AIVC 11091

4101 (RP-949)

A Critical Review of Displacement Ventilation

Xiaoxiong Yuan, Ph.D.

Qingyan Chen, Ph.D.
Member ASHRAE

Leon R. Glicksman, Ph.D.
Member ASHRAE

ABSTRACT

This paper reviews several aspects of the performance of displacement ventilation: temperature distribution, flow distribution, contaminant distribution, comfort, energy and cost analysis, and design guidelines. Ventilation rate, cooling load, heat source, wall characteristics, space height, and diffuser type have major impacts on the performance of displacement ventilation. Some of the impacts can be estimated by simple equations, but many are still unknown.

Based on current findings, displacement ventilation systems without cooled ceiling panels can be used for space with a cooling load up to 13 Btu/(h·ft^2) (40 W/m^2). Energy consumed by HVAC systems depends on control strategies. The first costs of the displacement ventilation system are similar to those of a mixing ventilation system. The displacement system with cooled ceiling panels can remove a higher cooling load, but the first costs are higher as well.

The design guidelines of displacement ventilation developed in Scandinavian countries need to be clarified and extended so that they can be used for U.S. buildings. This paper outlines the research needed to develop design guidelines for U.S. buildings.

INTRODUCTION

Since the energy crisis in the 1970s, the insulation of buildings has been improved and the ventilation rate has been reduced in order to save energy. However, such a reduction of air supply may cause an increase in the concentration of indoor pollutants. In the United States, about 800,000 to 1,200,000 commercial buildings with 30 to 70 million people have problems related to indoor air quality (IAQ) (Woods 1989). Draft (thermal comfort problems) and "sick-building" syndrome (IAQ problems) are very familiar ailments that are the direct results of the poor distribution of airflow, temperature, and contaminant concentrations. Dissatisfaction with the working environment could result in reduced productivity and economic loss. Since up to 90% of a typical person's time is spent indoors, indoor air quality is increasingly recognized as an essential factor for the prevention of human diseases and the promotion of people's comfort and welfare. Therefore, a good ventilation system that can provide good IAQ and save energy is crucial.

Since displacement ventilation was first applied to a welding industry in 1978 (Belin 1978), it has been increasingly used in Scandinavia as a means of ventilation in industrial facilities to provide good IAQ and save energy. Recently, its use has been extended to ventilation in offices and other commercial spaces where, in addition to air quality, comfort is an important consideration. In 1989, Nordic countries estimated that displacement ventilation accounted for a 50% market share in industrial applications and 25% in office applications (Svensson 1989).

A typical displacement ventilation system for cooling, as shown in Figure 1, supplies conditioned air from a low side wall diffuser to the occupied zone. The supply air temperature

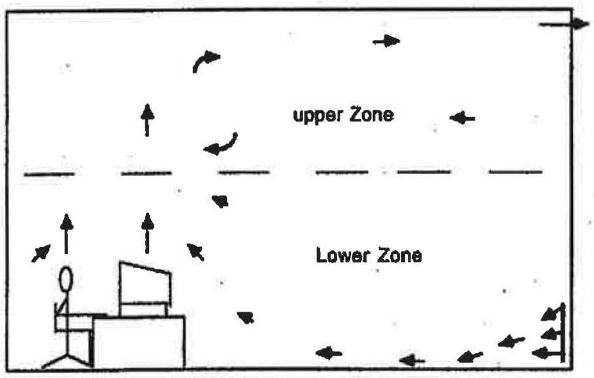

Figure 1   Sketch of displacement ventilation.

Xiaoxiong Yuan is a post-doctoral associate, **Qingyan (Yan) Chen** an assistant professor, and **Leon R. Glicksman** a professor in the Building Technology Program, Massachusetts Institute of Technology, Cambridge.

BACK TO PAGE ONE

is slightly lower than the desired room air temperature and the supply air velocity is low (lower than 100 fpm or 0.5 m/s). The supply air is spread over the floor and then rises as it is heated by the heat sources (e.g., persons, computers) in the occupied zone. Heat sources create upward convective flows in the form of thermal plumes. These plumes bring heat and contaminants that are less dense than air from the surrounding occupied zone to the upper space of the room. Exhausts are located at or close to the ceiling. Through the exhausts, the warm and polluted air is removed from the room. The air volume in a plume increases as it rises because the plume entrains ambient air. A stratification level exists where the airflow rate in the plumes equals the supply airflow rate. Two distinct zones are thus formed within the room; one lower zone below the stratification level with no recirculation flow (close to displacement flow) and one upper zone with recirculation flow. In an ideal situation, the occupied zone is within the lower zone with high IAQ. Hereafter, this ventilation system, without cooled ceiling panels, is referred to as the *displacement ventilation system.*

Research on displacement ventilation has been mainly conducted in Scandinavian countries. Since U.S. buildings have different layouts and higher cooling loads than Scandinavian buildings and there are differences in comfort sensation between American and European, direct applications of the Scandinavian results for U.S. design are not feasible. The objectives of this paper are to:

- review the state of the art of displacement ventilation research and design, including (1) temperature distribution, (2) flow distribution, (3) contaminant distribution, (4) comfort, (5) energy and cost analysis, and (6) design guidelines;
- outline research needed to develop design guidelines for the displacement system for U.S. buildings.

TEMPERATURE DISTRIBUTION

Since displacement ventilation systems supply cold fresh air directly to the occupied zone, potential draft exists in the floor level. In addition, the large temperature stratification that exists in a space with displacement ventilation may also cause discomfort. Therefore, a designer needs information about the air temperature distributions in order to achieve comfortable spaces with displacement ventilation.

Dimensionless Temperature and Vertical Temperature Gradient

Researchers found that the air temperature is nearly constant in horizontal directions except in the region near the supply diffusers. Figure 2 plots the vertical temperature profiles in offices obtained from several different investigations. The dimensionless temperature of air near the floor,

$$\theta_f = \frac{T_f - T_s}{T_e - T_s}$$

where T_f, T_s, and T_e stand for the air temperature near the floor, at the supply, and at the exhaust, respectively, varies from 0.2 to 0.7, and the air temperature gradient is not the same for different investigations. The discrepancies could be due to different thermal and flow conditions, such as ventilation rate, cooling load, heat source type and position, wall temperature distribution, wall radiative characteristics, space height, and diffuser type. The air temperature gradient is assumed to be constant in current design guidelines (Nielsen 1993). Since the temperature difference between head and feet is a critical criterion of thermal comfort, it is desirable to have a model to predict the vertical temperature gradient in the occupied zone.

