SIMULATION METHODOLOGY OF RADIANT COOLING WITH ELEVATED AIR MOVEMENT

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ABSTRACT

In this paper, we present a new method of combining ASHRAE Room Energy Balance Method with 3-d airflow modeling to estimate the thermal comfort conditions of the emerging air-conditioning system that combines radiant cooling and desiccant dehumidification. The room cooling load at a desired operative temperature is calculated, and so are the room surface temperatures and convective and radiative heat fluxes, which are used as input for the simulation of 3-d room airflow and radiant temperature distributions, which are subsequently combined to calculate the Predicted Mean Vote (PMV). The simulation results indicate that raising air supply temperature can provide better thermal comfort in terms of thermal field uniformity, and that thermal comfort indices directly linked with local body-surface heat transfers may be more useful in estimating uniformity associated with elevated air velocities.

INTRODUCTION

Various cooled ceiling technologies are emerging air-conditioning alternatives (Mertz 1992; Wilkins and Kosonen 1992, Niu et al 1995, Benhe 1999). With this hydronic cooling system, air is supplied mainly for ventilation and humidity control purposes. This presents the opportunity for desiccant applications (Waugaman et al, 1993, Niu 1999). When desiccant system is combined with chilled-ceiling system, the two basic functions of an air-conditioning system, i.e., humidity and temperature controls are 'de-coupled'. Chilled-ceiling provides cooling, and air supply from the desiccant unit provides ventilation and moisture removal. Desiccant dehumidified air is essentially a warm air, which will provide the flexibility of supply air temperature control. The advantage is that this air can be supplied at near isothermal or slightly warm conditions. This will make it possible to locate the air supply nozzle more close to the occupants, so that room air movement can be raised to the upper limit within the thermal comfort range. Consequently, room air temperature can be raised. The perceived benefits include improved thermal comfort, reduced space cooling load, raised COP of the chiller system due to higher evaporator temperatures required. In this paper, the new characteristics with regard to thermal comfort of such an integrated system will be explored.

SIMULATION METHODS

In this paper, a method of estimating the thermal comfort conditions by combining whole room-simulation with 3-d airflow modeling is presented. An enhanced dynamic cooling load program ACCURACY (Chen and Kooi 1988, Niu and Kooi 1993, Niu et al. 1995, 1997; Niu and Burnett 1998), which is based on ASHRAE Room Energy balance Method, is used to calculate the room cooling load at a desired operative temperature condition. In the program ACCURACY, the percentage of active chilled ceiling panels to the ceilings can be specified. The program will calculate the required ceiling panel surface temperature to maintain the room operative temperature. This can be done at different air supply flow rate and temperature conditions, so that the impacts of different air supply parameters on the system energy use and thermal comfort can be estimated. The simulation sequence in the program is that, at a given air supply flow rate, the required air supply temperature is calculated. If the supply temperature is below the preset minimum, the ceiling panels will be 'activated', and the activated panel surface temperature will be calculated, and consequently, the required chilled-water temperature are calculated using a heat-exchanger model. The simulation also provides the room surface temperatures and convective and radiative heat fluxes. Then the convective heat fluxes are used as the input boundary condition for the room airflow simulations, and the surface temperatures are used for the calculation of 3-d room radiant temperature distribution. For the air flow calculation, standard k-ε turbulence model (Launder and Spalding 1974) is used and Boussinesq assumption (Tritton 1988) is used to account for the buoyancy effects due to temperature difference. Also caution was taken in locating the discretization grids in the near wall region to compensate for the deficiencies of the standard k-ε model (Niu and Kooi 1992; Niu 1999).

For the calculation of the 3-dimensional mean radiant temperature (t_{mr}) distribution, the so-called composite radiosity model (Spalding 1980, 1994) is used. Using the composite radiosity model, it is also possible to calculate the plane radiant temperature t_{pr}, and subsequently the radiant temperature asymmetry Δt_{pr}. In this paper, we focus on the uniformity at different positions in the room, so only t_{mr} will be discussed. The computed airflow and radiant fields are combined to calculate the Predicted Mean Vote (PMV) (Fanger 1982). Also local percentage dissatisfied due to draft (PD) are calculated. This methodology can also be witnessed in a publication by Negrao et al. (1999). It should be noted that the PMV model was originally developed to describe the whole body thermal sensation at steady conditions, and the PD model to describe draft feeling at the neck region (Fanger 1988). Furthermore, in the PMV model, turbulence effect is not counted, and in the PD model, radiant cooling effect is not counted. These model limitations will be taken into account when the simulation results are examined.

