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# Models for Prediction of Temperature Difference and Ventilation Effectiveness with Displacement Ventilation

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## ABSTRACT

*Displacement ventilation may provide better indoor air quality than mixing ventilation. Proper design of displacement ventilation requires information concerning the air temperature difference between the head and foot level of a sedentary person and the ventilation effectiveness at the breathing level.*

*This paper presents models to predict the air temperature difference and the ventilation effectiveness, based on a database of 56 cases with displacement ventilation. The database was generated by using a validated CFD program and covers four different types of U.S. buildings: small offices, large offices with partitions, classrooms, and industrial workshops under different thermal and flow boundary conditions.*

*Both the maximum cooling load that can be removed by displacement ventilation and the ventilation effectiveness are shown to depend on the heat source type and ventilation rate in a room.*

## INTRODUCTION

Displacement ventilation has been widely used in Scandinavia during the past twenty years as a means to improve indoor air quality. A traditional displacement ventilation system for cooling, as shown in Figure 1, supplies conditioned air from a low side wall diffuser. Because it is cooler than the room air, the supply air spreads over the floor and then rises along heated plumes set up by heat sources in the room. Therefore, there is vertical temperature stratification in the room air. These heat sources (e.g., persons and computers) create upward convective flows in the form of thermal plumes and bring contaminants from the lower zone to the upper zone. An exhaust located at or close to the ceiling extracts the warm and contaminated room air. Hence, displacement ventilation provides better indoor air quality in the lower zone (occupied zone) than conventional mixing ventilation.

Many researchers have reported that displacement ventilation generally provides an acceptable comfort level in the room. However, a risk of draft exists at the floor level because of the high air velocity and low air temperature. In addition, the temperature difference between the head and foot level may be too large due to the vertical temperature stratification. 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% dissatisfied people due to draft. They found that 40% of the locations had a temperature difference between the head and feet larger than 5°F (3 K), the limit defined by *ANSI/ASHRAE Standard 55-1992, Thermal Environmental Conditions for Human Occupancy* (ASHRAE 1992). Obviously, these displacement ventilation systems were not properly designed. The computational fluid dynamics (CFD) technique and a full-scale experimental rig can be used to determine the temperature and velocity distribution in

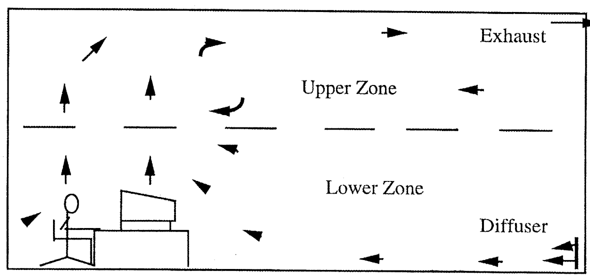


***Figure 1*** *Sketch of displacement ventilation.*

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a room with displacement ventilation. Nevertheless, the tools are not generally available for most designers. Designers need a simple model to predict the temperature difference between the head and foot level.

Moreover, there is no simple model available to estimate the ventilation effectiveness at the breathing level in a room with displacement ventilation. Displacement ventilation has higher ventilation effectiveness than mixing ventilation. The required amount of fresh air can be reduced for displacement ventilation to save energy. Most designs use an assumption of complete mixing to estimate the amount of fresh air needed. Such a calculation would lead to a substantial error.

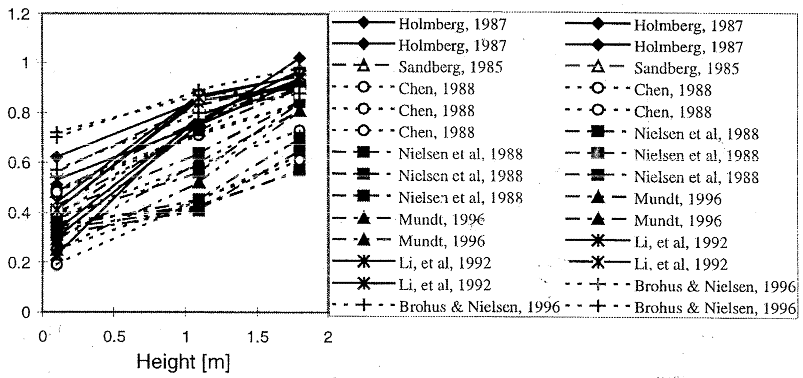


***Figure 3*** *Temperature profiles in offices obtained by different investigators.*

Therefore, the objective of the present study is to develop simple models to estimate the air temperature difference between the head and foot level in a room and the ventilation effectiveness at the breathing level.

## PREVIOUS WORK

In a space with displacement ventilation, the air temperature is nearly constant in the horizontal direction except near the supply diffusers. The air temperature near the floor, $T_f$, is higher than the supply air temperature, $T_s$, because of heat transfer from the floor to the air and the entrainment of the surrounding air by the supply air. Figure 2 presents a simplified temperature profile in a room with displacement ventilation, where $T_e$ is the exhaust air temperature. This linear profile from floor to ceiling is widely used in design. Further, Skistad (1994) suggested

$$T_f - T_s = (T_e - T_s)/2. \tag{1}$$

However, the vertical temperature profile in a room is not linear. Figure 3 plots the vertical temperature profiles in offices obtained by experimental measurements by different investigators; θ, $T - T_s / T_e - T_s$, is defined as the dimensionless vertical temperature normalized by the air temperature difference between the exhaust and supply.

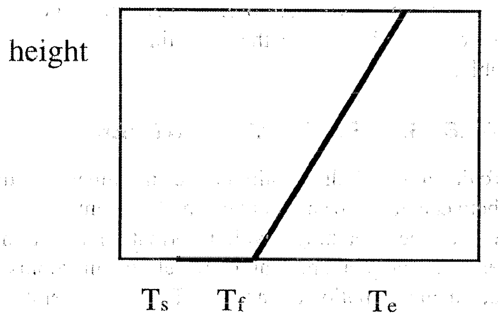


***Figure 2*** *Simplified vertical temperature profile in a room with displacement ventilation.*

The dimensionless air temperature near the floor, $\theta_f$, $T_f - T_s / T_e - T_s$, varies from 0.2 to 0.7 and is not a constant of 0.5, as estimated by Equation 1. Figure 3 shows that the air temperature does not vary linearly from the floor to the ceiling in most cases. Although the use of a linear vertical temperature profile and $\theta_f$ equal to 0.5 are close to the average of the data in Figure 3, average data cannot be used in the design of displacement ventilation.

Sandberg (1985), Nielsen (1988), Chen et al. (1988), Mundt (1990), Li et al. (1992), and others showed that the dimensionless air temperature near the floor decreases as the ventilation rate increases. Mundt (1990) further assumed (1) that the convective heat transfer from the floor to air raises the air temperature from $T_s$ to $T_f$, and (2) that the radiative heat transfer from the ceiling to the floor maintains the energy balance on the floor surface. Then she developed a formula to calculate the $\theta_f$ as

$$\theta_f = \frac{1}{\frac{V\rho C_p}{A}\left(\frac{1}{\alpha_r} + \frac{1}{\alpha_{cf}}\right) + 1}, \tag{2}$$

where

$V$ = ventilation flow rate,

$\rho$ = air density,

$C_p$ = specific heat of air at constant pressure,

$A$ = floor area,

$\alpha_r$ = radiative heat transfer coefficient from the ceiling to the floor,

$\alpha_{cf}$ = convective heat transfer coefficient from the floor to the room air.

