INDOOR ENVIRONMENT IN AN OFFICE FLOOR WITH NOZZLE DIFFUSERS: A CFD SIMULATION

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ABSTRACT

Computational Fluid Dynamics (CFD) simulation technique was used to study the effect of air distribution and supply parameters on ventilation performance and comfort of occupants in a government office building in Ottawa, Canada. The floor studied had two separate ceiling-based air supply systems, a slot system and a nozzle system with personal environmental control capability. In situ measurements were used to validate the results of the CFD simulation. Good agreements between the measured and predicted data were observed. The range of system configuration and parameters investigated include: (1) seasonal operating conditions (spring vs. summer); (2) supply airflow rate; (3) supply room temperature; (4) air throw orientation; and (5) diffuser location. Indoor air temperature, air velocity, mean age of air, Predicted Mean Vote (PMV), and Air Change Effectiveness (ACE) were estimated to investigate the performance of the ventilation system during occupancy.

INTRODUCTION

One of the main challenges facing Public Works and Government Services Canada, the custodian of federal government buildings, is to provide healthy and productive work environments to its building occupants while optimizing space use and minimizing energy waste. Open-plan floors and related workstation configurations do optimize space usage but make it increasingly difficult for conventional HVAC systems with common Air Handling Units (AHUs) to provide satisfactory indoor environmental conditions to each individual occupant with different comfort preferences. Providing each individual office worker with the capability to adjust own immediate workspace environment has been claimed to enhance personal comfort and productivity, on the one hands (Wyon, 1996) and lowering the operational, maintenance and energy costs of the building, on the other (Seem and Braun, 1992; Bauman et al., 1993). The different types of the localized ventilation systems evaluated in the above referenced studies were floor-based, desktop-based, and partition-based.

Most common and conventional ceiling mounted air supply systems use either slot or square diffusers. These diffusers provide a relatively uniform air velocity, temperature, humidity and air quality throughout the space (ASHRAE, 2001). The office studied, the eighth floor of a Public Works and Government Services building, is an open plan office that originally had ceiling mounted slot diffusers. The slot diffusers were adapted to incorporate nozzle diffusers. Each workstation was thus supplied with a single nozzle to provide the occupant with a personal environment control (PEC) system allowing each office worker the option to adjust individual air supply volume and air delivery angle.

Better understanding of the ventilation efficiency, thermal comfort and building energy consumption requires detailed description of airflow, temperature distribution, and age of air dispersion. As a powerful tool for fluid flow analysis, CFD can be used to predict air flow patterns, heat transfer, and species transportation in controlled environments. CFD techniques have been widely used to study indoor air and ventilation fields in non-industrial indoor microenvironments (ROOMVENT 2000). However, many studies on localized and personalized ventilation systems are mostly based on experimental data (Fisk et al,. 1991; Haghighat et al. 1996; Bauman et al., 1998; Nakamura et al., 1999; Faulkner et al., 2002). The purpose of this investigation is to generate a computational model for the 8th floor office mentioned above using a commercial CFD software package (Airpak by Fluent Inc., 2002) and use the results of the CFD simulations to investigate the effects of terminal type, air supply temperature, air supply volume, and air throw angle on ventilation, comfort, and potential energy costs.

In this study the air change effectiveness (ACE), PMV (Predicted Mean Vote), air temperature, velocity, and mean age of air are used as indices to quantify and compare the influence of air terminal device and air supply parameters.

ACE is a parameter used to compare the effective ventilation rate in the breathing zone to that occurs throughout the room provided the indoor air is well mixed. ASHRAE (1997), defines the ACE based on age s of air as: ACE = t_n / t_{avg}

Where t_n is the nominal ventilation time which equals the mean age of air exiting the building and t_{avg} is the mean age of air in the breathing zone.

