Displacement Ventilation for Industrial Applications

Types, applications, and design strategy

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Almost all U.S. ventilation and air-conditioning systems are of the mixing (dilution) type. Fresh, outdoor air is mixed with room air, resulting in fairly uniform temperatures, humidities, and contaminant concentrations throughout all areas and levels of the room. Displacement ventilation differs in that it creates stratified levels of temperatures and contaminant concentrations within a room.

CLASSIFICATION

Different types of air-supply systems are said in their literature to employ "displacement ventilation." To avoid confusion, the following classification is suggested:

- *Systems utilizing unidirectional low-turbulence flow, with air supplied at a low velocity, and supply diffusers and exhaust openings with large surfaces, such as perforated panels.* Air flow in these systems, which typically are referred to as unidirectional-flow or piston-air-distribution systems, can be either vertical (air is supplied from the ceiling and exhausted through the floor or vice versa, as in figures 1 and 2) or horizontal (air is supplied through one wall and exhausted through returns located on the opposite wall, as in Figure 3). The outlets are uniformly distributed over the ceiling, floor, or wall to provide a low-turbulent "plug"-type flow across the entire room. This type of system is used mainly for ventilating clean rooms, of which the main objective is to remove contaminants within the room or in halls with high heat and/or contaminant loads and a high air-change rate.
- *Systems utilizing underfloor air supply (Photo A), with air diffusers creating fast velocity and temperature decay along jets.* Heated by internal sources, air rises and is exhausted from the upper zone of the space.
- *Low-impulse cooled-air supply systems (Photo B) with air diffusers located either at the ceiling level*[1,2,3] *or at a height of about 10 ft.*[4,5] Under the influence of buoyancy, cold air flows toward the occupied zone. With some entrainment of the ambient air, it then spreads across the floor and floods the lower zone of the room. Air heated by internal sources rises and is exhausted from the upper zone. Low air entrainment into supply jets allows limited mixing of contaminated warm air of the upper zone with occupied-zone air. These systems, which sometimes are referred to as active-thermal-displacement sys-

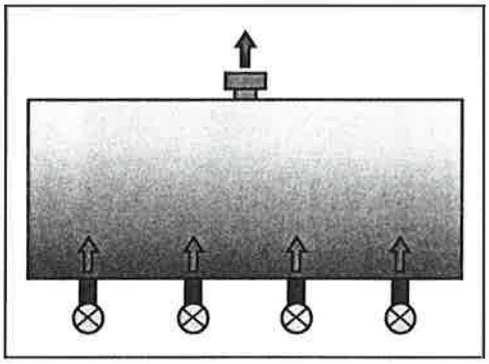

FIGURE 1. Vertical air flow in a unidirectional-flow or piston-air distribution system. Reproduced from AIR-IX, 1987, and LVIS, 1996.

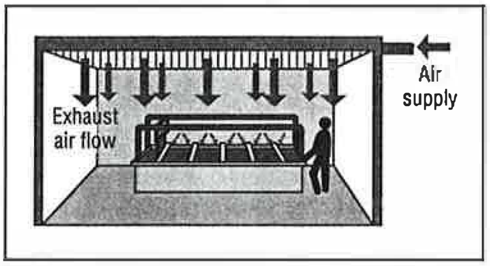

FIGURE 2. Vertical piston air distribution through a perforated ceiling in a manufacturing facility. This approach can be used for the ventilation of rooms with galvanic baths. Reproduced from AIR-IX, 1987.

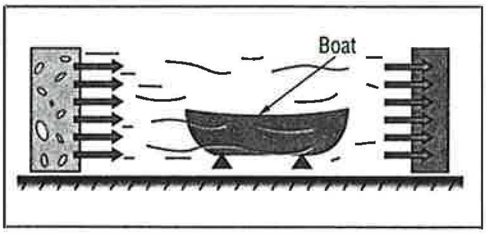

FIGURE 3. Horizontal air flow in a unidirectional-flow or piston-air-distribution system. This approach can be used for the application of epoxy in the production of glass-fiber boats. Reproduced from AIR-IX, 1987, and LVIS, 1996.

[1] *Superscript numerals indicate references listed at end of article.*

tems, remove heat and contaminants more effectively than do total mixing air-distribution systems.

• *Systems in which low-impulse cooled air is supplied through ducts with special nozzles located above the occupied zone, with air exhausted at floor level.*[6] Polluted air of the occupied zone is suppressed by an overlying air cushion that displaces the contaminated air toward floor-level exhausts (Figure 4). This system creates temperature and contaminant stratification above the air-supply level and prevents the mixing of air within the occupied zone. These systems also are sometimes referred to as active-thermal-displacement systems.