SIMULATION CASES

Room Description

A hypothetical office room is simulated as the object of this explore. The room is 5.1 m in length, 3.6 m in width and 2.64 m in height. The south facing façade has 30% glazing equipped with venetian blinds. 70% of the ceiling is loaded with chilled ceiling panels. There are two occupants (each occupant generating a convective heat of 50 watts and radiant heat of 25 watts). Heat gains also come from other internal heat sources in convective form. The convective heat is assumed to be 459 watts, generated by 4 large electrical appliances. The cooled panels will directly absorb part of the radiative heat gains. The displacement ventilation is adopted as the air supply method.

Figure 1 shows the configuration of the office room. The room is symmetrical in its layout, so only a half of the chamber is simulated to reduce the workload of calculation.

Simulated System Parameters

Three cases of the combining chilled-ceiling with floor level air supply were calculated. Air supply temperatures are set at 18 °C and 24 °C respectively, and the required chilled-ceiling temperatures are calculated accordingly to meet the required room operative temperature of 25°C. The temperature of 18 °C is believed to be the minimum supply air temperature to avoid large vertical temperature differences between head and ankles. At the supply temperature of 24 °C, two supply air velocities are investigated under the same supply flow rate of 26.7 l/s, or an air change (ACH) rate of 2 times per hour. This supply airflow rate is only around 30% above the minimum ventilation rate as specified in the ASHRAE ventilation standard, which is much lower than what's encountered in typical all-air-conditioning system designs in the hot and humid climate. It should be noted that, with the condensation-based air-conditioning system, supplying air at the temperature range 18°C to 24 °C would require the energy-wasteful reheating for humidity control purposes in the hot and humid climates. However, with desiccant dehumidification controlling the dew-point of the supply air, the supply air temperature can be raised by less-cooling independently without affecting the supply air humidity, which is set at 9.9 g/kg dry air to meet the moisture load of 0.184 kg/hour person in the current simulation.

Case 1: Baseline Case

An air supply temperature 18°C, air velocity 0.1m/s was simulated first. This is very much close to typical European displacement ventilation design parameters. Given in Table 1 are the initial conditions used as input for ACCURACY. The hour-by-hour meteorological weather data of Hong Kong is used.

Case 2: Raised Supply air temperature

In this case, at the precondition of maintaining the required room operative temperature the same as Case1, the air supply temperature is raised to = 24 °C. The other conditions are the same as Case 1. As the supply air temperature is changed, ACCURACY calculates the consequent load-extraction proportion of fresh air and chilled ceiling, and also the convective heat fluxes and surface temperatures.

Case 3: Raised Supply air velocity

While the other conditions are kept the same as Case 2, we just increase the air supply velocity from 0.1 m/s to 0.3 m/s by reducing the outlet area of the supply grille. The results calculated by ACCURACY is not changed under the assumption that the operative temperature is not affected by the increased air movement. The air movement effects are consequently examined in the following CFD analysis of the airflow details, air temperature and radiant temperature distribution and PMV.

RESULTS AND ANALYSIS

Table 1 presented the simulation results by ACCURACY for the two air supply temperatures. Since the results from ACCURACY are transient, the data are from the particular hour of the simulation year, i.e., 13:00 pm, July 1, 1989. It can be seen that, with the supply air temperature raised from 18°C to 24°C, the required heat extraction rate by the ceiling panels is increased from 335 watts to 488 watts. In terms of the percentages of the total cooling load, the panel cooling is increased from 60% to 90% of the total cooling load. This is achieved by lowering the chilled water temperature from 21.9 °C to 20.3 °C. These temperatures are still far above the dew point temperature of the room air. These results also clearly indicate that the required evaporator temperature for such a combined systems can be much raised from the conventional 7°C, which will significantly raise the system COP of the chiller. With regard to the impacts on system energy use, we are coupling with equipment models to do further analysis. In this paper, we focus on the consequent impacts on thermal comfort.