PMV is the most common quantitative method for evaluating thermal comfort, which indicates the mean vote concerning the global thermal sensation of a large group of people. It is expressed on a seven-point thermal sensation scale: -3 (cold), -2 (cool), -1(slightly cool), 0 (neutral), +1 (slightly warm), +2 (warm), +3 (hot). The simulation program uses the ISO 7730 standard to calculate PMV based on computed air temperature, air velocity, and mean radiant temperature along with reasonably assumed values of relative humidity, metabolic level, and clothing insulation level.

MEASUREMENTS AND SIMULATIONS

Measurements took place in a five-workstation area located in PWGSC office 8B1, with the dimension of L x W x H = 9.6m × 6.2m × 2.5m, see Figure 1a. The whole area is furnished with two background lights, one is at a fixed power of 128w, and the other one is adjustable from 0w to 128w. Each workstation contains heat sources of a typical office, including: 64 W (2 x 32w) task lights adjustable from 0 output to 100%; a personal computer and a monitor with a total power of 150W; an occupant performing typical office work releasing 75W sensible heat. Also, there are solar heat gains and heat losses through the south-facing window and external wall.

As for the ventilation system in use, each workstation is equipped with a nozzle diffuser, and Figure 1b provides the picture for which; the temperature of conditioned air is set by the central AHU within the range from 13°C to 16°C; while the supply volume through each nozzle is adjustable from 16 to 51L/s and the air throw angle can be set manually (360° rotation). Along the exterior walls and windows, there are re-circulation air induction units, sharing part of the cooling/heating load. Air returns through the vents at the ceiling level, which are located in workstations near the windows, as shown in Figure 1.

A motion sensor is installed in each workstation. If no motion is detected in the workstation after 10min, the sensor will request the control system to lower the airflow rate to 16L/s (minimum) and the task light to 12.8 w (20% of total power), provided these two values were not set manually.

Measurements were conducted under two conditions: one was recorded at 15:00 on April 6th 2003 (corresponding to spring condition) and the other was at 15:00 on September 24th 2004 (corresponding to summer condition), see Table 1 for details. These two sets of field measurements were used as the inputs and boundary conditions for simulation.

CFD simulations were performed by applying the aforementioned boundary conditions. Based on some tentative solutions by applying the four turbulence models, namely, the mixing length zero-equation model, indoor zero-equation model, standard k-ε model, and Re-Normalization (RNG) k-ε model to this problem, it was found that convergent solution with similar accuracy can be obtained by indoor zero-equation model with less computing effort. Mixing length model generated solution with relatively large discrepancy, while standard k-ε model required approximately 200% more computing time. With RNG k-ε model, it's hard to get convergent solution for this specific case. Thus, indoor zero-equation model was chosen for simulation, which is also recommended by Chen's work (Chen and Xu, 1998). Other modeling methods and calculation conditions are described in Table 2.

MODEL VALIDATION

When preliminary results are available, validation can be conducted first by comparing the predicted airflow patterns created by nozzles (Figure 2) with those in literature (Hu, 2003), with supply volumes at 51L/s and 47.2L/s. Similar velocity distribution pattern and magnitude can be observed. Then the predicted temperature was compared with the field measured data. Figure 3 presents a comparison between the readings of four wireless sensors (located in WS 1, 2, 4, and 5, respectively) and the predicted temperatures.

Good agreement between the predicted and measured data can be observed for spring case. The largest discrepancy, at sensor 4 location, was less than 3.5%. Such a good agreement demonstrated that the selected physical models and boundary conditions were appropriate in terms of describing the heat transfer and airflow phenomena within this office; thus, reliable solutions to this indoor air problem can be obtained by the model created.

While for the summer data, discrepancies appeared for all four-sensor locations, the largest value being around 6.5%. The reason may lie in the fact that, under the summer condition, occupants are more sensitive to the microenvironment and are more willing to adjust the perceived temperature; consequently, frequent fluctuations in the space airflow and temperature are inevitable. However, the problem was defined as a steady-state one and thus caused some discrepancies.

