = airflow rate along the jet, cfm
Q_o = initial supply airflow rate, cfm
K_1 = centerline velocity decay coefficient[*]
A_o = free area, core area or neck area of air diffusor, ft^2
H_o = height of the slot, ft
X = distance from the face of the outlet, ft

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[*] See Table 4-2 for examples using generic outlets. For specific air diffusers, refer to manufacturers' catalogs.

Equations 5-7 show that, for the same jet type, entrainment ratio is less with a large K_1 than it is with a small K_1. Radial and conical diffuser jets have a smaller entrainment ratio than compact and incomplete radial jets with the same K_1 value. Linear diffuser jets have a smaller entrainment ratio than radial and conical jets.

Non-Isothermal Jets

The heating and/or cooling of spaces can be coupled with their ventilation. When the temperature of supplied air is different from the room air temperature, the behavior of the diffuser air jet is affected by thermal buoyancy due to the air density difference. Buoyant flow can be classified as

- Positive buoyant jets, when a buoyant force acts in the direction of the jet supply velocity at the origin, i.e., upward-projected heated air jet and or downward-projected cooled jet
- Negative buoyant jets when the buoyant force acts in the opposite direction, i.e., downward-projected heated air jet or upward-projected cooled air jet
- Nonbuoyant jets when the effect of buoyancy is negligible
- Plume when the buoyant force completely dominates the flow, as for flow generated with a heat source.

Along with a constant velocity zone (zone 1) there is a constant temperature zone in the jet. Heat diffusion in a jet is more intense than momentum; therefore, the core of constant temperatures fades away faster than that of constant velocities and the temperature difference profile is flatter than the velocity profile. The length of the zone with constant temperatures (Figure 6-2) is therefore shorter than the length of the constant velocity zone (zone 1) (Abramovich 1940, Koestel 1954, Grimitlyn 1970).

The centerline temperature differential within a zone of fully established turbulent flow (zone 3) of a non-isothermal horizontal jet can be derived by:

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \frac{\sqrt{A_o}}{X}$$

(for compact and radial jets)   (19)

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \sqrt{\frac{H_o}{X}}$$ (for a linear jet)   (20)

where:

t_x = centerline air temperature
$t_{o.z.}$ = average occupied zone air temperature
K_2 = centerline temperature decay coefficient[*]
t_o = discharge air temperature
A_o = free area, core area or neck area of air diffusor
H_o = height of the slot
X = distance from the face of outlet.

According to Abramovich (1940), Regenscheight (1959) and Shepelev (1961), the relation between velocity distribution and temperature distribution in the cross-section of non-isothermal compact linear or radial jets within zone 3 can be expressed as

$$\Delta t / \Delta t_x = (t - \overline{t_\infty})/(t_x - \overline{t_\infty}) = \sqrt{V/V_x} \,.$$

(21)

To characterize the relationship between the buoyancy forces and momentum flux in different cross-sections of the non-isothermal jet at some distance, X, Grimitlyn (1994) proposed a local Archimedes number:

$$Ar_x = (g \, X \, \Delta t_x)/(V_x^2 \, T_\infty)$$

(22)

where g is the acceleration due to the buoyancy.

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[*] See Table 4-2 for examples using generic outlets. For specific air diffusers, refer to manufacturers' catalogs.

Equations for the local Archimedes number were derived for different types of air jets are:

$$Ar_x = K_2/K_1{}^2 \, (Ar_o) \left(\frac{X}{\sqrt{Ao}}\right)^2 \text{ (for compact and radial jets)} \tag{23}$$

$$Ar_x = K_2/K_1{}^2 \, (Ar_o) \left(\frac{X}{Ho}\right)^{1/2} \text{ (for linear jets)} \tag{24}$$

where the Archimedes number at the outlet is:

$$Ar_o = (g \, \sqrt{Ao} \, \Delta t_o)/(V_o{}^2 \, T_\infty) \tag{25}$$

where T_∞ = average room air temperature, °K.

For practical use, the influence of buoyant forces on temperature and velocity decay in vertical non-isothermal jets, as proposed by Grimitlyn (1982), can be accounted for by the coefficient of non-isothermality:

For compact jets:

$$\frac{V_x}{V_o} = K_1 \frac{\sqrt{A_o}}{X} K_n \tag{26}$$

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \frac{\sqrt{A_o}}{X} \frac{1}{K_n} \tag{27}$$

where K_n be computed as:

$$K_n = \sqrt[3]{1 \pm 2.5\frac{K_2}{K_1^2}Ar_o\left(\frac{X}{\sqrt{A_o}}\right)^2} \tag{28}$$

For linear jets:

$$\frac{V_x}{V_o} = K_1 \sqrt{\frac{H_o}{X}} \, K_n \tag{29}$$

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \sqrt{\frac{H_o}{X}} \frac{1}{K_n} \tag{30}$$

where K_n be computed as:

$$K_n = \sqrt[3]{1 \pm 1.8\frac{K_2}{K_1^2}Ar_o\left(\frac{X}{H_o}\right)^{1.5}} \tag{31}$$

or applying the Ar_x criterion equations for K_n, can be transformed into

$$K_n = \sqrt[3]{1 \pm aAr_x} \tag{32}$$

where a = 2.5 for compact and incomplete radial jets and a = 1.8 for linear jets.

The plus sign in equation 32 corresponds to the situation when the direction of buoyancy and inertia forces coincide, whereas the minus sign corresponds to their counteraction.

The introduction of the local Archimedes criterion helps to clarify when buoyant forces need to be considered and when they can be ignored for practical reasons. Grimitlyn (1982) suggested critical local Archimedes number values, **$Ar_{x.crit}$**, below which a jet can be considered practically unaffected by buoyancy forces: **Ar_x** < 0.1 for a compact jet and incomplete radial jets; **Ar_x** < 0.15 for a linear jet.

The throw, Z_{max}, of downward-projected heated jets can be derived for upward-projected chilled air jets from equations 20 by considering K_n equal to some value, e.g., 0.1:

For compact and incomplete radial jets:

$$\frac{Z_{max}}{D} = \frac{0.63\ K_1}{\sqrt{K_2\ Ar_o}} \tag{33}$$

For linear jets

$$\frac{Z_{max}}{H_o} = 0.67\ \frac{K_1^{4/3}}{(K_2\ Ar_o)^{2/3}} \tag{34}$$

As previously mentioned, buoyancy forces influence the trajectory of horizontally projected air jets or air jets supplied under some angle to the horizontal plane (Figure 6-3). Most non-isothermal air jets studies were devoted to horizontally projected compact air jets. Summary of results of these studies in (Zhivov 1983) shows that non-isothermal air-jet trajectory can be described as

$$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}} tg\alpha_o \pm \psi\ \frac{K_2}{K_1^2}\ Ar_o\ \left(\frac{X}{\sqrt{A_o}}\right)^3 \tag{35}$$

Where Z is a vertical distance from the outlet and X is a horizontal distance. The main difference in most of the equations for the jet trajectory proposed by different researchers is the value of the coefficient ψ. Extensive experimental data obtained for different types of grilles and nozzles and an analysis of the literature indicate that the value of the coefficient ψ can be approximated (Zhivov 1993) as 0.47.

This equation allows the location of the vortex (X_v, Z_v) of the heated or chilled air jet to be calculated as:

$$X_v = \frac{K_1\cos\alpha_o\sqrt{\sin\alpha_o}}{\sqrt{K_2\ Ar_o}} \tag{36}$$

$$Z_v = \frac{2}{3}\frac{K_1\cos\alpha_o(\sin\alpha_o)^{\frac{3}{2}}}{\sqrt{K_2 Ar_o}} \tag{37}$$

and the horizontal distance, X_o, between the center of supply outlet to the point where chilled or heated air is supplied at the angle α_o upward or downward will cross the level of the supply outlet at:

$$X_o = \frac{\sqrt{3}K_1\cos\alpha_o\sqrt{\sin\alpha_o}}{\sqrt{K_2 Ar_o}} \tag{38}$$

Jet Attachment

A jet discharging close to the plane of the ceiling or the wall is common in ventilation practice. The presence of adjacent surface restricts air entrainment from the side of this surface. This results in a pressure difference across the jet, which curves toward the surface. The curvature of the jet increases until it attaches to the surface. This phenomenon is usually referred to as a "Coanda" effect. The attached jet (commonly called a "wall jet") can result from air supply through the outlet with one edge that coincided with a plane of the wall or the ceiling. A jet supplied at some distance from the surface or at some angle to this surface also can become attached (Figure 6-3).
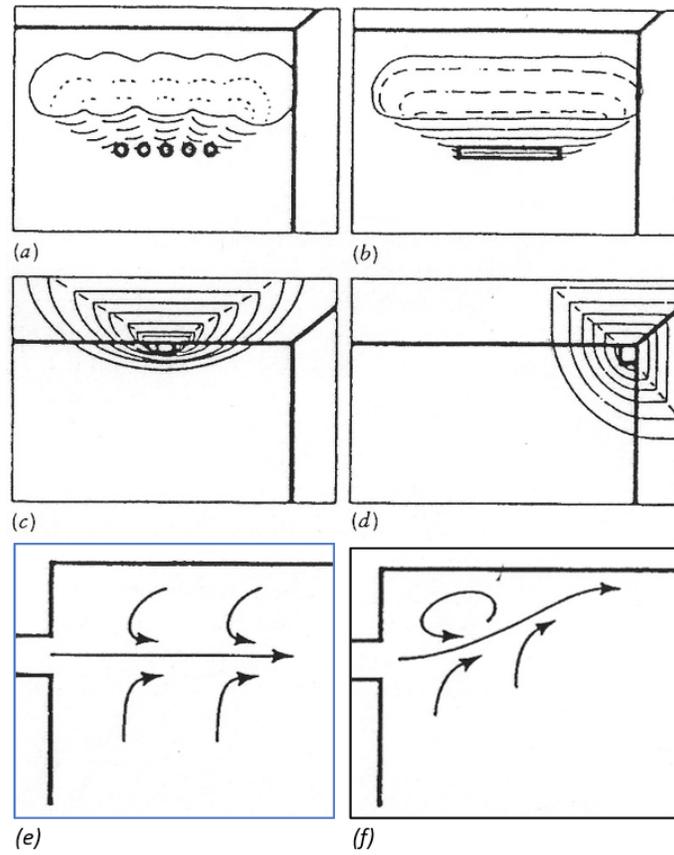
Figure 6-3. Jet attachment with air supply through outlets located at some distance from the surface: (a) multiple jets, (b) long slot, (c) rectangular jet, (d) rectangular corner jet, (e) general schematic. Reproduced from Nielsen (1981)

The results of experimental studies of compact, linear, and radial wall jets summarized by Grimitlyn (1982) indicate that the parameter K_{wal}/K_1, which reflects the influence of the wall on the velocity decay along the jet, increases from 1 to 1.4 with a distance from the outlet. For example, for a compact jet $K_{wal}/K_1 = 1$ when $X < 5do$; for a linear jet K_{wal}/K_1 reaches its maximum value equal to 1.4 only at $X < 20Ho$, where H_0 is an outlet width.

It is not uncommon to supply air to the room with jets attached both to the ceiling and to the wall surfaces (Nielsen 1981, Grimitlyn 1982). Air jets can be parallel to both surfaces or can be directed at some angle to one or both surfaces (Figure 6-3). Studies of compact wall jets supplied parallel to both surfaces reported by Grimitlyn (1994) show that the correction factor value is in the range from 1.6 to 1.7, which means that restriction of entrainment from two sides reduces velocity decay by 20% to 30% compared to the case with only a wall jet.

When a jet is supplied at some distance from the surface, the attachment occurs when the distance between the outlet and the surface is below a critical distance, otherwise, the jet will propagate as a free jet (Awbi 1991). If the jet attaches to the surface, the flow downstream of the attached point is similar to that of a wall jet. For a compact isothermal jet, the critical distance for jet attachment to the surface is $Lcrit = 6A,/^2$ (Farquharson 1952). For $L_{crit} < 6A/^2$, the velocity decay coefficient K_1 becomes greater than it would be in the case of free jet such that it should be corrected to compensate for surface proximity.