The surface convection heat flows and surface temperatures listed in Table 1 are used as boundary conditions for the due CFD analysis. Plotted as Figure 2, 3 and 4, are the distributed air temperature, radiant temperature, air velocity, PD due to draft, and PMV results in one selected vertical plane in the simulated room, where the hypothetical occupant is seated as indicated in Figure 1.

Case 1: Baseline Case

The simulated velocity distribution is typical of displacement ventilation pattern, which is characterized by low force convection flow, and buoyancy induced hot plumes around the human body and the convective internal heat sources (Figure 2 a). As typical with displacement ventilation, the air temperature near the floor level tends to be low (Figure 2 b), and there is a vertical air temperature difference between the feet/ankle level and the head level, which is undesired for human body thermal comfort. In the plotted vertical plane, the temperature difference between the 10 cm and 110 cm level appears to be very high, and reaches almost 6 K. While this may reflect the known problems with displacement ventilation, the magnitude may be exaggerated by the simulation methods employed here. In the ACCURACY simulation, the vertical air temperature difference is neglected, and the heat flux calculated for the floor will be biased accordingly. When this heat flux was used for the airflow simulation, the air temperature near the floor level tended to be lower-biased. Nevertheless, the vertical temperature difference is mainly produced by the low temperature supply air.

Around the supply nozzle, the PD due to draft (Figure 2 c) tends to be high, due to the combined effects of low temperature, relatively high velocity, and the existence of turbulence. At certain distance away from the nozzle, the draft risks will normally diminish. The PD distributions in the occupied plane appear to agree with general observations for displacement ventilation.

The mean radiant temperature distribution pattern (Figure 2 d) is dominated by the warm window surface and the cooled ceiling. In general, the radiant temperatures near the window are around 5 °C higher than those in the internal areas. Due to the ceiling effect, the radiant temperature around the occupant in the internal area appears to be fairly uniform. It should be noted that in our simulation, only the radiant temperature effects due to internal temperature differences are calculated, but the direct radiative heat from electrical appliances to people are not calculated. In office environment this is most likely the case.

PMV, which incorporated human metabolic level, clothing, air temperature and humidity, mean velocity, and radiant temperature, is used to assess the overall thermal comfort uniformity, although it was originally developed for whole-body energy balance analysis. In the plotted vertical plane (Figure 2 e), it appears that the vertical differences between angle level and head level are fairly large. While the head level is neutral, the foot level is slightly cool according the PMV scale. This may primarily be attributed to the large undesired vertical air temperature gradient in the occupied zone.

Case 2: Raised Supply Air Temperature

In the occupied plane, no obvious differences can be seen in the air velocity as from Case 1 (Figure 3 a). The vertical air temperature gradients in the occupant plane are much reduced. The difference $\Delta t_{0.1-1.0}$ is around 2 K (Figure 3 b). The PD due to draft around the air supply region is much reduced (Figure 3 c). In the occupied vertical plane, PD values are very low, indicating very low draft risks. The radiant temperature distribution becomes more uniform (Figure 3 d). Only very close to the window, radiant temperatures are still high. This clearly indicates that lower ceiling surface temperature tend to improve radiation asymmetry that would normally exist in a convection-cooled room. Because the supply air temperature is increased, the chilled ceiling affords a majority of the sensible cooling load. The surface temperature of chilled ceiling is lower than that in Case 1, and consequently the mean radiation temperature is lower. Also, the radiation temperature has generally negative gradients in the vertical direction. The cooled-ceiling panel surface is directly exposed to the head of the occupant. This is favorable to the reduction of thermal discomfort associated with the usually positive vertical temperature difference (Fanger et al 1985, Hodder et al 1998). It is a common sense that keeping the foot warm and the head cool is a desired thermal comfort situation. When the air supply temperature is raised, the vertical air temperature difference is also reduced. When the two effects are combined, the vertical PMV stratification is reduced, and more favorable for better thermal comfort (Figure 3 e).