• *Systems that directly supply the occupied zone with cooled air at low velocities and that exhaust air from the upper zone (Figure 5).* These systems sometimes are referred to as passive-thermal-displacement systems. The air from the diffusers spreads along the floor, creating a relatively cool layer of fresh air near the floor. Heat sources within the occupied zone create thermal plumes of rising air, which entrain the air near the floor. The warm contaminated air forms a stratified region in the upper zone of the room, which is exhausted

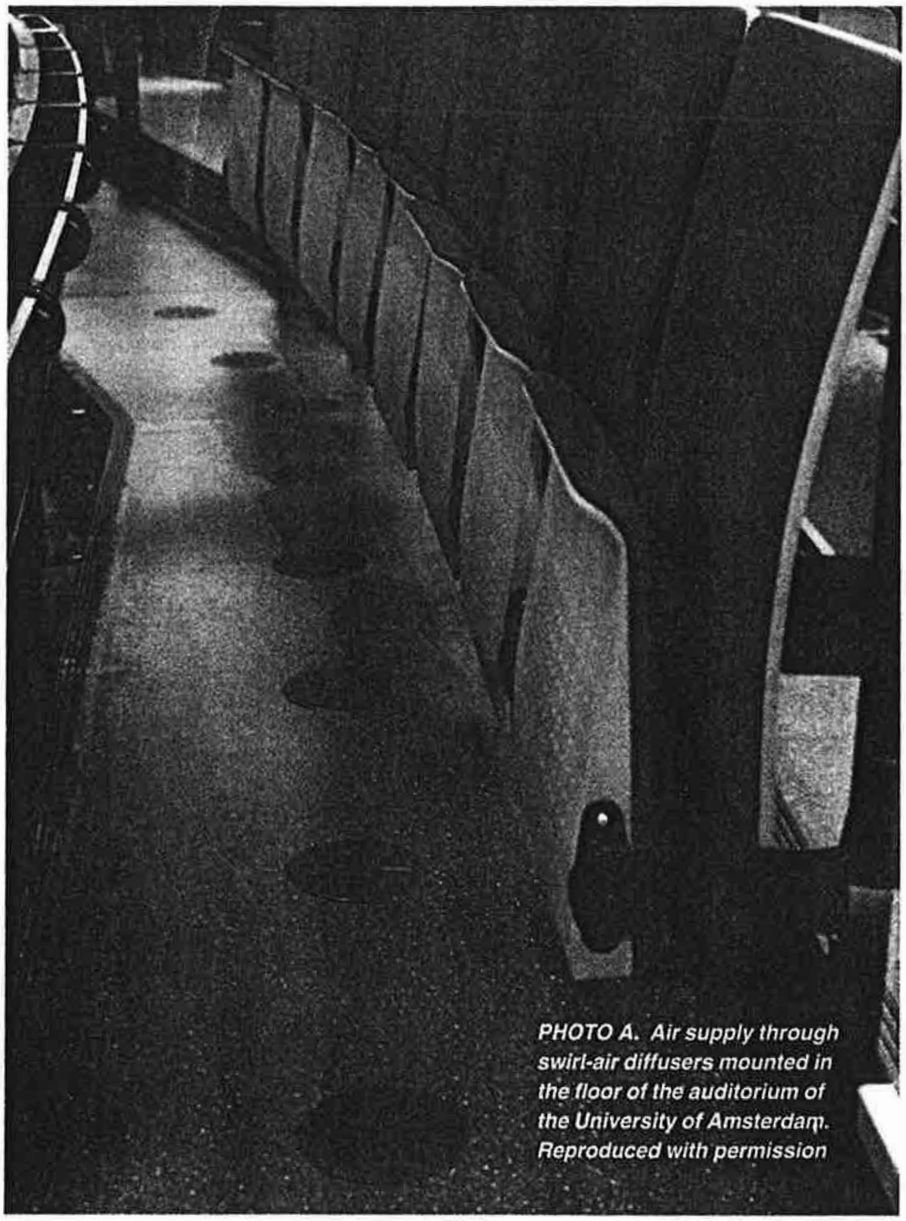

PHOTO A. Air supply through swirl-air diffusers mounted in the floor of the auditorium of the University of Amsterdam. Reproduced with permission