As shown in Figure 4a, the prediction by Equation 2 agrees with most of the measured data cited in Figure 3 when $\alpha_r = 0.9$ Btu/(h·ft$^2$·°F) (5 W/m$^2$·K) and $\alpha_{cf} = 0.7$ Btu/(h·ft$^2$·°F) (4 W/m$^2$·K). Point 1 corresponds to a case where the walls were covered with aluminum in the experiment and the radi-

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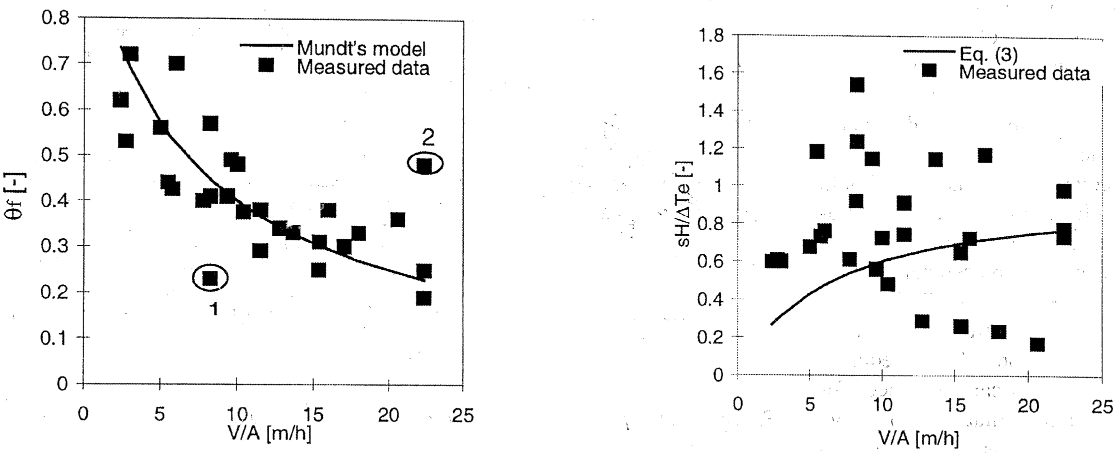


(a) (b)

***Figure 4*** *Performance of Mundt's model: (a) dimensionless temperature near the floor vs. supply flow rate, (b) dimensionless temperature gradient vs. supply flow rate.*

ative heat transfer to the floor was small. Point 2 is for another case where the cooling load was small and the total temperature difference was only 5°F (3 K) in the experiment.

Although not explicitly stated, Equation 2 accounts for the impact of cooling load on $\theta_f$ because the ventilation rate and cooling load are interrelated.

If the temperature varies linearly with elevation, the temperature gradient, $s$, can be estimated as

$$s = \frac{T_e - T_f}{H} = \frac{(1-\theta_f)(T_e - T_s)}{H} \tag{3}$$

or

$$\frac{sH}{T_e - T_s} = 1 - \theta_f = 1 - \frac{1}{\frac{V\rho C_p}{A}\left(\frac{1}{\alpha_r} + \frac{1}{\alpha_{cf}}\right) + 1}, \tag{4}$$

where $H$ = room height.

Unfortunately, Equation 4 does not accurately predict the temperature gradient of the measured data, as shown in Figure 4b. The data used in Figure 4b are the same as those in Figure 3.

Nielsen (1988) investigated the gradient of the vertical air temperature. He found that in a room with a constant cooling load from a concentrated heat source, the temperature gradient decreases slightly as the Archimedes number ($g\beta h\Delta T_e / u_s^2$) increases. The gradient is strongly related to the surface temperature of the heat sources (Nielsen 1992). It seems difficult to predict the nonlinear temperature profile. This is because many parameters, such as ventilation rate, heat source type and position, wall temperature and wall radiative characteristics, space height, and diffuser type contribute to it (Yuan et al. 1998). Actually, it is not necessary to predict the whole vertical profile. Only the air temperatures between the head and foot level are required in the design of comfort conditions.

If the air temperature at the head level of a sedentary person is defined as the room design temperature, it is determined by comfort and is known in the design. If we can establish an accurate model to calculate the air temperature difference between the head and foot level, we can calculate the air temperature near the floor, $T_f$. With Mundt's model relating $T_s$ and $T_f$ (Equation 2) and the steady-state room energy balance equation,

$$Q_t = \rho C_p V (T_e - T_s), \tag{5}$$

we can determine the air supply temperature $T_s$ and exhaust temperature $T_e$. Thus, we can determine the important temperatures needed to design a displacement ventilation system.

On the other hand, the primary purpose of displacement ventilation is to improve indoor air quality. A designer needs a model to estimate the ventilation effectiveness in a room with displacement ventilation. However, the literature review (Yuan et al. 1998) and our studies show that the distributions of contaminant concentration and ventilation effectiveness are strongly influenced by the position of the contaminant sources and the heat sources. Therefore, no general model for the ventilation effectiveness at the breathing level is presently available.

## A DATABASE OF DISPLACEMENT VENTILATION

To develop models to estimate the air temperature difference between the head and foot level and the ventilation effectiveness at the breathing level, information about the air temperature and contaminant distributions in rooms with displacement ventilation are needed. The air temperature and contaminant distributions are needed for a large number of cases to develop an accurate simplified model. This requires a database of the air temperature and contaminant distributions for rooms with various kinds of geometric, thermal, and flow boundary conditions.

There are two approaches to establish a database of air temperature and contaminant distributions: direct measurements and numerical simulations. Direct measurements in rooms with different geometric, thermal, and flow boundary conditions give the most realistic information. However, they are very expensive and time consuming for many difficult cases, and the control of thermal and flow boundary conditions is also difficult. The use of numerical simulations seems a good choice at present.

To establish a large database, the present investigation uses the CFD technique to simulate the air temperature and contaminant concentrations in different rooms with displacement ventilation. The CFD program has been validated by comparison to seven sets of measured data from a small office, a large office with partitions, a classroom, and an industrial workshop (Yuan et al. 1998). The measured data were obtained under different thermal and flow boundary conditions. These data are representative but are not sufficient to develop a comprehensive model. The validated CFD program has been used to expand the database to 56 cases for four types of indoor spaces in the U.S.— small offices, large offices with partitions, classrooms, and industrial workshops.

Figure 5 shows typical configurations of the four types of spaces. The 56 cases break down into 18 cases of small offices (SO), 12 cases of large offices with partitions (LO), 14 cases of classrooms (CR), and 12 cases of industrial workshops (WS). The thermal and flow conditions for the cases are summarized in Table 1. The cases vary the space height ($H$), ventilation rate ($n$), heat generated by occupant ($Q_o$), heat generated by equipment ($Q_e$), heat generated by overhead lighting ($Q_l$), heat from transmitted solar radiation ($Q_{slr}$), and heat from the building envelope other than the transmitted solar radiation ($Q_{wl}$). The table also summarizes the total cooling load ($Q_t$) per floor area.

These thermal and flow boundary conditions cover a wide range of U.S. buildings:

- 8 ft $\leq$ room height $\leq$ 18 ft (2.43 m $\leq$ room height $\leq$ 5.5 m),
- 2 ACH $\leq$ ventilation rate $\leq$ 15 ACH,
- 6.6 Btu/(h·ft²) $\leq Q_t/A \leq$ 38 Btu/(h·ft²) (21 W/m² $\leq Q_t/A \leq$ 120 W/m²),
- $0.08 \leq Q_{oe}/Q_t \leq 0.68$,
- $0 \leq Q_l/Q_t \leq 0.43$,
- $0 \leq Q_{ex}/Q_t \leq 0.92$,

where

$Q_t$ = total cooling load in the room,

$A$ = floor surface area,

$Q_{oe}$ = heat generated by the occupant and equipment,

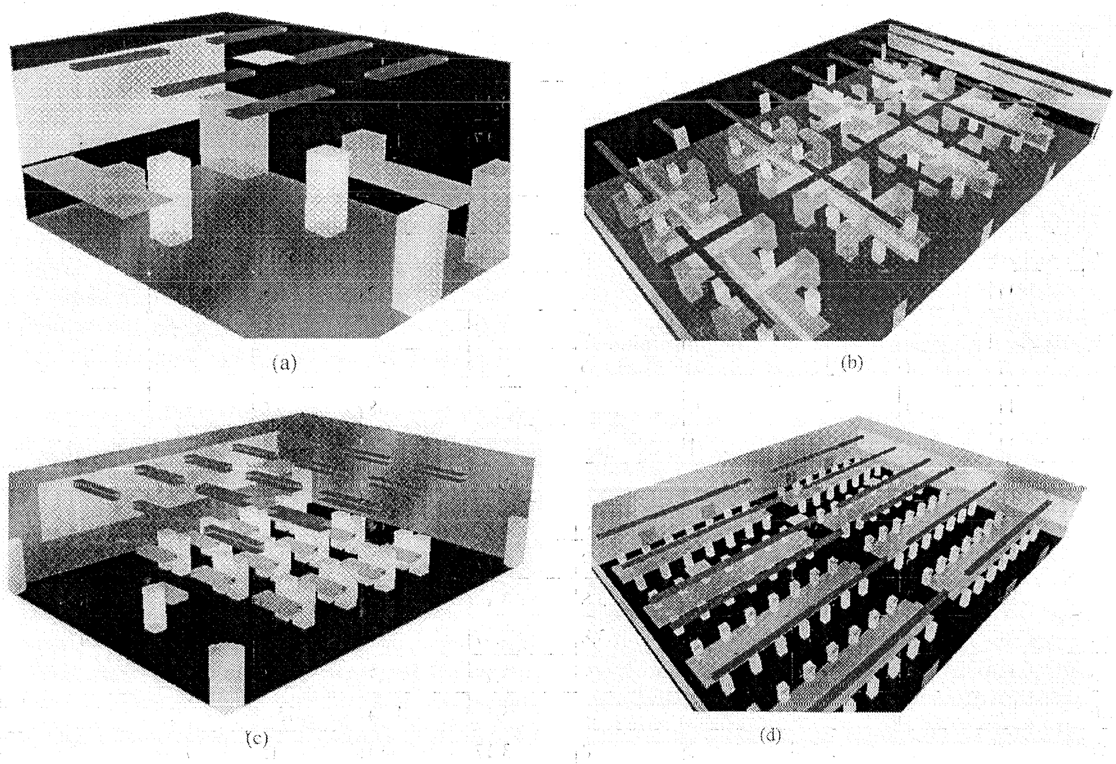


***Figure 5*** *Typical rooms studied: (a) small office, (b) large office with partitions, (c) classroom, and (d) workshop.*