RESULTS AND DISCUSSIONS

Based on the quasi-validated model, the following cases are designed to simulate and analyze the effects of air terminals and air supply parameters on the workspace environment and occupants' thermal sensations. Table 3 lists all the cases studied for the five-workstation layout. Case 1, which is under spring conditions, is used as the base case. Due to the large amount of simulation cases and results, a limited number of case studies have been selected from Table 3 to be presented next. The emphasis of the data presented here is on the thermal conditions near the occupants and right under the diffusers. The thermal condition studied for near the occupant was taken as the average of four points around the occupant at each required height. The following parameters have been selected to evaluate the microenvironment in each workstation:

1. The air velocity, temperature, and mean age of air at the following different heights: 0.1m, 0.6m, 1.1m, 1.7m, 2.0m and 2.4m. The height level at 0.1 m, 0.6m and 1.1m, correspond to recommended heights for seated subjects with the 1.1m being the breathing level. The 0.1m 1.1m and 1.7m levels correspond to heights recommended for standing subjects with the 1.7m being the breathing level. Furthermore, two other height level at 2.0m and 2.4m were also used since the air supply terminals and return grills are located on the ceiling.
2. Predicted Mean Vote, PMV, at the height of 1.1m. Air Change Effectiveness.

Spring (base case) vs. summer conditions

In Table 3, cases 1 and 8 demonstrate spring and summer operation conditions for the ventilation system, respectively. Figures 4a and 4b present the average temperature and air velocity results near the occupants in each of the workstations. These parameters are the most critical parameters to analyze the risk of occupant's thermal discomfort. Table 4 lists the PMV values near the occupants at the height of 1.1m. For a sedentary person performing moderate office work, the metabolic rate is 1.2 ($70W/m^2$), the clothing insulation level is assumed to be 1.0 clo ($0.155\ m^2 \bullet K/W$), and relative humidity is set to about 40%. For the spring case, all the five workstations are occupied; while in the summer case, except for workstation 4, all the other four workstations are unoccupied, thus the sensible heat from human body and reduction in the task lighting power are not included in the heat load within those unoccupied workstations. Figure 5a and 5b plot the distribution of mean age of air near the five occupants. Observations are as follows:

1. In summer, the AHU and induction units supply air at low temperature, compounding this condition, the low heat load due to the low occupancy results in a relatively low temperature in all the five workstations: on average, 2.5°C lower than those in spring; in both cases, the vertical temperature differences between 0.1m level and 1.7m level are less than 3°C, which is the maximum temperature difference recommended by the ASHRAE Standard 55-1992 and ISO7730.

2. Although the flow rate of the supply air varies from workstation to workstation in both cases, air velocity difference between the two cases at all the heights is small; near all the five occupants in both cases, the air velocities are less than 0.25 m/s, which is within the comfort limit.

3. In the case of spring, PMV values near the five occupants at the height of 1.1m are all near neutral, except for workstation 2, where the supply volume is maximum--51L/s; while in summer, PMV are negative in all the five workstations, due to the low supply temperature, especially in the unoccupied workstation 1, 2, and 3, the supply volume from the corresponding nozzles are not at minimal. As can be expected, the relatively high airflow rate bring the PMV in these three workstations down to -0.275, -0.19, and -0.13, indicating a typical office worker wearing typical summer cloth may feel "slightly cool". It can be concluded that the cooling capacity has not been utilized efficiently, and this kind of energy waste deserves serious consideration.

4. In summer, for the unoccupied workstation 5, although the air is supplied at minimum flow rate (16L/s), the thermal environment is not very pleasant (PMV=-0.142). Such a result suggests that the minimum flow rate (when a workstation is unoccupied) can be set to lower than the present value---16L/s. For the occupied workstation 4, even setting the airflow rate to minimum (16L/s), the user still ends up with temperature and PMV no higher than those values in the unoccupied workstations. This may indicate that within a partitioned zone, the interactive influence between each of the workstations is not negligible. The ventilation airflow rate in workstations 2 and 5 are 36L/s and 26L/s, respectively; however, only minor difference can be observed from the velocity and temperature in these two workstations; additionally, the local supply temperature in workstation 4 is 1.8°C higher than that to workstation 5, but this only causes little difference in the velocity and temperature between the two workstations. Thus, the supply air volume and temperature in a workstation do have impacts on the microclimate in adjacent workstations.