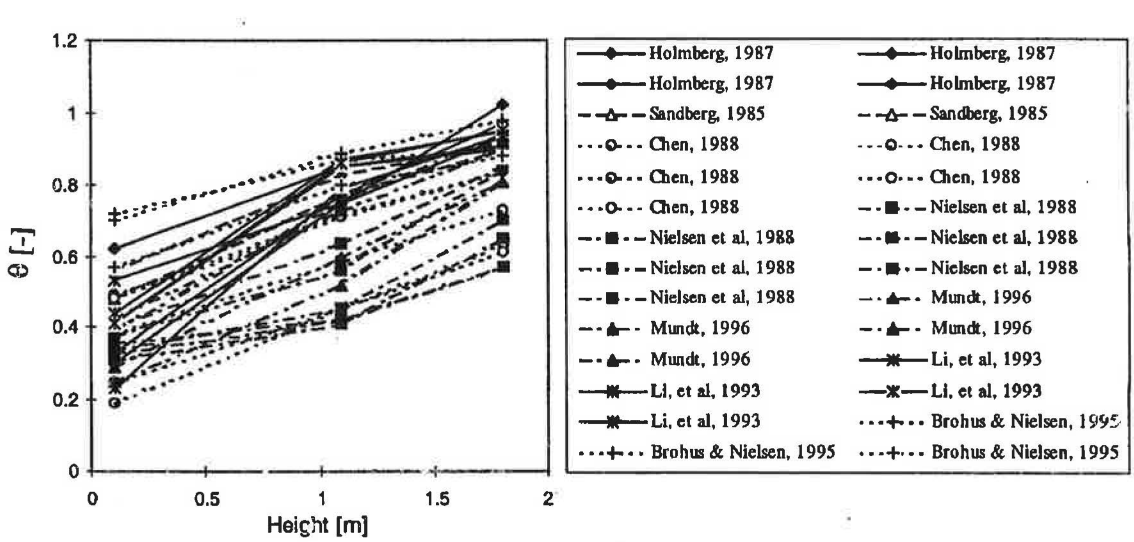

Figure 2 Temperature profiles in office rooms.

BACK TO PAGE ONE

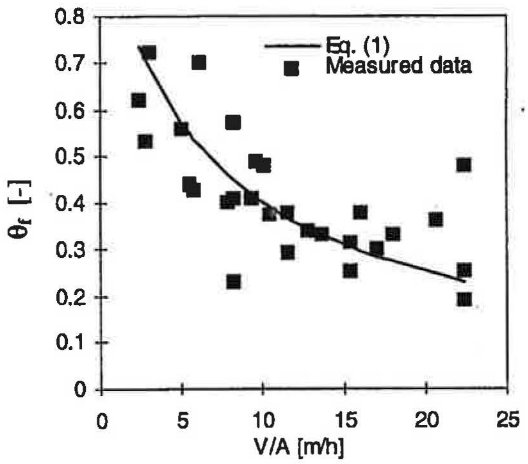

Figure 3 *Dimensionless temperature near the floor vs. supply flow rate.*

Impact of Ventilation Rate and Cooling Load

The experimental data obtained by Sandberg (1985) shows that θ_f is between 0.56 and 0.48 when the air change rate is between 2 and 4 ACH. Chen et al. (1988) reported that θ_f decreases from 0.4 to 0.2 when the air change rate increases from 3 to 5 ACH. Similar results can be found from Mundt (1990), Li et al. (1992), etc.

Mundt (1990) developed a formula to calculate θ_f based on a simple model of ceiling to floor heat transfer. θ_f is a function of ventilation rate given as:

$$\theta_f = \frac{1}{\frac{V\rho C_p}{A}\left(\frac{1}{\alpha_r} + \frac{1}{\alpha_{cf}}\right) + 1} \tag{1}$$

where,

V = ventilation flow rate (ft^3/h or m^3/h),

ρ = air density (lb/ft^3 or kg/m^3),

C_p = specific heat at constant pressure (Btu/[lb·°F] or J/[kg K]),

A = floor area (ft^2 or m^2),

α_r = radiative heat transfer coefficient from ceiling to floor (Btu/[h· ft^2· °F] or W/[m^2 · K]),

α_{cf} = convective heat transfer coefficient from the floor to room air (Btu/[h· ft^2 · °F] or W/[m^2· K]).

As shown in Figure 3, Equation 1 is in good agreement with most measured data in the literature (the same references cited in Figure 2), when α_r = 0.9 Btu/(h·ft^2·°F) (5 W/[m^2 K]) and α_{cf} = 0.7 Btu/(h· ft^2·°F) (4 W/[m^2 K]). This equation accounts for the impact of cooling load on θ_f because ventilation rate and cooling load are interrelated.

Impact of Heat Source Type and Position and Wall Characteristics

Nielsen (1996) pointed out that the vertical air temperature gradient is strongly related to the surface temperature of the heat sources. Based on experimental results obtained in rooms with a variable height from 8.3 ft to 15 ft (2.5 m to 4.5 m), Nielsen presented a chart to determine the air temperature near the floor.

It is a common practice to assume adiabatic thermal conditions for internal walls. In many cases, the "internal walls" are not adiabatic. Since wall area is large, a small temperature difference between the walls and room air could result in a significant downflow (if the walls are colder) or upflow (if the walls are warmer). In addition, the temperature along a vertical line of an internal wall is not a constant and there is a temperature gradient in room air. Heat transfer occurs between room air and internal walls. Most of the investigations neglect the impact of internal wall temperature on the vertical temperature gradient.

A study conducted by Jarnyr (1982) showed vertical temperature profiles at five different times of day in a workshop. The temperature gradient increased from 0.23°F/ft (0.38 K/m) in the early morning to 0.42°F/ft (0.7 K/m) at noon and then decreased slightly to 0.39°F/ft (0.65 K/m) in the evening. The nondimensional temperature near the floor, θ_f, varied from 1/3 in the morning to 1/7 in the afternoon. It is clear that heat from external walls contributes to the temperature gradient.

Li et al. (1992) showed that heat conduction through walls and radiation between room surfaces, particularly between ceiling and floor, make a significant contribution to the vertical temperature profile. Li et al. (1992) suggested a four-node model and a multinode model to include the contribution of radiative heat transfer and conduction through walls.

Mundt (1996) recently extended her early model (Mundt 1990) to consider the influences of heat transfer through the building enclosure and the heat sources on the vertical temperature profile. The new model relates air temperatures near the floor and ceiling to room geometry and the heat transfer among the room air, heat sources, floor, and ceiling.

The models of both Mundt (1996) and Li et al. (1992) could predict the vertical temperature profiles. However, the models do not closely predict the results of various investigators. Since both models assume the air temperature gradient to be constant, they need improvements.

Cooled ceiling panels with displacement ventilation are often used in spaces with a high cooling load. The vertical temperature gradients in the spaces with the cooled ceiling panels are smaller than those without the panels. The temperature distributions are almost uniform in the upper zones, as reported by Skistad (1994) and Taki et al. (1996). If the panel temperature is too low, the displacement ventilation could be transformed into mixing ventilation.

BACK TO PAGE ONE

Impact of Space Height

Displacement ventilation is more suitable for high spaces, such as concert halls and workshops. Skistad (1989) studied temperature profiles in a concert hall with supply openings under chairs. The temperature rises rapidly from the supply air temperature at the floor to the elevation where people are located. Above the people, there is only a slight temperature gradient up to the elevation where the lights are located. At that level, another temperature jump occurs, which brings the air temperature up to the exhaust air temperature at the ceiling level.

Recently, Niemela and Koskela (1996) made measurements in a large industrial hall with a height of 90 ft (27 m). Their results show that the temperature increases with elevation in the zone lower than 23 ft (7 m). In the upper zone, the temperature is almost a constant. These measurements confirm again that vertical temperature gradient is not a constant. Large spaces may be divided into a few zones for determining the temperature distribution.

Impact of Diffuser Type

Skaret (1986) and Nielsen et al. (1988) investigated the impact of supply diffusers on the temperature distribution. It is better to increase the entrainment of room air so as to decrease the temperature gradient in the occupied zone. The performance of diffusers is critical to avoid draft near the diffusers. Recently, manufacturers have developed new products such as aspirating diffuser and modulating diffuser. The performance data can be found from product catalogs.

FLOW DISTRIBUTION

One important feature in displacement ventilation is to properly control and design the airflow distribution. Proper distribution will ensure good air quality and comfort level in the space. For example, well-designed displacement ventilation can achieve a one-dimensional displacement flow in the occupied zone and bring the contaminants to the upper zone. Both thermal plumes and supply air from diffusers play an important role in the airflow distribution.