Case 3: Raised Supply Air Velocity

As can be expected, the velocities in the outlet plane are raised. However, the velocities in the occupied plane do not appear to be much affected (Figure 4 a), and neither the temperatures (Figure 4 b). The temperatures in the outlet plane become more uniform, due to the stronger mixing introduced by the higher velocities. The PD due to draft in the occupied plane also do not very much affected, although in the outlet plane the supply jet tends to generate PD up to around 16% (Figure 4 c). The radiation temperature distribution is essentially the same as in Case 2 (Figure 4 d). The combined effect on PMV is that PMV in the occupied plane is slight lowered than in Case 2, and tends to be more evenly distributed in the occupied zone (Figure 4 e), indicating that, by raising the supply air velocity, it is possible to raise the set-point for the air temperature to achieve the same or even better comfort level.

DISCUSSIONS

As a more direct comparison, in Figure 5, the vertical variations of the air velocity and temperature, mean radiant temperature, and PMV for all the three cases are plotted at one location in the occupied vertical plane. In the plotted location in the occupied plane (Figure 5 a), the velocities are elevated in Case 3, while Case 2 exhibits the lowest velocity. Due to buoyancy effects, velocities in Case 1 are higher than those in Case 2 in the near floor region. The differences in temperature between occupant's head and foot (Figure 5 b) $\Delta T_{1.1\text{-}0.1}$ are 5.5 °C, 2.4 °C, 1.2 °C, respectively for Case 1, 2 and 3. The differences in PMV between occupant's head and foot (Figure 5 e) $\Delta PMV_{1.1\text{-}0.1}$ are 1.0, 0.3, and 0.1, respectively for the three cases. The vertical gradients for the radiant temperature are negative, the $\Delta T_{mr,\,0.1-1.1}$ are -1.2°C and -1.5°C respectively for the supply air temperature 18 °C and 24 °C (Figure 5 d).

Also, plotted in Figure 5c is the vertical variation PD due to draft in the mid-plane, in a location downstream of the supply jet. This plotting indicates that the PD values are increased by 10% in the near floor level. However, in the angle level, 5% less the calculated PD values should be expected according to Fanger's (1988) report. On the other hand, the PMV values for Case 3 are actually raised in comparison with Case 1 (Figure 5 e), indicating a feeling toward warm side. Coming back to the fact that the draft model did not take into account radiant temperature, and PMV did not take into account turbulence, the overall conclusion appear to be that Case 3 should be a more favorable comfort conditions.

In summary, supplying warm air make it possible to locate supply more close to occupants, and this can reduce the mixing of the fresh air and the indoor 'old' air, therefore, increase the personal air quality. What is more significant is that, coupled with increased air supply temperature, the fear that directly delivering the high velocity cold fresh air to occupants make them uncomfortable becomes redundant. Within certain extent, increasing the air velocity could help raise the preferred indoor air temperature, which is advantageous for energy saving.

CONCLUSIONS

Chilled ceiling combined with displacement ventilation and desiccant system could provide preferable indoor air quality and thermal comfort condition. The chilled ceiling could reduce the vertical temperature difference, and the desiccant dehumidification could take off the limitation on supply air temperature because of its independent outdoor air humidity control. To fully assess these potentials, a new simulation method is adopted, which fully takes into account the panel radiant effects in 3-dimension manner.

Raising supply air temperature could avoid the vertical asymmetry created with low supply air temperature. It will be easier to achieve thermal comfort condition to use higher supply air temperature then lower air temperature. The results clearly indicate that at higher air supply temperature and lower ceiling temperature conditions, the thermal comfort uniformity is much desired – the system tends to 'keep your head cool while keeping your feet warm'. It can be concluded that the conceived system can provide better thermal comfort in terms of thermal field uniformity, although the currently available thermal comfort indices might not be adequate to quantify this.

Further, raising supply air velocity could obtain lower PMV index under the condition of same heat load and operative temperature. This makes it possible to achieve the same PMV index when air temperature is increased. A special investigation on energy saving will be carried-out in the next step, by including a PMV-based feed-back control algorithm. The concern on the elevated PD due to draft may become redundant as air temperature is further raised. Nevertheless, a more refined draft model might be needed for this assessment.

