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INDOOR AIRFLOW WITH COOLING PANEL AND RADIATIVE/CONVECTIVE HEAT SOURCE

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ABSTRACT

This paper numerically studies the effects of a radiative heat source and a cooling panel on indoor airflow, temperature stratification, and dispersion of contaminants in a furnished office with displacement ventilation. The air supply device simulated is a quarter-cylinder displacement diffuser. The percentage of radiative heat is considered to be 20% and 80%, respectively, and the ventilation rate changes from 2.1 ach to 4.2 ach. Four cases, with and without a cooling panel, are examined.

In order to determine the radiative heat transfer between surfaces of the room, a simplified analytical method is proposed to estimate the temperature of an adiabatic wall so that computer time can be saved without a significant sacrifice in accuracy.

It has been found that a ceiling-mounted cooling panel can not only reduce the vertical temperature stratification but also strengthen the air movement. With the same total heat, the increase in the portion of radiative heat may enhance air movement and contaminant dispersion. Also, the air temperature in the central region becomes lower.

INTRODUCTION

Air movement in buildings plays an important role in energy savings, thermal comfort, and indoor air quality because the distributions of temperature and contaminant concentration are directly dependent on the airflow pattern. Air movement in buildings is affected by many parameters, such as the locations of air supply and return openings, the strength and location of heat sources, the percentage of radiative heat of the heat sources, and the condition of the supply air. It is difficult to deal with the variations of each parameter in a single experimental study. With the development of modern computers, the numerical simulation has been widely used in indoor airflow study (Haghighat et al. 1990; Nielsen 1989; Rhodes 1989; Murakami et al. 1988; Chen and van der Kooi 1988; Jones and Sullivan 1985; Whittle 1986) because of its flexibility in dealing with different boundary conditions and its ability to provide an overview of flow field and distributions of flow properties. The capability of numerical models to simulate heat and contaminant sources enables one to estimate the indoor air quality in occupied zones at a certain ventilation level. A number of numerical simulations have been conducted to investigate the airflow affected by indoor heat sources. The following are a few examples.

Markatos and Cox (1984) predicted the development of a fire and the contaminant concentration distribution within a shopping mall. The fire was simulated by point heat and contaminant sources. Both steady and transient cases were predicted, either for a fixed heat release rate or for a linearly growing fire reaching the fixed rate at three minutes from ignition.

Chen (1990) studied energy consumption and thermal comfort in a ventilated office equipped with a radiant panel, radiator heating, and a warm-air heating system, respectively. The thermal boundary condition used in his computation of room airflow was coupled with the heat conduction through walls. Although radiative heat transfer between walls was studied in this work, the heat sources in the room were assumed to be convective.

Jiang et al. (1991) applied the k-ϵ model of turbulence to study the velocity and temperature distributions in a partitioned enclosure with mixed convective flow. With the assumption that the solar radiation through windows and the heat generated by a computer were all considered to be convective, they investigated the effects of door location and supply air temperature on velocity field, contaminant dispersion, and draft risk.

In previous works, heat sources in rooms were considered mostly as convective sources, which means that the heat emitted from a source is all carried away by the adjacent air through convection. However, the heat sources in rooms, in general, are neither 100% convective nor 100% radiative. A heat source usually has a higher surface temperature than walls. A net radiative heat flow would take place whenever there is a temperature difference between objects. Therefore, heat transfer in buildings occurs not only by convection, but also by radiation. When a cooling panel is installed, the radiative

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heat transfer becomes even more important. The same amount of heat from a source may result in different temperature distributions, even in a different airflow pattern in a room, if the portion of radiative heat changes. There is still a considerable degree of uncertainty about the fractions of heat released by radiation and convection and about the aerodynamic effects of these fractions. Numerical simulation may address some of these questions.

The aim of the present study is to examine the dependence of the airflow pattern, velocity levels, and distributions of temperature and contaminant concentration on the percentage of radiative heat and the ventilation flow rate in a large office ventilated by a quarter-cylinder displacement diffuser. The effects of a cooling panel on room airflow are investigated as well.

The research is conducted by numerical solution of the conservation equations for three-dimensional turbulent flow with the k-ϵ two-equation turbulence model (Launder and Spalding 1974). An airflow program (Rosten and Spalding 1984) that has been well verified in room airflow simulation is employed in the computation.

DETERMINATION OF RADIATIVE HEAT TRANSFER

Airflow in rooms is directly dependent on boundary conditions at walls. The boundary condition for the energy equation is expressed by giving either temperature or heat flux at the walls.

For ventilated rooms, the assumption of adiabatic conditions at inner walls seems to be close to reality and, therefore, is usually adopted for the sake of simplicity. Moreover, it is easier to achieve an adiabatic boundary condition in experiments. With this assumption, the boundary condition for the energy equation is

$$q = 0 \tag{1}$$

where q is the net heat gain at the wall.

As discussed in the preceding section, heat sources in rooms are not all convective. Radiative heat transfer occurs between heat sources and walls. When a cooling panel is placed on the ceiling, the temperature difference between walls and ceiling is not negligible; thus, the radiative heat transfer between walls should be taken into consideration. When computing the radiative heat transfer between source, ceiling, and walls, the surface temperatures of the heat source and walls must be known. However, if a constant temperature is assigned to an adiabatic wall, it could be in contradiction with the boundary condition of zero heat flux at the wall. Thus, iteration, which is costly because of a large amount of computing time, is required to deal with this problem. The iteration consists of the following steps:

1. guess the wall temperature,
2. use the guessed wall temperature to compute the radiative heat transfer between the cooling panel and walls,
3. use the computed radiative heat flux to/from walls that provides the boundary condition to compute the distributions of air velocity and temperature in the room, and
4. use the resulting air temperature and the guessed wall temperature to check whether the adiabatic condition at the wall is satisfied. If not, modify the guessed wall temperature and start another iteration from the second step.