Impact of Thermal Plumes

The thermal plume generated by a heated object will increase its volume with the height above the object. In an environment with temperature stratification, such as a space with displacement ventilation, the air temperatures in the plume and surrounding it are identical at a certain level. Higher than this level, no buoyancy force exists in the plume. Therefore, the thermal plume can only reach a maximum height in an environment with temperature stratification. Mundt (1992) found that the flow rate of a thermal plume in a space with a vertical temperature gradient is a little smaller than that without the gradient. The maximum height of the plume is significantly shorter. If the height of a plume is less than the height of the occupied zone (6 ft or 1.8 m from the floor), the contaminants within the plume will spread in the occupied zone and cannot reach the upper zone. Therefore, the maximum height of a plume is an important design parameter. Mundt (1992) presented the following equations to calculate the flow rate and the maximum height of a plume in a space with air temperature gradient:

$$V = 0.00238\, Q_c^{3/4} s^{-5/8} (0.004 + 0.039\, y_1 + 0.380\, y_1^2 - 0.062\, y_1^3) \tag{2}$$

$$y_{max} = 0.98 Q_c^{1/4} s^{-3/8} - y_0 \tag{3}$$

$$y_1 = 2.86 (y + y_0) Q_c^{-1/4} s^{3/8} \tag{4}$$

where

V = flow rate in a plume (m^3/s),
y_{max} = maximum height of the plume (m),
s = vertical air temperature gradient (K/m),
Q_c = convective heat emission (W),
y = height above the heated object (m),
y_0 = distance between the virtual origin of the plume and the heated object (m).

The calculated results agree reasonably with the measured data (Mundt 1992; Kofoed and Nielsen 1990) for a plume around a sedentary person, as shown in Figure 4. The equations may also be applied to fluorescent lamps and personal computers. If a desk lamp and a computer have the same power input, the plume flow rate from the lamp is much smaller than that from the computer (Mundt 1992) because the lamp has a smaller area than the computer. A heat source with a small area has a plume with a lower entrainment and a lower flow rate.

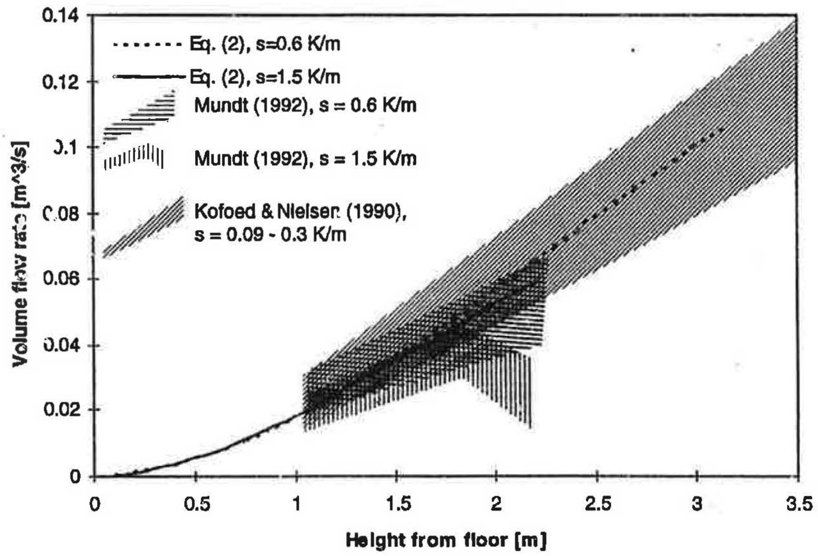

Figure 4 *Volume flow rate around and above a person.*

BACK TO PAGE ONE

In many cases, a heated object is placed close to a wall. Due to the Coanda effect, a plume close to a wall can be considered as one half of the flow in a free plume with a double convective heat emission 2 Q_c. The flow rate of a plume close to a corner is about one quarter of the rate in a free plume with 4 Q_c. The plumes generated by a number of sources near each other may form a large plume with a flow rate of about $N^{1/3}$ V, where N is the number of the heat sources and V is the flow rate in a free plume (Nielsen 1993).

Impact of Walls

Buoyancy will drive airflow up (or down) along a hot (or cold) vertical surface, such as a wall. The flow rate in the turbulent boundary layer may be calculated from Nielsen (1993):

$$V = 0.0028 \Delta T_w^{2/5} y^{6/5} l \qquad (5)$$

where

V = flow rate in the boundary layer (m^3/s),

ΔT_w = temperature difference between room air and the wall surface (K),

y = length measured from the leading edge (m),

l = horizontal width of the surface (m).

The up or down airflow along a wall is a typical wall layer. Heiselberg (1993) presented a formula to calculate the maximum velocity in the layer. For a modest temperature difference of a few degrees between the wall and room air, the flow along the wall may be as large as that from several heat sources in the room, such as people or equipment.

Impact of Diffusers

Since relatively cold air is supplied directly to the occupied zone, the velocity has to be well controlled to avoid draft. The velocity near a diffuser depends on the flow rate from the diffuser, the temperature difference between the supply and exhaust, and the diffuser type. Figure 5 shows a typical velocity distribution near a diffuser (Nielsen 1993).

The distance from a displacement diffuser to the 40 fpm (0.2 m/s) velocity contour along the center line, l_n, is an important parameter. To make l_n smaller is a primary goal for diffuser manufacturers. Normally, the manufacturers provide charts to determine l_n and velocity distribution near the diffuser in their product catalogs.

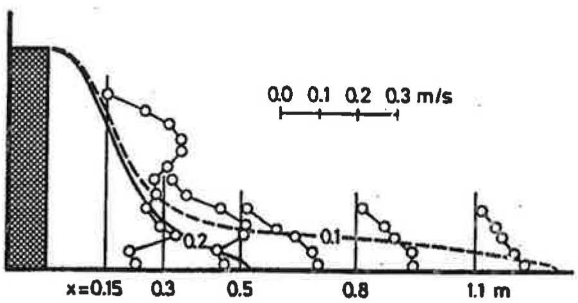

Figure 5 *A typical velocity distribution near a diffuser.*

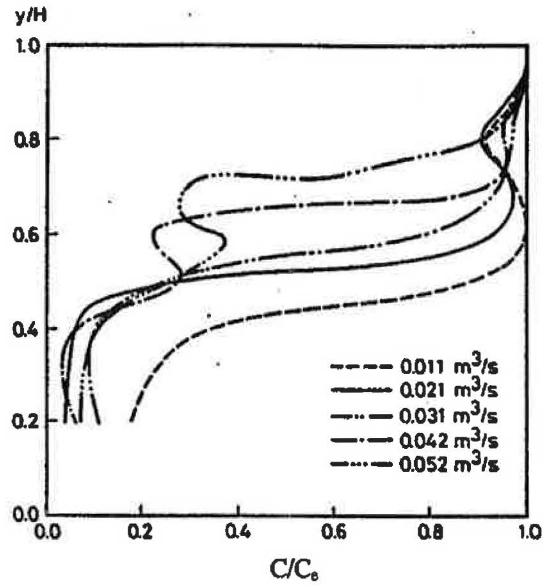

Figure 6 *Typical profiles of the contaminant concentration,* C, *vs. different ventilation rates;* C_e *is the concentration at exhaust.*

CONTAMINANT DISTRIBUTION

The advantage of displacement ventilation is that it may provide better indoor air quality in the occupied zone than mixing ventilation. It is, therefore, important to study the impact of different parameters, such as contaminant source type and location, human body convection, wall surface temperature, and space height, on the contaminant distribution.