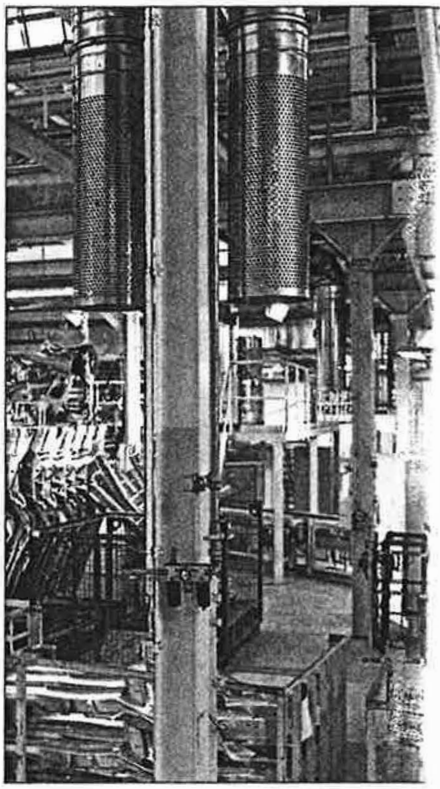

PHOTO B. Low-impulse air supply into the Volkswagen AG production facility through perforated air diffusers installed on the column at a height of 10 ft. Reproduced with permission from KRANTZ-TKT.

by high-level air returns. Low-turbulent supply air flows entrain little ambient air and do not facilitate mixing between the upper and lower regions. Stratification of contaminant concentrations makes providing higher-quality air in the occupant breathing zone without an increase in system operating costs possible.

Passive-thermal-displacement systems, which were the first displacement-ventilation systems introduced, have been common for the ventilation of industrial facilities in Scandinavia for the past 30 years. They still are the most commonly used displacement ventilation systems in Europe. More recently, however, their use has been expanded to the ventilation of office and other commercial spaces where, in addition to air quality, comfort is an important consideration.

SELECTING AN AIR-DISTRIBUTION METHOD

Among the criteria used to select an air-distribution method are heat- an

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contaminant-removal effectiveness, K_t and K_c:

$$K_t = \frac{t_{exh} - t_o}{t_{o.z.} - t_o}$$

$$K_c = \frac{C_{exh} - C_o}{C_{o.z.} - C_o}$$

where:

t = air temperature.

C = contaminant concentration in the supply (o), occupied zone (o.z.), and exhaust air (exh).

Other criteria are heating/cooling loads and air-flow rate, which can be supplied to a room without causing thermal discomfort (drafts, high air-temperature gradient in the occupied zone, etc.) or disturbing the process (reducing local exhaust-capture performance, blowing away the shielding-gas layer from arc welding, etc.).

Piston air distribution provides the greatest heat- and contaminant-removal effectiveness (K_t and K_c greater than 2). Correctly designed and applied passive-thermal-displacement systems may result in heat- and contaminant-removal effectiveness between 1.8 and 2.5. Heat- and contaminant-removal effectiveness using active-thermal-displacement systems can be between 1.2 and 1.8. Correctly designed mixing air-supply systems result in the uniform distribution of air temperature and contaminant concentration throughout the ventilated room, with heat- and contaminant-removal effectiveness equal to 1.

Figure 6 compares air-flow rates and heating/cooling loads, which can be supplied to a ventilated space with mixing, active-thermal-displacement, and passive-thermal-displacement air-distribution systems.

Because of its initial success, displacement ventilation was specified for a wide variety of applications in Europe—regardless of whether or not it offered advantages and provided better indoor air quality compared with mixing systems. The misuse of displacement ventilation can be reduced by applying current knowledge about these systems and by performing a life-cycle cost analysis. (For an economic analysis of displacement ventilation, see Seppanen et al, 1989[7]; Zhivov et al, 1998[8]; and Hu et al, 1999.[9])

This article discusses applications and suggests a practical design strategy for "conventional" displacement-ventilation systems.