The number of iterations required is obviously dependent on how close the guessed temperature is to the final one that satisfies the adiabatic condition at the wall. An arbitrarily guessed temperature would greatly increase the iteration number and thus the computing time. This is why it is difficult to do the computation including the radiative heat transfer between adiabatic walls. When simulating a real ventilation supply device and a number of inside heat sources, a large number of grid nodes is required. It takes about two hours of CPU time in a supercomputer to compute the flow field in a room without iteration of the wall boundary condition. An arbitrarily guessed wall temperature would make the iteration method far too expensive, especially when many parameters affecting room airflow must be studied. In the present study, a simplified analytical approach is introduced to estimate the temperature at an adiabatic wall and to control the convergence of the iteration so computing time would be shortened. The four steps of the iteration will now be explained.

Guess the Wall Temperature

The simplified analysis is based on the overall heat balance of the wall, ceiling, and ventilation air. Assuming that

1. the cooling panel remains at a constant temperature;
2. all boundary surfaces of the room are at a uniform constant temperature except the cooling panel in the ceiling, which has a different constant temperature; and
3. the convective heat transfer coefficient between the wall and air, based on the temperature difference between the wall and the center of the room, is constant and known,

then the equations of heat balance can be written as (see Figure 1):

at adiabatic walls,

$$Q_{s-w} = Q_{w-c} + h(T_w - T_a) A_w \tag{2}$$

at the constant-temperature ceiling,

$$Q_{ceiling} = h(T_a - T_c) A_{ceiling} + Q_{w-c} + Q_{s-c} \tag{3}$$

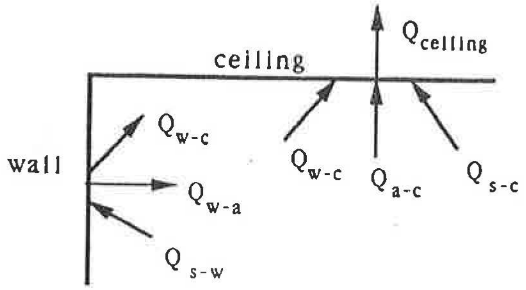


Figure 1 *Heat balance at wall and ceiling.*

for ventilation air,

$$\dot{m}C_p(T_{ex} - T_{in}) = h(T_w - T_a)A_w + h(T_c - T_a)A_c + Q_{conv} \quad (4)$$

total heat balance,

$$Q_{source} = \dot{m}c_p(T_{ex} - T_{in}) + Q_{ceiling} \quad (5)$$

radiation heat transfer between walls and ceiling,

$$Q_{w-c} = A_c\,\sigma(T_w^4 - T_c^4)/(A_c/A_w(1/\varepsilon_w - 1) + 1/\varepsilon_c) \quad (6)$$

where

A_w	=	area of wall,
A_c	=	area of ceiling,
$\dot{m}$	=	ventilation flow rate,
h	=	convective coefficient based on the temperature difference between the wall surface and the air at the center of the room,
Q_{s-w}	=	radiation from heat source to wall,
Q_{w-c}	=	radiation from wall to ceiling,
Q_{s-c}	=	radiation from source to ceiling,
$Q_{ceiling}$	=	total heat absorbed by cooling panel on the ceiling,
Q_{source}	=	total heat emitted from heat sources,
Q_{conv}	=	convective heat from heat sources,
T_a	=	air temperature at the center of the room,
T_{in}	=	supply air temperature,
T_{ex}	=	exhaust air temperature,
T_w	=	wall temperature,
T_c	=	ceiling temperature.

In a real situation, the radiation from the heat source is absorbed and reflected by the enclosure and the furniture in the room. It is complicated to determine the amount of the radiation absorbed by the enclosure. Therefore, the radiative heat from the heat source is assumed to be uniformly distributed to the enclosure. If the convective heat transfer coefficient, h = 4 W/m·K, suggested by ASHRAE (1989) is adopted, there are five unknowns: T_a, T_w, T_{ex}, $Q_{ceiling}$, and Q_{w-c}. With the above five equations Equations (2 through 6), these unknowns can be determined.

Determine the Radiative Heat Transfer Between Walls

The wall temperature obtained from the overall balance analysis is then used as an input in a computer program developed by Chen and van der Kooi (1988). The program computes the net radiative heat flux at each constant-temperature wall including multi-reflection between the walls.

Compute Airflow Field

When computing the room airflow, the precalculated radiative heat flux between the walls and ceiling and the radiative heat from the heat source are used to determine the boundary condition of constant heat flux at the adiabatic wall, that is,

$$Q_{w-a} = Q_{s-a} - Q_{w-c}. \quad (7)$$

This equation means the same as Equation 2. The term q_{w-a} represents the heat flux from the wall to room air through convection. With the thermal boundary conditions fixed, the airflow field is determined accordingly by the airflow computer program.

Check the Boundary Condition at Walls

The computed room air temperature at the center of the room is used to check that the wall temperature is properly assigned so the adiabatic boundary condition at the walls can be satisfied. In other words, the convective heat transfer from walls to air, based on the given convective coefficient,

$$Q_{w-a} = h(T_w - T_a)A_w, \quad (8)$$

must be satisfied. If not, the wall temperature must be adjusted and the radiation heat transfer between ceiling and walls recalculated as well. The modified wall temperature is expected to yield a new boundary condition and thus a better temperature field, which would satisfy Equation 8, the overall energy conservation at adiabatic walls, more accurately. In such an iteration process, the overall heat balance at walls is taken as the criterion. With the analytically based guess of the wall temperature and an overall heat balance as the iteration criteria, less CPU time would be required than when arbitrarily guessing the wall temperature, and the energy conservation is checked at each of the control volumes adjacent to the boundary. This method results in a flow field with a certain degree of accuracy close to the more sophisticated method since the slight difference in the temperature boundary condition would not cause a significant difference in the flow field (Chen and van der Kooi 1988).