Impact of Contaminant Source Type and Location

Typically, the occupied zone with displacement ventilation has a lower contaminant concentration level than that in the upper zone, as shown in Figure 6 (Heiselberg and Sandberg 1990). Chen et al. (1988) showed that both the energy and ventilation efficiencies of displacement ventilation are higher than those of mixing ventilation when the contaminant source is combined with a heat source. The ventilation efficiency increases as the ventilation rate increases or the cooling load decreases.

Stymne et al. (1991) showed that the contaminant concentration level varies significantly in both vertical and horizontal directions, depending on the position of pollutant sources related to the thermal plumes. As illustrated in Figure 7, the contaminant concentration in the occupied zone is high when the contaminant is combined with a weak heat source. The thermal plumes are too weak to reach the upper zone.

BACK TO PAGE ONE

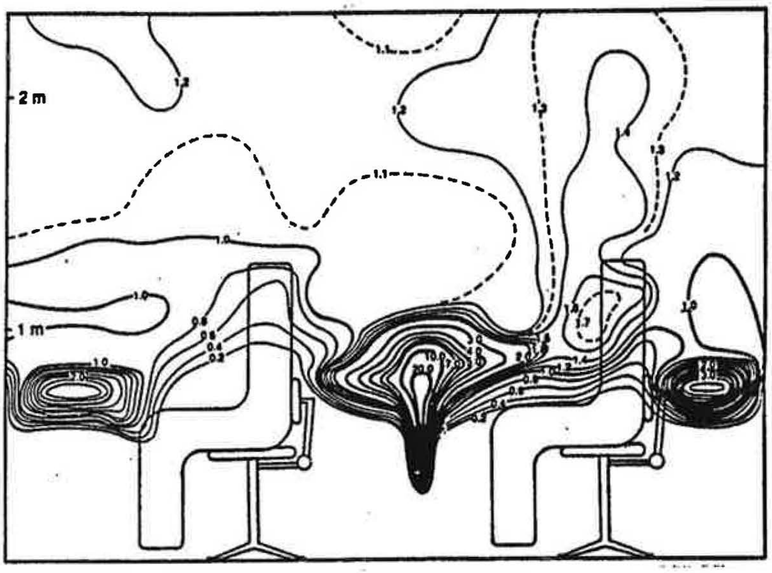

Figure 7 *Concentration contours in a room with a tracer gas emitted above a 4 W heat source at a low level.*

Mundt's (1996) measurements showed that the local air quality is good when a tracer gas source is placed above a heat source that produces a plume that can reach the ceiling. When the tracer gas source is placed outside the thermal plume, the local air quality depends strongly on whether the tracer gas has a positive or negative buoyancy and on the room flow pattern. In this case, the occupied zone might have a high contaminant concentration level. The conclusions are similar to those of Stymne et al. (1991).

Impact of Convection from Human Bodies

Holmberg et al. (1987) found that a free convection flow around a person may protect the breathing zone from surrounding contaminants at the head level, but it may also bring contaminants from the source below the breathing zone. Saeteri (1992) showed that the concentration of a tracer gas in the air inhaled is lower than that at the same elevation some distance from the person because the convective flow around the human body brings fresher air from the floor level directly to the breathing zone. This has been confirmed by Murakami et al. (1997) through a detailed computational-fluid-dynamics simulation. Brohus and Nielsen (1996) found the concentration of inhaled contaminants may be expressed as a linear function of the stratification height.

Impact of Wall Surface Temperature

Nielsen (1993) pointed out that the downdraft caused by a cold wall or window may bring polluted air from the upper zone to the lower zone and reduce ventilation efficiency.

Skistad's (1994) measurements showed that in a displacement ventilated room with cooled ceiling panels, the contaminant concentration increased quickly in the region from the floor to the elevation of 3 ft (1 m), and the concentration in the breathing zone was almost the same as that near the ceiling. A downfall of polluted air from the upper part into the occupied zone was seen by Kruehne and Fitzner (1993) when the cooled ceiling panel temperature was low. However, when Niu (1994) placed contaminant sources within thermal plumes in a space with cooled ceiling panels, he found that the concentration profile would be similar to that without the cooled panel, if the panel surface temperature was kept at 68°F (20°C).

Impact of Space Height

The benefits of displacement ventilation are more likely to be realized in spaces with high ceilings, such as industrial spaces, than those with low ceilings. Skistad (1989) measured the concentration of carbon monoxide emitted by a silicon carbide furnace in a workshop. A clean occupied zone was found in the measurements. Niemela and Koskela's (1996) measurements in a large industrial hall indicated that the concentration of hexavalent chromium in the occupied zone was two or three times lower than that in the upper zone, whereas opposite results were observed for dust.

Impact of Other Parameters

The distribution of contaminants is sensitive to disturbances in room airflow, such as those caused by the opening or closing of doors and the movement of people. Mattsson and Sandberg (1994) showed that both air change efficiency and contaminant removal effectiveness increase when a person simulator moves forward and backward at velocities less than 60 fpm (0.3 m/s). However, when the velocity increases beyond this point, the efficiency will decrease and the displacement ventilation may instead take the form of mixing ventilation. Brohus and Nielsen (1996) found that the movement of people causes an increase of the concentration of inhaled contaminants due to the disturbance to the free convection flow around people. This flow transports fresh air from the floor level to the breathing zone.

Fukao et al. (1996) conducted measurements in two larger offices with different ventilation systems. The results indicated that the air quality with the floor-mounted displacement system is better than that with a ceiling-mounted mixing system, while the thermal environments are almost the same between the two systems. Tanabe and Kimura (1996) measured the age of air in an office room with three different ventilation systems. They concluded that a wall-mounted displacement system provides better air quality than a floor-mounted displacement system, and the floor-mounted system is better than a ceiling-mounted mixing system.

COMFORT

The primary reason for using displacement ventilation is to achieve a high level of IAQ. However, the ventilation must maintain an acceptable comfort level. Previous investigations showed that large vertical temperature gradient and draft are

BACK TO PAGE ONE

the two main causes of discomfort with displacement ventilation. To reduce the temperature gradient, the supply flow rate must be increased. This will lead to a high air velocity at the floor level and to a high draft risk. It is also not feasible to increase the ventilation rate because of energy concerns.

Draft Risk Assessment

Many researchers (Chen 1988; Sandberg and Blomqvist 1989; Kegel and Schulz 1989; Olesen et al. 1994; Akimoto et al. 1995; Taki et al. 1996) reported that displacement ventilation may generally provide a good thermal comfort environment in various spaces. However, the draft risk in the floor level seems rather high in spaces with displacement ventilation. Melikov and Nielsen (1989) evaluated the thermal comfort condition in 18 displacement ventilated spaces. Within the occupied zone, they found that 33% of measured locations had higher than 15% of dissatisfied people due to draft. Also, 40% of the locations were found to have a temperature difference between head and foot larger than 5.4°F (3.0°C).

Some measures are available to reduce the discomfort level caused by temperature gradient. Glicksman et al. (1996) used low-flow-rate fans in the floor level to reduce the temperature difference between the ankle and breathing level of a seated person. The measure does not affect the flow in the upper zone in a room with displacement ventilation if the vertical momentum of the fan exhaust is kept low enough.