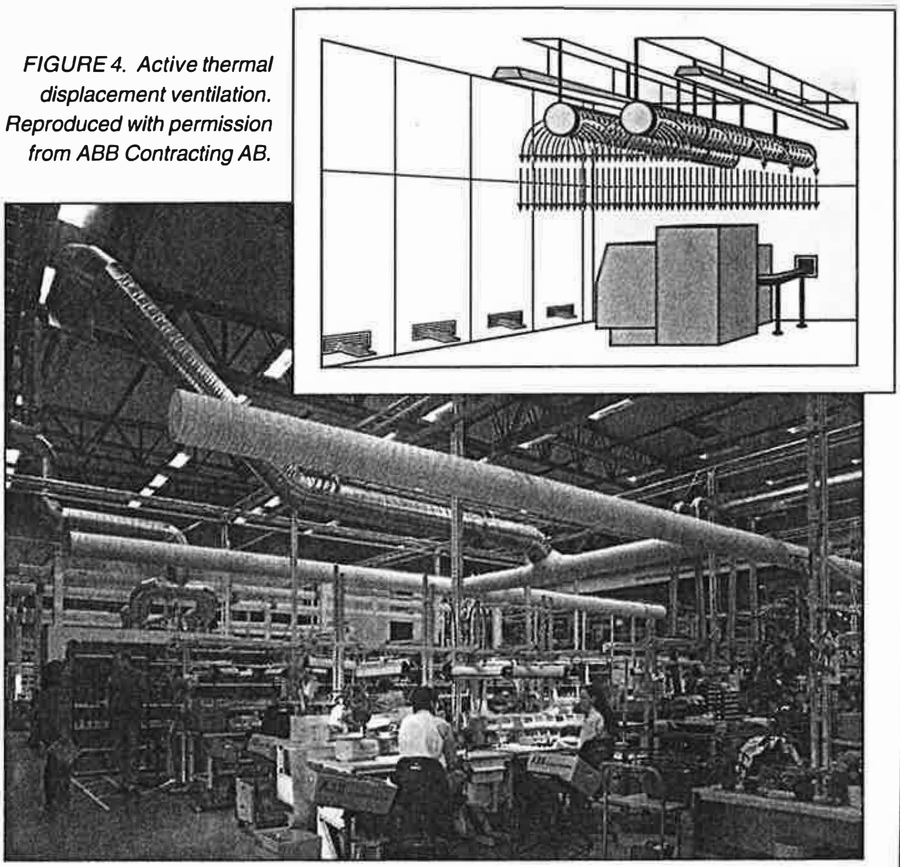


FIGURE 4. Active thermal displacement ventilation. Reproduced with permission from ABB Contracting AB.

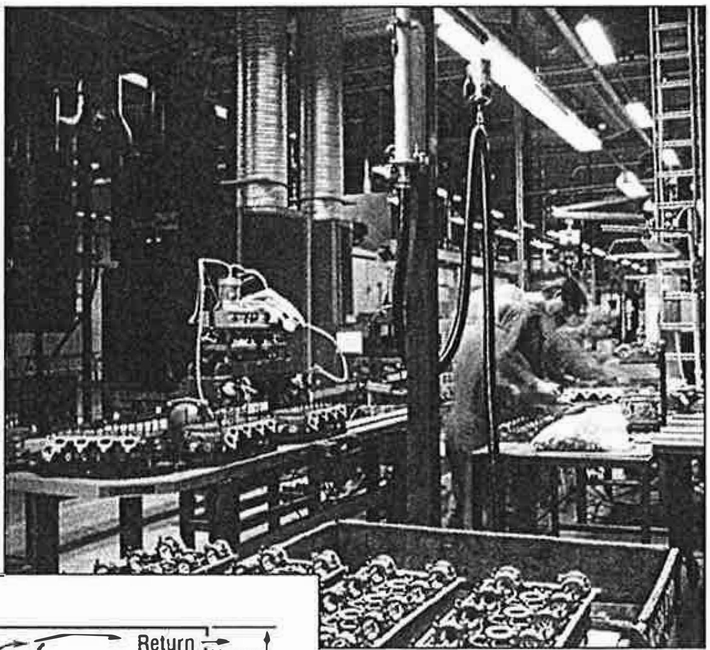
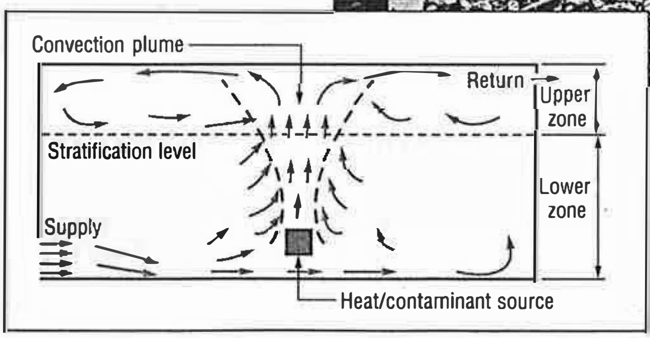


FIGURE 5. Thermal passive-displacement ventilation. Reproduced with permission from ABB Contracting AB.