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**TABLE 1a**
**Case Specification (I-P units): SO—Small Offices, LO—Large Offices, CR—Classrooms, and WS—Industrial Workshops**

| Case | $H$ ft | $n$ ACH | $Q_o/A$ Btu/h·ft$^2$ | $Q_e/A$ Btu/h·ft$^2$ | $Q_l/A$ Btu/h·ft$^2$ | $Q_{slr}/A$ Btu/h·ft$^2$ | $Q_{wl}/A$ Btu/h·ft$^2$ | $Q_t/A$ Btu/h·ft$^2$ | $T_{fs}$ °F | $T_{cs}$ °F | $T_s$ °F |
|---|---|---|---|---|---|---|---|---|---|---|---|
| SO1 | 9.2 | 4 | 2.52 | 3.71 | 3.17 | 3.17 | 2.14 | 14.7 | 76.3 | 82 | 60.8 |
| SO2 | 8 | 4 | 2.52 | 3.71 | 3.17 | 3.17 | 2.14 | 14.7 | 75.2 | 81.7 | 57.7 |
| SO3 | 11 | 4 | 2.52 | 3.71 | 3.17 | 3.17 | 2.14 | 14.7 | 77.4 | 82.4 | 64.4 |
| SO4 | 9.2 | 3 | 2.52 | 3.71 | 3.17 | 3.17 | 2.14 | 14.7 | 75.7 | 82.8 | 55.2 |
| SO5 | 9.2 | 6 | 2.52 | 3.71 | 3.17 | 3.17 | 2.14 | 14.7 | 76.8 | 81.3 | 66 |
| SO6 | 9.2 | 4 | 1.26 | 3.71 | 3.17 | 3.17 | 2.16 | 13.5 | 76.1 | 81.5 | 62.1 |
| SO7 | 9.2 | 4 | 2.52 | 0 | 3.17 | 3.17 | 2.23 | 11.1 | 76.1 | 79.9 | 64.4 |
| SO8 | 9.2 | 4 | 2.52 | 1.85 | 3.17 | 3.17 | 2.19 | 12.9 | 76.1 | 81 | 62.6 |
| SO9 | 9.2 | 4 | 2.52 | 3.71 | 0 | 3.17 | 2.23 | 11.6 | 75.7 | 80.4 | 64 |
| SO10 | 9.2 | 4 | 2.52 | 0 | 0 | 3.17 | 2.33 | 8.02 | 75.6 | 78.3 | 67.5 |
| SO11 | 9.2 | 4 | 2.52 | 3.71 | 3.17 | 0 | 0 | 9.41 | 76.3 | 80.1 | 66 |
| SO12 | 9.2 | 4 | 2.52 | 3.71 | 3.17 | 6.34 | 2.04 | 17.8 | 76.6 | 83.5 | 57.7 |
| SO13 | 9.2 | 4 | 1.26 | 0 | 0 | 12.7 | 2.13 | 16.1 | 76.5 | 82 | 59.4 |
| SO14 | 9.2 | 6 | 2.52 | 3.71 | 3.17 | 6.34 | 2.07 | 17.8 | 77.4 | 82.6 | 64 |
| SO15 | 9.2 | 6 | 2.52 | 3.71 | 3.17 | 14.1 | 1.91 | 25.4 | 78.6 | 85.6 | 59.2 |
| SO16 | 9.2 | 8 | 2.52 | 3.71 | 3.17 | 14.1 | 1.91 | 25.4 | 79.3 | 84.9 | 63.5 |
| SO17 | 9.2 | 9 | 2.52 | 3.71 | 3.17 | 14.1 | 1.91 | 25.4 | 79.5 | 84.6 | 64.9 |
| SO18 | 9.2 | 15 | 2.52 | 3.71 | 3.17 | 27 | 1.63 | 38 | 83.1 | 88 | 66.2 |
| LO1 | 9.8 | 4 | 2.07 | 3.04 | 3.8 | 3.17 | 1.19 | 13.3 | 78.1 | 81.7 | 63 |
| LO2 | 11 | 4 | 2.07 | 3.04 | 3.8 | 3.17 | 1.19 | 13.3 | 78.4 | 81.9 | 64.6 |
| LO3 | 13 | 4 | 2.07 | 3.04 | 3.8 | 3.17 | 1.19 | 13.3 | 79 | 81.9 | 66.9 |
| LO4 | 11 | 3 | 2.07 | 3.04 | 3.8 | 3.17 | 1.19 | 13.3 | 78.3 | 82.4 | 60.4 |
| LO5 | 11 | 6 | 2.07 | 3.04 | 3.8 | 3.17 | 1.19 | 13.3 | 78.6 | 81.1 | 68.5 |
| LO6 | 11 | 4 | 2.07 | 0 | 3.8 | 3.17 | 1.29 | 10.3 | 77.9 | 80.2 | 66.9 |
| LO7 | 11 | 4 | 2.07 | 1.52 | 3.8 | 3.17 | 1.22 | 11.8 | 78.3 | 81 | 65.7 |
| LO8 | 11 | 4 | 2.07 | 6.09 | 3.8 | 3.17 | 1.1 | 16.2 | 79 | 83.5 | 62.4 |
| LO9 | 11 | 4 | 2.07 | 3.04 | 0 | 3.17 | 1.32 | 9.6 | 77.5 | 79.7 | 67.6 |
| LO10 | 11 | 4 | 2.07 | 0 | 0 | 3.17 | 1.38 | 6.62 | 77 | 78.1 | 69.8 |
| LO11 | 11 | 4 | 2.07 | 3.04 | 3.8 | 0 | 0 | 8.91 | 77.2 | 79.7 | 68.4 |
| LO12 | 11 | 4 | 2.07 | 3.04 | 3.8 | 6.34 | 1.06 | 16.3 | 79.9 | 83.5 | 61.9 |
| CR1 | 11 | 3 | 5.64 | 0 | 3.8 | 3.17 | 1.17 | 13.8 | 78.6 | 82.2 | 59.7 |
| CR2 | 9 | 3 | 5.64 | 0 | 3.8 | 3.17 | 1.2 | 13.8 | 77.5 | 81.7 | 55.2 |
| CR3 | 13 | 3 | 5.64 | 0 | 3.8 | 3.17 | 1.17 | 13.8 | 79.3 | 82.4 | 63 |
| CR4 | 11 | 5 | 5.64 | 0 | 3.8 | 3.17 | 1.2 | 13.8 | 79 | 81.3 | 66.6 |
| CR5 | 11 | 3 | 4.28 | 0 | 3.8 | 3.17 | 1.2 | 12.5 | 78.3 | 81.5 | 61.2 |
| CR6 | 11 | 3 | 7 | 0 | 3.8 | 3.17 | 1.14 | 15.1 | 79 | 82.8 | 58.3 |
| CR7 | 11 | 3 | 5.64 | 1.2 | 3.8 | 3.17 | 1.14 | 15 | 78.8 | 82.9 | 58.5 |