Impact of Cooling Load and Cooled Ceiling Panel Temperature

Figure 8 shows the range of cooling load per floor area investigated by some researchers. Most of the studies show that the displacement ventilation system can only provide acceptable comfort if the corresponding cooling load is less than about 13 Btu/(h·ft^2) (40 W/m^2). With higher ceiling heights, the displacement system is capable of removing larger cooling loads (Skistad 1994).

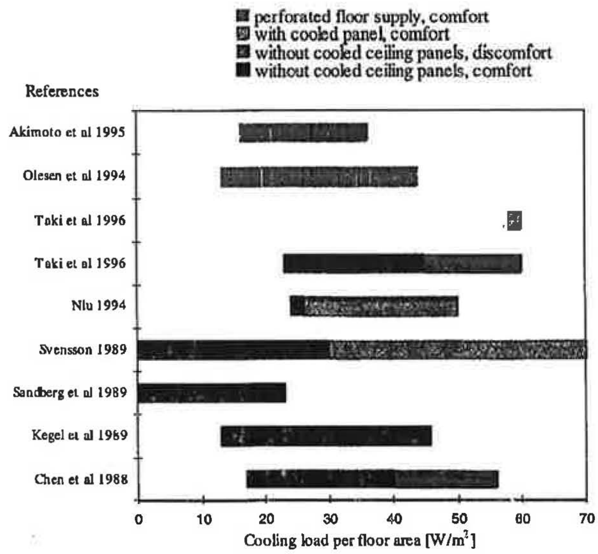

Figure 8 *Ranges of cooling load per floor area investigated by some researchers.*

By increasing the area of the air supply outlet (e.g., supplying air through a perforated floor) or by providing additional heat removal capacity (e.g., using cooled ceiling panels), displacement ventilation may be applied to a space with higher cooling load. Olesen et al. (1994) found that no thermal comfort problems existed under the tested conditions with the cooling loads up to 14 Btu/(h·ft^2) (44 W/m^2) in a room with a perforated floor. Niu (1994) showed that the displacement ventilation combined with cooled ceiling panels may provide a comfort environment at a cooling load up to 16 Btu/(h· ft^2) (50 W/m^2).

Taki et al. (1996) measured the vertical temperature profiles for four different cooling loads with and without cooled ceiling panels. The results showed a significant influence of the panel temperature on the air temperature distribution in the room. The cooled ceiling panel may create downdrafts in the occupied zone. To avoid it, the surface temperature should be higher than 59°F (15°C). The minimum surface temperature is also required to avoid condensation on the panel surface.

ENERGY AND COST ANALYSIS

Annual energy consumption, first costs, and operation and maintenance costs over a life-cycle are important criteria for the evaluation of a ventilation system. Almost all the energy analyses in the literature were done by numerical simulation because it is too expensive and time consuming to conduct hour-by-hour measurements for a building based on a yearly basis.

Energy Analysis

Seppanen et al. (1989) evaluated the energy performance of displacement ventilation systems and mixing ventilation systems in U.S. office buildings. The study is for south, north, and core zones with four representative U.S. climates (Minneapolis, Seattle, Atlanta, and El Paso). They compared different control strategies, such as a variable-air-volume system and constant-air-volume system, and systems with different components, such as recirculation, economizer, and heat recovery device. The energy consumption was found to depend very much on the control strategies and air-handling systems. The energy consumed by displacement systems with heat recovery and variable-air-volume flow control is similar to that of mixing systems.

Chen and Kooi (1988) pointed out the significant impact of the vertical temperature gradient on energy consumption in a room with displacement ventilation when they analyzed a Dutch office with different ventilation systems. The conclusions are similar to those of Seppanen et al. (1989) although the approaches and weather data are different between the two investigations.

Niu's (1994) calculation showed that the annual energy consumption of displacement ventilation with a water-cooled ceiling system is almost the same as that of an all-air system. His investigation used a variable-air-volume system.

Previous studies show that both the supply air temperature and the exhaust temperature in displacement ventilation are higher than those of mixing ventilation. The air temperature difference between the supply and the exhaust is nearly the same between the two ventilation systems. According to Skistad (1994), the temperature difference for displacement ventilation can be larger for high spaces and, therefore, supply airflow rate can be reduced considerably. Note that displacement ventilation may use more natural cooling, since the supply air temperature is 4°F - 6°F (2°C - 3°C) higher than that of mixing-type ventilation.

First Cost Analysis

Seppanen et al. (1989) found that the first cost of a system is difficult to estimate. Their investigation indicated that the first costs of displacement systems are substantially higher than those of mixing systems when cooled ceiling panels are required. Without cooled ceiling panels, the costs of the displacement system are similar to those of a mixing system. Skistad (1994) also reported that there is no significant first cost difference between the two systems, except that the cost of diffusers in the displacement ventilation is higher than that in mixing ventilation.

DESIGN GUIDELINES

According to the analysis in the previous sections, the following parameters are most important in the design of the displacement system:

- Supply airflow rate and temperature.
- Air temperature at floor level.
- Vertical temperature gradient.
- Maximum air velocity at floor level.
- Stratification height (lower zone height) or contaminant concentration gradient.
- Energy consumption.
- First costs and maintenance costs.

The most complete design guidelines available are those developed by Skistad (1994). He used a three-step approach.

1. Determine the required airflow rate for removal of surplus heat based on the cooling load and the air temperature difference between supply and exhaust openings. The temperature difference is calculated assuming $\theta_f = 0.5$ and there is a constant vertical temperature gradient.
2. Find the required airflow rate for removal of pollutants according to ventilation standards.
3. Choose the larger of the two flow rates determined at Steps 1 and 2 as the ventilation rate. Choose supply diffusers according to the data provided by manufacturers in order to avoid draft.

Despite the simple design guidelines, there are problems. Figure 2 shows that θ_f varies from 0.2 to 0.7, and the vertical temperature gradient is not a constant. The above design guidelines may underestimate the vertical temperature gradient. Too large a gradient may cause discomfort.

FURTHER RESEARCH

Research Needed

The above review shows that many results are available in the literature but further research is needed:

- to develop a universal but simple equation for determining vertical temperature gradient;
- to study general diffuser performance to outline essential information needed from the manufacturers;
- to predict correctly contaminant distribution with a simple method and to calculate the stratification height;
- to conduct energy and cost analysis for U.S. buildings.

For different types of buildings, the design guidelines will be different. It is appropriate to conduct research for several types of buildings in which the displacement ventilation system is more feasible and likely to be used.

Displacement ventilation was initially used successfully in industrial workshops in Scandinavia. The impact of climatic conditions may not be significant on the indoor environment of industrial workshops. The applications of the displacement ventilation to U.S. industrial workshops should be supported, though we may not have confidence in using the available design guidelines.

Applications of displacement ventilation systems to other buildings must be in harmony with architectural design since architects play a key role in building design and spatial arrangement. A national survey among many leading architectural firms has been conducted to determine whether architects would use displacement ventilation in their designs. The survey results, illustrated in Figure 9, show that architects are

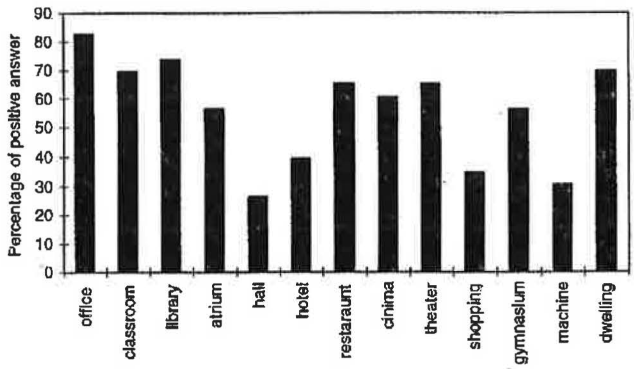

Figure 9 *The survey results from the architects.*

BACK TO PAGE ONE

rather interested in displacement ventilation. About 83% and 70% of architects would consider using displacement ventilation for offices and classrooms, respectively. In addition to industrial workshops, offices and classrooms are likely to have more indoor air quality problems than other types of buildings. Therefore, design guidelines should be developed for U.S. industrial workshops, offices, and classrooms.