SIMULATION OF A QUARTER-CYLINDER DISPLACEMENT DIFFUSER

The dimension and structure of a quarter-cylinder diffuser are presented in Figure 2. At the surface of the diffuser, a two-dimensional airflow can be assumed. The conditioned air is radially emitted from the quarter-cylinder diffuser, and, at each horizontal section, the supply air is described by eight control volumes adjacent to the diffuser surface along the radius, as shown in Figure 2. The ventilation airflow from the diffuser is then simulated by discrete flows in eight directions. The mass flow of supply air, as well as the momentum in the x and y directions, are assigned to the adjacent control volumes.

CASE STUDY

The length, width, and height of the office are 7.26 m, 6 m, and 2.4 m, respectively. The office being examined is symmetrical in geometry, layout of furniture, and source locations. Thus, only half of the room, shown in Figure 3, is studied. For half a room, the width is 3 m. The field distributions presented in the case study are all for half a room. The locations of the convective and radiative heat sources are all specified in Figure 3. Four cases, listed in Table 1, are computed.

In all cases, the temperature of the supply air is 18°C. The occupants are considered as both heat and contaminant sources. The strength of each contaminant source is normalized as 0.01 mL/s to indicate CO_2 emitted from each sitting occupant at mouth level.

The computed three-dimensional velocity vector and the contours of temperature and contaminant concentration for the four cases are all presented in two horizontal sections (z = 0.1 m and 1.35 m) and one vertical section in the y direction. The velocity levels at three horizontal sections—z = 0.1 m, 1.35 m, and 1.8 m—are demonstrated in Figure 4. They are chosen because these sections represent the ankle level and the head levels of sitting and standing occupants. The vertical coordinate in Figure 4 represents the percentage of the area (in each of the three horizontal sections) in which the air velocity is within the velocity bands specified on the horizontal coordinate.

TABLE 1
Case Description

Case (ach)	1	2	3	4
Flow Rate	2.1	2.1	4.2	4.2
Convective Heat	80%	80%	80%	20%
Radiative Heat	20%	20%	20%	80%
Total Heat	1306	1306	1306	1306
Cooling Panel	Yes	No	No	No

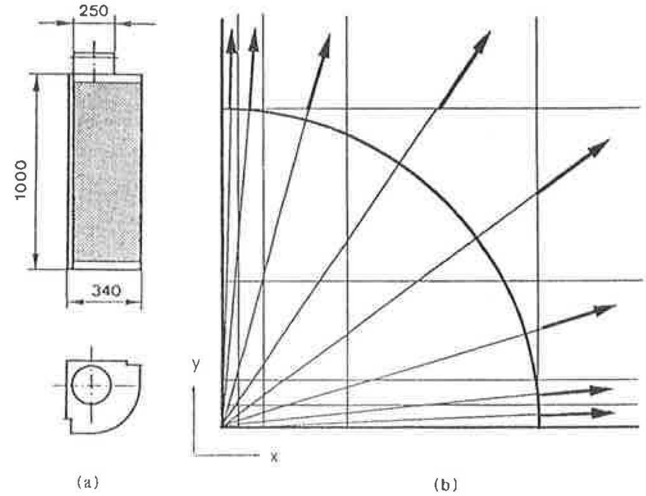
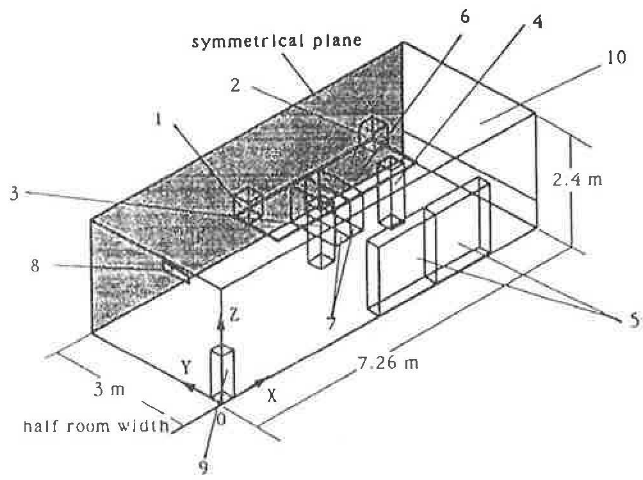


Figure 2 *Simulation of a quarter-cylinder displacement diffuser.*



1. Heat source 1: 130 W convective heat and 130 W radiative heat
2. Heat source 2: 130 W convective heat
3. Heat and contaminant sources 3: 81.5 W convective heat and 0.01 mL/s contaminant (occupant 1)
4. Heat and contaminant sources 4: 81.5 W convective heat and 0.01 mL/s contaminant (occupant 2)
5. Book shelves
6. Desk
7. Cabinets
8. Exhaust opening
9. Air diffuser
10. Window (heat source 5: 100 W)

Figure 3 *Configuration and layout for half a room.*

Case 1

Case 1 is one where a cooling panel with a constant temperature of 19°C is installed on the whole ceiling.