ACKNOWLEDGEMENTS

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NOMENCLATURE

ACH= Air change rate
CFD= Computational fluid dynamics
COP = Coefficient of performance
PD = percentage dissatisfied of people
PMV = Preidcted Mean Vote
t_{mr} = mean radiant temperature
t_{pr} = plane radiant temperature
Δt_{pr} = radiant temperature asymmetry
$\Delta T_{0.1\text{-}1.1}$ = vertical temperature difference between the ankle and head level

Table 1. The simulation results by ACCURACY for the hour13:00, July1, 1989

	Temperatures (°C)				Cooling load (W)		
	Air Supply	**Water supply**	**Cooled Ceiling Surface**	**Indoor Air**	**Cooling load by air (W)**	**Cooling load by panel (W)**	**Total cooling load (W)**
Case 1	18	21.9	22.7	25.2	231.5	335.4	566.9
Case 2 (3)	24	20.3	21.4	25.4	45.6	487.8	533.4
	Convective heat flow (W)				**Wall surface temp. (°C)**		
	Case1		Case 2 & 3		Case 1	Case 2 & 3	
Rear wall	-3.14		-17.19		25.2	25	
Floor	-51.68		-72.11		24.7	24.3	
Ceiling	-7.5		-10.89		23.3	24.4	
Cooling panel	-245.5		-356.66		22.7	21.4	
South wall	-3.29		-13.76		25.1	25	
Façade 1	-19.7		-22.75		25.9	23.6	
Façade 2	-3.89		-4.51		25.9	23.6	
Window	121.72		118.49		37.6	34.6	

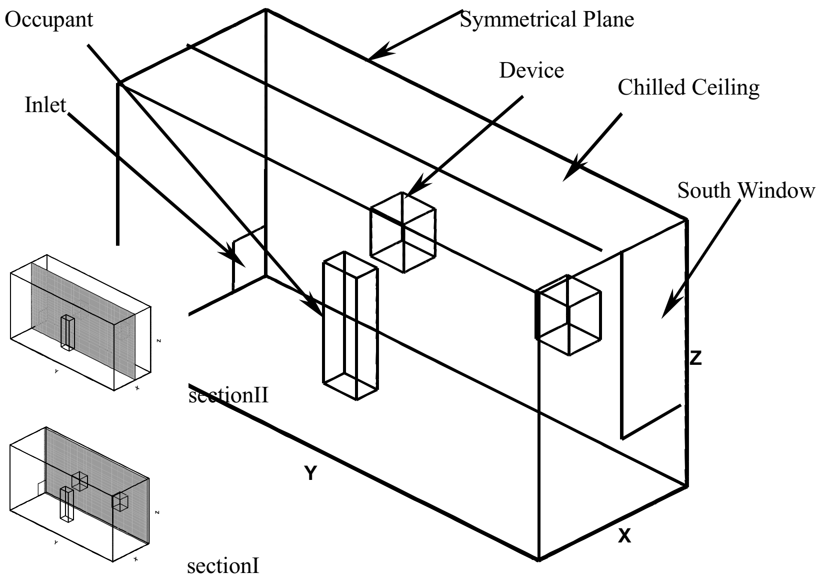
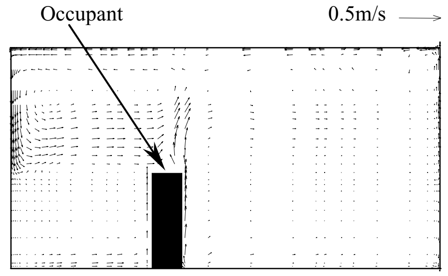
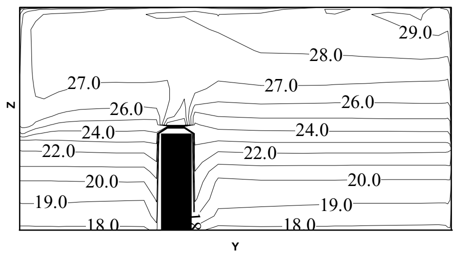


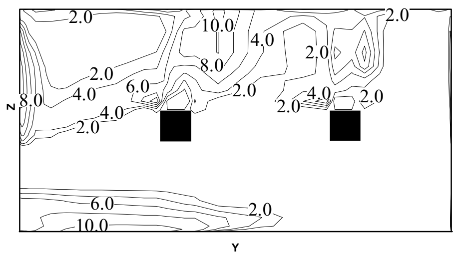
Figure 1. Configuration of the simulated office room