The design guidelines available in the literature may not be suitable for U.S. buildings. The major differences between Scandinavian and U.S. offices and schools are the cooling load and spatial arrangement. Most U.S. cities have higher temperatures in summer than Scandinavian cities. Also, offices in the U.S. may have more lighting and equipment. Therefore, the cooling load could be higher in the U.S. than in Scandinavian countries. Since the cooling load limits the applications of the displacement system, it is necessary to investigate typical cooling load in U.S. offices and schools.

Seppanen et al. (1989) investigated the cooling and heating loads of offices in south, north, and core zones in four representative U.S. climates. In the core space, the average cooling load is 4.4 Btu/(h·ft^2) (14 W/m^2) and the maximum load is 7.5 Btu/(h·ft^2) (24 W/m^2). The cooling load in the perimeter zone in most areas of the U.S. is larger than 13 Btu/(h·ft^2) (40 W/m^2), which is considered the maximum limit for displacement ventilation. A field survey on the internal heat gains in different buildings was conducted recently in the greater Boston area. The internal heat gains, including the heat generated by the occupants, lights, computers, printers, copiers, and fax machines, are shown in Table 1. The minimum values for offices are mainly for core zones and the maximum values are mainly for perimeters.

From the survey and the results of Seppanen et al. (1989), the displacement system can be applied to the core zones that have relatively small cooling loads and do not need heating, whereas, in the perimeter zones, the cooling loads seem too high to use the displacement system, although the cooling loads are usually smaller than heat gains. Combined with a cooled ceiling panel, displacement ventilation might be applied to some perimeter zones. However, the first cost

TABLE 1
Internal Heat Gains Found in Different Buildings in Greater Boston

		Single offices	Cubicle offices	Classrooms
No. of buildings surveyed		10	4	7
Minimum	Btu/(h· ft^2)	7	16	10
	W/m^2	21	52	33
Maximum	Btu/(h· ft^2)	28	23	22
	W/m^2	89	71	69
Average	Btu/(h· ft^2)	15	18	16
	W/m^2	49	58	51

would be significantly higher. On the other hand, heating and cooling are required in the perimeter zones. In Scandinavian countries, a radiator is often used for heating in the winter and fresh air is supplied by the displacement system. This implies that the supply air temperature in winter can still be somewhat lower than the room air temperature and a stratified flow can be maintained. However, in many U.S. office buildings, air-conditioning systems are often used for both heating and cooling and there is no radiator available. If displacement ventilation is used in the perimeter spaces, there must be a separated heating system that does not disturb the flow pattern. Convectors, baseboard heaters, radiant panels, or resistance wires could be used. However, the first costs and operating costs with two systems would be very different. Further investigation is needed.

Proposed Design Guidelines

With the proposed research results, it is possible to develop design guidelines for U.S. buildings. The format of the design guidelines may be as follows:

1. Determine the required flow rate of thermal comfort for summer cooling, V_h:
   According to *ASHRAE Standard 55-1992*, choose the air temperature difference between head and foot levels, $T_h - T_f$ (≤5.4 °F or 3 K). Determine the vertical temperature gradient θ_{hf} by applying Equation 6, which needs to be developed:

$$\theta_{hf} = \frac{T_h - T_f}{T_e - T_s} = \text{f (room geometry, heat source type, etc.)} \quad (6)$$

   Let $\Delta T_e = T_e - T_s$, and one may obtain:

$$\Delta T_e = (T_h - T)/\theta_{hf} \quad (7)$$

$$V_h = Q_c/(\rho C_p \Delta T_e) \quad (8)$$

2. Determine the required flow rate for acceptable indoor air quality, V_p, according to ASHRAE Standard 62R.
3. Choose the greater one from V_h and V_p as the ventilation rate, V.
4. Calculate supply air temperature, $T_s = T_f - (\Delta T_e\, \theta_f)$, where $T_f = T_h - (T_h - T_f)$, and T_h is the design room temperature. θ_f can be calculated by Equation 1.
5. Calculate the stratification height and/or contaminant concentration gradient by using a model that will be developed from the proposed research.
6. Select air supply diffusers according to V, ΔT_e, and the information on diffuser performance that will be defined from the proposed research.
7. Check if the displacement system works in winter.
8. Calculate annual energy consumption, annual maintenance costs, and the difference of the first costs between the displacement system and mixing ventilation system. (Using

BACK TO PAGE ONE

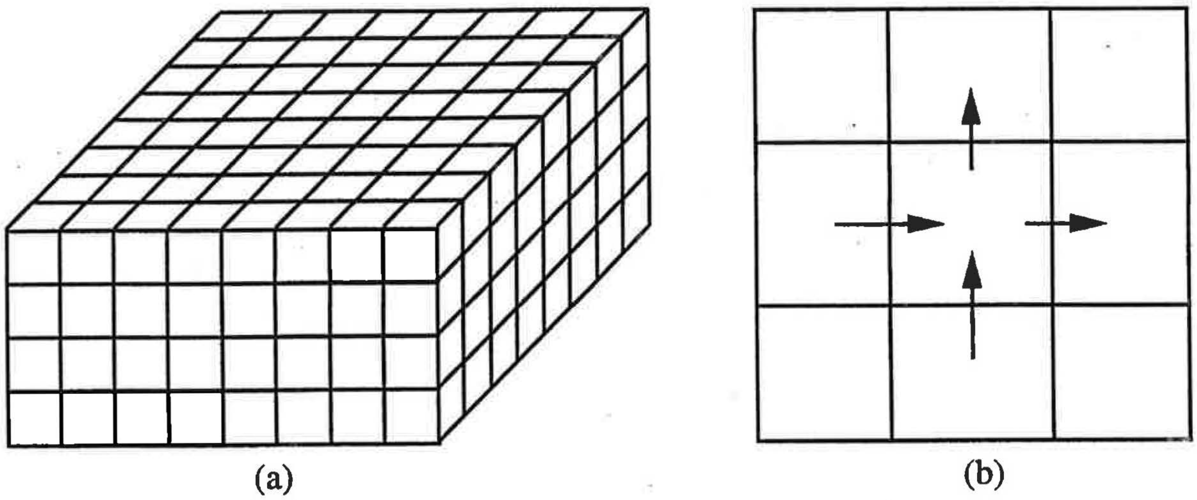

Figure 10 *(a) Typical cell distribution used for room flow simulation with the new model; (b) sketch of mass balance within a cell.*

the relative first costs may avoid difficulty in the estimation of the absolute costs.)

Computer Tools to Aid Design

It would be beneficial if a computational tool were available to aid the design. Recently, Chen and Xu (1997) developed a new flow model for the prediction of indoor airflows. The model subdivides the interior space into a modest number of cells (Figure 10a). The minimum number is about 6 x 6 x 6 for a three-dimensional calculation, but 15 x 15 x 15 would yield a much better prediction. For each cell, conservation of mass is satisfied so that the sum of mass flow into or out of a cell from all its neighbors (Figure 10b) is balanced to zero. Similarly, the exchange of momentum by flow into or out of a cell must be balanced in each direction with pressure, gravity, viscous shear, and momentum transport by turbulent eddies. A similar energy and mass balance for each species of pollutant, e.g., water vapor, must be made at each cell. For turbulent flow, which normally exists in room flow, a zero-equation model is used to describe the turbulent transport of momentum, energy, and different species.