Jet Separation

When the temperature of the attached air jet is lower than the temperature of the ambient air, this jet will remain attached to the ceiling until the downward buoyancy force becomes greater than the upward static pressure (Coanda force). At this point, the jet separates from the ceiling and begins a downward curving trajectory (Miller 1991) (Figure 6-4).
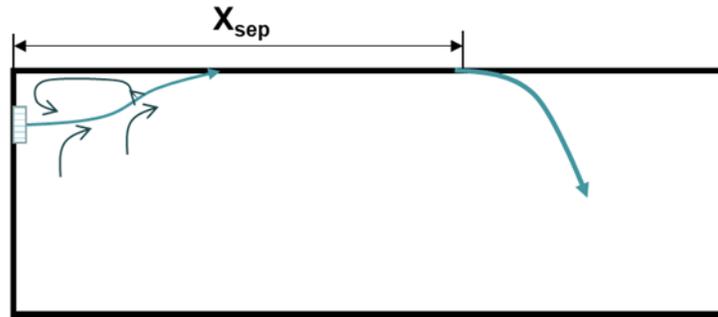
Figure 6-4. Jet separation from the ceiling under the buoyancy forces.

Studies of non-isothermal jets (Grimitlyn 1970, Schwenke 1976, Nielsen and Moller 1987, Miller 1991, Anderson et al. 1991, Kirkpatrick et. al. 1991) showed that the distance to the point of jet's separation can be computed using

$$\frac{X_{sep}}{\sqrt{A_o}} = \frac{a}{(Ar_o)^b} \tag{39}$$

For linear diffuser jets (Rodahl 1977), a = 2.5H$_o$ and b = ⅔. For the compact diffuser jet, a = 1.6 $\sqrt{A_o}$ and b = 0.5 (Anderson et al. 1991; Kirkpatrick 1991). According to theoretical analysis and experimental data collected by Grimitlyn (1970), the separation distance of jets could be expressed by the following equations similar to equation 27 considering diffuser characteristics K_1 and K_2.

For compact and incomplete radial jets:

$$X_{sep} = \frac{0.55\, K_1\sqrt{A_o}}{\sqrt{K_2 Ar_o}} \tag{40}$$

For linear jets

$$X_{sep} = \frac{0.4 K_1^{\frac{4}{3}} H_o}{(K_2 Ar_o)^{\frac{2}{3}}} \tag{41}$$

For radial jets

$$X_{sep} = \frac{0.45\, K_1\sqrt{A_o}}{\sqrt{K_2 Ar_o}} \tag{42}$$

Depending on the type of the mechanical system used, a change in internal thermal and contaminant loads and in outside air conditions will result in a change of supply air flow and correspondingly supply air velocity, a change in supply air temperature, or in a cycling performance of the air conditioner that periodically turns on and off.

Cooled supply air is typically drier than the room air. The attached jet cools the surface it is attached to, but the surface temperature above the jet remains below the dew point temperature. With the rapid change in the airflow or temperature of the supply air, the separation point of the jet will change, which will result in the cold ceiling surface being surrounded by the room air having higher humidity. This can result in moisture condensation on this part of the surface and mold growth (Figure 6-5a). When the AC system is designed for cycling performance, condensation will occur not only on the ceiling surface, but also on the air supply diffuser and internal part of the supply ductwork (Figure 6-5b,c), which will result in mold growth on the ceiling and water damage caused by condensed water dripping on the floor.

(a)

(b)

(c)
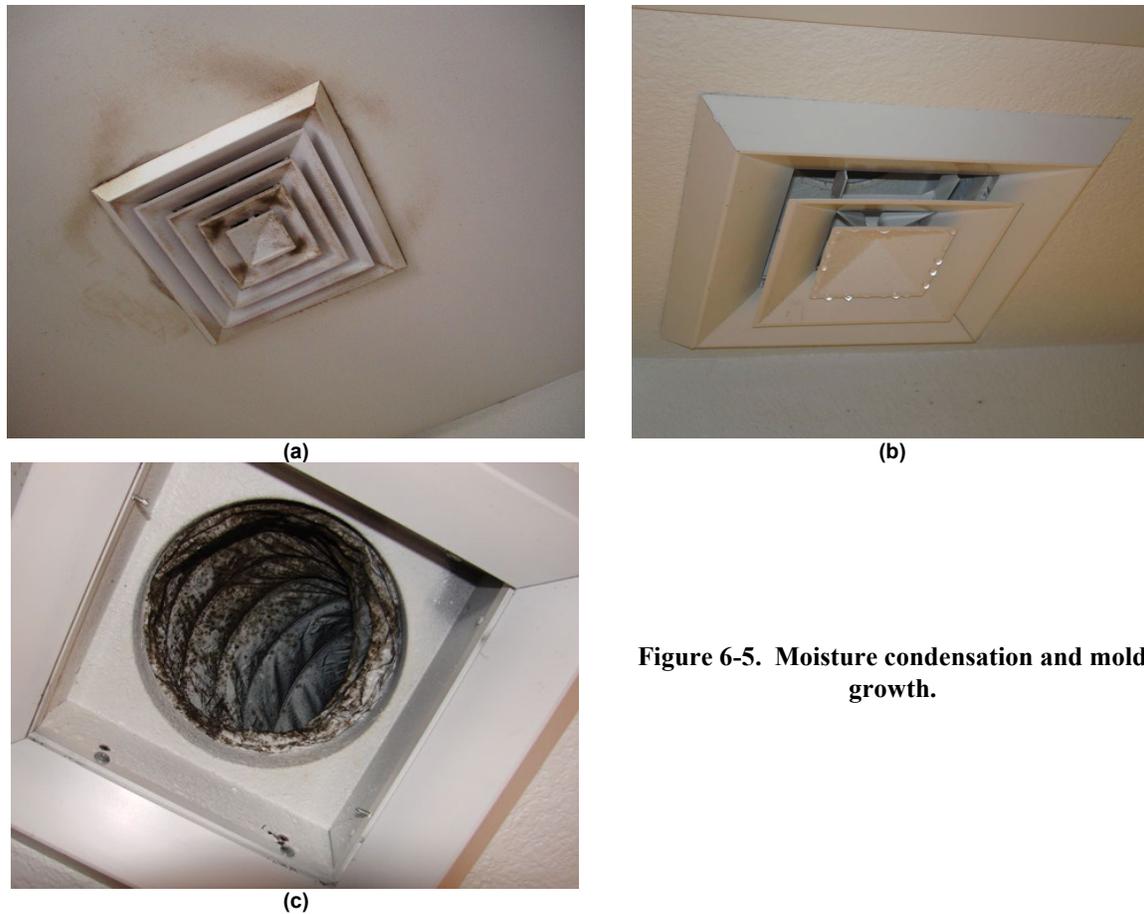
Figure 6-5. Moisture condensation and mold growth.

To avoid condensation and mold growth due to unstable air jet behavior, the following measures are recommended:

- Reduce supply air temperature difference using a dedicated outdoor air supply (DOAS) with a complementing cooling system.
- Increase return air to reduce supply air temperature and increase supply air velocity. Make sure that the return air is filtered to eliminate room contaminants returning back to the room, use ASHRAE Epidemic Task Force recommendation on the level of return air filtration.
- In hot and humid climates, avoid air supply using attached air jets.
- When possible and economical, use displacement, underfloor or localized air supply. Due to higher heat and contaminant removal efficiency, these methods reduce required supply airflow and reduce contaminant movement across the ventilated room.

Figure 6-6. Textile air dispersion diffusers can eliminate the probability of condensation by using porous fabric diffusing the air (courtesy of Duct Sox Co.).

Air Diffusers and Their Performance Characteristics

A wide range of air diffusion devices can be used with different air distribution methods (e.g., Figure 6-6). Diffuser manufacturers use ASHRAE Standard 70, *Method of Testing the Performance of Air Outlets and Air Inlets* (ASHRAE 2021b) to acquire listed product data. This section lists the most typical air diffusers.

Grilles are one of the most universal types of air diffusers (Figures 6-7 and 6-8). They can have one or two rows of vanes: vertical or horizontal and different aspect ratios and vane ratios. Vanes affect grill performance if their depth is at least equal to the distance between the vanes.

A grille discharging air uniformly forward (vanes in a straight position) has a spread of 14 to 24 degrees, depending on the type of diffuser, duct approach, and discharge velocity. Turning the vanes influences the direction and the throw of the discharged airstream. The parallel horizontal vanes direct the air stream vertically within 45 degrees. Note that, if the vane ratio is less than two, the jet inclination will be smaller than the angle of the vanes.

Vertical vanes can be used to spread the air horizontally, and horizontal vanes can be used to spread the air vertically. A grille with diverging vanes (vertical vanes with uniformly increasing angular deflection from the centerline to a maximum at each end of 45 degrees) has a spread of about 60 degrees, which reduces the throw considerably. With increasing divergence, the quantity of air discharged by a grille for a given upstream total pressure, decreases.

A grille with converging vanes (vertical vanes with uniformly decreasing angular deflection from the centerline) has a slightly higher throw than a grille with straight vanes, but the spread is approximately the same for both settings. The airstream converges slightly for a short distance in front of the outlet and then spreads more rapidly than air discharged from a grille with straight vanes.
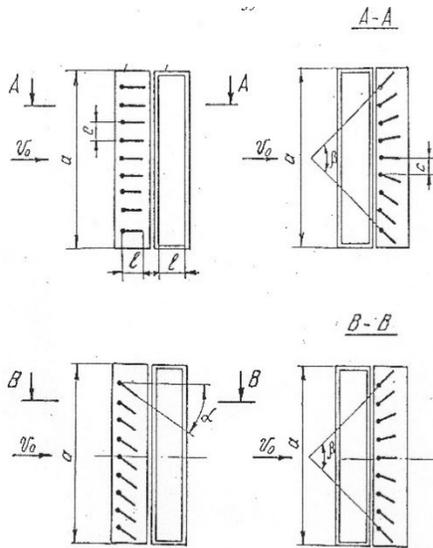
Figure 6-7. Schematic of the double grill with a straight frame of both jet ccharacteristicsand jet direction controls: c/l = 1, wide edge toward the airflow: top α=0, β = 90deg; bottom α= 30deg, β = 90deg.
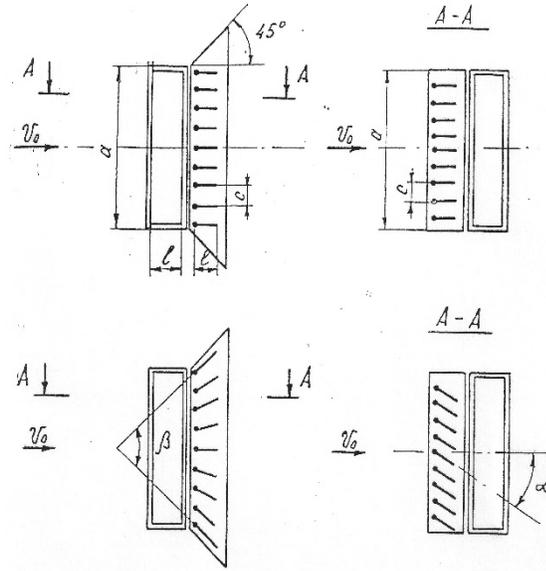
Figure 6-8. Schematic of the double grill with a straight frame of the jet direction control and aone-wayy diffusor-type frame of the jet characteristics control: c/l = 0.67, wide edge toward the airflow: top α=0, β = 0; bottom α= 30deg, β = 90deg.

Ceiling-mounted air diffusers can be round, rectangular, or linear and can have outlets covered with a grille, perforation, plaque, vanes forming multiple slots, or a swirl insert (Figure 6-9). There are controlled and non-controlled ceiling-mounted air diffusers. Depending on their design, they can form attached radial, concentrated, or linear jets as well as non-attached conical or concentrated air jets. Rectangular air diffusers with triangle or four-sided grilles form non-uniform circular flow and can be considered as two, three or four separate jets.