APPLICATION

Among the major advantages of displacement-ventilation systems is better air quality in the breathing zone and lower air velocities throughout most of the occupied zone. Displacement ventilation is especially effective in removing contaminants from the occupied zone when a contaminant source is combined with a heat source. When the contaminant source does not generate heat and is located outside of the thermal plumes or when the thermal plumes are not strong enough to rise above the stratification level in the presence of a temperature gradient, high concentrations will occur in the occupied zone.

When considering displacement ventilation, it is important to know that:

- It works best in rooms with a height of 10 ft or more.
- It is not for applications in which contaminants do not have a heat source nearby to create thermal plumes with enough air-flow capacity to carry the contaminants to the upper zone of the room.
- The supply air cannot be heated above the desired room air temperature. Thus, when heating is required, a displacement-ventilation system should be complimented by a separate heating system, such as a hydronic hot-water or steam system with radiators or convectors or a gas-fired system with overhead radiant panels.
- High cooling capacity cannot be achieved because of the limitation of the supply air cooling. The difference between the supply-air temperature and the desired occupied-zone air temperature enables the maintenance of the allowable vertical temperature gradient in the occupied zone: 1 F per ft in commercial buildings with primarily standing activity and 1.4 F per ft with primarily sitting activity. Also, the supply-air-temperature differential is limited to 5.4 F in commercial buildings and to 11 F in industrial spaces with moderate activity to prevent abnormal air velocity (drafts) near the floor level. Through experience, manufacturers have found that the cooling load through the air supply typically should not exceed 12.7 Btuh per sq ft for commercial spaces and 25.4 Btuh per sq ft for industrial spaces with a moderate activity level when regular displacement-ventilation air diffusers are used and can be increased to 19 Btuh per sq ft and 31.7 Btuh per sq ft, respectively, with induction-type air diffusers. Other cooling systems, such as cooling ceilings, may be needed in some climates.
- Considerable physical activity in a ventilated space decreases the heat- and contaminant-removal effectiveness of displacement ventilation. Practical experience has shown that displacement ventilation is not effective in body shops or welding shops that heavily use robotics, as moving car bodies and robotic arms distract temperature and contaminant stratification along room height and, thus, negate the advantages of displacement air supply.[10]

DESIGN APPROACHES

Displacement-ventilation systems are designed either based on the analytical model or using Computational Fluid Dynamic (CFD) codes. The analytical approach is used far more often in designing displacement systems. However, CFD codes can be useful in designing a system for a large room, because the dimensions of the room may be too large for full-scale measurements, which are important to support an analytical method, and because a design for such a room often is unconventional. The use of CFD codes for practical three-dimensional computations requires expertise and computational power that usually are unavailable to designers. Also, the prediction of velocities and temperatures in rooms with displacement ventilation using CFD codes generally is inaccurate.

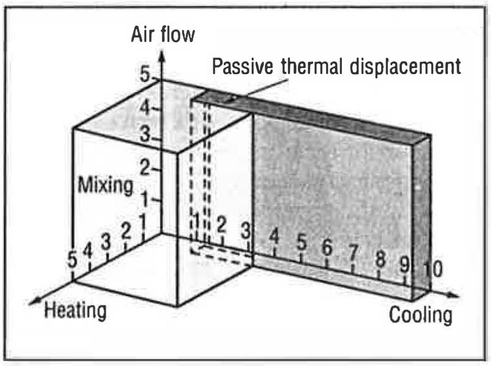


FIGURE 6. Matrix for evaluating air-flow rate and heating- and cooling-load range with mixing, active-thermal-displacement, and passive-thermal-displacement air supply. One unit along the air-flow axis equals 0.38 cfm per sq ft, while one unit along the heating- and cooling-load axis equals 4.8 Btuh per sq ft. Reproduced with permission from ABB Contracting AB.

Experience has shown that an analytical approach can lead to a simple and practical design method that produces good results for most applications. To use an analytical approach, a designer must consider:

- Air flows created by supply devices.
- Temperature and contaminant stratification.
- Convective plumes above heat sources in the stratified environment.
- The ventilation effectiveness of displacement ventilation.

PRIMARILY HEAT-REMOVAL REQUIREMENTS

In designing a displacement-ventilation system for a room with primarily heat-removal requirements, calculations are conducted to obtain the:

- Heat-removal-efficiency coefficient value, K_r.
- Supply air-flow rate for heat-removal purposes, G_{or}.
- Supply-air temperature, t_o.
- Exhaust-air temperature, t_{exh}.
- Vertical temperature gradient, $\Delta t/H_r$.