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**TABLE 1a (Continued)**
**Case Specification (I-P units): SO—Small Offices, LO—Large Offices, CR—Classrooms, and WS—Industrial Workshops**

| Case | $H$ ft | $n$ ACH | $Q_o/A$ Btu/h·ft² | $Q_e/A$ Btu/h·ft² | $Q_l/A$ Btu/h·ft² | $Q_{slr}/A$ Btu/h·ft² | $Q_{wl}/A$ Btu/h·ft² | $Q_t/A$ Btu/h·ft² | $T_{fs}$ °F | $T_{cs}$ °F | $T_s$ °F |
|---|---|---|---|---|---|---|---|---|---|---|---|
| CR8 | 11 | 3 | 5.64 | 0 | 1.9 | 3.17 | 1.24 | 11.9 | 78.1 | 81.1 | 61.9 |
| CR9 | 11 | 3 | 5.64 | 0 | 5.71 | 3.17 | 1.11 | 15.6 | 79.2 | 83.3 | 57.7 |
| CR10 | 11 | 3 | 5.64 | 0 | 3.8 | 0 | 0 | 9.45 | 77.4 | 79.9 | 64.9 |
| CR11 | 11 | 3 | 5.64 | 0 | 3.8 | 6.34 | 1.05 | 16.8 | 80.1 | 84 | 55.9 |
| CR12 | 11 | 3 | 5.64 | 0 | 3.8 | 9.51 | 0.92 | 19.9 | 81.5 | 85.8 | 52.2 |
| CR13 | 11 | 3 | 5.64 | 0 | 0 | 9.51 | 1.05 | 16.2 | 80.4 | 83.7 | 56.3 |
| CR14 | 11 | 4 | 5.64 | 0 | 3.8 | 3.17 | 1.17 | 13.8 | 78.8 | 81.7 | 64 |
| WS1 | 15 | 3 | 4.85 | 1.94 | 3.17 | 3.17 | 1.14 | 14.3 | 74.8 | 79.9 | 62.2 |
| WS2 | 10 | 3 | 4.85 | 1.94 | 3.17 | 3.17 | 1.17 | 14.3 | 72.9 | 79.7 | 54.5 |
| WS3 | 18 | 3 | 4.85 | 1.94 | 3.17 | 3.17 | 1.2 | 14.3 | 75.2 | 79.7 | 64.6 |
| WS4 | 15 | 2 | 4.85 | 1.94 | 3.17 | 3.17 | 1.11 | 14.2 | 74.7 | 81.3 | 56.5 |
| WS5 | 15 | 4 | 4.85 | 1.94 | 3.17 | 3.17 | 1.17 | 14.3 | 74.8 | 79.2 | 64.9 |
| WS6 | 15 | 3 | 2.42 | 0.97 | 3.17 | 3.17 | 1.4 | 11.1 | 73.9 | 78.1 | 64.4 |
| WS7 | 15 | 3 | 4.85 | 0 | 3.17 | 3.17 | 1.39 | 12.6 | 74.5 | 78.8 | 63.3 |
| WS8 | 15 | 3 | 4.85 | 1.94 | 0 | 3.17 | 1.49 | 11.4 | 73.9 | 77.9 | 64.2 |
| WS9 | 15 | 3 | 4.85 | 1.94 | 3.17 | 0 | 0 | 9.95 | 73.9 | 77.9 | 64.9 |
| WS10 | 15 | 3 | 4.85 | 1.94 | 3.17 | 6.34 | 1.3 | 17.6 | 75.6 | 81.7 | 60.1 |
| WS11 | 15 | 3 | 4.85 | 1.94 | 3.17 | 9.51 | 1.2 | 20.7 | 76.5 | 83.5 | 57.9 |
| WS12 | 15 | 3 | 4.85 | 1.94 | 0 | 9.51 | 1.3 | 17.6 | 75.6 | 81.3 | 60.1 |

**TABLE 1b**
**Case Specification (SI units): SO—Small Offices, LO—Large Offices, CR—Classrooms, and WS—Industrial Workshops**

| Case | $H$ m | $n$ ACH | $Q_o/A$ W/m² | $Q_e/A$ W/m² | $Q_l/A$ W/m² | $Q_{slr}/A$ W/m² | $Q_{wl}/A$ W/m² | $Q_t/A$ W/m² | $T_{fs}$ °C | $T_{cs}$ °C | $T_s$ °C |
|---|---|---|---|---|---|---|---|---|---|---|---|
| SO1 | 2.8 | 4 | 7.96 | 11.7 | 10 | 10 | 6.74 | 46.4 | 24.6 | 27.8 | 16 |
| SO2 | 2.4 | 4 | 7.96 | 11.7 | 10 | 10 | 6.74 | 46.4 | 24 | 27.6 | 14.3 |
| SO3 | 3.3 | 4 | 7.96 | 11.7 | 10 | 10 | 6.74 | 46.4 | 25.2 | 28 | 18 |
| SO4 | 2.8 | 3 | 7.96 | 11.7 | 10 | 10 | 6.74 | 46.4 | 24.3 | 28.2 | 12.9 |
| SO5 | 2.8 | 6 | 7.96 | 11.7 | 10 | 10 | 6.74 | 46.4 | 24.9 | 27.4 | 18.9 |
| SO6 | 2.8 | 4 | 3.98 | 11.7 | 10 | 10 | 6.82 | 42.5 | 24.5 | 27.5 | 16.7 |
| SO7 | 2.8 | 4 | 7.96 | 0 | 10 | 10 | 7.04 | 35 | 24.5 | 26.6 | 18 |
| SO8 | 2.8 | 4 | 7.96 | 5.84 | 10 | 10 | 6.9 | 40.7 | 24.5 | 27.2 | 17 |
| SO9 | 2.8 | 4 | 7.96 | 11.7 | 0 | 10 | 7.04 | 36.7 | 24.3 | 26.9 | 17.8 |
| SO10 | 2.8 | 4 | 7.96 | 0 | 0 | 10 | 7.34 | 25.3 | 24.2 | 25.7 | 19.7 |
| SO11 | 2.8 | 4 | 7.96 | 11.7 | 10 | 0 | 0 | 29.7 | 24.6 | 26.7 | 18.9 |
| SO12 | 2.8 | 4 | 7.96 | 11.7 | 10 | 20 | 6.44 | 56.1 | 24.8 | 28.6 | 14.3 |
| SO13 | 2.8 | 4 | 3.98 | 0 | 0 | 40 | 6.72 | 50.7 | 24.7 | 27.8 | 15.2 |
| SO14 | 2.8 | 6 | 7.96 | 11.7 | 10 | 20 | 6.54 | 56.2 | 25.2 | 28.1 | 17.8 |
| SO15 | 2.8 | 6 | 7.96 | 11.7 | 10 | 44.4 | 6.04 | 80.1 | 25.9 | 29.8 | 15.1 |
| SO16 | 2.8 | 8 | 7.96 | 11.7 | 10 | 44.4 | 6.04 | 80.1 | 26.3 | 29.4 | 17.5 |

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**TABLE 1b (Continued)**
**Case Specification (SI units): SO—Small Offices, LO—Large Offices, CR—Classrooms, and WS—Industrial Workshops**