For VAV applications with considerable air volume and initial temperature differential range, air diffusers with controlled outlet area and/or direction of air supply, as well as those with a variable induction of room air, create better performance within the year-round cycle of system operation. For application in industrial and commercial facilities with high ceilings, these diffusers can be mounted on duct drops (with an installation height of 10 to 15 ft [3 to 9.4m]) supplying compact conical or non-attached radial jets. Non-attached radial and conical jets typically collapse into conical or compact jets under the influence of buoyant forces.
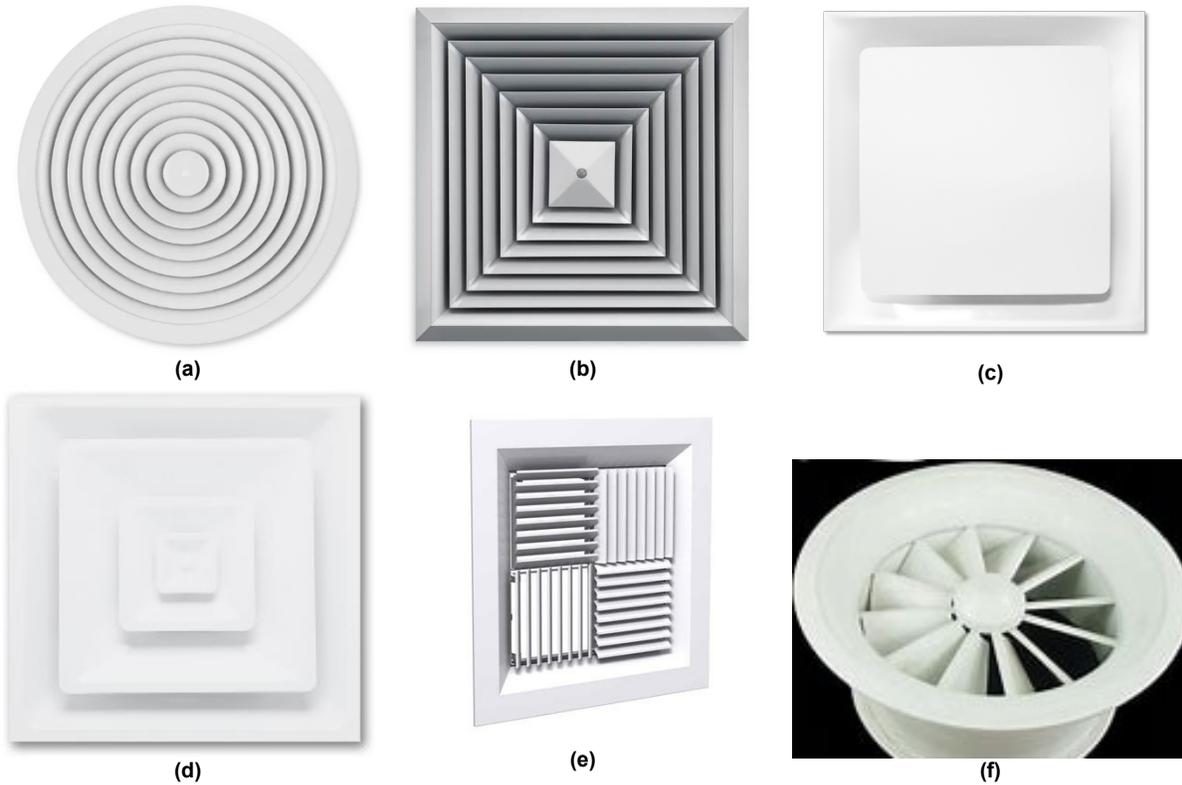
Figure 6-9. Ceiling air diffusers: (a) with fixed air control blades for horizontal air supply with radial jets (TROX); (b) with fixed air control blades for horizontal air four-way air supply (TROX); (c) with an adjustable faceplate for horizontal radial and conical jet supply (Krueger); (d) with fixed concentric cones for conical jet supply (KRUEGER); e - adjustable diffuser for four-way air supply (PRICE); (f) round diffuser with a swirl insert (VENTECH).

Round, square, and rectangular nozzles with an outlet size from 4 in. to 4 ft (10.2 cm to 1.2 m) are commonly used for different applications: from small residential rooms to large atriums, sports halls, and industrial buildings. Converging nozzles form air jets with considerably higher throw and lower noise levels compared to other air diffusers (Figure 6-10). Diverging nozzles are used to supply compact jets with increased angles of divergence and reduced throw. Typically, the latter type of jet can be achieved either by placing two or more concentric cones at the supply side of the air diffuser or by placing a swirl insert the straight nozzle.
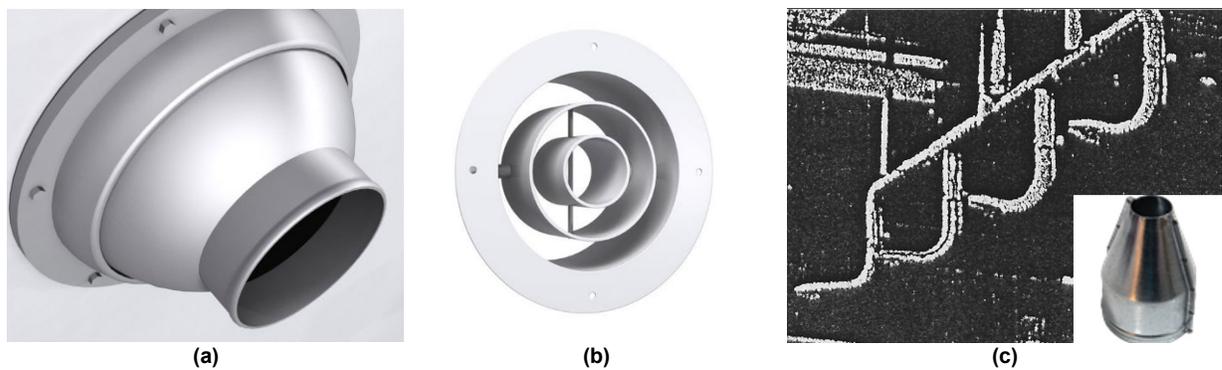
Figure 6-10. Nozzle diffusers: (a) adjustable round diffuser (PRICE); (b) diffuser with concentric nozzles; (c) nozzles for Dirivent (see Fig 3-3c) air supply system (Courtesy of Flakt Group).

Perforated panels are used to supply air either directly into the occupied zone for displacement ventilation systems, or vertically downward. (Figure 6-11). They discharge air at low velocities (40 to 100 fpm [0.2 to 0.5 m/s]) and low turbulence. A diffuser panel is much more than a simple perforated plate or a filter mat. Simple perforated plates usually produce an irregular flow of supply air and cannot be used as supply air devices. Panels using filter mats have a problem; the filter mats are blackened by particles in the supply air. Recently a new generation of perforated panels was introduced:

1. with induction chambers that allow mixing supply air with room air inside air diffuser housing. This design allows to supply air with a greater air temperature difference without causing discomfort in the occupied zone, and

2. with internal deflectors to adjust the flow direction. These panels are capable of decreasing the restricted zone (zone with abnormal velocities) in front of the air diffuser.

(a)   (b)   (c)

(d)
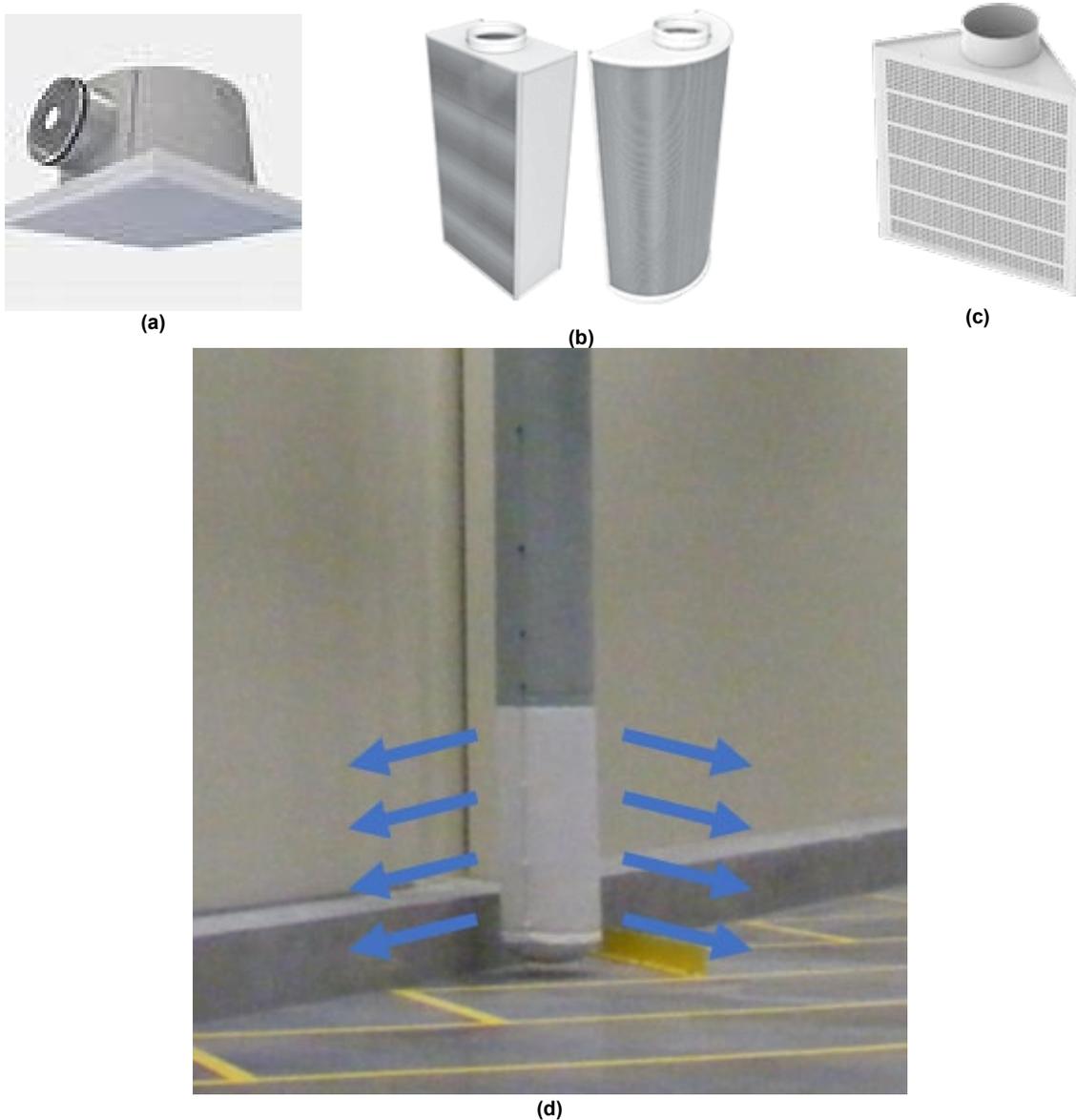
Figure 6-11. Perforated panels: (a) ceiling perforated panel for hospital operating room/cleanroom (Halton); (b) flat and half-round displacement ventilation diffusers (Metalaire), (c) flat face diffuser for corner application (Titus); (d) Round Textile Air Dispersion Diffuser using 100% textile porosity for air diffusion DuctSox).

Perforated ducts are round or rectangular with partial or complete perforated/slotted walls. They are primarily used to supply air in spaces where a high air change rate is required and where air velocities in the occupied zone are limited due to process limitations (e.g., to prevent contaminant spillage from local exhausts). The supply surface may be created either by perforating the duct wall, or by cutting the incomplete holes in the wall and bending metal peaks inside/outside the duct (to deflect air jets in the right direction from the duct surface), or by stamping converging nozzles in the desired areas of the sheet metal bend that is used to form the spiro duct. A similar effect can be achieved using textile permeable ducts or textile ducts with nozzles or with groups of holes (also known as "textile air dispersion systems") (see Figure 6-12).