5. From the airflow pattern (Figure 2), nozzle diffusers deliver air down to the breathing zone and create vertical jet. ACE values listed in Table 4 are higher than unity, indicating that the nozzle diffuser does a good job of mixing the room air and no evident short-circuiting exists. A short-circuiting flow pattern increases the room air age and thus causes ACE to be less than unity.

6. It is very interesting to look at the distribution of mean age of air, Figure 5. The largest value always exists in workstation 3, while the lowest values

appear in workstations 1 and 4, regardless of the ventilation airflow rate. From Figures 1 and 2, one can see that the return grills are located above workstations 1 and 4, and workstation 3 is the remotest to these vents. It can be inferred that the distance from a workstation to the vent is a critical factor affecting the mean age of air.

Effects of air supply volume and temperature

Since the supply air volume and temperature are critical for energy saving and comfort level, cases 3, 4, 5, 6 and 7 are designed to check the effects of these two parameters. The five different combinations of supply air volume and temperatures are: case 3, low temperature (13°C) and low volume (16L/s); case 4, low temperature (13°C) and medium volume (35L/s); case 5, high temperature (15.8°C) and low volume (16L/s); case 6, high temperature (15.8°C) and medium volume (35L/s); and case 7, high temperature (15.8°C) and high volume (51L/s). All the other parameters are identical to those in the spring case. Figures 6a and 6b present the average velocity and temperature results near the occupants in WS 2, and Table 5 presents the PMV values. Observations are as follows:

1. In all the five cases, velocities decrease with increase in height below 1.1m, while increase slightly with increase at height above 1.1m. Higher supply volume creates relatively higher velocities, but the maximum difference between cases at all the heights above 0.1m is no greater than 0.1m/s.

2. In all the five cases, temperature varies with height in a consistent pattern. In general, the maximum temperature difference between the two cases is no greater than 3°C. The lowest temperatures are provided by the medium supply volume combined with a low supply temperature (case 4), while the highest temperatures are produced by low volume combined with high supply temperature (case 5). The other cases (high volume with high temperature case 7, medium volume with high temperature case 6, and low volume with low temperature case 3) provide similar intermediate temperature results.

3. Comparison between case 3 and case 5 indicates that when supply volume is low (16l/s), supply temperature has relatively small influence on the temperature distribution; when supply volume is increased to 35l/s, increasing the supply temperature from 13°C (case 4) to 15.8°C (case 6) results in a 1.8°C rise in the temperature near occupant.

4. From Table 5, PMV values indicate that high supply temperature combined with high/medium airflow rate produce more comfortable environment under the specific conditions. Much higher COP (coefficient of performance) of the central air conditioning equipment can be achieved by increasing the supply air temperature and the supply volume can be related to the energy consumption of fan. Thus, choosing the optimal combination of these two parameters for the ventilation system deserve further consideration. In this specific scenario, it is reasonable to choose medium supply volume and high supply temperature within the controllable range.

Effects of diffuser location

The area of workstation is about 7m^2 in this office environment, the distance between occupant and diffuser has important influence on both occupants' thermal sensation and ventilation effectiveness. In cases 9a, 9b and 9c, occupants are sitting under the diffusers; while in cases 5, 6, and 7, diffusers are located away from the occupants. All the other conditions for these two sets of cases are identical. Figures 7a, 7b, and 7c present the temperature, average velocity, and mean age of air near occupant in WS 2 in both cases 9s and in cases 5, 6 and 7. Table 6 presents the PMV and ACE values. Observations are as follows:

1. Between the height of 0.1m and 1.7m, there is only slight difference in the magnitude and distribution pattern of temperature when changing the nozzle location. Above 2m, placing diffusers above the occupants results in low temperature.