| Case | $H$ m | $n$ ACH | $Q_o/A$ W/m$^2$ | $Q_e/A$ W/m$^2$ | $Q_l/A$ W/m$^2$ | $Q_{slr}/A$ W/m$^2$ | $Q_{wl}/A$ W/m$^2$ | $Q_t/A$ W/m$^2$ | $T_{fs}$ °C | $T_{cs}$ °C | $T_s$ °C |
|---|---|---|---|---|---|---|---|---|---|---|---|
| SO17 | 2.8 | 9 | 7.96 | 11.7 | 10 | 44.4 | 6.04 | 80.1 | 26.4 | 29.2 | 18.3 |
| SO18 | 2.8 | 15 | 7.96 | 11.7 | 10 | 85.2 | 5.14 | 120 | 28.4 | 31.1 | 19 |
| LO1 | 3.3 | 4 | 6.54 | 9.6 | 12 | 10 | 3.76 | 41.9 | 25.8 | 27.7 | 18.1 |
| LO2 | 3 | 4 | 6.54 | 9.6 | 12 | 10 | 3.76 | 41.9 | 25.6 | 27.6 | 17.2 |
| LO3 | 3.9 | 4 | 6.54 | 9.6 | 12 | 10 | 3.76 | 41.9 | 26.1 | 27.7 | 19.4 |
| LO4 | 3.3 | 3 | 6.54 | 9.6 | 12 | 10 | 3.76 | 41.9 | 25.7 | 28 | 15.8 |
| LO5 | 3.3 | 6 | 6.54 | 9.6 | 12 | 10 | 3.76 | 41.9 | 25.9 | 27.3 | 20.3 |
| LO6 | 3.3 | 4 | 6.54 | 0 | 12 | 10 | 4.06 | 32.6 | 25.5 | 26.8 | 19.4 |
| LO7 | 3.3 | 4 | 6.54 | 4.8 | 12 | 10 | 3.86 | 37.2 | 25.7 | 27.2 | 18.7 |
| LO8 | 3.3 | 4 | 6.54 | 19.2 | 12 | 10 | 3.46 | 51.2 | 26.1 | 28.6 | 16.9 |
| LO9 | 3.3 | 4 | 6.54 | 9.6 | 0 | 10 | 4.16 | 30.3 | 25.3 | 26.5 | 19.8 |
| LO10 | 3.3 | 4 | 6.54 | 0 | 0 | 10 | 4.36 | 20.9 | 25 | 25.6 | 21 |
| LO11 | 3.3 | 4 | 6.54 | 9.6 | 12 | 0 | 0 | 28.1 | 25.1 | 26.5 | 20.2 |
| LO12 | 3.3 | 4 | 6.54 | 9.6 | 12 | 20 | 3.36 | 51.5 | 26.6 | 28.6 | 16.6 |
| CR1 | 3.3 | 4 | 17.8 | 0 | 12 | 10 | 3.7 | 43.5 | 26 | 27.6 | 17.8 |
| CR2 | 2.7 | 3 | 17.8 | 0 | 12 | 10 | 3.8 | 43.6 | 25.3 | 27.6 | 12.9 |
| CR3 | 3.9 | 3 | 17.8 | 0 | 12 | 10 | 3.7 | 43.5 | 26.3 | 28 | 17.2 |
| CR4 | 3.3 | 5 | 17.8 | 0 | 12 | 10 | 3.8 | 43.6 | 26.1 | 27.4 | 19.2 |
| CR5 | 3.3 | 3 | 13.5 | 0 | 12 | 10 | 3.8 | 39.3 | 25.7 | 27.5 | 16.2 |
| CR6 | 3.3 | 3 | 22.1 | 0 | 12 | 10 | 3.6 | 47.7 | 26.1 | 28.2 | 14.6 |
| CR7 | 3.3 | 3 | 17.8 | 3.8 | 12 | 10 | 3.6 | 47.2 | 26 | 28.3 | 14.7 |
| CR8 | 3.3 | 3 | 17.8 | 0 | 6 | 10 | 3.9 | 37.7 | 25.6 | 27.3 | 16.6 |
| CR9 | 3.3 | 3 | 17.8 | 0 | 18 | 10 | 3.5 | 49.3 | 26.2 | 28.5 | 14.3 |
| CR10 | 3.3 | 3 | 17.8 | 0 | 12 | 0 | 0 | 29.8 | 25.2 | 26.6 | 18.3 |
| CR11 | 3.3 | 3 | 17.8 | 0 | 12 | 20 | 3.3 | 53.1 | 26.7 | 28.9 | 13.3 |
| CR12 | 3.3 | 3 | 17.8 | 0 | 12 | 30 | 2.9 | 62.7 | 27.5 | 29.9 | 11.2 |
| CR13 | 3.3 | 3 | 17.8 | 0 | 0 | 30 | 3.3 | 51.1 | 26.9 | 28.7 | 13.5 |
| CR14 | 3.3 | 3 | 17.8 | 0 | 12 | 10 | 3.7 | 43.5 | 25.9 | 27.9 | 15.4 |
| WS1 | 4.5 | 3 | 15.3 | 6.11 | 10 | 10 | 3.59 | 45 | 23.8 | 26.6 | 16.8 |
| WS2 | 3 | 3 | 15.3 | 6.11 | 10 | 10 | 3.69 | 45.1 | 22.7 | 26.5 | 12.5 |
| WS3 | 5.5 | 3 | 15.3 | 6.11 | 10 | 10 | 3.79 | 45.2 | 24 | 26.5 | 18.1 |
| WS4 | 4.5 | 2 | 15.3 | 6.11 | 10 | 10 | 3.49 | 44.9 | 23.7 | 27.4 | 13.6 |
| WS5 | 4.5 | 4 | 15.3 | 6.11 | 10 | 10 | 3.69 | 45.1 | 23.8 | 26.2 | 18.3 |
| WS6 | 4.5 | 3 | 7.63 | 3.05 | 10 | 10 | 4.42 | 35.1 | 23.3 | 25.6 | 18 |
| WS7 | 4.5 | 3 | 15.3 | 0 | 10 | 10 | 4.4 | 39.7 | 23.6 | 26 | 17.4 |
| WS8 | 4.5 | 3 | 15.3 | 6.11 | 0 | 10 | 4.69 | 36.1 | 23.3 | 25.5 | 17.9 |
| WS9 | 4.5 | 3 | 15.3 | 6.11 | 10 | 0 | 0 | 31.4 | 23.3 | 25.5 | 18.3 |
| WS10 | 4.5 | 3 | 15.3 | 6.11 | 10 | 20 | 4.09 | 55.5 | 24.2 | 27.6 | 15.6 |
| WS11 | 4.5 | 3 | 15.3 | 6.11 | 10 | 30 | 3.79 | 65.2 | 24.7 | 28.6 | 14.4 |
| WS12 | 4.5 | 3 | 15.3 | 6.11 | 0 | 30 | 4.09 | 55.5 | 24.2 | 27.4 | 15.6 |

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$Q_l$ = heat generated by the overhead lighting,

$Q_{ex}$ = heat from exterior walls and windows and the transmitted solar radiation.

Since there is a temperature stratification in a room with displacement ventilation, the ceiling and floor surface temperature are unknown. In this investigation, we use the two temperatures in the CFD program to calculate the air temperature and contaminant distributions. Following is a discussion of a procedure used to estimate the temperatures.

In a space with displacement ventilation as shown in Figure 6, the steady-state heat balance on the surfaces of the floor and the ceiling can be expressed as

$$Q_{af} = Q_{sf} + Q_{rf} + Q_{of}, \tag{6}$$

$$Q_{ac} = -Q_{sc} + Q_{rc} + Q_{oc}\,, \tag{7}$$

where

$Q_{af}$ = convective heat transfer from the floor to the air,

$Q_{sf}$ = radiative heat transfer from the heat sources to the floor,

$Q_{rf}$ = radiative heat transfer from the ceiling and walls to the floor,

$Q_{of}$ = heat transfer from the space under the floor to the floor surface,

$Q_{ac}$ = convective heat transfer from the air to the ceiling,

$Q_{sc}$ = radiative heat transfer from the heat sources to the ceiling,

$Q_{rc}$ = radiative heat transfer from the ceiling to the walls and floor,

$Q_{oc}$ = heat transfer from the ceiling surface to the space above the ceiling.

Further, Newton's law reads:

$$Q_{af} = \alpha_{cf}\,(T_{fs} - T_f)A, \tag{8}$$

$$Q_{ac} = \alpha_{cc}\,(T_c - T_{cs})A, \tag{9}$$

where

$\alpha_{cf}$ = convective heat transfer coefficient on the floor,

$T_{fs}$ = floor surface temperature,

$T_f$ = air temperature near the floor,

$\alpha_{cc}$ = convective heat transfer coefficient on the ceiling,

$T_{cs}$ = ceiling surface temperature,

$A$ = floor/ceiling area.

The convective heat transfer on the floor causes an air temperature increase from the supply temperature to the air temperature on the foot level. Therefore,

$$Q_{af} = \rho C_p\, V(T_f - T_s). \tag{10}$$

The radiative heat transfer from the heat sources to the floor and the ceiling, respectively, may be estimated by

$$Q_{sf} = \sum_j r_{fj} Q_j\,, \tag{11}$$

$$Q_{sc} = \sum_j r_{cj} Q_j\,, \tag{12}$$

where

$Q_j$ = heat emitted by $j$th heat source, including transmitted solar radiation;

$r_{fj}$ = fraction of radiative heat transfer from $j$th heat source to the floor;

$r_{cj}$ = fraction of radiative heat transfer from $j$th heat source to the ceiling.

The $r_{fj}$ and $r_{cj}$ need to be estimated from the room geometry.