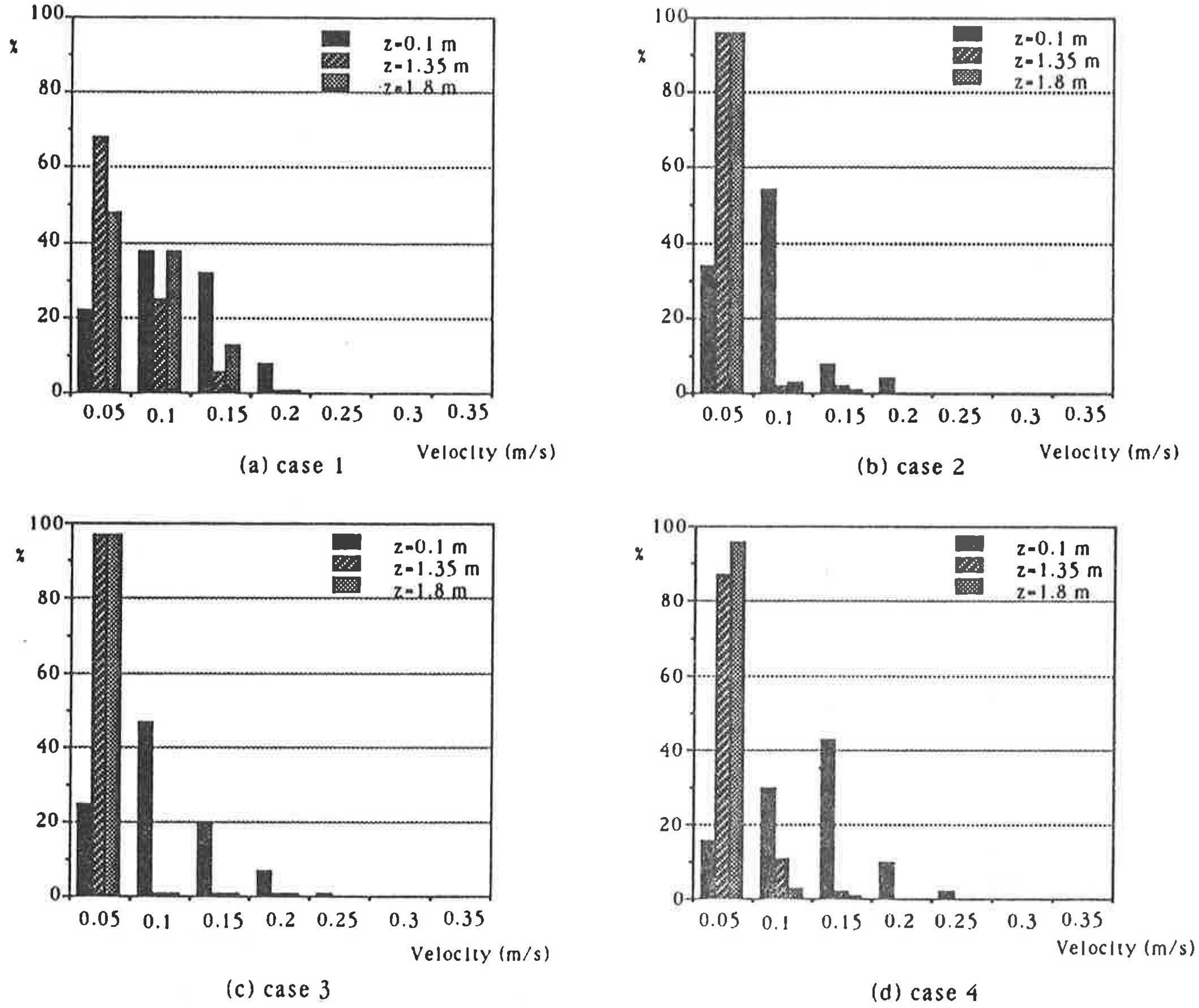


Figure 4 *Velocity levels at three horizontal sections.*

(The distribution of the heat sources in the room for cases 1 through 3 is presented in Figure 3.)

Figure 5 presents the velocity vectors, air temperature distribution, and contaminant concentration distribution for case 1.

At the horizontal section $z = 0.1$ m, the air velocity is quite high and uniform. Air moves downward near the walls because the wall temperature is lowered by the cooling panel, which absorbs the radiative heat from the walls. About a 2°C temperature variation along the x direction at this section is observed. In most areas, the air temperature is higher than 20°C. About 10% of the area in this section has an air velocity of 0.2 m/s or more, as shown in Figure 4a.

At the higher horizontal section, $z = 1.35$ m, the head level of a sitting occupant, the downward movement of the air near the walls becomes stronger, while the velocity in the central region decreases. In the region near the window, the air flows upward because of the buoyancy caused by the heat gain from the window. The air temperature is nearly 2°C higher than that at ankle level ($z = 0.1$ m). It may make a sitting occupant feel uncomfortable. The contaminant concentration in this section is not significantly higher than that in the lower section, although the contaminant source is located at this level. This may be attributed to the strong downward velocity near the walls, which transfers the contaminated air to the lower level.

Figure 5c shows that the air velocity near the floor and ceiling is much larger than in the central region. The vertical temperature stratification is about 3°C. Since the sitting occupants act as heat and contaminant sources, the air above them moving upward due to buoyancy brings the contaminant to the upper region of the room. There-