(a)

(b)

(c)

(d)
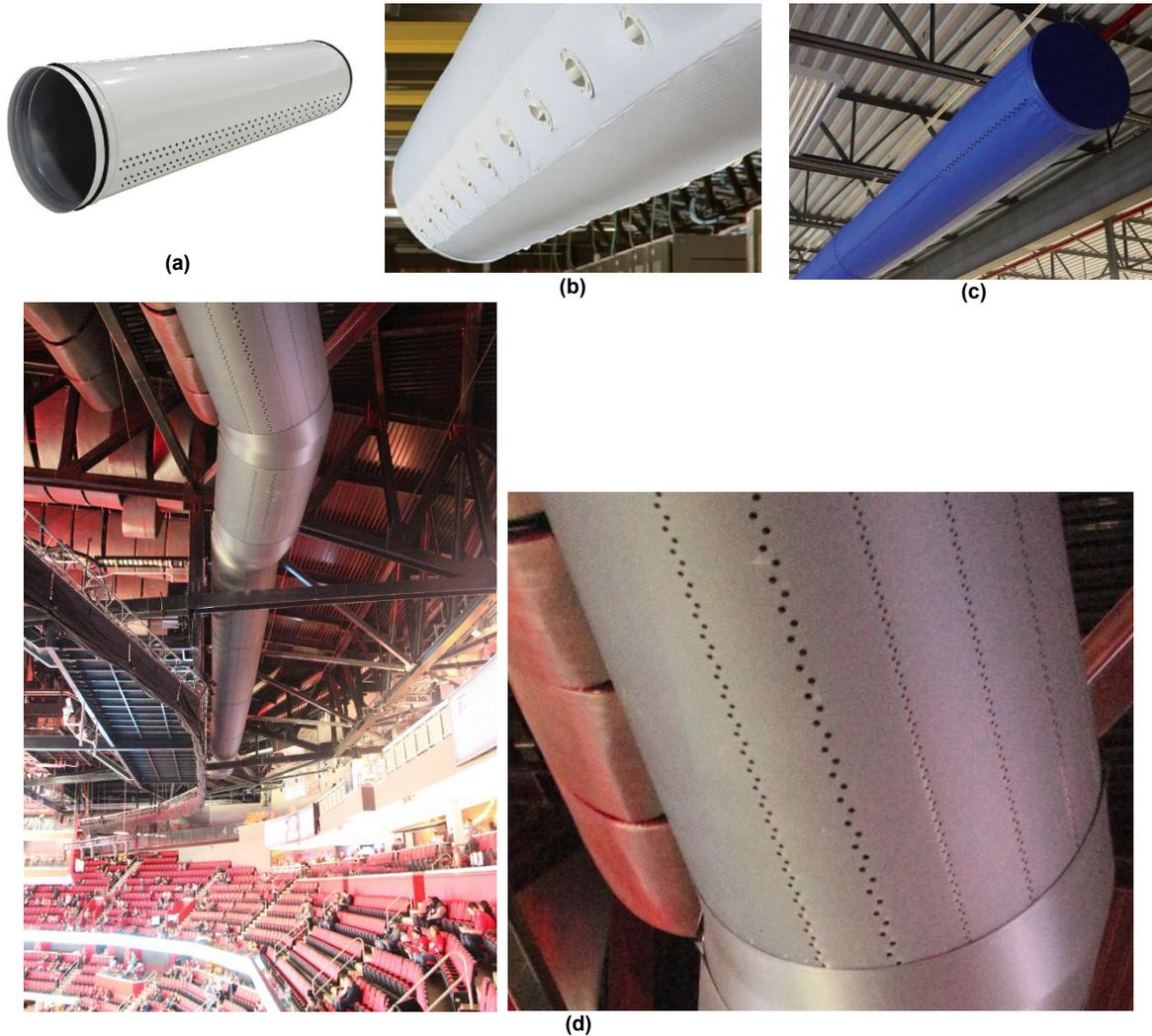
Figure 6-12. Metal and fabric air distribution ducts: (a) metal perforated duct (NAD Klima); (b) fabric duct with nozzles (DUCTSOX); (c) fabric ducts with holes (KEFIBERTEC); (d) various sized outlets allow textile air dispersion systems to target different occupant areas with different air jet throws.

Table 6-1 summarizes typical applications for air supply diffusers.

Table 6-1. Typical applications for air supply diffusers.

Air diffuser type	Air diffuser performance characteristics		Method of air distribution	Application
	K_1	K_2		
Large grills	2-6	1.8-5.1	Fig. 2.1, Fig. 2.3, Fig. 2.4	Large shops
Sidewall grills, Grills mounted on duct drops	2-6 2-4	1.8-5.1 1.8-3.5	Fig. 2.2 Fig. 2.5d	Low rooms (<20 ft [<6.1 m]) Large shops
Circular diffusers	1-3	0.9-3.2	Fig. 2.5a,b,c Fig. 2.9a	Low rooms (<20 ft [<6.1 m]) Large shops
Square diffusers	1-2.8	1.2-3.2	Fig. 2.5 a,b,c Fig. 2.9a	Low rooms (<20 ft [<6.1 m]) Large shops
Linear diffusers	2.5	2	Fig. 2.2	Low, small rooms
Perforated panels (round, half/quarter round, flat) mounted on or near the floor			Fig. 2.7, Fig. 2.8b Fig. 2.10b	Rooms higher than 12 ft (3.7 m) with surplus heat or combined heat and contaminant emissions
Perforated panels, mounted on duct drops	2.1	1.7	Fig. 2.6(3), Fig. 2.10c	Shops with surplus heat and a few workplaces or shops obstructed by process equipment
Perforated ducts	0.5	1.2	Fig. 2.6(1)	Low industrial rooms with surplus heat, high air change rate and requirements for low velocities in the occupied zone
Perforated ceiling-mounted panels or perforated panels	2.1	1.7	Fig. 2.6(2)	Same special applications (e.g., clean rooms)
Nozzles converging diverging or with swirl inserts	6-6.8 1-2.5	4.2-4.8 0.8-2.0	Fig. 2.1, Fig. 2.3 Fig. 2.4, Fig. 2.10a	Large shops Large shops; air supply into ails

* Approximate ranges are given for air diffuser characteristics: K_1 = coefficient of velocity decay along the jet; K_2 = coefficient of temperature decay along the jet. For actual values of these characteristics, consult manufacturers' guides.

Impact of Partitions and Process Equipment in Ventilated Spaces on Air Distribution Effectiveness and Efficiency

The results of most of the analytical and experimental studies on different methods of air distribution were obtained for empty rooms and do not reflect the influence of obstructions, e.g., partitioned areas, furniture, and process equipment, on the air distribution in spaces and ventilation (heat/contaminant removal) efficiency. In spaces of many commercial and industrial buildings, partitioned areas, furniture, and process equipment may occupy a significant part of the floor area or the space height. Also, workplaces can be located either within 6 ft (1.8m) of the floor level or at different heights for operating and servicing process equipment or assembling workpieces. Therefore, the requirements to the occupied zone thermal conditions and air quality should be extended also to those locations.

Information on the influence of obstructions on room air distribution ventilation efficiency is limited. Obstructions result in an aerodynamic change of the space and a change in the airflow pattern, and therefore velocity and contaminant distribution (Nielsen 1988, Jiang and Haghighat 1993, Lee and Awbi 1998).

Literature review field and laboratory studies (Zhivov 1983) show that large size obstructions affect airflow patterns and reduce ventilation efficiency. To evaluate the influence of process equipment, Zhivov proposed the use of following parameters:

- ratio of the vertical cross-sectional area of the obstructions to the cross-sectional area of the room σ_1
- ratio of the floor area, occupied by obstructions to the area of occupied zone σ_2
- ratio of the volume of the technological equipment to the room volume σ_3.

For commercial and industrial buildings, the above ratios can reach: $\sigma_1 = 0.7$, $\sigma_2 = 0.6$ and $\sigma_3 = 0.4$.

Figure 6-13 illustrates some conventional methods of air supply currently used in spaces with large size process equipment:

a. concentrated air supply with horizontal jets (attached or not attached to the ceiling) above obstructions

b. concentrated air supply with horizontal jets into the isles between obstructions: e.g., telephone exchange stations, warehouses with multistory racks, welding shops, and assembly plants

c. with inclined jets supplied from the upper zone (above 20 ft [6.1 m])

d. through air supply panels with low velocity toward workplaces located at different heights

e. vertical upward air supply through spiral, slotted, or perforated air diffusers on the floor

f. by inclined jets supplied close to workplaces on different heights

g. by inclined jets supplied close to workplaces from the height of 10 to 15 ft (3 to 4.6m) above the floor level

h. vertical downward with compact jets into the isles between obstruction

i. with horizontal jets into the occupied zone.
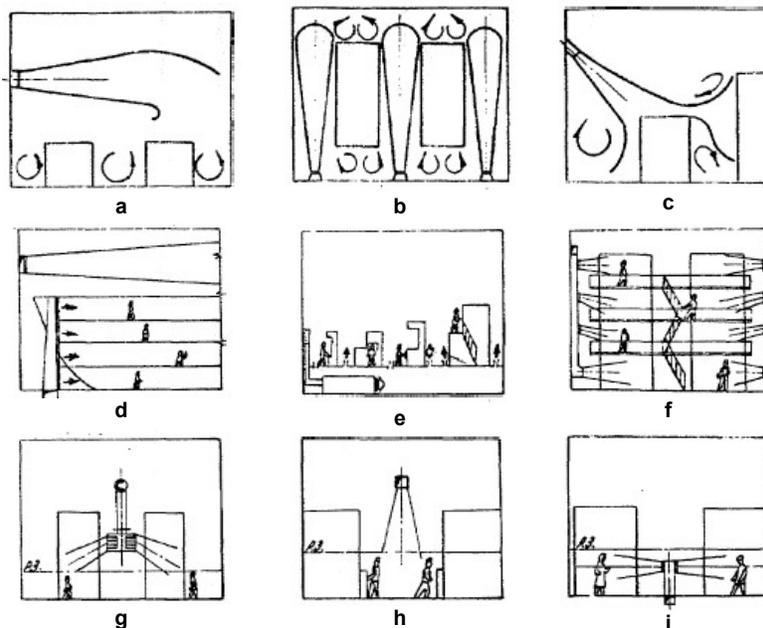
Figure 6-13. Examples of conventional air supply methods into obstructed spaces of industrial buildings (Reproduced from Zhivov 1983).

Air distribution in rooms is influenced the most by obstruction of the occupied zone when concentrated air supply and air supply with inclined jets from a height greater than 12 ft (3.7 m) above the floor level, are used (Zhivov 1983). Results of experimental studies of concentrated air supply in significantly obstructed spaces (Curilev and Pechatnikov 1966, Pechatnikov 1967; Regescheit 1964; Schwenke 1975; Gunes 1970) show that obstruction of the room's vertical cross- section area results in increased confinement of the supply air jets and thus reduces their throw (Figure 6-14). It also shows that concentrated air supply into obstructed spaces results in an increase in the reverse airflow velocity compared to the case of air supply into empty rooms.
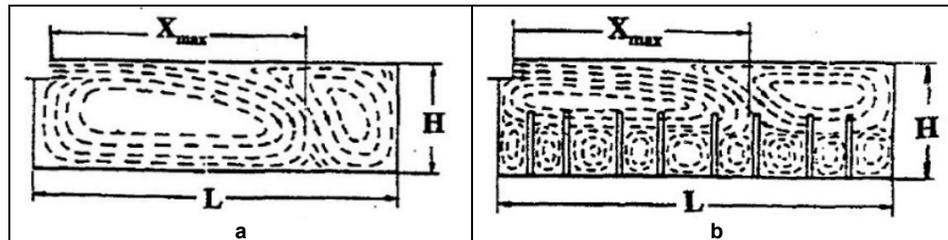
Figure 6-14. Influence of obstructions on horizontal jet throw: (a) empty room, (b) obstructed space.

When air is supplied into the isles between the process equipment from a height greater than 10 ft (3m), stagnant (poorly ventilated) zones occur behind this equipment (Figure 6-13b). Air supply above the process equipment results in poor ventilation of the isles between this equipment as well (Figure 6-13a)

Experimental studies of air supply with inclined jets into spaces significantly obstructed by process equipment (Zhivov 1993) showed a significant increase in maximum airflow velocities near the floor. Space obstruction by process equipment also decreases an inclined jet's throw when air is supplied from the upper zone or into the corridors between the equipment (Figure 6-13c).

Data from experimental studies of air supply provided by inclined jets into workshops with 33 ft (10m) high process equipment that occupies 55% of the room floor area, show the influence of this equipment on the distribution of temperatures and velocities across the occupied zone and the decrease of air distribution efficiency.

This discussion shows that neglecting the influence of the obstructions on the room air distribution may cause a dramatic overestimation of air distribution efficiency. A distributed method of air supply close to the workplace (e.g., see Figure 6-13g,h,i) was shown to be the most commonly used in obstructed spaces since they create better indoor air quality (IAQ) and thermal comfort at workplaces (Zhivov 1983).