According to Mundt (1996), the radiative heat transfer from the ceiling and walls to the floor, $Q_{rf}$, and the radiative heat transfer from the ceiling to the floor and walls, $Q_{rc}$, can be estimated via

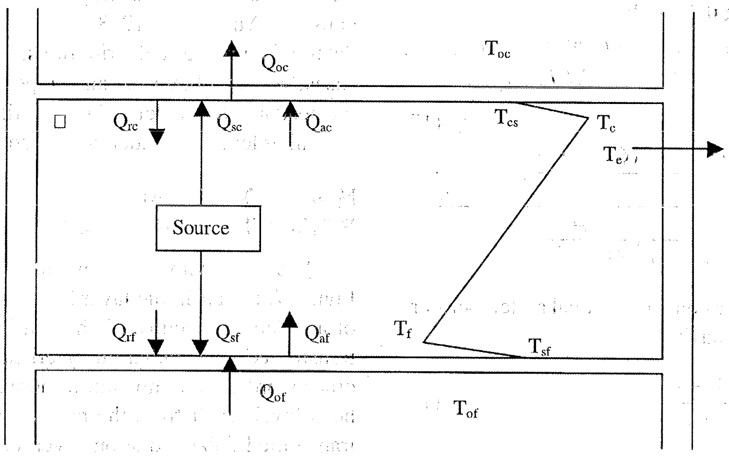


***Figure 6*** *Heat transfer in a space with the displacement ventilation.*

$$Q_{rf} = \alpha_r A\ (T_{cs} - T_{fs})\ Y_1, \tag{13}$$

$$Q_{rc} = \alpha_r A\ (T_{cs} - T_{fs})\ Y_2, \tag{14}$$

where

$\alpha_r$ = the radiative heat transfer coefficient,

$Y_1$ and $Y_2$ = coefficients.

The values of $Y_1$ and $Y_2$ depend on the distribution of surface temperatures and the geometry of the room envelope. Mundt (1996) showed that the $Y_1$ and $Y_2$ are between 0.6 to 0.8 for rooms with displacement ventilation. The lower value corresponds to rooms with a high H/A (the ratio of the room height to floor area).

The heat transfer from the space under the floor to the floor surface, $Q_{of}$, and the heat transfer from the space above the ceiling to the ceiling surface, $Q_{oc}$, can be expressed as

$$Q_{of} = A\ (T_{of} - T_{fs})/R_f, \tag{15}$$

$$Q_{oc} = A\ (T_{cs} - T_{oc})/R_c\ , \tag{16}$$

where

$T_{of}$ = temperature of the space under the floor,

$T_{oc}$ = temperature of the space above the ceiling,

$R_f$ = thermal resistance from the space under the floor to the floor surface,

$R_c$ = thermal resistance from the space above the ceiling to the ceiling surface.

The total cooling load is offset by the ventilation system, i.e.,

$$Q_t = \rho C_p V(T_e - T_s), \tag{17}$$

where

$\rho$ = air density,

$C_p$ = specific heat of air,

$V$ = volume flow rate from the supply,

$T_e$ = air temperature at the exhaust,

$T_s$ = air temperature at the supply.

From the above equations, Mundt (1996) developed the following equation to calculate $\theta_f = T_f - T_s\ /\ T_e - T_s$:

$$\theta_f = \frac{\dfrac{(Q_{sc} - Q_{oc})Y_1 + (Q_{sf} + Q_{of})Y_2}{Q_t Y_1 \alpha_{cc}} + \dfrac{(Q_{sf} + Q_{of})}{Q_t Y_1 \alpha_r} + \dfrac{AH}{\rho C_p V H_e}}{\dfrac{Y_2}{Y_1 \alpha_{cc}} + \dfrac{1}{Y_1 \alpha_r} + \dfrac{1}{\alpha_{cf}} + \dfrac{AH}{\rho C_p V H_e}} \tag{18}$$

With the assumption of a constant vertical air temperature gradient in the space, $s$, we have

$$s = \frac{T_e - T_f}{H_e} \tag{19}$$

where

$H_e$ = the exhaust elevation.

The combination of $\theta_f$ with Equations 17 and 19 leads to

$$s = \frac{Q_t(1 - \theta_f)}{\rho C_p V H_e}. \tag{20}$$

Once $\theta_f$ and $s$ are obtained, the temperatures can be determined for a given design as follows. The air temperature near the floor is

$$T_f = T_h - s\ H_h \tag{21}$$

where $T_h$ = desired design room temperature at the head level of a sedentary person and $H_h$ = the head elevation.

The exhaust air temperature is

$$T_e = T_f + s\ H_e. \tag{22}$$

The supply air temperature is

$$T_s = T_e - Q_t\ /(\rho C_p V). \tag{23}$$

The floor surface temperature is

$$T_{fs} = T_f + \rho C_p V\ (T_f - T_s)/(A\alpha_{cf}). \tag{24}$$

The air temperature near the ceiling is

$$T_c = T_e + (H - H_e)\ s. \tag{25}$$

The ceiling surface temperature is

$$T_{cs} = \frac{\alpha_{cc} T_c + \alpha_r Y_2 T_f + (Q_{sc} - Q_{oc})/A}{\alpha_{cc} + \alpha_r Y_2}. \tag{26}$$

Since $Q_{of}$, $Q_{oc}$, and $Q_t$ depend on $T_{fs}$ and $T_{cs}$, iterations are necessary between Equations 15 and 26. The calculated temperatures are listed in Table 1.

The above derivation assumes a constant vertical temperature gradient in the room air. This assumption is only used for the estimation of the surface temperatures on the floor and the ceiling and the supply air temperature. The experimental data from the literature show that the gradient is not a constant in many cases, but the average value of the data is close to constant (Yuan et al. 1998). The CFD program will provide detailed temperature distributions. From the results of the 56 cases, we can obtain the air temperature difference between the head and foot level and the ventilation effectiveness at the breathing level of a sedentary person.

## MODEL OF THE AIR TEMPERATURE DIFFERENCE BETWEEN THE HEAD AND FOOT LEVEL

In a room with displacement ventilation, as illustrated in Figure 7, the air in the layer between the head and foot level of a sedentary occupant is heated by occupants, equipment, transmitted solar radiation, overhead lighting, and walls. In other words, the temperature increases from the foot to the head level result from the heat from occupants, equipment, transmitted solar radiation, overhead lighting, and the heat gain/loss through the exterior walls/windows. Obviously, the heat from the occupants and equipment contributes more

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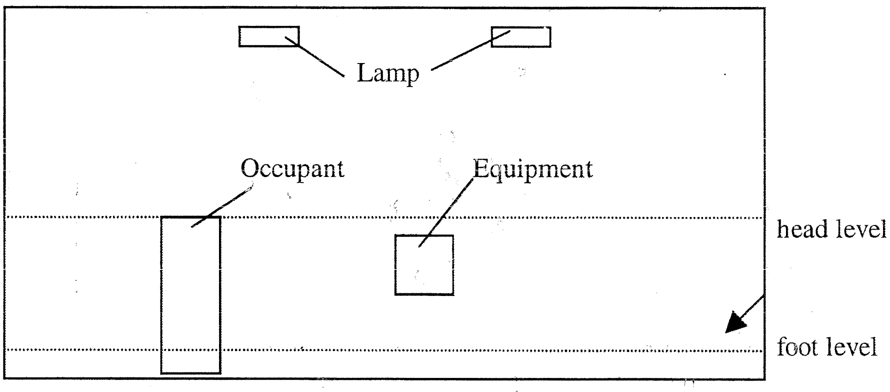


***Figure 7*** *Sketch of a room with the displacement ventilation.*

significantly to the temperature increase in this layer than that from overhead lighting. This is because the occupants and equipment are located in this layer.

We can assume the heat transfer to the air between head and foot level by

$$\Delta T_{hf}\rho C_p V = a_{oe}Q_{oe} + a_l Q_l + a_{ex}Q_{ex} \tag{27}$$

where

$Q_{oe}$ = heat generated by occupants, desk lamps, and equipment, $(Q_o+Q_{dl}+Q_e)$;

$Q_l$ = heat generated by overhead lighting;

$Q_{ex}$ = heat from the exterior wall and window surfaces and the transmitted solar radiation;

$a_{oe}$, $a_l$, and $a_{ex}$ = weighting coefficients for the contribution of the convective heat to the air between head and foot level.

Since

$$V = nHA, \tag{28}$$

where

$V$ = the supply flow rate,

$n$ = the air change rate,

$H$ = the room height,

$A$ = the floor area,

Equation 27 turns to

$$\Delta T_{hf} = \frac{a_{oe}Q_{oe} + a_l Q_l + a_{ex}Q_{ex}}{\rho C_p nHA} \tag{29}$$

Equation 29 is a model to calculate the air temperature difference between the head and foot level of a sedentary person in a room with displacement ventilation. Based on the database for the cases listed in Table 1, the coefficients that gives the best agreement are

$$a_{oe} = 0.295, \quad a_l = 0.132, \quad a_{ex} = 0.185. \tag{30}$$

The model should only be applied to cases within the range of the present database. This range covers most U.S. buildings except large spaces such as theaters and atria. We recommend the use of a validated CFD program or experimental measurements to design displacement ventilation systems in large spaces.