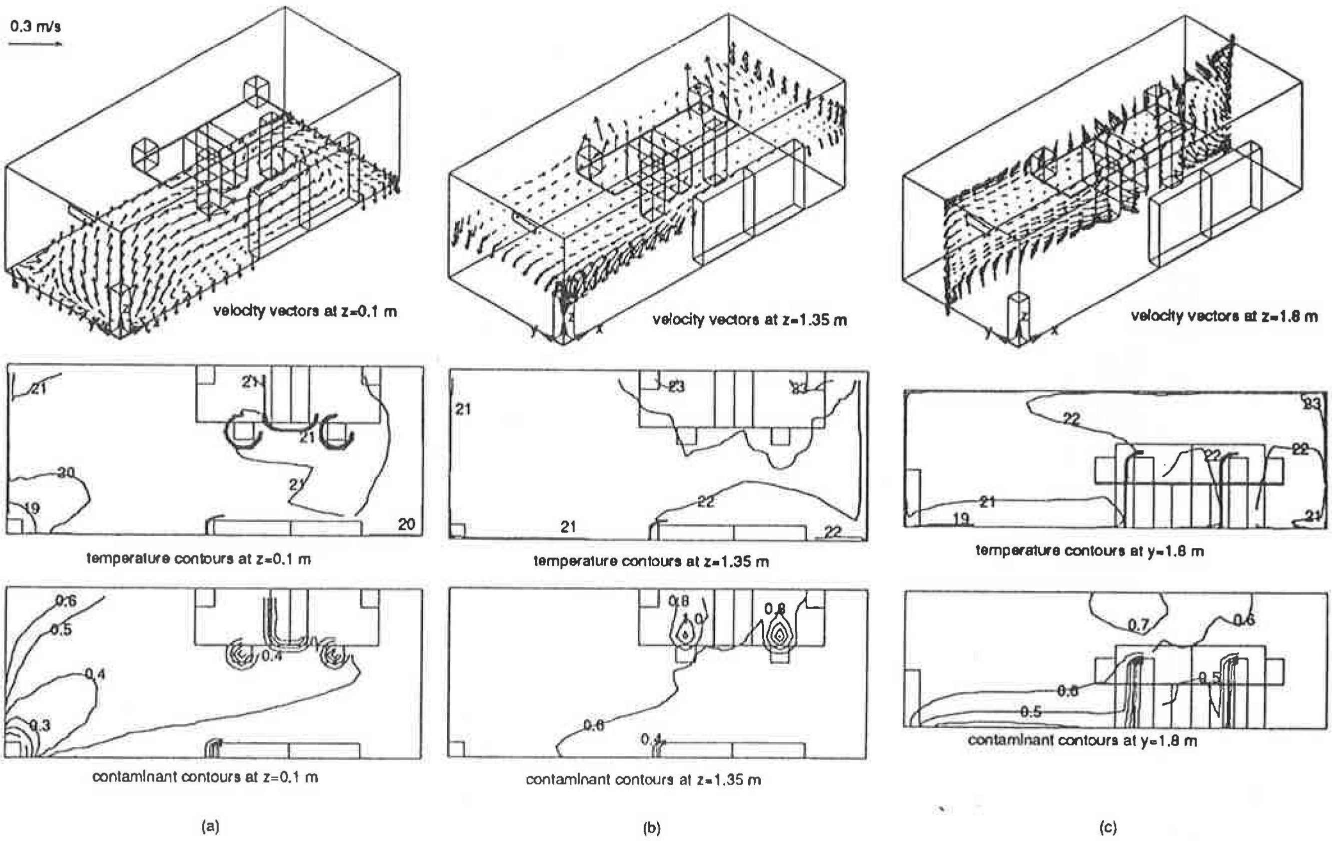


Figure 5 *Velocity vectors and contours of temperature and concentration for case 1 (ach = 2.1, total heat source = 1306.8 W: convective 80%, radiative 20%).*

fore, the contaminant concentration in that region is higher. In general, the contaminant concentration in the room is relatively high because of the low ventilation rate.

Case 2

Case 2 involves removing the cooling panel while all the other parameters remain the same as in case 1. Without the cooling panel, the air velocity in the room is significantly decreased, as shown in Figure 6, although the ventilation rate remains unchanged. The downward airflow near the walls disappears. Alternately, the air near the wall has a tendency to move upward. The radiative heat from the heat source is absorbed by the walls, making the wall temperature higher than that of the adjacent air.

At horizontal section $z = 0.1$ m (ankle level), the air velocity in most areas is less than 0.15 m/s, as presented in Figure 4b. The air temperature at this section is 1°C higher on average than in the former case, and the contaminant concentration is lower (see Figure 6a). The explanation may be that in case 1, the downward air movement along the walls brings the contaminant down from higher levels, where the sources are located, to the region near the floor. The temperature rises by 2°C along the x direction, the same as in case 1.

The air temperature at the $z = 1.35$ m level is quite uniform with little horizontal gradient, but it is quite high. It is about 3°C higher than that at ankle level, which could result in discomfort. The contaminant concentration is higher than in case 1 because of the lower air velocity around the contaminant sources (the sitting occupants). As shown in Figure 4b, the area with a velocity equal to or below 0.05 m/s in this section is almost 97%.

Figure 6c illustrates that the airflow near the ceiling and floor is strong compared to that in the central region, and the vertical temperature stratification can be more than 5°C. It is also shown that with the same contaminant emission rate, occupant 1 suffers a higher contaminant concentration than occupant 2. It may be due to the lower air velocity above occupant 1.