2. Much more evident effects on the air velocity distribution can be observed. Placing diffusers above the occupants generates higher velocity. In the upper unoccupied zone the air velocity difference is large; in the breathing zone, this difference decreases but still exists. In cases 9a, 9b, and 9c, the air velocities at 1.1m height are higher than 0.25 m/s, and thus may cause dissatisfaction.

3. Although temperatures are almost in the same magnitude, the high velocities in cases 9b and 9c cause negative PMV values, as shown in Table 6. It can be inferred that nozzle diffuser maintains the ambient temperature warmer and undisturbed while supplying cold air directly to the breathing zone. Thus, nozzle diffusers with medium and high airflow rates should be installed at a reasonable distance from the workstation location to prevent draft and discomfort; otherwise, the air jet angle should be adjusted to avoid uncomfortable conditions.

4. When occupant is sitting under nozzle, there is a reduction in the mean age of air near him at the height of 1.1m or above, and thus ACE values increase (Table 6). In cases 9a, 9b, and 9c, increasing supply volume results in higher ACE. A consistent increase or decrease in the age of air above the floor would be an indication of a general upward or downward airflow pattern. Thus, occupant will probably find downward airflow when sitting under the nozzle, and may feel upward airflow when staying away from the diffuser.

Effects of air throw angle

All the conditions for cases 1 and 2 are identical, except that the nozzles deliver air downward in case 1 and toward the occupants in case 2. Table 7 presents PMV values at 1.1m height under nozzle diffusers in WS2, WS3, and WS5. In both cases, airflow rates are 51L/s (high), 23L/s (medium), and 16L/s (low) for nozzle 2, 3 and 5, respectively.

1. As expected, at locations under the three nozzles, comfort level has been improved. PMV values under nozzles 2, 3, and 5 increased from -0.79 to -0.31, from -0.5 to -0.38, and from -0.3 to -0.2, respectively; consequently, adjusting the air supply angle is effective in terms of preventing local discomfort in jet area.

2. When air is delivered toward the occupants, the PMV values near the occupants decrease. Higher supply flow rate results in lower PMV, as can be observed near occupant in worksation 2.

CONCLUSION

1.Commercially available Computational Fluid Dynamics (CFD) simulation software, Airpak, has been successfully applied to simulate the airflow pattern, temperature distribution, and thermal comfort in a PWGSC office. Good agreement between in-situ measurements and simulation results indicated that the physical model and boundary conditions have been accurately implemented in the software, and the simulation results are accurate and representative of the actual office.

2. Nozzle diffuser maintains the ambient temperature warmer and undisturbed while supplying cold air directly to the breathing zone and thus possesses high energy efficiency potential. However, nozzle diffuser may cause a “slightly cold” region beneath it, thus the installation location and air throw angle should be chosen with cautious.

3. In this office under summer conditions, waste of cooling capacity may exist to some extend, more field measured data are necessary to identify this problem. For those unoccupied workstations, minimum supply flow rate can be lower than the present value (16L/s), while the supply temperature can be higher than the present up limit (15.8°C).

4. In addition, an unoccupied workstation with low air flow rate and task lighting level has some impact on the microclimate in the adjacent workstations.

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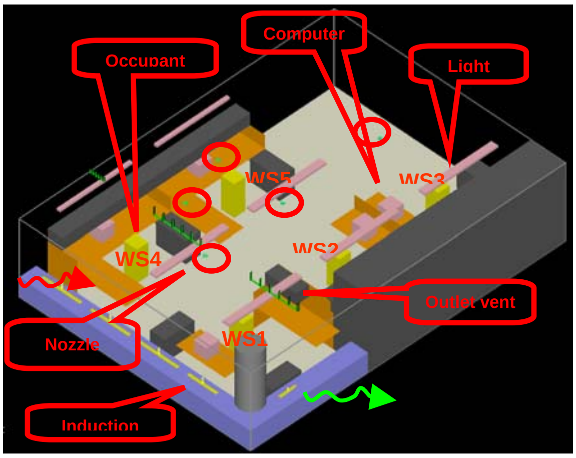