For people and unhooded equipment, the cooling load factor is about 0.75. The model (Equation 29) indicates that one-third of the cooling load enters the space between foot and head level. The other two-thirds enters the upper space by the thermal plume. The radiative heat from the overhead lighting to the building envelope is about 20% of the total energy input to the lamps. About two-thirds of the radiative heat is projected to the floor and the lower part of the wall. This eventually heats the air between the foot and head level. Mundt (1996) measured the air temperature profile in a room with displacement ventilation. The only head source is a simulated person. The temperature difference between the head and foot level of the occupant over the temperature difference between the return and supply air is 0.3, which is in excellent agreement with the $a_{oe}$ value. This suggests the values of the weighting coefficients, $a_{oe}$, $a_l$, and $a_{ex}$, are physically sound.

Figure 8 compares the air temperature difference obtained from the database and calculated by the model (Equation 29). The temperature differences calculated with the assumption of a constant vertical temperature gradient, $(T_e - T_f)/H$, are also presented in the figure for comparison. The correlation between the model and database is very good. This implies the temperature differences calculated with the model

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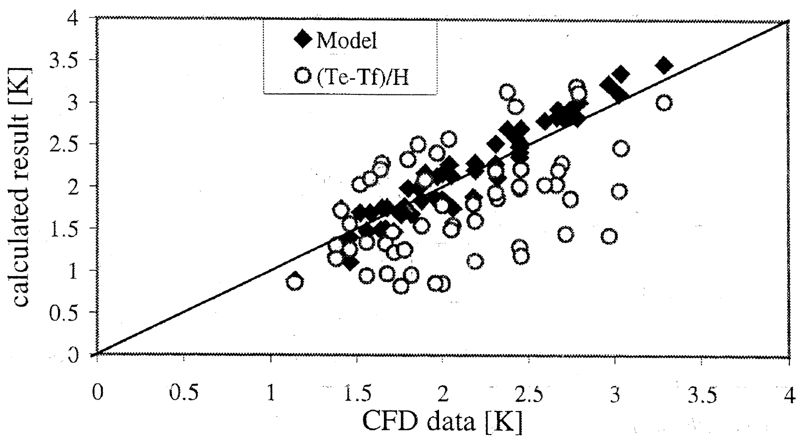


***Figure 8*** *Correlation of the air temperature difference between the head and foot level.*

are close to those from the CFD simulation. It is not surprising that the average values calculated by the model agree with the simulation since the values of the coefficients were obtained from the same simulations. It is gratifying that the model does accurately capture the influence of individual parameter variations on the foot to head temperature difference. However, the assumption of a constant temperature gradient from floor to ceiling is not very good. Straub (1962) has provided a good explanation of how temperature gradient is formed. Obviously, the constant gradient assumption neglects many factors because of its simple form.

Figure 9 provides a more detailed case-by-case comparison for the small offices, large offices with partitions, classrooms, and industrial workshops, respectively. The results show that the assumption of a constant temperature gradient is more problematic when the ceiling height is high, as in the workshops shown in Figure 9d.

Figure 10 compares the $\Delta T_{hf}$ obtained by the model and the constant temperature gradient assumption with the measured data from the literature. The agreement between the model and the data is less satisfactory in some cases, such as those from Holmberg et al. (1987) and Nielsen et al. (1988). No information about temperature on the walls is available from Holmberg et al. (1987). The temperature difference from head to foot will be influenced by heat transfer from the vertical walls. There is no indication of wall temperatures to determine if the wall adds or removes heat from the room air. In Nielsen's cases, there was a large glass wall in the test room for which no temperature information is available. Among the three cases from Brohus and Nielsen (1996), the calculated values agree with the last two cases but not with the first one. The measured $\Delta T_{hf}$ in the first case should be much larger than that in the third case because the heat sources are almost the same in the two cases and the ventilation flow rate in the first case is less than half of that in the third case. Thus, some other unreported changes in room conditions may have occurred. For all other cases, the $\Delta T_{hf}$ calculated by the model is close to the measured data. Figure 10 shows that the model estimation of the temperature difference between the head and foot level is much better than the assumption of a constant temperature gradient from floor to ceiling.

The model (Equation 29) shows that a large cooling load can cause a large $\Delta T_{hf}$. The head to foot temperature difference, $\Delta T_{hf}$, should be less than 3.6°F (2 K) for comfort consideration. Therefore, the cooling load has an upper limit for acceptable comfort with displacement ventilation. However, the model suggests that the air temperature difference between the head and foot level not only depends on the total cooling load but also on the type of heat gains. The maximum cooling load is not a fixed value for thermal comfort in displacement ventilation. If a majority of the cooling load is from overhead lighting or other heat sources above the stratification level, displacement ventilation can operate with a much higher cooling load and still provide comfortable conditions.

The model indicates that an increase in the ventilation rate, $n$, may reduce $\Delta T_{hf}$. However, the airspeed from the diffuser cannot be too high. To maintain a thermally comfortable environment requires a large diffuser area when the ventilation rate increases. Therefore, the maximum cooling load depends on the area available for installing diffusers and the distribution of heat sources. Note that a higher ventilation rate will consume more energy from the fan and requires a larger air-handling unit.

## VENTILATION EFFECTIVENESS MODEL

It is difficult to derive a general model for ventilation effectiveness with displacement ventilation. We restrict our efforts to a model for rooms where the contaminant sources are associated with the heat sources.

The database was used with a technique similar to that for the model of the air temperature difference (Equation 29). The model for prediction of ventilation effectiveness, $\eta$, at the breathing level of a sedentary person in a displacement ventilated room is

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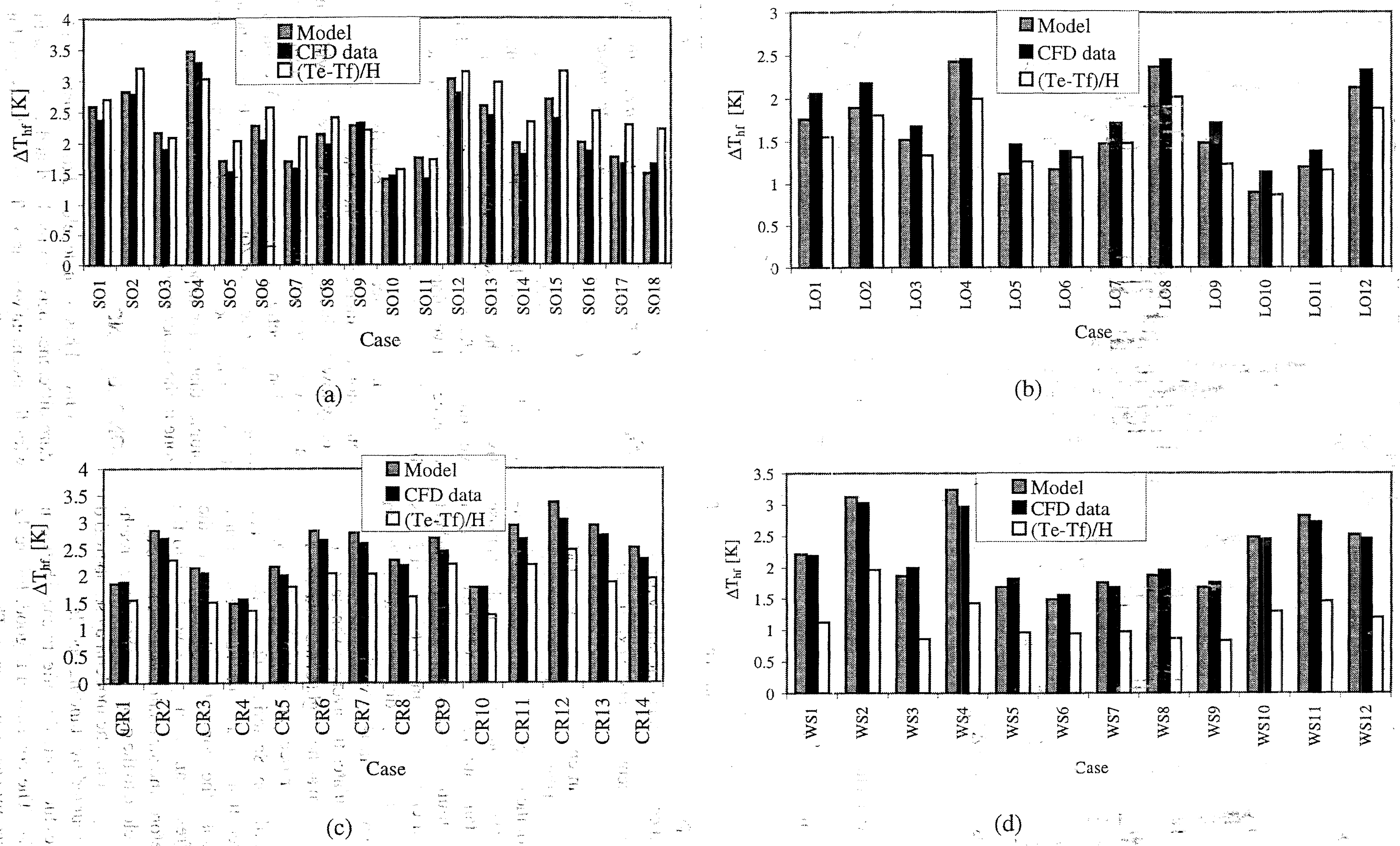