Case 3

In case 3, there is no cooling panel, but the ventilation rate is doubled from that in case 2. As mentioned in the previous sections, when the ceiling-mounted cooling panel is removed, room temperature increases by 2 to 3°C on average, with the ventilation flow rate remaining the same. In order to remove more heat from the room, the ventilation rate should be increased. Case 3 is designed to examine the effects of ventilation rate on the

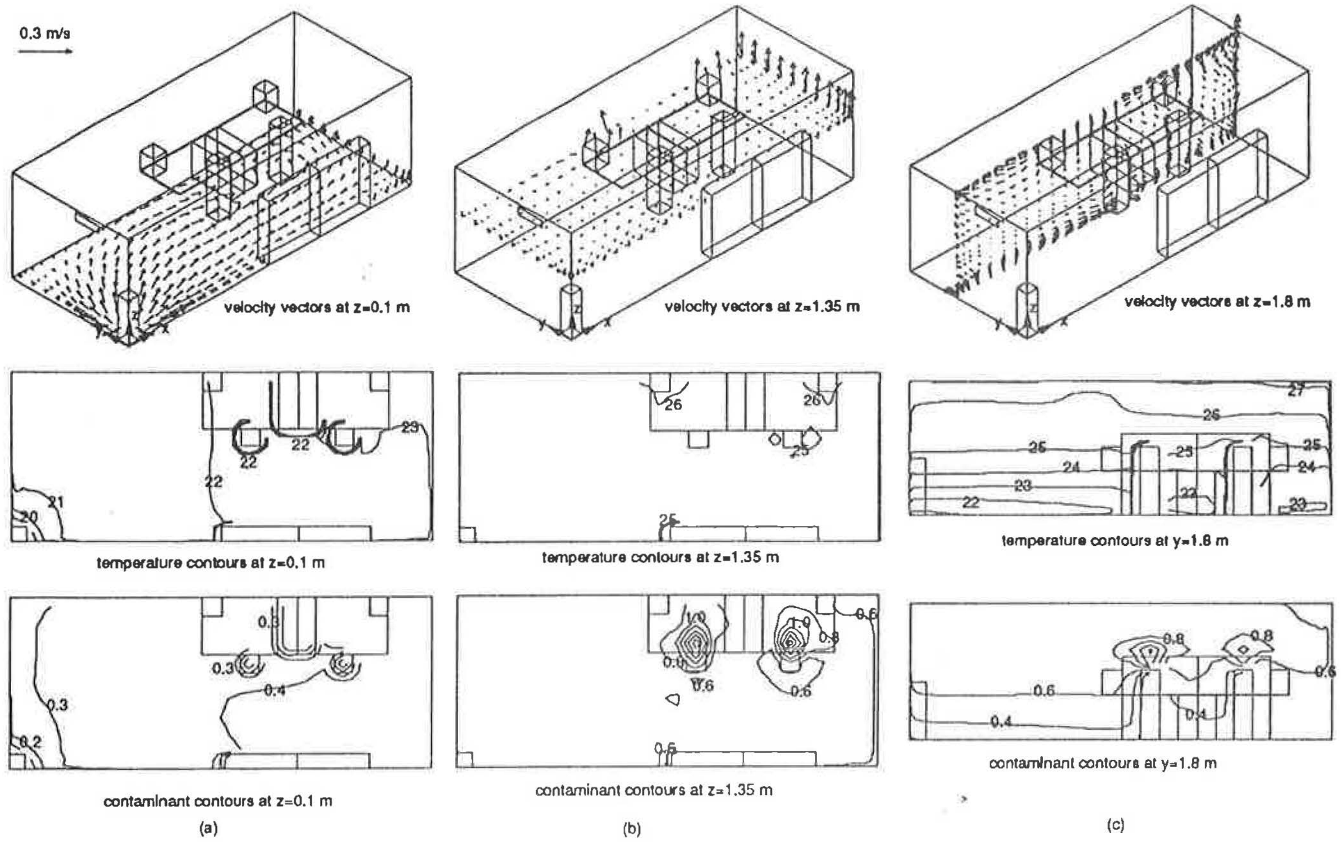


Figure 6 *Velocity vectors and contours of temperature and concentration for case 2 (ach = 2.1, total heat source = 1306.8 W: convective 80%, radiative 20%).*

airflow pattern, and the removal of heat and contaminants. In case 3, the ventilation flow rate is increased from 2.1 ach to 4.2 ach. The other parameters are the same as in cases 1 and 2.

It is observed from Figures 6 and 7 that the airflow patterns in cases 2 and 3 are similar in the region near the floor (the air velocity in case 3 is higher). The area where the air velocity is 0.15 m/s or higher is increased from about 17% in case 2 to about 30% due to the higher ventilation rate. The average level of air temperature is close to that of case 1 (with a cooling panel but a lower ventilation rate). The horizontal temperature difference along the x direction, observed only in the region near the floor, as shown in Figure 7a, is about 3°C. The contaminant concentration in that section is nearly one-third that in case 2. This is because the supply diffuser is placed at floor level, and the region near the floor benefits directly from the increase of ventilation rate.

At horizontal section z = 1.35 m, the air velocity is as low and uniform as in case 2 despite the ventilation rate being doubled. From Figure 4c one can see that the area where the air velocity is equal to or lower than 0.05 m/s is about 97%, the same as in case 2. Figures 6b and 6c show that when the ventilation rate increases from 2.1 ach to 4.2 ach, the room temperature is 1 to 2°C lower, but there is still a 5°C vertical temperature stratification. In the upper region, the horizontal temperature stratification is negligible. The temperature difference between the ankle level and head level of a sitting occupant is 3°C, with no improvement from case 2, but the contaminant removal is enhanced significantly. Almost everywhere, the contaminant concentration becomes half of that in case 2. However, in the area near the sitting occupants' heads, the concentration is still quite high, almost the same as in case 2. It means that in the central area, the ventilation efficiency is not affected much by ventilation flow rate.

Case 4

The radiative heat is increased to 80% of total heat. To study the effects of radiative heat on room air movement, the radiative heat is increased from 20% to 80% in case 4. Therefore, the heat distribution differs from that shown in Figure 3. The two sitting occupants (heat sources 3 and 4) are still assumed to be convective heat sources, but with an emission rate of 65 W from each instead of 81.5 W. The 523 W radiative heat (for half a room) is equally distributed to heat sources 1 and 2 (261 W for each). The other parameters are the same as in case 3.