Figure 1a Layout of the office

Figure 1b Nozzle diffuser

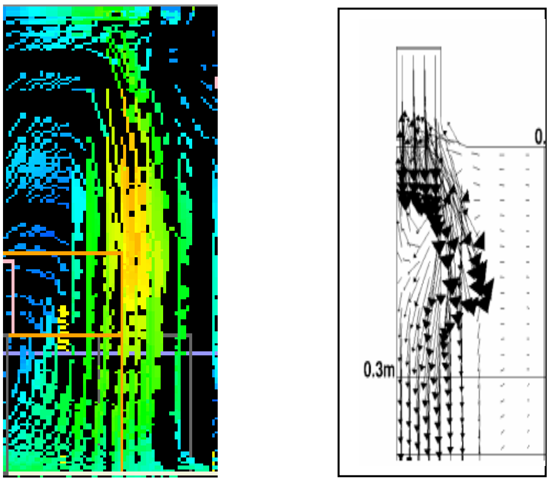

Figure 2 Airflow pattern from nozzle diffuser: simulation (left), literature (right)

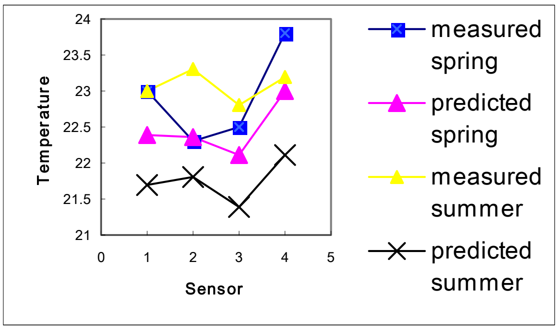

Figure3 Comparison of temperatures at four sensor locations (spring and summer)