Air supply systems with directing jets are commonly used in obstructed shops of industrial buildings in European countries. In spaces with large size obstructions in the occupied zone, horizontal directing jets increase the main streams throw. Vertical directing jets inject the clean and tempered air of the main streams and deliver it to workplaces located in the aisles between the process equipment, or compensate for the unwanted temperature gradient when heated air is supplied (Zhivov 1982, 1985, 1994).

Also, numerous applications of displacement ventilation in commercial and industrial spaces show that presence of partitions, furniture, and process equipment does not create a negative effect on temperature and velocity distribution in the occupied zone. These applications consequently result in a highly effective contaminant and heat removal from the occupied zone. In terms of energy consumption, displacement ventilation can achieve an indoor set-point temperature in the partitioned spaces about two times faster than mixing ventilation. Under mixing ventilation, the time to achieve a set-point temperature was notably reduced when each partitioned space is served by its own diffuser.

Air Distribution with Variable-Air-Volume (VAV) Systems

VAV systems are those that supply air at a variable airflow rate and a variable temperature to meet varying heating, cooling, and ventilation needs of different building zones. Some benefits associated with using VAV systems include a reduction in fan energy and energy for heating and cooling. However, limitations presented by air distribution systems may impose restrictions on the system operation in optimal operations modes. This section analyses the limitations imposed by some traditional and special methods of air supply.

When air is supplied from the ceiling-mounted air diffusers, the distribution of temperatures and velocities within the occupied zone depends on the ratio of the jet's cross-sectional area at the point of its entering the occupied zone and the area of the occupied zone that it ventilates (A_j/ $A_{o.z}$). Based on studies conducted by Grimitlyn et al. (1986) the most uniform distribution of temperatures and velocities in the occupied zone occurs when the ratio $A_j/A_{o.z}$ is between 0.55 and 0.6 for radial jets; from 0.3 through 0.6 for conical jets, and from 0.3 to 0.5 for compact jets. Figure 6-15 presents a graphic interpretation of these limitations.
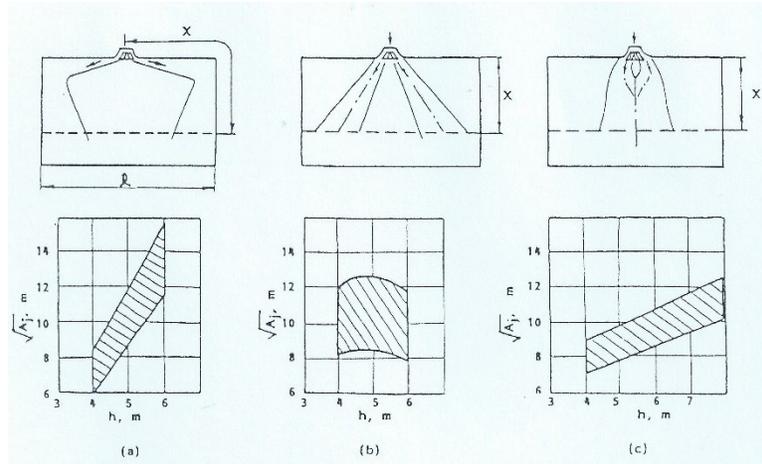
Figure 6-15. Limitations of VAV systems.

When air is supplied through a ceiling-mounted air diffuser, it is recommended to use diffusers that can supply the maximum airflow rate with radial jets; the reduced airflow rate with the radial jet that may collapse into a conical or a compact jet. Results of simulation conducted of different heating and cooling loads and supply air temperature differences show that, to maintain acceptable uniformity of velocities and temperatures distribution in the occupied zone, the reduction of airflow rate with air supply through ceiling-mounted air diffusers cannot exceed 35% from the maximum value (Zhivov 1990).

When cooled air is supplied through wall-mounted grills attached to the ceiling jets, the separation point of the jet from the ceiling surface X_{sep} should not be more than 50% of the length of the room served by one grill. To achieve the most uniform distribution of temperatures and velocities in the occupied zone with such an air distribution method, the airflow rate can be reduced by up to 50% when grills with fixed louvers are used. Special air diffusers with pneumatic or electric drives designed for VAV HVAC systems that are used primarily for air supply in small offices allow changes in the area of the air-discharge outlets (Connor, TROX), and the use of air diffusers with an injector that changes the ratio of the primary air (supplied to the diffuser) to the air injected from the space into the mixing box of the air diffuser (Carrier). The use of such diffusers allows the supply airflow rate to be reduced by up to 70% without negatively affecting air distribution uniformity in the occupied zone.

When air is supplied by inclined jets, the most uniform distribution of temperature and velocity is achieved when air circulates according to the design patterns shown in Figure 6-16 (Zhivov 1990). The use of grilles with fixed air supply direction and velocity decay characteristics controls can reduce the supply airflow rate by 50 to 65% without jeopardizing the uniformity of air distribution in the occupied zone. This strategy can be implemented using an air diffuser such as those shown in Figure 6-17.
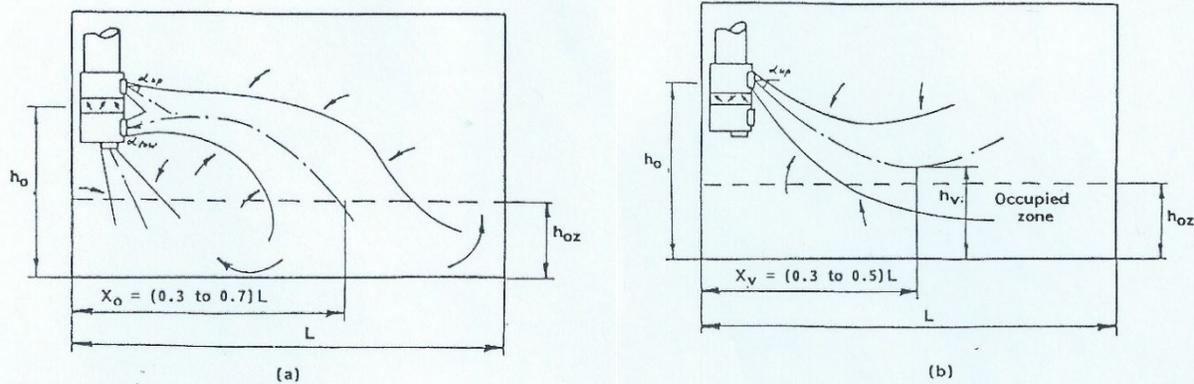
Figure 6-16. Design patterns that support the uniform distribution of temperature and velocity of air circulation.

The air diffuser is comprised of two tiers of ventilation grilles mounted on round air ducts with an electric drive-equipped damper between them. The number of ventilation grilles that can be attached to each tier may be one, two, or four depending on the diffuser location: one in the corner, two at the wall, and four at the columns.

The bottom piece of the air diffuser can have stamped grilles attached to the end face of the air duct in the lower tier through which 10% of the total airflow is supplied at an angle of 30 degrees to the horizontal direction. The vanes of the direction control section of the grilles in the upper tier are placed at some angle downward and those for the grilles in the lower tier at some angle upward. The maximum airflow (in cooling season) is discharged through the upper and lower tiers of grilles and through the bottom of the air diffuser; the reduced airflow is supplied only through the upper tier of grilles. Such a concept allows the airflow rate to be reduced by 75 to 85%.
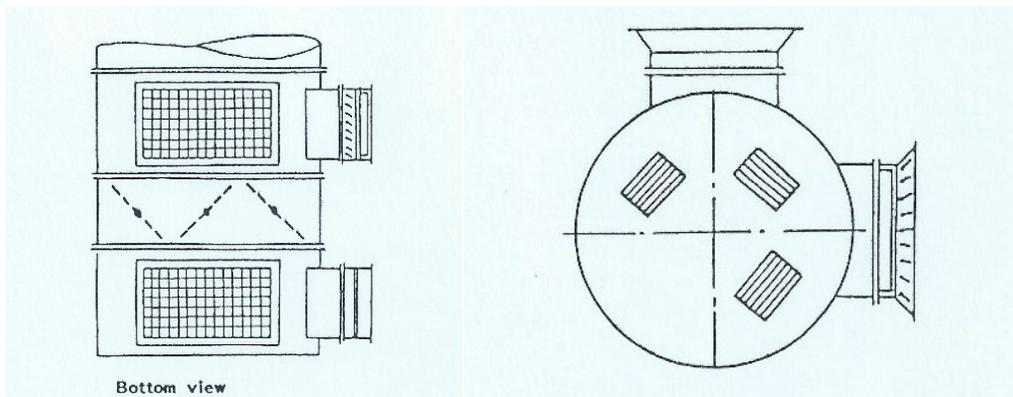
Figure 6-17. Air diffuser allowing direction and jet characteristics control for air supply with inclined jets in VAV systems.

Systems of air distribution systems with directing jets (Figure 6-18) are used to supply air in industrial and commercial buildings. Zhivov (1983) showed that air circulation within the space is caused mainly by the energy of the directing jets, which is an order of magnitude greater than the energy of the main airstreams. That is why variations in the air volume supplied by the mainstream do not affect the circulation pattern. Thus, with reduced thermal or contaminant loads in the ventilated space, it is possible to reduce the supply airflow rate by 70 to 90%.
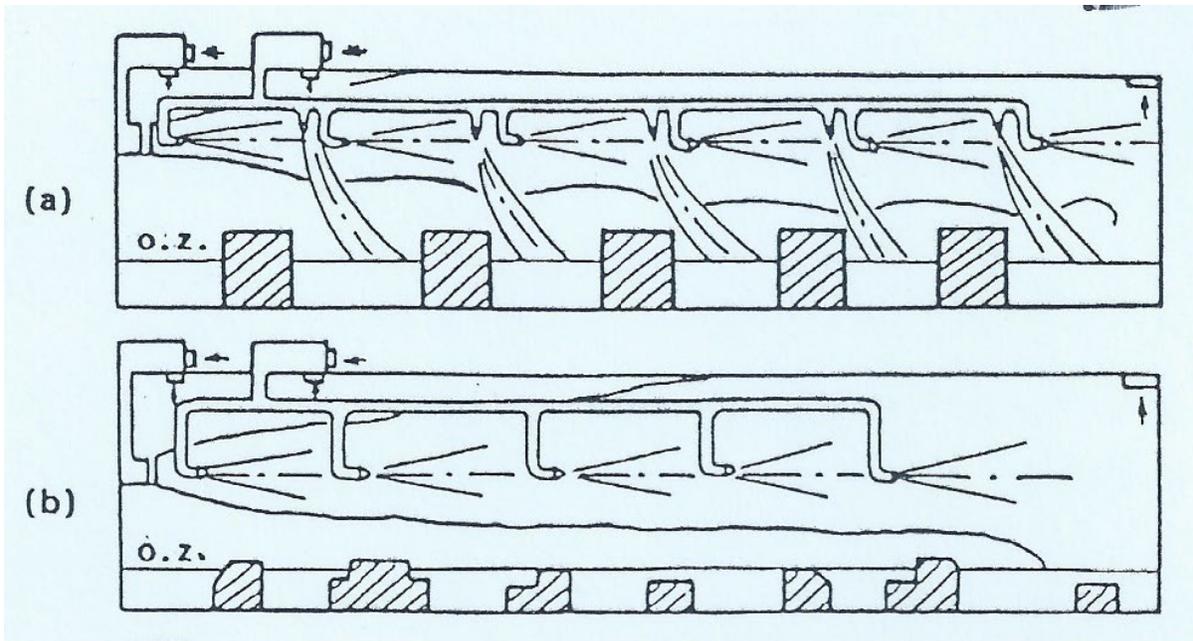
Figure 6-18. Air distribution systems with directing jets.