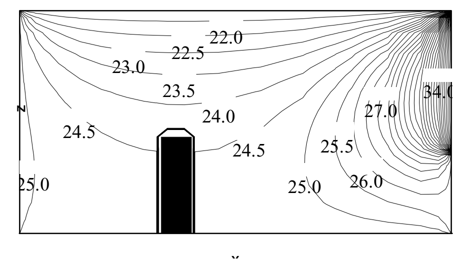
(a) Indoor air velocity (at X = 0.8)



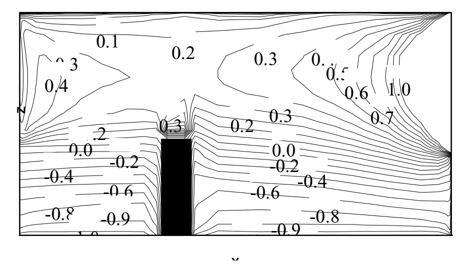
(b) Indoor air temperature (at X = 0.8)



(c) PD due to draft (at X = 0.07)

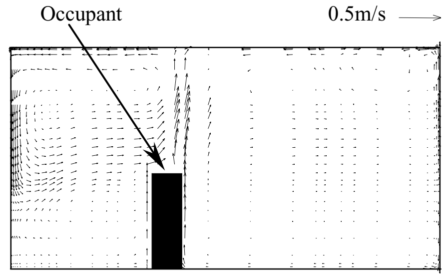


(d) Radiation temperature (at X = 0.8)

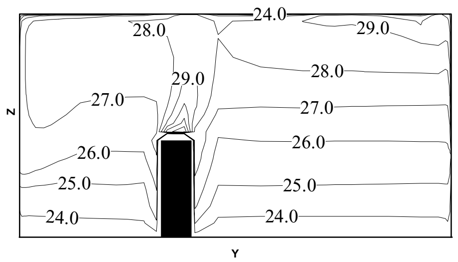


(e) PMV distribution (at X = 0.8)

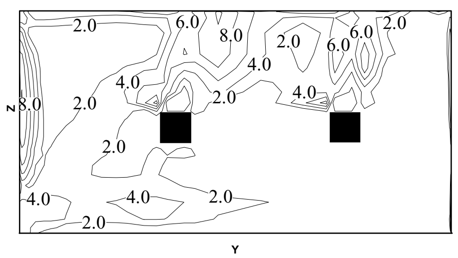
Figure 2. Simulated field distribution of Case 1



(a) Indoor air velocity (at X = 0.8)



(b) Indoor air temperature (at X = 0.8)



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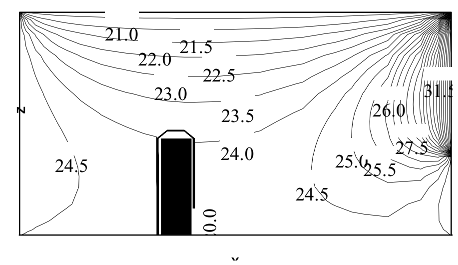
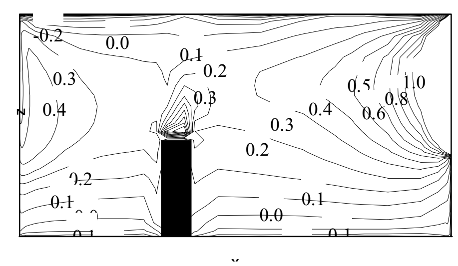
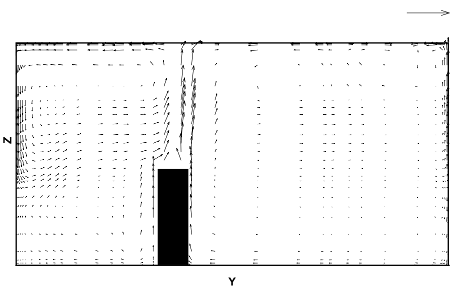


(d) Radiation temperature (at X = 0.8)

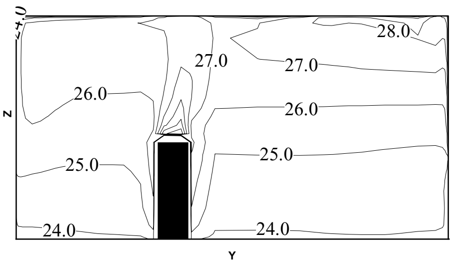


(e) PMV distribution (at X = 0.8)