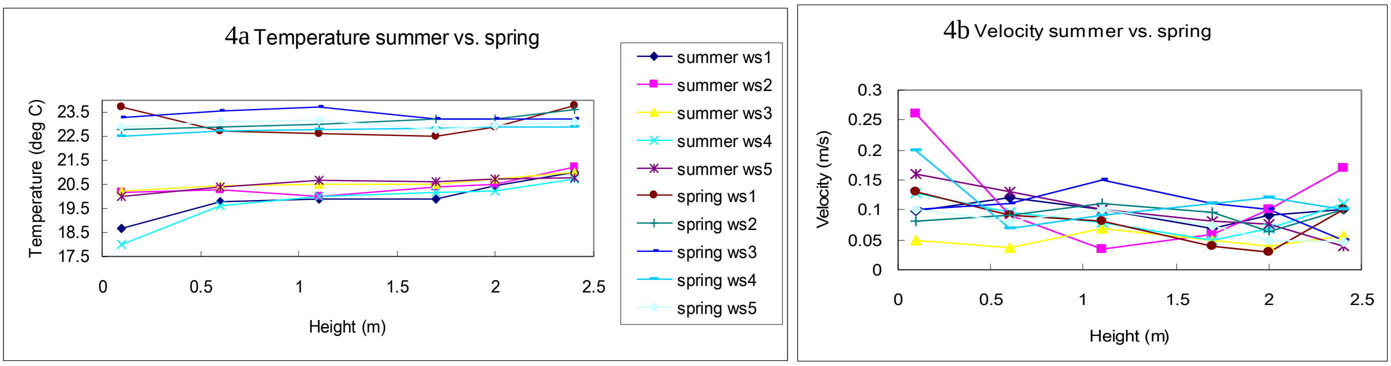

Figure 4 Temperature results and Velocity: summer and spring

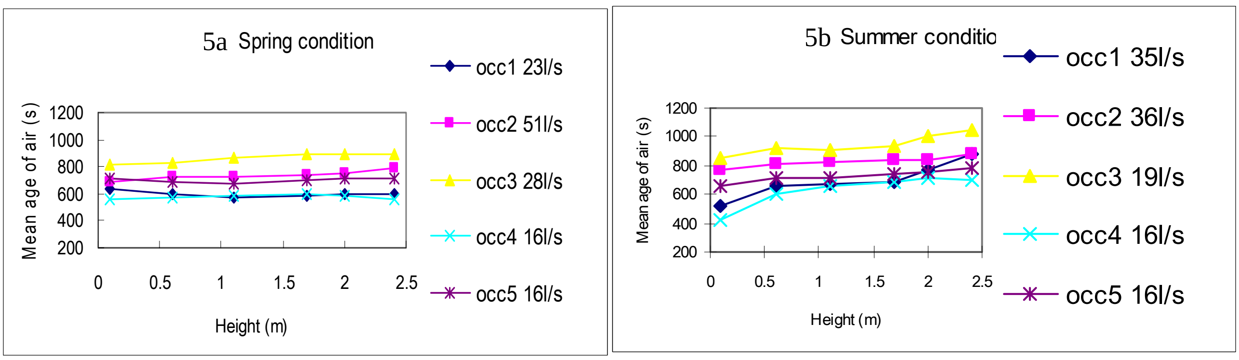

Figure 5 Distribution of mean age of air: summer and spring

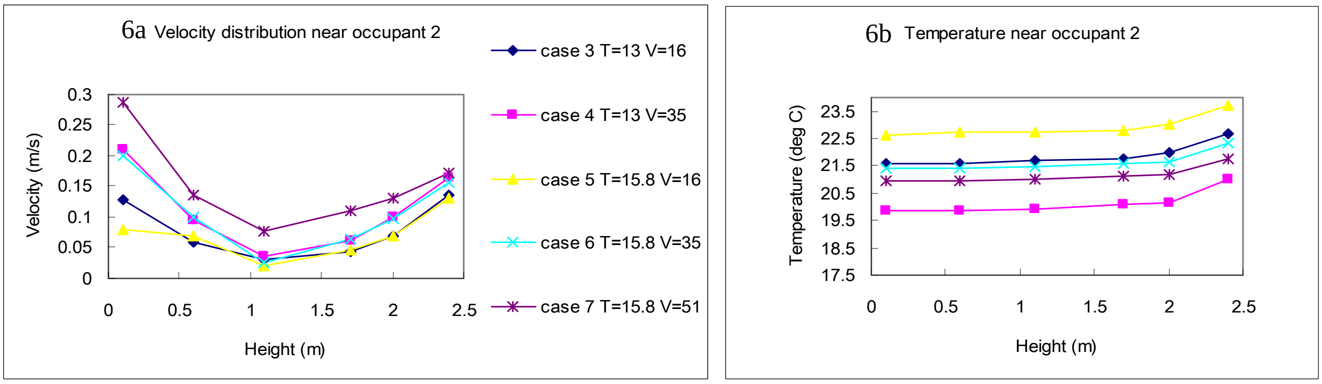

Figure 6 Velocity and Temperature near occupant 2 (effects of supply volume and temperature)