Assumptions

In designing a displacement-ventilation system for a room with primarily heat-removal requirements, the following is assumed:

- Temperature stratification is a linear function (there is no step stratification, as there is with a contaminant-concentration distribution), $\Delta t = (t_{exh} - t_{floor})/H_r$.
- Heat balances and radiant and turbulent heat exchange are calculated for two zones: the lower zone, which is limited by the height of the occupied zone, and the upper zone, which is above the occupied zone.
- Occupied-zone temperature is the air temperature at the height of $h_{o.z.}$—that is, 3.6 ft for spaces with predominant seating activity and 5.9 ft for spaces with standing activity. Occupied-zone temperature at a certain height is considered to be the same

throughout the occupied-zone area outside of the direct influence of the supply air flow.

• The temperature difference between the head level ($h_{o.z.}$ = 3.6 ft or 5.9 ft) and the ankle level (h_{floor} = 4 in.) is limited to 3.6 to 5.4 F to avoid discomfort. This results in restriction of temperature gradient ($\Delta t/H$) along the room height by 1.1 to 1.4 F per ft with seating activity and 0.7 to 1 F per ft with standing activity.

Heat-removal-coefficient evaluation is based on the model described in detail in Shilkrot and Zhivov, 1992, 1996.[11,12]

Suggested design procedure

In designing a displacement-ventilation system for a room with primarily heat-removal requirements, do the following:

Step 1. List all heat sources in the room.

Step 2. Calculate the average convective heat component, ψ, based on the individual source strength, W (Btu per hr); the convective-heat-component individual source strength, W; and the convective heat component, ψ_i, as follows:

$$\psi = \frac{\Sigma(W_i \times \psi_i)}{\Sigma W_i} \quad (1)$$

Step 3. Calculate the averaged radiant-heat component into the occupied zone based on the individual source strength and the radiant-heat component into the occupied zone, as follows:

$$\varphi = \frac{\Sigma(W_{rad'i} \times \varphi_i)}{\Sigma W_{rad\,i}} = \frac{\Sigma[\varphi_i \times (1-\psi_i)\times W_i]}{\Sigma[W_i \times (1-\psi_i)]} \quad (2)$$

Step 4. Calculate the heat-removal-efficiency coefficient used for the first iteration, K_{to}, as follows:

$$K_{to} = \frac{1}{\varphi(1-\psi)} \quad (3)$$

Step 5. Select the supply-air-temperature difference, $\Delta t_o = t_{o.z.} - t_o$, based on air-diffuser performance data, type of human activity, and the distance between the air diffuser and the nearest person.

Step 6. Calculate the preliminary value of the supply air-flow rate, G_{ot} lbs per min, using $K_t = 0.5\ K_{to}$ for the first iteration:

$$G_{ot} = \frac{\Sigma W_i}{C_p \Delta t_o K_i} \quad (4)$$

Step 7. Evaluate the heat-removal-efficiency coefficient, K_t^*, using the procedure and supporting graphs from Zhivov *et al*, 1997.[13]

Step 8. Compare K_t^*, calculated in Step 7, with the K_t calculated as 0.5 K_{to}. If $(K_t^* - K_t)/K_t^*$ is less than 0.1, proceed to Step 9. If it is greater than 0.1, assume that $K_t = K_t^*$ and repeat the calculations in Step 6.

Step 9. Calculate the exhausted air temperature ($t_{exh} = t_o + K_t\,\Delta t_o$).

Step 10. Calculate the supply-air temperature, t_o, given the occupied zone temperature, $t_{o.z.}$, as follows:

$$t_o = t_{o.z.} - \Delta t_o \quad (5)$$

Step 11. Calculate the temperature gradient, Δt per H, along the room height, as follows:

$$\Delta t/H = \frac{t_{exh} - t_{o.z.}}{H_{room} - h_{o.z.}} = \frac{\Delta t_o(K_t - 1)}{H_{room} - h_{o.z.}} \quad (6)$$

If Δt per H is greater than the Δt per H prescribed to achieve thermal comfort, decrease Δt_o and repeat calculations starting with Step 6.

Step 12. Calculate the supply air-flow rate, G_o, with the final values of K_t and Δt_o, as follows:

$$G_o = \frac{\Sigma W_i}{C_p \Delta t_o K_t} \quad (7)$$

HEAT- AND CONTAMINANT-REMOVAL REQUIREMENTS

In designing a displacement-ventilation system for a room with heat- and contaminant-removal requirements, calculations are conducted to obtain the:

• Contaminant-removal-efficiency coefficient value, K_c.
• Supply air-flow rate for contaminant- and heat-removal purposes, G_o.
• Occupied-zone air concentration, $C_{o.z.}$.
• Exhaust-air concentration, C_{exh}.
• Inhalation-zone concentration, C_e.