***Figure 9*** *Comparison of the* $\Delta T_{hf}$ *obtained by the model, the CFD data, and the constant temperature gradient assumption. (a) small offices, (b) large offices with partitions, (c) classrooms, and (d) industrial workshops.*

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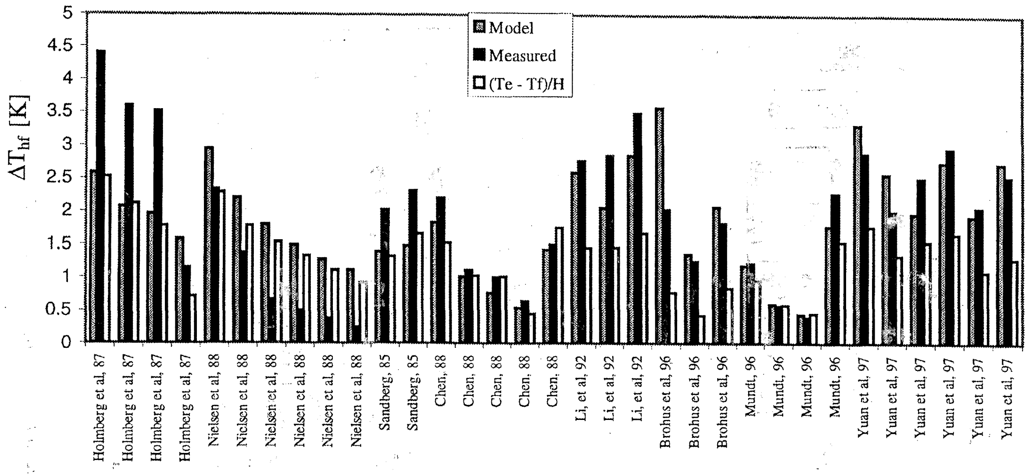


***Figure 10*** *Comparison of the* $\Delta T_{hf}$ *obtained by the model, measured data, and constant gradient assumption.*

$$\eta = 2.83(\quad)(\quad)/Q_t \tag{31}$$

where $\eta$ = ventilation rate and

$$\eta = \frac{c_h - c_s}{c_e - c_s} \tag{32}$$

where

$c_h$ = mean contaminant concentration at the head level of a sedentary person,

$c_s$ = contaminant concentration at the supply air,

$c_e$ = contaminant concentration at the exhaust air.

Equation 31 is purely an empirical best fit with the data. However, the ratios between the coefficients for $Q_{oe}$, $Q_l$, and $Q_{ex}$ are the same as those for the model of the air temperature difference (Equation 30). The ventilation effectiveness in the database is for all cases with contaminants released from the location of the occupants—the contaminant sources are combined with heat sources. The ventilation effectiveness model (Equation 31) is only valid for the same conditions. Additionally, the ventilation effectiveness is not a constant in both vertical and horizontal directions. The model calculates the average ventilation effectiveness throughout the room at the height of the breathing level.

One must also consider occupants standing in a space with displacement ventilation. Saeteri (1992) and Brohus and Nielsen (1996) showed that the air inhaled originates at a lower elevation because the convective flow around the human body brings fresher air from the lower level to the breathing level. Therefore, the air quality inhaled by a standing person is probably close to the quality of that at the breathing level of a sedentary person. The model may still be valid for spaces primarily occupied by standing people.

Figure 11 compares the ventilation effectiveness between the model and database. The correlation is good. Figure 12 provides a detailed case-by-case comparison. The values of

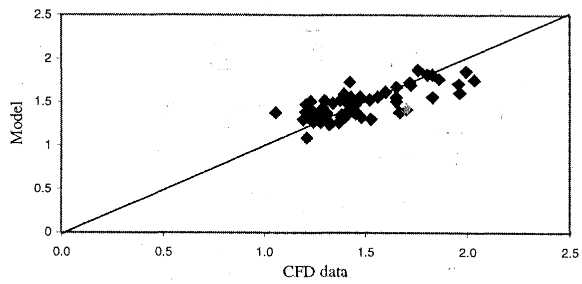


***Figure 11*** *Correlation of the ventilation effectiveness at the breathing level between the model and CFD data.*

ventilation effectiveness predicted by the model are generally in good agreement with the CFD results. The ventilation effectiveness for the 56 cases varies between 1.2 and 2. Since the ventilation effectiveness for perfect mixing ventilation is 1.0, displacement ventilation does provide better indoor air quality.

The model indicates that the effectiveness increases as the ventilation rate increases. When the ventilation rate is sufficiently low, the increase of the effectiveness with ventilation rate is very pronounced. The model also suggests that the ventilation effectiveness is high when the fraction of the heat sources in the occupied zone ($Q_{oe}$) over the total heat source is large. This is because a large $Q_{oe}$ generates strong thermal plumes that can transport the contaminants from the occupied zone to the upper zone.

## CONCLUSIONS

A model has been developed to estimate the air temperature difference between the head and foot level in a space with displacement ventilation. The model was developed from a database of 56 displacement ventilation conditions by use of

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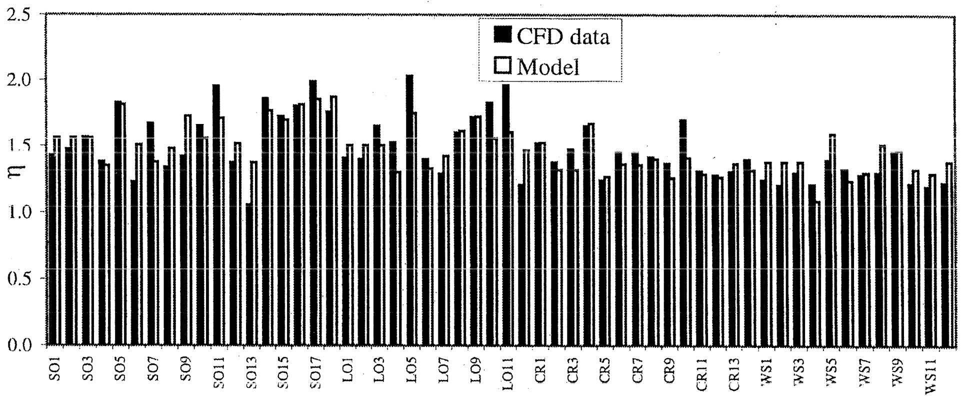


***Figure 12*** *Comparison of the ventilation effectiveness between the model and CFD data.*

a validated CFD program. The 56 cases cover four different types of buildings: small offices, large offices with partitions, classrooms, and industrial workshops under different thermal and flow boundary conditions normally found in U.S. buildings. The model should not be applied to large spaces such as theaters and atria until it is validated for such conditions.

This investigation shows that the maximum cooling load in a room with displacement ventilation is not a fixed value. The cooling load depends on the distribution of the heat sources and the ventilation rate of the indoor space.

Based on the same database, a model of the ventilation effectiveness at the breathing level of a sedentary person has also been developed. The model is applicable to indoor spaces where the contaminant sources are associated with heat sources. The study also shows that the ventilation effectiveness is high when a large fraction of the total heat sources is in the occupied zone.

## ACKNOWLEDGMENTS

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