The wall temperature increases by absorbing more radiative heat from the heat sources. As a result, the

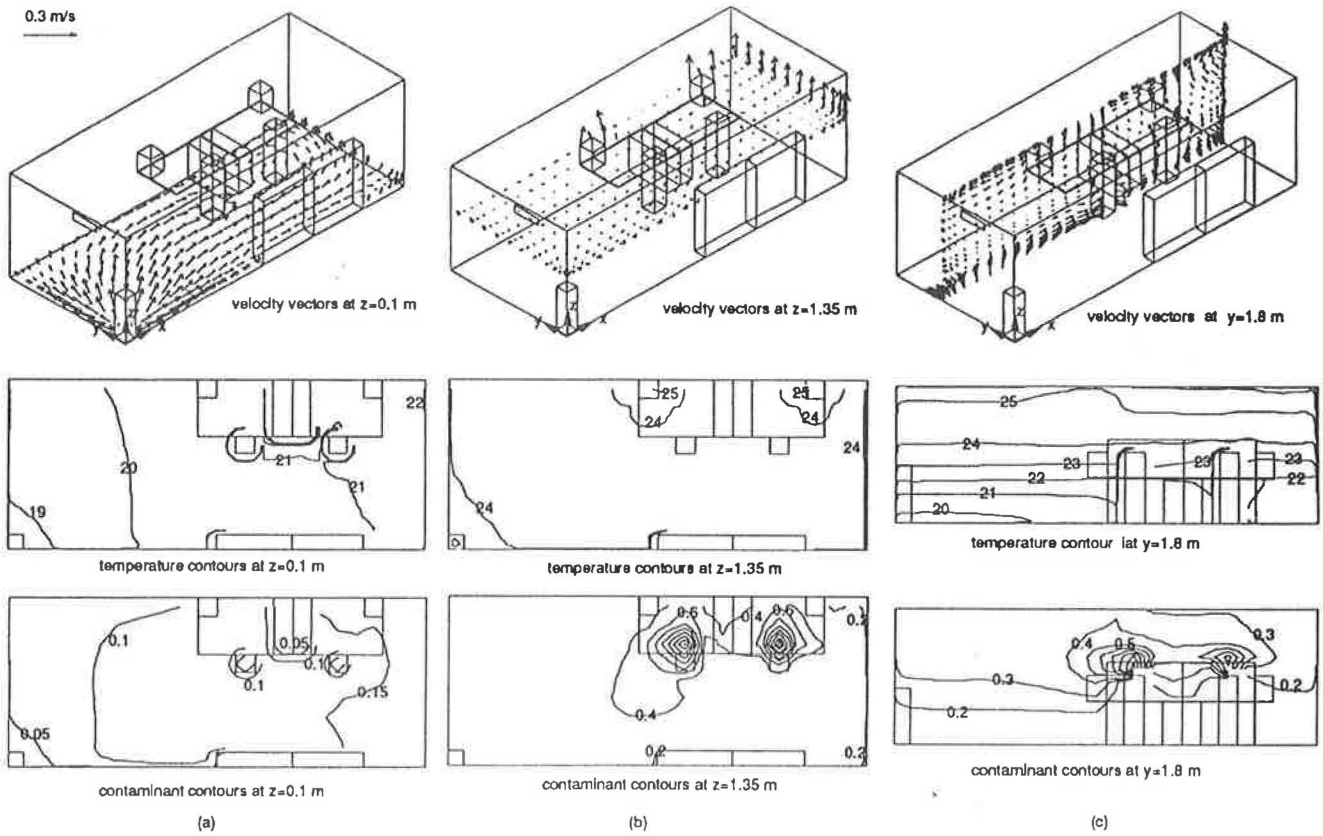


Figure 7 *Velocity vectors and contours of temperature and concentration for case 3 (ach = 4.2, total heat source = 1306.8 W: convective 80%, radiative 20%).*

upward velocity of air near the walls increases due to buoyancy, as presented in Figure 8. At horizontal sections $z = 0.1$ m and $z = 1.35$ m, the air velocity is higher than in case 3, although the ventilation rate is the same. This means that when radiative heat increases, the air movement in most areas becomes stronger. It can be explained as follows.

With the same amount of heat, if the percentage of convective heat is higher, the buoyancy is stronger only in the small region above the heat sources, and the high velocity due to the buoyancy is easier to be diffused into the adjacent air. When the percentage of radiative heat increases, the temperature at all walls rises by absorption. As a result, the air velocity is increased along all the walls, which may enhance the air circulation in the room, including the air movement near the floor, as observed in Figures 8a and 8b. The percentage of area with velocity higher than 0.15 m/s at section $z = 0.1$ m is about 60%, while this figure is only about 30% in case 3, as shown in Figures 4c and 4d. Figure 8c demonstrates that in the region close to the ceiling, the air velocity is not as high as that in case 3. This may be due to the reduction of convective heat; therefore, the buoyancy effect in the central area at this section becomes correspondingly smaller. The upward air movement is minimized, which could result in a lower velocity at ceiling level. Another reason may be due to the upward flow along the wall opposite the window, which is counterflow to the buoyancy flow along the window.

The vertical temperature stratification in case 4 is even larger than that in case 3 (about 7°C). However, the gradient is steep only in the region close to the ceiling. The temperature at the ceiling can be as high as 28 to 29°C. In the occupied zone (from floor to height $z = 1.8$ m), the vertical temperature variation seems to be close to that in case 3. At section $z = 1.35$ m, the air temperature is about 1°C lower than in case 3 because the convective heat source located in this section is reduced from 80% to 20%. In horizontal sections, the air temperature is quite uniform except at the ankle level, where there is a 3°C air temperature difference along the room length.