Assumptions

In designing a displacement-ventilation system for a room with heat- and contaminant-removal requirements, the following is assumed:

• The distribution of contaminant concentration along the room height is described by a step function. The height of the step, called stratification level (h_{st}), is equal to the height of convective plumes above floor level, where the total air-flow rate in all convective plumes, ΣG_i, is equal to the supply air-flow rate, G_o.

• The stratification level exceeds 3.3 ft in rooms with primarily sedentary activity and 4.9 ft in rooms with standing/walking activity.

• Contaminants released into the space from unheated (isothermal) sources are passive. They can be released below the stratification level into the lower zone in the amount of Q_p^l or above the stratification level into the upper zone in the amount of Q_p^{up}. If the maximum height of a convective plume above the heat and contaminant source is lower than the stratification level because of the temperature-gradient effect, contaminants released by this source are considered passive, with the convective plume above this source not accounted for in the stratification-level calculation. The maximum height of a convective plume can be calculated from the data presented in the report in tabular form using the temperature gradient calculated according to the procedure described in Step 4 of the previous section. Contaminants released into the space from heated sources, Q_{conv}, are transported into the upper zone with a convective plume if the thermal-plume rise is calculated considering that temperature stratification exceeds the stratification level.

• Contaminant concentration in the occupied zone, $C_{o.z.}$, is limited by target level (TLV) or a portion of it (aTLV).

• Background concentration in the outdoor air and insufficient return-air cleaning can lead to contaminated supply air, C_o.

Suggested design procedure

In designing a displacement-ventilation system for a room with heat- and contaminant-removal requirements, do the following:

Step 1. Calculate air-flow rate, G_o, and the temperature gradient, Δt per H, along the room height according to the procedure in Step 4 of the previous section with only heat-removal requirements.

Step 2. Using tabulated data[13] for typical heat sources, calculate stratification height, h_{str}, considering all heated sources located in the occupied zone. The total air-flow rate in convective plumes at the stratification level should be equal to the G_o calculated with Equation 7. Air-flow rates in each convective plume should be calculated ac-

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counting for temperature gradient, Δt_o.

Step 3. Obtain K_c coefficient using the procedure and supporting graphs in Zhivov *et al*, 1997.[14]

Step 4. Calculate exhausted air concentration, C_{exh}, as follows:

$$C_{exh} = C_0 + \frac{Q_p^l + Q_p^{up} + Q_{conv}}{G_0} \qquad (8)$$

Step 5. Calculate occupied-zone-air concentration, $C_{o.z.}$, as follows:

$$C_{o.z.} = C_0 + \frac{C_{exh} - C_0}{K_c} \qquad (9)$$

Step 6. Calculate contaminant concentration, C_e, in the inhalation zone, as follows:

$$C_e = C_{o.z.}$$

when:

h_e is greater than h_{str} (10)

$C_e = C_{o.z.} (Kc [1 - h_{str}/h_e] + h_{str}/h_e)$

when:

h_e is greater than h_{str} (11)

Step 7. If C_e is less than aTLV, proceed with air-diffuser selection. If C_e is greater than aTLV, increase air-flow rate, G_o, with a corresponding decrease in supply-air-temperature difference, Δt_o, and temperature gradient, $\Delta t/H$. Recalculate stratification level, h_{str}.

AIR-DIFFUSER SELECTION

Selection of air diffusers is based on the following:

- Supply air-flow rate, Q_Σ, cfm.
- Supply-air-temperature difference, Δt_o, which typically is limited to 5.4 F for commercial applications when regular displacement air diffusers are used (without induction) and 11 F when induction-type air diffusers are used.
- Length of the "restricted near zone," l_{rest}, which is the area between the air diffuser and the nearest sitting or standing person. This can be different for different locations. The result is different types, sizes, and even shapes of air diffusers in the same room.
- Acoustical limitations (sound level).

Air-diffuser and exhaust/return location

Displacement air diffusers should be positioned so that large obstacles or walls at right angles to the direction of the propagation of the air are at least 3.3 ft beyond the restricted near zone of the unit. The recommended minimum distance between two air diffusers is greater than the sum of their near zones by 3.3 ft.

To decrease ductwork, it often is preferable to locate all air diffusers along one wall. However, the location of diffusers along different walls makes increasing the allowable cooling load provided by the displacement system possible.