The computer model has been demonstrated for simulating four types of indoor airflows: winter heating, neutral ventilation, summer cooling, and displacement ventilation. The results of the new model agree well with experimental data obtained from the literature (Chen and Xu 1997). For displacement ventilation, Figure 11 shows similar airflow patterns and the distributions of air temperature and tracer gas concentration (helium was used to simulate a contaminant) computed by the new model and the k-ε model (Launder and Spalding 1974). The results have been further compared with experimental data (Chen and Xu 1997). The agreement between the computed and measured results is reasonably good.

The computations can be done on a personal computer. With the same grid number as that used in a CFD simulation with the k-ε model (303 for a single-person office), the new model is 10 times faster than the k-ε model. If the numerical cell is reduced to 153, such as the one shown in Figure 11a-c, the computing cost is reduced by another order. The computing time becomes less than 10 seconds with minimum cell number (63).

A graphical data input interface for the program is under development. The executable program will be available in the public domain. The program could assist designers in determining the distributions of air velocity, temperature, and contaminant concentrations in a space.

CONCLUSIONS

The air temperature near the floor and vertical temperature gradient in the occupied zone of a space with displacement ventilation are important for the evaluation of thermal comfort. The two parameters need to be calculated at the design stage. The ventilation rate, cooling load, heat source type and location, wall radiative characteristics, and diffuser type have significant impacts on the two parameters. Models are available to determine the air temperature near the floor, but there are no simple equations for the calculation of the vertical air temperature gradient.

Airflow distribution depends on the flow rates of thermal plumes, wall layers due to buoyancy, and supply air. The flow rate of a thermal plume can be determined by the heat source type, location, and size. Simple equations are available to calculate the flow rates of thermal plumes and wall layers if the wall to air temperature difference is known. Manufacturers normally provide data to determine the air distribution near a diffuser.

BACK TO PAGE ONE

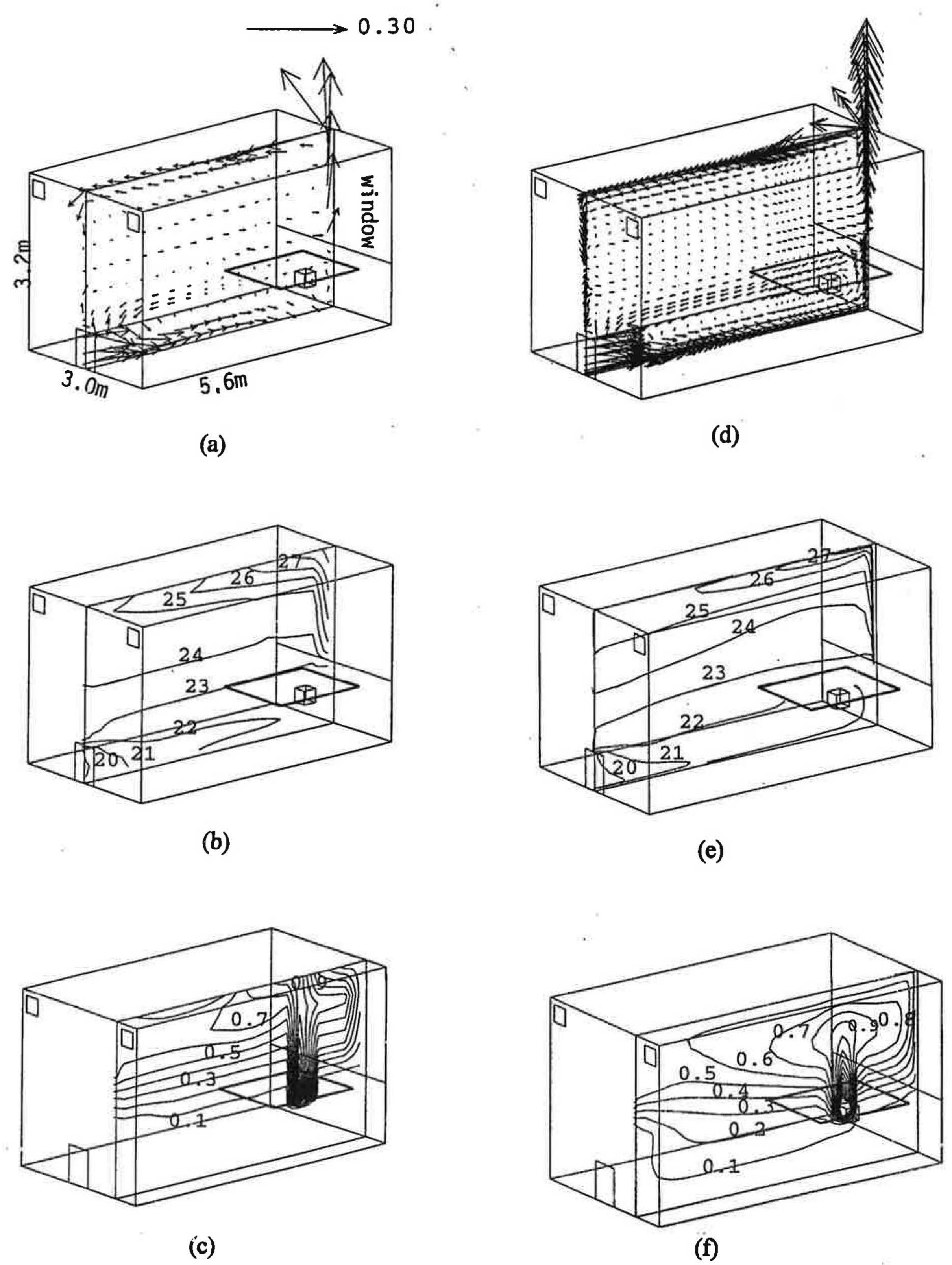

Figure 11 Comparison of the airflow patterns and distribution of air temperature (°C) and helium concentration (%) for (a), (b), and (c) new model and (d), (e), and (f) k-ε model.

BACK TO PAGE ONE

Contaminant concentration distribution depends on contaminant source type and location and associated plume strength. The contaminant concentration in the occupied zone becomes low when the plume can reach the upper zone. It is more beneficial to apply displacement ventilation for spaces with a high ceiling if the contaminants are buoyant gases. Due to upward buoyant convection around a person, the inhaled air is the air from the lower part of the room. Cold walls and/or cooled ceiling panels may generate a downflow that brings contaminants from the upper zone to the occupied zone. Prediction of contaminant distribution is more difficult than air temperature and flow distributions.

Comfort requires the vertical air temperature gradient in the occupied zone to be less than 1.2 °F/ft (2 K/m) for displacement ventilation. Draft possibly takes place near a diffuser. Based on a review of the literature, current limits for the displacement ventilation system indicate it is only suitable for spaces with a cooling load less than 13 Btu/(h·ft^2) (40 W/m^2). With cooled ceiling panels, displacement ventilation can remove a higher cooling load (16 Btu/(h·ft^2) or 50 W/m^2). Further investigation for U.S. designs will help refine these limits. It is important that the panel temperature should not be lower than 59°F (15°C) to avoid disturbance to the room airflow pattern and condensation on the panel surfaces.