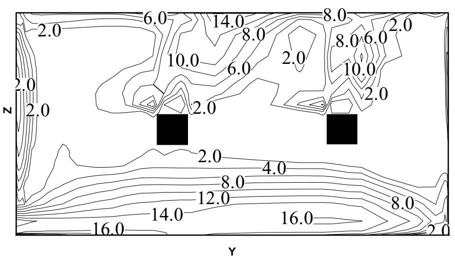
Figure 3 Simulated field distribution of Case 2



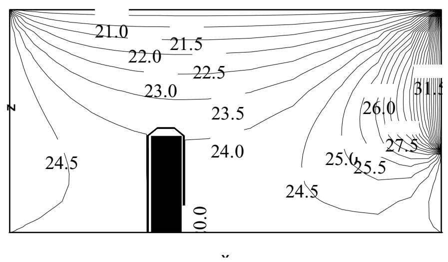
(a) Indoor air velocity (at X = 0.8)



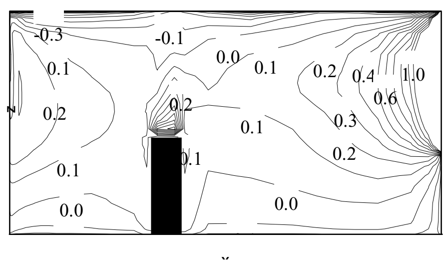
(b) Indoor air temperature (at X = 0.8)



(c) PD due to draft (at X = 0.07)

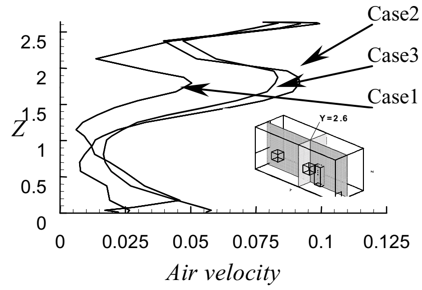


(d) Radiation temperature (at X = 0.8)

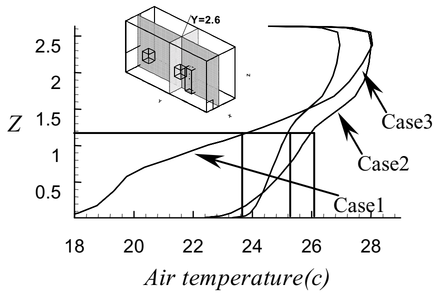


(e) PMV distribution (at X = 0.8)

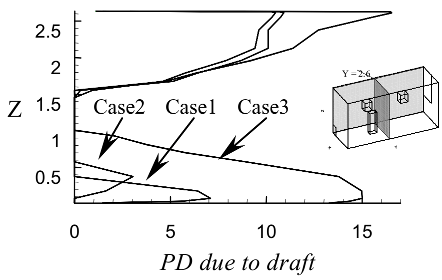
Figure 4 Simulated field distribution of Case 3



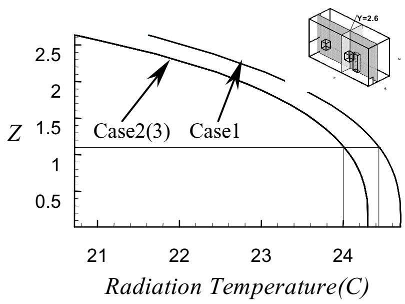
a). X = 0.8m, Y = 2.6m



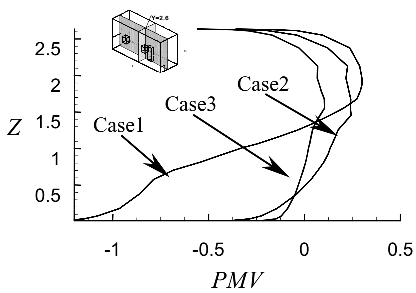
b). X = 0.8m, Y = 2.6m



c). X = 0.07m, Y = 2.6m



d). X = 0.8m, Y = 2.6m)



e). X = 0.8m, Y = 2.6m

Figure 5. Comparison of vertical stratifications