When choosing the location of air diffusers, the position of heat loads should be taken into account. Higher air volumes supplied near sources with heavy heat loads will reduce the spreading of excess heat within the occupied zone and, thus, increase heat-removal efficiency.

Air exhausts/returns should be positioned at or near the ceiling. The removal of both excess heat and contaminants can be made more effective by locating exhausts/returns immediately above major sources of heat. In restaurants with smoking and non-smoking areas, exhausting air from the smoking area and locating returns in the non-smoking area is recommended.

AIR-DIFFUSER-TYPE SELECTION

The design and shape of an air diffuser significantly affects thermal comfort in the occupied zone and the length of the restricted near zone. A poorly designed air-supply device will result in a high velocity zone (air velocities greater than 40 fpm) that extends several meters into the room.

In rooms with a significant cooling load, induction air diffusers can be considered. For the same cooling load, induction units allow reduced-size ductwork. In studies conducted at Aalborg University,[15] two air-supply scenarios involving the same cooling load were compared: (1) air supply through a regular air diffuser in the amount of 177 cfm with a temperature difference of 5.4 F and (2) air supply in amount of 71 cfm with a temperature difference of 13.5 F. In both cases, air velocity of 40 fpm was reached at a distance 6.6 ft from the air diffuser.

It is important to mention that induction units can create discomfort in connection with VAV systems. They require a certain primary air-flow rate to be able to induce the room air. As a result, there is a risk of low-temperature air supply to the room when the primary air-flow rate is reduced.

Special air diffusers with internal nozzles that can deflect air to the sides (along the walls) can be used to prevent drafts.

Number of air diffusers

For the best results regarding both comfort and efficiency, consider using several small units instead of a few large ones. Given a choice, one should select air diffusers of different shapes and makes.

Evaluating performance

The simple design approach assumes that the air diffusers selected for a room are of the same type and that their number is minimal. In this case, the minimum value of the restricted near-zone length is selected. Better results can be achieved when air diffusers are selected individually for different locations.

Contrary to mixing-type systems, the main concern for engineers specifying displacement-ventilation systems is the zone of the jet adjacent to the air diffuser. Although the velocity of supply air through a displacement air diffuser is significantly lower than that through a mixing system, a draft sensation may occur if someone is located close to an air diffuser. The combination of low supply-air velocity, supply-air-temperature difference in the range of 0.9 to 5.4 F, and a relatively large supply-air surface, results in supply Archimedes numbers that are substantially greater than those for mixing air jets. The buoyancy forces in air flow from a displacement-ventilation diffuser cause a transformation of jet profiles. A relatively uniform initial jet profile develops into a floor jet with a maximum velocity close to the floor surface. Because of the jet transformation, there is an initial velocity acceleration along the transformation zone, followed by velocity decay in the floor jet. A greater supply-air-temperature difference results in higher jet transformation and, thus, greater air velocity in the floor region.

Based on studies of different air diffusers conducted at Aalborg University, it was concluded that the major concern related to supply-air-temperature difference is its impact on the length of the restricted near zone. If the manufacturer's specification data for the given

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supply-air-temperature difference and the air-flow rate through one air diffuser are available, one can obtain the length of restricted near zone with boundary air velocity (V = 40 fpm). The approximate air velocity at the distance X from an air diffuser can be calculated using the following equation:

$$V_x = 0.2\frac{l_{rest}}{X} \quad (12)$$

When there is a lack of data from the manufacturer and the designer knows only the type of air diffuser, maximum velocity, V_x, can be calculated for the given air flow rate, G_o, and the supply-air-temperature difference ($T_o - T_{o.z.}$) using the following equation:

$$\frac{V_x}{G_0} = K\frac{1}{X} \quad (13)$$

where:

K is an air-diffuser characteristic that depends on the type and shape of the air diffuser and the parameter $(T_o - T_{o.z.})/G_o^2$. This parameter can be regarded as a modification of the Archimedes number.

Data collected at Aalborg University[16] indicates that the first generation of diffusers has a high level of *K* and a radial distribution of air flow. Some diffusers even have a directional flow along the axis at low Archimedes numbers, which results in an increased *K* coefficient. The new generation of air diffusers directs flow along the wall and creates lower velocities perpendicular to the wall. This results in lower *K* values. Equation 13 is valid for X values up to 4.9 ft from the air diffuser.

ACKNOWLEDGMENTS

While this paper is based on the project "Develop a Design Guide for Displacement Ventilation," which was funded by Philip Morris Management Corp., and the project "Ventilation Guide for the Automotive Industry," the views and opinions in this paper are solely those of the authors.

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