Chapter 7. Air and Contaminant Movement between Building Zones

The following classification of building zones is used (IVDG 2001, Ch. 7) to evaluate air and contaminant movement between zones within the building:

- Building areas separated by physical walls (e.g., halls, rooms, booths) located on the same level. Walls have either intentional apertures or leaks (Figure 7-1a)
- Building areas separated by physical partitions (e.g., slabs) located on different levels (floors). Air movement between these zones may occur (e.g., through stairways, air ducts or chases) Figure 7-1b
- Building areas separated by air jets located on the same level (e.g., jet-assisted hoods, air curtains) (Figure 7-1c)
- Building areas within the same room (with no physical partitions) having different requirements for air cleanliness ("clean" and "dirty" areas) located on the same level (Figure 7-1d)
- Zones located within the same room on different levels. These zones have different air temperatures and/or contaminant concentrations (Figure 7-1e)
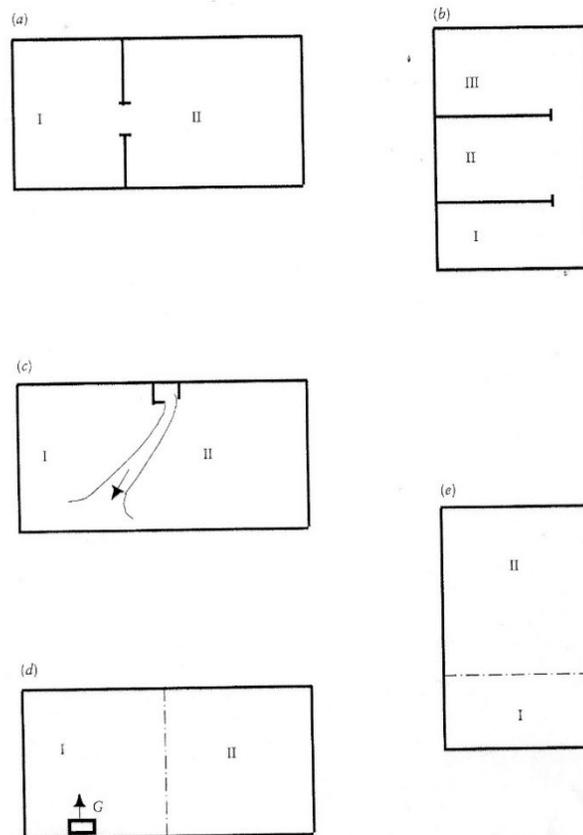
Figure 7-1. Zones within a building: (a) located on the same level and separated by physical partitions with apertures; (b) located on different levels and separated by physical partitions with apertures; (c) located on the same level and separated by air curtains; (d) located in the same room without physical separation; e – located in the same room on different levels without physical separation.

Air and contaminant movement between different zones may be caused by one of the following mechanisms:

- Static pressure difference between two zones resulting from the unbalanced air supply and exhaust in each zone. Air and contaminants move from the zone with high static pressure to the zone with lower static pressure (Figure 7-2a)
- Static pressure difference between two zones resulting from the wind effect on the building envelope (Figure 7-2b)
- Buoyancy forces creating vertical air movement along the passage between two rooms located on different levels, or thermal plumes creating temperature and contaminant differences between two zones located on different levels of the same room (Figure 7-2c)
- Turbulent exchange between air in different zones due to energy introduced by supply air jets, convective currents, or moving objects. In this case the resulting mass between the zones equals 0 (Figure 7-2d).
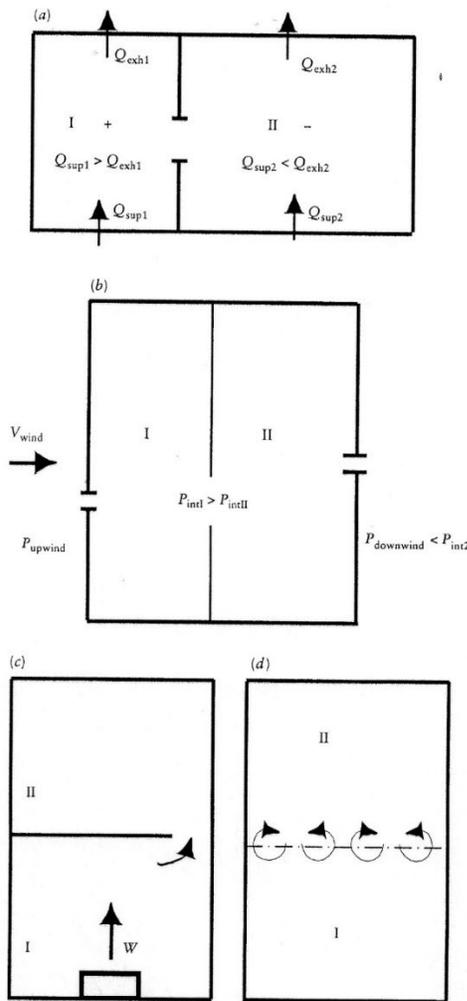
Figure 7-2. Mechanisms of contaminant movement between building zones: (a) difference in static pressure resulting from unbalanced air supply and return; (b) difference in static pressure resulting from wind effect; (c) buoyancy forces create vertical air movement along the passage between zones located on different levels; (d) turbulent exchange between air in different zones due to energy introduced by supply jets, moving objects, etc.

To control air and contaminant movement between zones, different construction, process-related, and ventilation techniques are used. "Clean" and "dirty" areas can be separated using solid walls, curtains, or partitions (Figure 7-3a). Ventilation techniques used to separate zones include:

- Pressure management in different zones (Figure 7-3b)
- Air oasis with specially organized local air supply and exhaust (Figure 7-3c)
- Natural or displacement ventilation systems creating temperature stratification (Figure 7-3d)
- Air curtains and jet-assisted hoods separating "dirty" zones from "clean" zones (Figure 7-3e).
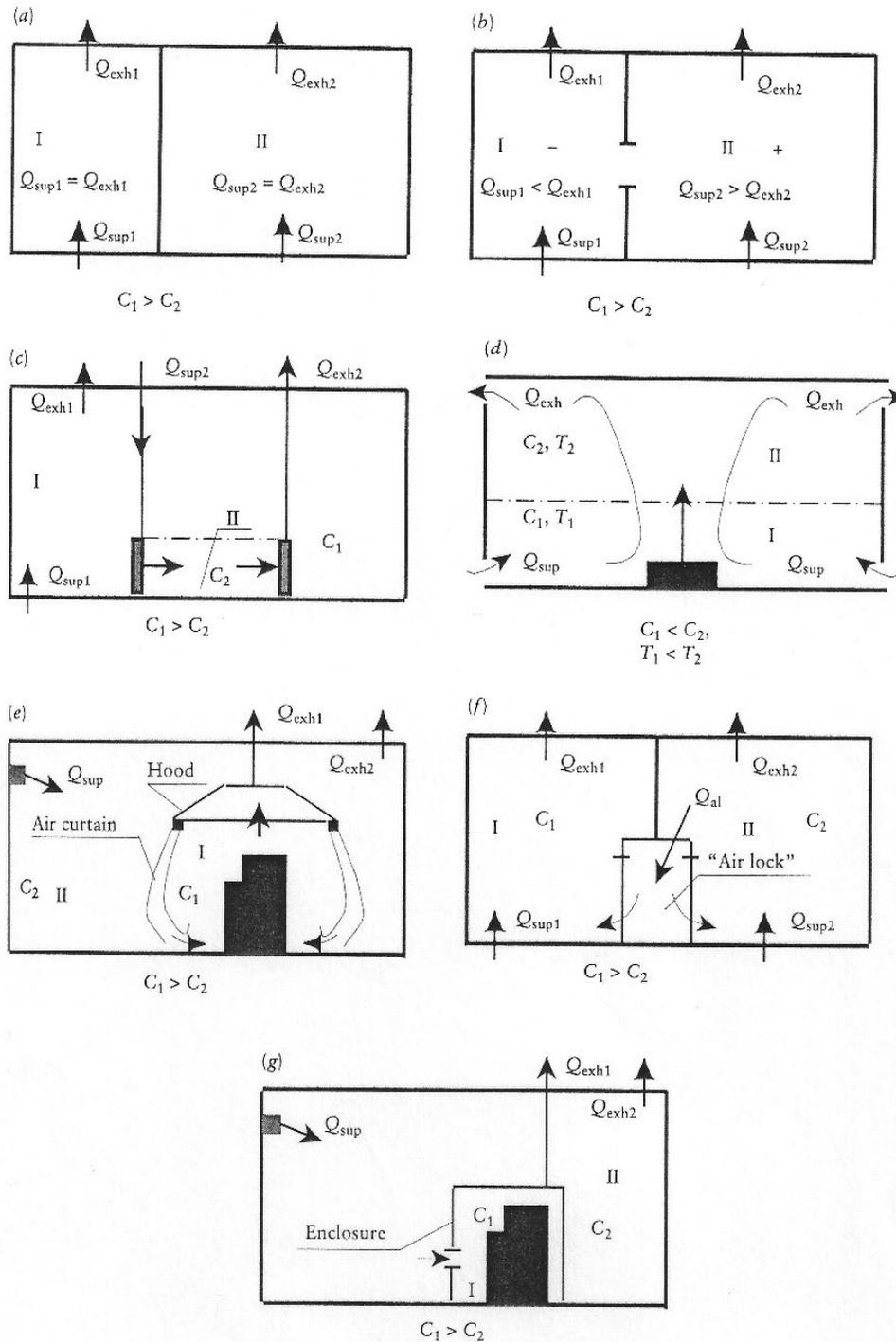
Figure 7-3. Construction, process-related, and ventilation techniques used to separate building zones:
(a) solid walls, (b) unbalanced supply and return airflow rates; (c) specially arranged local supply and return, creating an "air oasis" with cleaner and cooler air in the desired zone of the building; (d) temperature and contaminant stratification along the room height using a natural or displacement ventilation system; (e) air curtain supplied around the perimeter of the canopy hood, which separates the contaminated process zone; (f) an "air lock" located between two zones; (g) enclosing process equipment and extracting air from the enclosure.

Combined construction, process-related, and ventilation measures include "airlocks" between two zones (Figure 7-3f) and process equipment enclosures with air exhaust from enclosures (Figure 7-3g)

Regarding pressure management between two zones, it is important to mention that the airflow created by the pressure difference between the "clean" and "dirty" zones does not completely prevent contaminant movement from the "dirty" to the "clean" zone. Consider an example of a two-bay building with an air supply into the high "clean" bay and an air exhaust from the low "dirty" bay (Figure 7-3.a). The high bay has a higher air temperature than the lower bay. The stack effect created between the two bays creates contaminant movement from the "dirty" zone to "clean" zone through leaks and other openings in the walls separating these zones. To prevent contaminant movement between the "clean" and "dirty" zones, these zones can be separated by an air lock (Figure 7-4b).
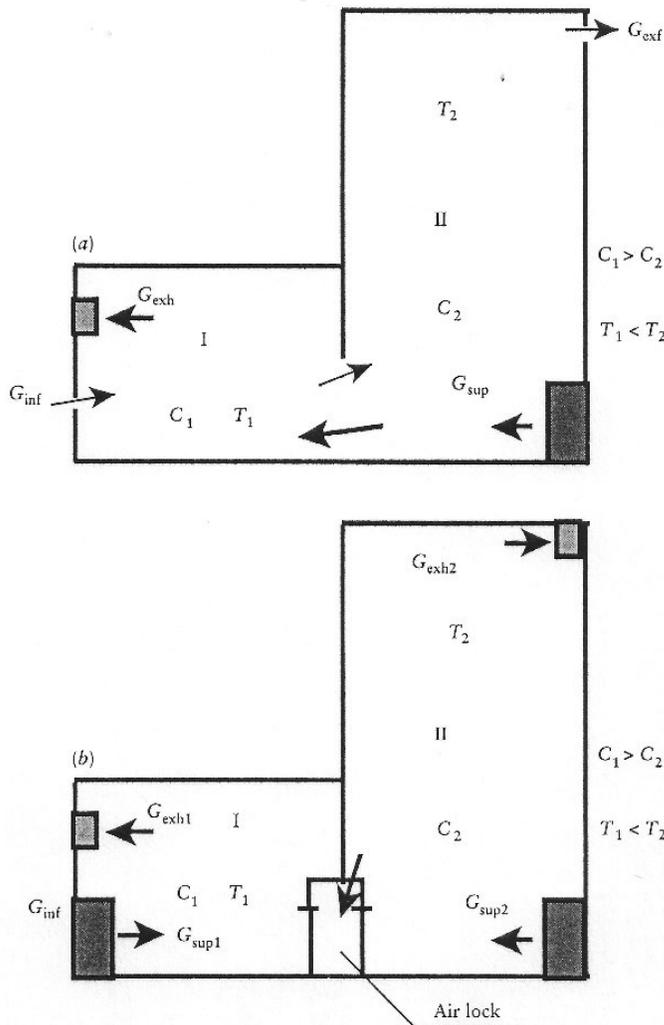
Figure 7-4. (a) contaminant movement between two building zones with different ceiling heights due to the combined effect of unbalanced mechanical systems and the "stack" effect; (b) prevention of contaminant movement between two zones using an "air lock".