The energy consumed by the displacement ventilation system with heat recovery and variable-air-volume control is similar to that by a mixing ventilation system. The first costs of the displacement system are nearly the same as those of the mixing system. If cooled ceiling panels are used, the first costs will increase significantly.

The existing design guidelines in Scandinavia cannot be used for U.S. buildings with confidence, since U.S. buildings have higher cooling loads, different building layouts, and different heating systems. Further investigations are necessary to develop design guidelines for U.S. buildings. This paper outlines the format of the design guidelines and the use of a computer tool to aid the design.

REFERENCES

Akimoto, T., T. Nobe, and Y. Takebayashi. 1995. Experimental study on the floor-supply displacement ventilation system. *ASHRAE Transactions* 101(2).

Belin, K. 1978, Allmaen ventilation med displacerande stroemning, BFR-rapport R 77:1978, Stockholm.

Brohus, H., and P. V. Nielsen. 1994. Contaminant distribution around persons in rooms ventilated by displacement ventilation. *Proceedings of ROOMVENT'94.*

Brohus, H., and P. V. Nielsen. 1996. Personal exposure in displacement ventilated rooms. *Indoor Air* 6: 157-167.

Chen, Q. 1988. Indoor airflow, air quality and energy consumption of buildings, Ph.D. thesis, Delft University of Technology, The Netherlands.

Chen, Q., and J. van der Kooi. 1988. ACCURACY—a computer program for combined problems of energy analysis, indoor airflow and air quality. *ASHRAE Transactions* 94(2): 196.

Chen, Q., J. van der Kooi, and A. Meyers. 1988. Measurements and computations of ventilation efficiency and temperature efficiency in a ventilated room. *Energy and Buildings* 12(2): 85-99.

Chen, Q., and W. Xu. 1997. Simplified method for indoor airflow simulation. *Proceedings of CLIMA 2000, Brussels.*

Fukao H., M. Oguro, K. Hiwatashi, and M. Ichihara. 1996. Environment evaluation in an office with floor-based air-conditioning system in an office building. *Proceedings of ROOMVENT'96* (3): 315-322.

Glicksman, L. R., L. K. Norford, G. M. Okutan, and K. J. Holden. 1996. Scale model studies of displacement ventilation. *Proceedings of ROOMVENT '96* (3): 347-354.

Heiselberg, P. 1993. Draught risk from cold vertical surfaces. *Proceedings of INDOOR AIR'93.*

Heiselberg, P., and M. Sandberg. 1990. Convection from a slender cylinder in a ventilated room. *Proceedings of ROOMVENT '90.*

Holmberg, R. B., K. Folkesson, L. G. Stenberg, and G. Jansson. 1987. Experimental analysis of office climate using various air distribution methods. *Proceedings of ROOMVENT '87.*

Jarmyr, R. 1982. Displacerande inblasning—Nagra erfarenheter. Tekn. Meddelanden o. 247, Inst. For Uppv.- o Vent. Teknik, KTH, Stockholm.

Kegel, B., and U. W. Schulz. 1989. Displacement ventilation for office buildings. *Proceedings of 10th AIVC Conference.* Vol. 1.

Kofoed, P., and P. V. Nielsen. 1990. Thermal plume in ventilated rooms. *Proceedings of ROOMVENT '90.*

Kruehne, H., and K Fitzner. 1993, Stroemungsuntersuchungen am Klimasystem Quellueftung und Deckenkuehlung. *Ki Klima-Kaelte-Heizung*, No.11.

Launder, B. E., and D. B. Spalding. 1974. The numerical computation of turbulent flows. *Comp. Meth. Appl. Mech. Energy* 3: 269-289.

Li, Y., M. Sandberg, and L. Fuchs. 1992. Vertical temperature profiles in rooms ventilated by displacement: Full-scale measurement and nodal modeling. *Indoor Air* 2: 225-243.

Mattsson, M., and M. Sandberg. 1994. Displacement ventilation ¾ influence of physical activity. *Proceedings of ROOMVENT '94.*

Melikov, A. K., and J. B. Nielsen. 1989. Local thermal discomfort due to draft and vertical temperature difference in rooms with displacement ventilation. *ASHRAE Transactions* 95(2).

Mundt, E. 1990. Convection flow above common heat source in rooms with displacement ventilation. *Proceedings of ROOMVENT '90.*

BACK TO PAGE ONE

Mundt, E. 1992. Convection flows in rooms with temperature gradients—theory and measurements. *Proceedings of ROOMVENT '92.*

Mundt, E. 1996. The performance of displacement ventilation system. Ph.D. thesis, Royal Institute of Technology, Sweden.

Murakami, S., S. Kato, and J. Zhen. 1997. Flow and temperature fields around human body with various room air distribution—CFD study on computational thermal manikin: Part I. *ASHRAE Transactions* 103(1).

Nielsen, P. V. 1993. Displacement ventilation—theory and design. Department of Building Technology and Structural Engineering, Aalborg University, Denmark.

Nielsen, P. V. 1996. Temperature distribution in a displacement ventilated room. *Proceedings of ROOMVENT '96* (3): 323-330.

Nielsen, P. V., L. Hoff, and L. G. Pedersen. 1988. Displacement ventilation by different types of diffusers. *Proceedings of the 9th AIVC Conference*, Warwick.

Niemela, R., and H. Koskela. 1996. Air flow patterns in a large industrial hall with displacement ventilation. *Proceedings of ROOMVENT '96* (3): 363-370.

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

Olesen, B. W., M. Koganei, G. T. Holbrook, J. Seelen, and J. E. Woods. 1994. Evaluation of a vertical displacement ventilation system. *Building and Energy* 29: 303-310.

Saeteri, J. 1992. A breathing mannequin for measuring local ventilation effectiveness. *Proceedings of ROOMVENT '92.*

Sandberg, M. 1985. Luftutbyteseffektivitet, ventilationseffektivitet, temperatureffektivitet I cellkontor. System med luft som energibaerare. Statens institut foer byggadsforskning, Meddelande M85:24.

Sandberg, M., and C. Blomqvist. 1989. Displacement ventilation systems in office rooms. *ASHRAE Transactions* 95(2).

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

Skaret, E. 1986. Hva er effectiv ventilasjon. *VVS & Energi* 5.

Skistad, H. 1989. Fortrengningsventilasjon I komfortanlegg med lavimpuls lufttilforel I oppholdssonene, Norsk VVS Teknisk Forening, Oslo.

Skistad, H. 1994. *Displacement Ventilation.* Taunton, Somerset, England: Research Studies Press Ltd.

Stymne, H., M. Sandberg, and M. Mattsson. 1991. Dispersion pattern of contaminants in a displacement ventilated room. *Proceedings of the 12th AIVC Conference.*

Svensson, A. G. L. 1989. Nordic experiences of displacement ventilation systems. *ASHRAE Transactions* 95(2).

Taki, A. H., D. L. Loveday, and K. C. Parsons. 1996. The effect of ceiling temperatures on displacement flow and thermal comfort—Experimental and simulation studies. *Proceedings of ROOMVENT '96* (3): 307-314.

Tanabe, S., and K. Kimura. 1996. Comparisons of ventilation performance and thermal comfort among displacement, underfloor and ceiling based air distribution system by experiments in a real sized office chamber. *Proceedings of ROOMVENT '96* (3): 299-306.

Woods, J. E. 1989. Cost avoidance and productivity in owning and operating buildings. *Occupational Medicine: State of the Art Reviews* 4(4): 753-770.