The contaminant concentration in front of occupant 1 is higher than that at occupant 2 at sitting and standing levels because occupant 1 is downstream of occupant 2. The average contaminant concentration is slightly higher than in case 3, but the highest contaminant concentration near the source is lower than that in case 3. It may be that the air movement in case 4 is stronger, therefore, with the same ventilation rate, the contaminant dispersion is enhanced.

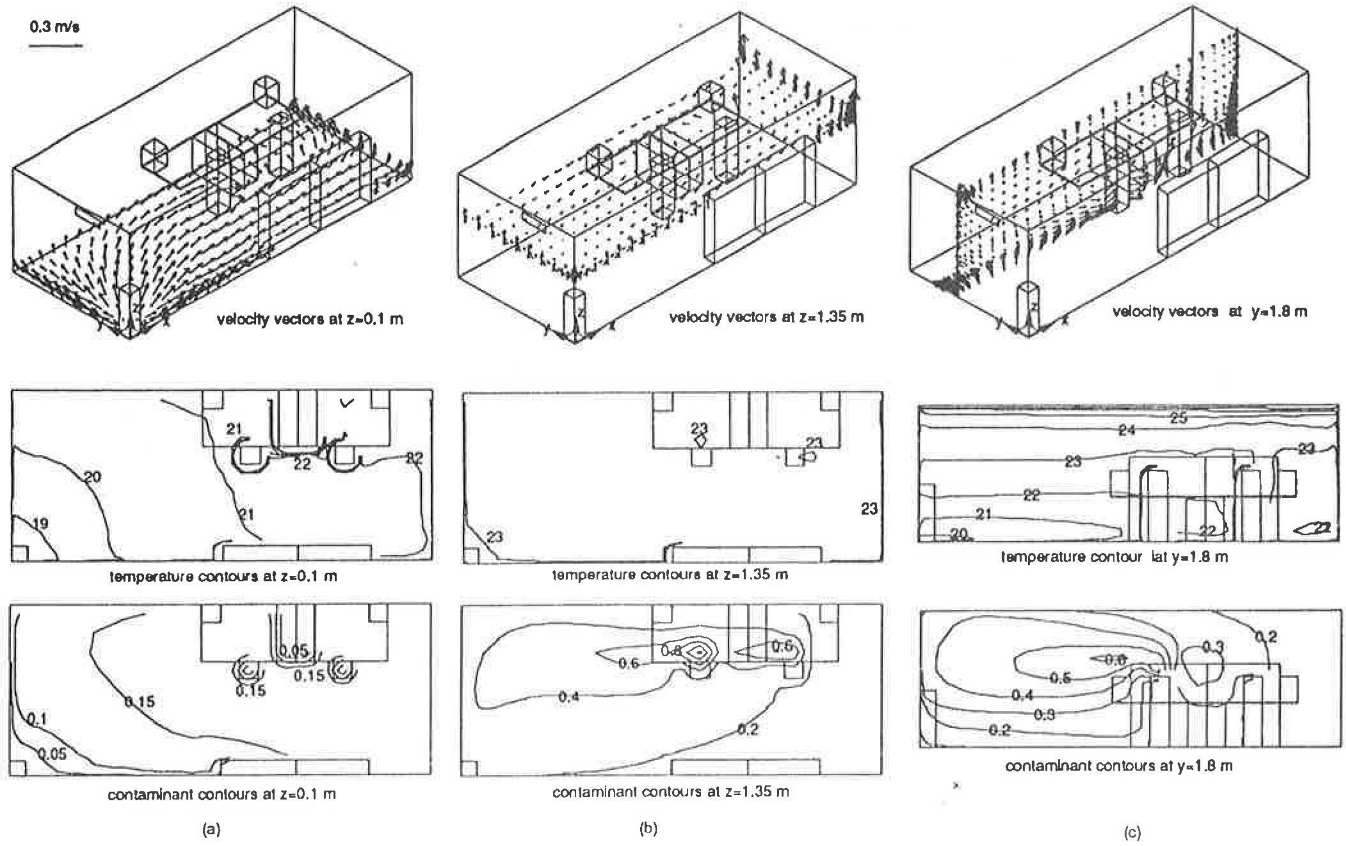


Figure 8 *Velocity vectors and contours of temperature and concentration for case 4 (ach = 4.2, total heat source = 1306.8 W: convective 20%, radiative 80%).*

CONCLUSIONS

The airflow pattern and the distributions of temperature and contaminant concentration in a large office with both convective and radiative heat sources are numerically investigated. The office is equipped with a quarter-cylinder displacement diffuser and a cooling panel. The effects of the parameters and the percentage of radiative heat on the airflow in the room are examined.

The following conclusions can be drawn from the present study:

- When the total heat emission rate remains constant, increasing the percentage of radiative heat indicates a shift of the heat source from its actual location to the boundaries of the room. In the present study, the radiative heat increase from 20% to 80% causes a considerable shift of heat source location. Thus, the changes in airflow pattern, temperature, and contaminant fields are significant. The changes include (1) air movement is strengthened, especially in the region near the walls; (2) the contaminant dispersion is increased; and (3) the air temperature in the occupied zone is lowered.
- Without a ceiling cooling panel, the vertical temperature stratification in the room could be as high as 7°C. When a cooling panel is placed in the ceiling, the air movement in the room is greatly enhanced due to the buoyancy effect, and the vertical temperature difference is reduced to 3°C. However, there is still a 2°C temperature difference between the ankle level and the head level of a sitting occupant.
- Increasing the ventilation flow rate can lower the room temperature, but is not a remedy for vertical temperature stratification. With displacement diffusers, contaminant removal in the middle region of the room cannot be improved by only increasing the ventilation rate.

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