Another factor influencing contaminant and heat transfer from dirty to clean zones against the stable airflow is a turbulent exchange between these zones. This process should be considered in the design of displacement or natural ventilation systems (Figure 7-5b) and in the evaluation of emission rate of contaminants from the encapsulated process equipment (Figure 7-5a). Turbulent exchange between the contaminated air under enclosure and the room can be decreased by reducing the opening size and the turbulence level under the enclosure. Turbulence exchange can be reduced by inserting a flow equalizer of a spigot into the enclosure opening.

Figure 7-5. Contaminant and heat transfer due to turbulent exchange between building zones: (a) contaminant movement against airflow near the vicinity of local exhaust; (b) heat and contaminant transfer between the lower and the upper zones of the building with displacement ventilation.

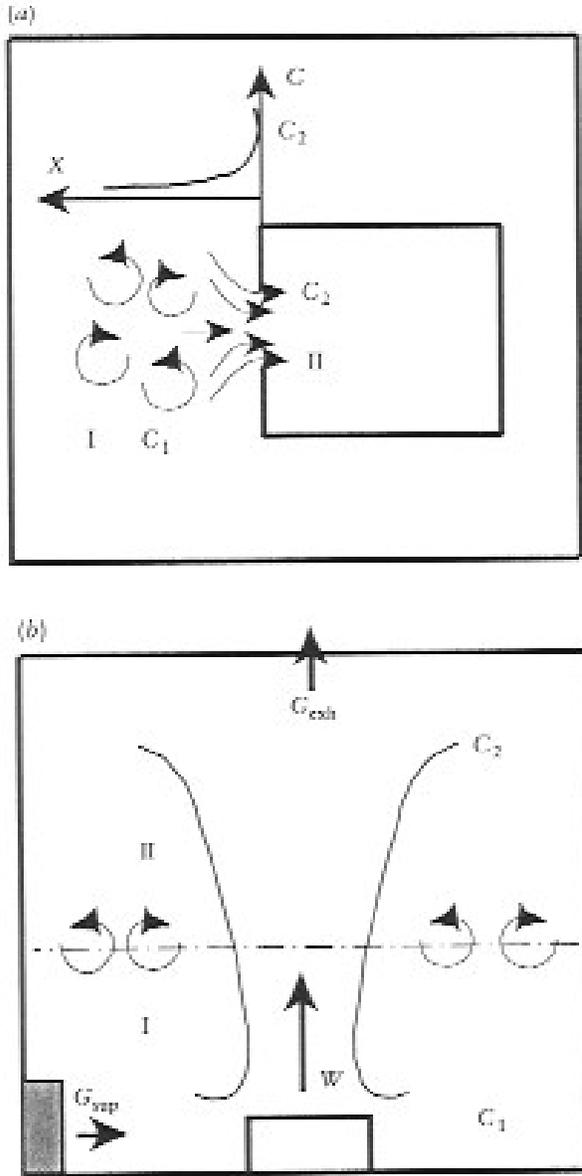# Chapter 8. Air Supply Strategies to Reduce Airborne Cross-Infection in Ventilated Spaces.

New knowledge obtained since the outbreak of the COVID 19 pandemic (REHVA 2020) demonstrates that there is a need to rethink air distribution system design strategies that can reduce cross-infection in ventilated spaces. This will offer a healthier workplace, encourage a return to a collaborative work pattern, and thereby improve workers' productivity and reduce absenteeism.

Unfortunately, research related to the air supply strategies' ability to protect occupants from airborne infectious disease transmission is limited. In the past, such research was applicable primarily to healthcare facilities (Nielsen et al. 2008, ASHRAE 2013). Most research on air distribution strategies has been focused on acceptable IAQ and on the reduction of energy consumption to satisfy environmental concerns related to the design of energy-efficient buildings. For these purposes, acceptable IAQ was assumed to be of sufficiently good quality so that a substantial majority of occupants express no dissatisfaction with odor or other sensory irritations, and so that it is unlikely to encounter contaminants at concentrations known to pose a health risk (ASHRAE 2019).

Since the worldwide severe acute respiratory syndrome (SARS) epidemic between November 2002 and June 2003, some additional research has been done regarding air distribution in residential and commercial buildings and its effect on the transmission of viral aerosols and their concentration in room air (e.g., Rudnick and Milton 2003). This section discusses specifics of air supply strategies in the indoor environments of commercial, administrative, and residential buildings where significant viral loads of pathogens may be released in the air when one or a few occupants are infected.

During breathing, talking, singing, and coughing, particles of different sizes (0.05-500μm) loaded with viruses are generated and expelled, especially by asymptomatic and pre-symptomatic infected people. Results of studies summarized by Melikov (2020) reveal that under typical indoor settings, the short-range exposure occurs up to 5 ft (1.5m) distance between infected and exposed persons (Ai et al. 2018). At a 5- to 6-ft (1.5–2m) distance, long-range exposure is predominant. The risk for short-range exposure is much higher than for long-range exposure. Airborne cross-infection in such spaces of public buildings as offices, meeting rooms, auditoriums, and dining establishments can be controlled using different air distribution strategies including mixing ventilation, vertical ventilation, displacement ventilation, and personal ventilation. The outside airflow rate to be supplied to the room must exceed minimum requirements by ASHRAE Std 62.1 (2019) and the airflow rate required for building pressurization. Research shows that while an increase in the outdoor air supply to occupied spaces can substantially reduce the risk of long-range exposure, it is less effective in reducing short-range exposure. The short-range exposure depends on the complex interaction of flows in the vicinity of the infected and exposed persons, including the transient flow of respiration, free convection flow existing around the human body, and flow of ventilation (Ai et al. 2018, 2019).

An air distribution strategy that has high ventilation effectiveness must be selected. Undisturbed mixing-type air distribution systems create complete mixing of contaminants and even air temperatures throughout the ventilated space unless partitions and interaction with other flows prevent the flow (Figure 8-1). While air movement is commonly treated as a thermally uncomfortable draft, in rooms with an infected person, air movement results in more serious consequences. Studies of a Guangzhou restaurant and some previous airplane infections (Lu et al. 2020, ECDPC), this phenomenon of spread by air movement is well known. A strong directed airflow toward an infected person may carry viral material in an aerosol toward a susceptible person in a very high concentration, which may propagate the virus within a specific part of the room.
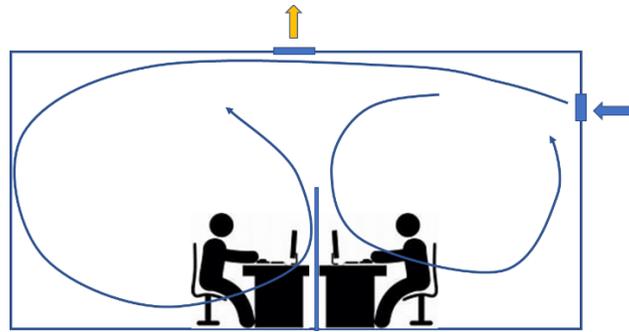
Figure 8-1. Mixing air supply into partitioned office space with horizontally projected jets.

Some systems will be more suitable for high flow rates than others. If effective air supply is missing, the limited airflow rate may prevent drafts in the occupied zone but, according to Nielsen et al. (2008), pollutants may take a shortcut in the microenvironment around persons standing or sitting at the distance smaller than 4 ft (1.2m). The probability of cross-infection increases if one person has an increased breathing frequency, is speaking, or especially if the person is coughing (Badeau et al. 2002). To alleviate the situation, a downward air supply from large diffusers with a low air velocity (Figure 8-2) to displace particles with the air return located near the floor will be highly efficient, but difficult to implement (Qian et al. 2006).
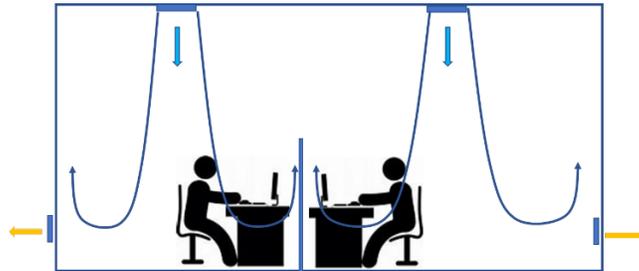
Figure 8-2. Mixing air supply into partitioned office space with downward-projected jets.

Displacement ventilation systems are highly effective against cross-infection due to thermal plumes created by people and other heat sources (Figure 8-3). Still, such effectiveness cannot be guaranteed in all situations, e.g., airflow created by exhalation or coughing where a contaminated jet can propagate horizontally toward another person (Nielsen 2009). In a well-designed office, the installation of partitions separating cubicles can minimize such problems.
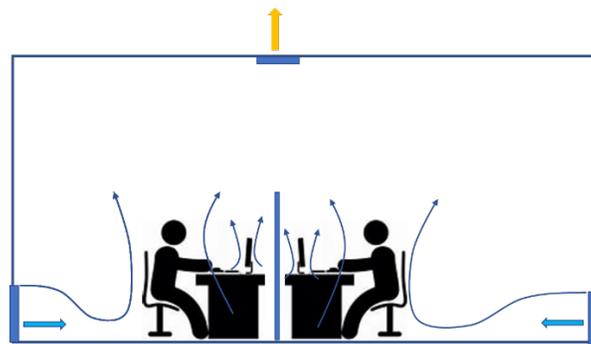
Figure 8-3. Displacement air supply into partitioned office space.

Partitions/barriers can physically separate spaces that are next to each other. When used for infection control, the barrier is intended to prevent someone on one side of the barrier from exposing a person on the other side of the barrier to infectious fluids, droplets, and particles. Whether a barrier interferes with improved ventilation depends on how it is installed. Protective barriers can sometimes help improve ventilation, sometimes hinder ventilation, and sometimes have no effect on ventilation (CDC 2021).

Protective barriers (Figure 8-4) can help to improve ventilation when used to facilitate directional airflows or desired pressure differentials between clean and less-clean spaces. The barrier can be aligned with the intended airflow to help direct it toward the desired location, such as an HVAC return air grille or a portable air cleaner inlet.

Clear mobile acrylic guard (nationalbusinessfurniture.com)

Cubicle Workstations (Scutchi Designs)
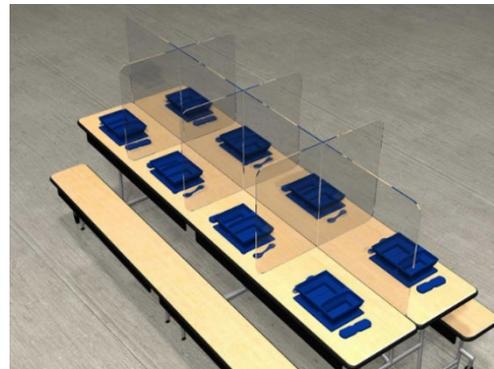
Figure 8-4. Barriers reducing cross-infection in office spaces

Alternatively, a barrier might be placed between two areas to better isolate one side of the barrier from the other. In this configuration, the barrier can also assist the HVAC design scheme in establishing a desired pressure differential between the adjacent spaces. If necessary, small pass-through openings or a retractable panel incorporated into the barrier can allow the transfer of physical objects from one side to the other. This type of barrier application might be applied, for example, at a receptionist's desk or a ticket booth.

When not carefully installed, barriers can sometimes hinder good ventilation. Barriers can unintentionally interrupt the airflow distribution within a space, thus allowing a concentration build-up of human-associated (or other) aerosols that may remain suspended in the air for minutes to hours. In this case, people could be exposed to higher concentrations of infectious aerosols than they would without the barriers in place. The larger the barrier, the greater the likelihood that this may occur. To reduce this likelihood, ensure that barriers are correctly positioned for the

anticipated occupancy, and ensure that they are no larger than necessary to prevent the direct transfer of respiratory droplets that could "spray" directly from one person onto another.

The use of localized air supply (Figure 8-5) may reduce the risk of cross-infection. Partitions will further enhance the ventilation efficiency and reduce cross-infection with the use of localized air supply.

Source: VITAL Environments

Figure 8-5. Use of localized air supply.

Chapter 9. Recommendations for Selection of Air Supply Methods

The economic efficiency and hygienic effectiveness of general ventilation systems significantly depend on the proper choice of the air supply method. ASHRAE (1995) recommends the following methodology for the selection process:

1. Select several feasible methods of air supply suitable for the given space. Specify requirements specific to thermal comfort in the occupied zone, the level of the occupied zone obstruction with furniture, partitions or process equipment, internal loads, and desirable operation modes of the HVAC system throughout the year-round cycle (variable or constant airflow systems).

2. Determine the airflow requirements for each method based on ventilation effectiveness and limitations on supply air temperature difference, and determine a maximum supply air velocity based on the requirements of temperature and uniform velocity distribution in the occupied zone.

3. Determine the dimensions (length and width) of the occupied zone area that can be served by one air diffuser, the height of its installation, and the position of controls for the design mode of the system operation (e.g., the highest cooling load in summer).

4. Check the airflow pattern and the prediction of air temperature and air velocity distribution in the occupied zone for other characteristics of HVAC system operation modes (e.g., in a heating mode).

5. Compare and down-select air supply methods based on the LCCA.

Air supply method selection is subject to analysis in each specific case. However, there are some general recommendations that are applicable to various situations.

The air supply method and the number of air diffusers to be used and their characteristics (velocity and temperature decay) should be selected based on the size of the space (L x B x H), supply air change rate (ACH), heating and/or cooling loads to be satisfied by the air system, requirements for IAQ and thermal comfort in the occupied zone or at workplaces, operation modes of the ventilation system (VAV or CAV), the level of the occupied zone obstruction by partitions or process equipment and the desire to prevent cross-contamination/infection of the ventilated space from random or fixed sources. Other considerations for the air supply method include interior design consideration and space available for the ductwork. Life cycle comparison of different alternatives should be made based on first costs and operating costs, which include the airflow requirements for each method based on ventilation effectiveness, and limitations on supply air temperature difference and a maximum supply air velocity satisfying requirements for temperature and velocity distribution uniformity in the occupied zone.

Air supply methods to be considered include **mixing-type, thermal and active displacement, unidirectional, and localized**.

Mixing-type air distribution methods include air supply with jets projected vertically downward, inclined jets, jets directed vertically upward, and horizontal jets, which should be selected and sized for spaces with air change rates below 6, insignificant heating and cooling loads, CAV systems and VAV systems with air flow rate reduction not exceeding 60%, spaces with low partitions and other obstruction of the occupied zone by process equipment.

In hot and humid climates, cold air supply with attached jets should be avoided with the VAV systems and systems that use on/off operation modes. Cold surfaces of air diffuser and adjacent ceiling may result in condensation when in contact with the room air.

In spaces with the risk of cross-infection, air supply with horizontal and inclined jets should be avoided except for individually occupied rooms.

For the mixing-type systems with air supply using horizontal jets, the length of the room ventilated by a single jet should not exceed $L < 0.62 \, K_1 \, (B \times H)^{1/2}$ to assure that the space is ventilated by the jet without creating secondary vortices; also, the width, B, should be less or equal to 3H (Figure 9-1).

For the air supply with ceiling-mounted air diffusers creating radial, conical or compact air jets, room height should not exceed 20 ft (6.1 m), and the occupied zone area, L x B, can be selected using the illustrations in Figure 9-2. In industrial spaces where the height exceeds 20 ft (6.1 m), these air diffusers can be installed on the duct drops.
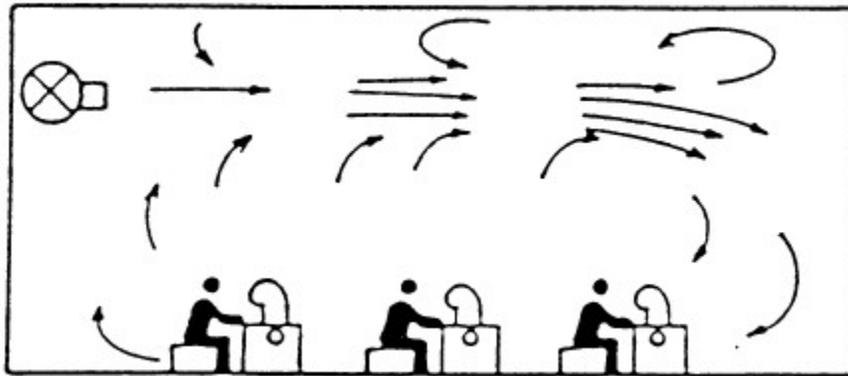
Figure 9-1. Air supply with horizontal jets (AFS 1990).
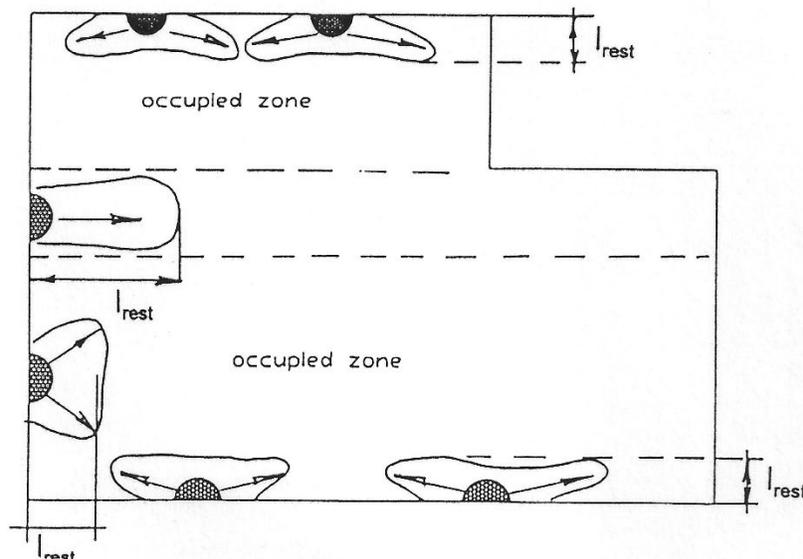
Figure 9-2. Example of air diffusers layout and restricted zone l$_{rest}$ selection.

Unidirectional air supply methods using supply diffusers and exhaust openings with large surfaces are recommended for spaces with a high air change rate (>5 ACH). They should be considered for clean rooms that require a low level of pollutants such as dust, airborne microbes, aerosol particles, and chemical vapors. Unidirectional air supply reduces the risk of cross-infection, especially in workplaces separated by partitions that reduce short-range exposure. In the underfloor air supply systems, air temperatures should be kept above 61 F (16 °C) to minimize the risk of condensation and subsequent mold growth in the underfloor cavity. Downward and upward unidirectional air supply should be considered for obstructed spaces to allow a significant range of airflow change.

Thermal displacement ventilation systems should be considered for spaces higher than 10 ft (3m) with a predominant cooling load and a moderate air change rate. Limitations on supply air velocity and temperature are based on the size of the restricted zone around air diffusers. Displacement air supply should be considered for obstructed rooms including rooms with partitions to reduce the risk of cross-infection, especially in workplaces separated by partitions that reduce short-range exposure.

The shape and design of air diffusers for displacement ventilation significantly affect thermal comfort in the occupied zone and the length of the restricted near zone with air velocity exceeding 40 fpm (12.2 m/min).

An active displacement system supplies air with a lower velocity than one with a mixing-type of air supply, but with a higher velocity than with thermal displacement ventilation, which allows limited use of this system with heated air. This system is primarily designed for industrial applications.

Localized air supply systems should be considered for spaces with a limited number of workplaces. They significantly improve air quality and thermal comfort directly in the workplace and reduce the risk of cross-infection.

In addition to the dimensions of the ventilated space, the selection of the supply air jet type and characteristics of the air diffuser (K_1 and K_2) should be based on the requirements for maximum velocity in the occupied zone and for the uniformity of velocity and temperature in the occupied zone, and for the need to satisfy heating and cooling loads with the air supply system (i.e., to avoid drafts with the cold air supply and reduced heating and ventilation efficiency due to buoyancy forces).

Note that velocity and temperature decay characteristics significantly impact induction of ambient air into the jet and therefore its mixing with the ambient air. Consider the limitation of different air supply methods on the range of airflow rate reduction with VAV systems, the impact of partitions and process equipment on performance of air distribution systems, and the potential for cross-infection associated with different air supply strategies.

The amount of energy used by the system (supply airflow rate, heating, cooling, and electricity) depends on the effectiveness of ventilation, heat, and moisture removal specific to the air supply method selected. Table 9-1 lists supporting information.

Figure 9-3 compares airflow rates and heating and cooling loads that can be supplied to a ventilated space with appropriately selected mixing and displacement air distribution systems. For more information regarding economic comparison between displacement and mixing ventilation see Sepannen (1989), Zhivov et al. (2000), and Hu et al. (1999).
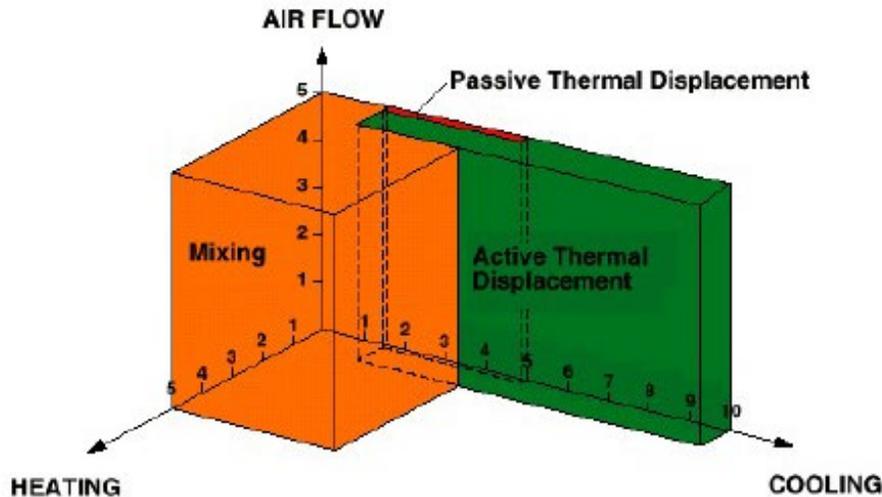
Figure 9-3. Matrix for the airflow rate, heating and cooling load range evaluation with different methods of air supply: one unit along the airflow axis equals ~7 m³/h x m² (0.37 cfm/ft²), one unit on heating and cooling axis equals 7 W/m² (2.2 Btu/hr ft²) (Reproduced with permission from ABB).

Table 9-1. Recommendations for selection of the air supply method.

Characteristics of Ventilated Space						Recommended Air Supply Methods
Height, ft (m)	Air Change Rate, ACH	Activity type	Requirements For Parameter Uniformity in The Occupied Zone	Obstructions	HVAC System Type	Recommended Air Supply Methods
>20 ft (>6.1m)	5-7	Moderate and heavy	No specific requirements	Insignificant	CAV	Concentrated air supply by horizontal jets, by horizontal jets assisted by directing jets, air supply with vertical compact jets, displacement air supply, spiral vortex supply
					VAV	Air supply by horizontal jets assisted by directing jets, air supply by inclined jets, downward air supply with compact jets.
				>10 ft (>3.0 m)	CAV	Concentrated air supply by horizontal jets assisted by directing jets, air supply by inclined jets, displacement air supply
					VAV	Concentrated air supply by horizontal jets assisted by directing jets, air supply by inclined jets
>20 ft (>6.1m)	>5	Sedentary and light	Low and Moderate degree of uniformity	Insignificant	CAV	Air supply with vertical radial, conical or compact jets, air supply with horizontal attached or non-attached jets, displacement air supply, localized air supply, unidirectional air supply
					VAV	Air supply with vertical radial or compact jets, air supply with horizontal attached or non-attached jets
				>7 ft (>2.1 m)	CAV	Air supply with vertical compact jets, air supply with horizontal attached or non-attached jets into aisles between partitions/equipment, displacement air supply, unidirectional air supply, air supply with vertical compact jets, air supply with horizontal attached or non-attached jets into aisles between equipment, localized air supply, localized air supply
					VAV	Air supply with vertical compact jets, air supply with horizontal attached and non-attached jets into aisles between equipment, localized air supply

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