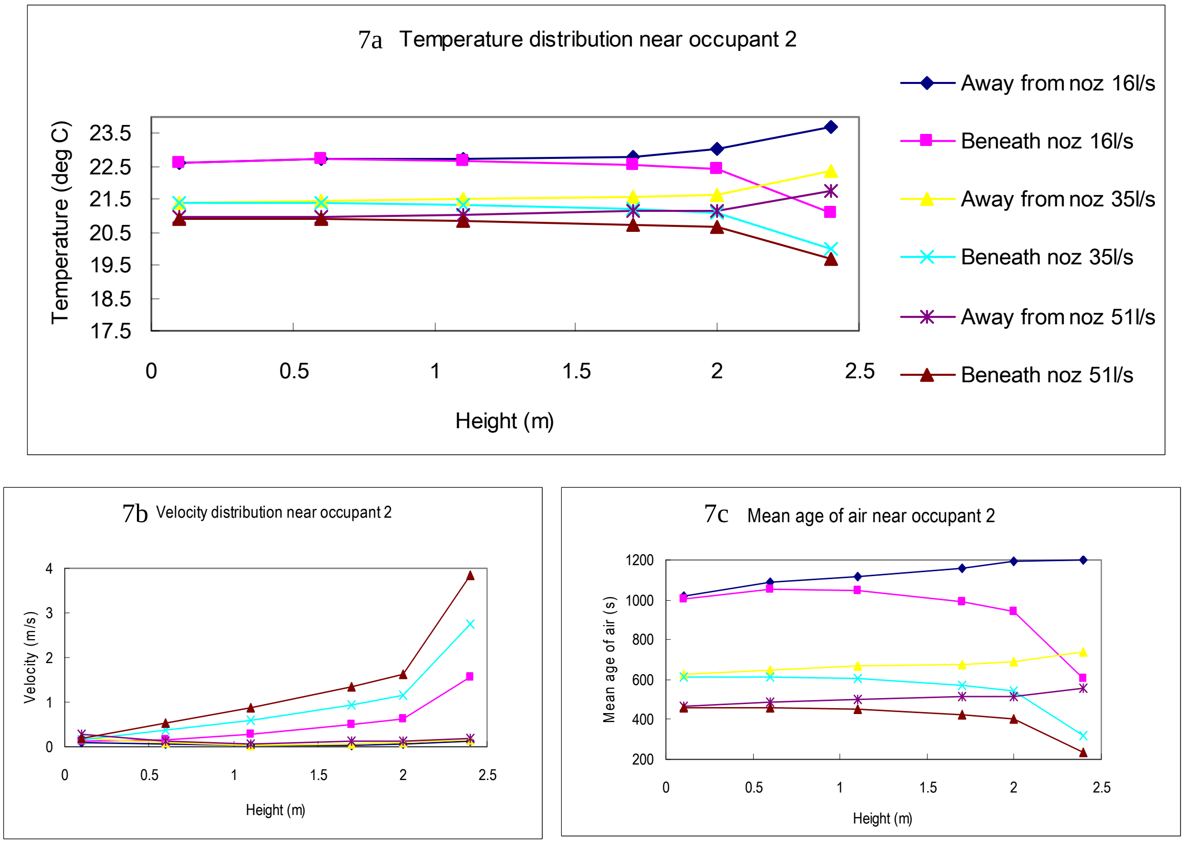

Figure 7 Temperature, velocity and mean age of air near occupant 2 (effects of diffuser location)

Table 1 Measurements as boundary conditions

Measurements/Boundary conditions		Spring conditions (case 1)					Summer conditions (case 2)				
Window/wall inner surface temperature (°c)		21					27				
Ambient temperature (°c)		23					23				
Air supply temperature (°c)		15.8					17	15.4	15.8	18	16.2
Local supply volume through nozzle diffusers(L/s)		23 ws1	51 ws2	28 ws3	28 ws4	16 ws5	35 ws1	36 ws2	19 ws3	16 ws4	16 ws5
Task lighting level (watt)		12.8	64	64	12.8	64	12.8	12.8	12.8	12.8	12.8
Ambient lighting level (watt)		128 (Near window)/128(Interior)					102.4 (Near window)/128(Interior)				
Induction Unit	Temperature (°c)	21(supply)/22(return)					18.3(supply)/22.4 (return)				
	Flow rate (L/s)	150					150				

Table 2 modeling method and meshing size

Turbulence model	Radiation model	Diffuser modeling method	Maximum meshing size (m)			Number of cells
Indoor zero-equation model	Surface to surface model	Momentum method	0.3(X)	0.1(Y)	0.5(Z)	250×10^3

This method assumes the airflow from a diffuser can be predicted by the isothermal axisymmetric jet formula.

Table 3 case studies

Case No.	Window inner face temperature	Temperature (°C)		Supply Volume (L/s)					Air throw angle
		Room	Supply‖	(ws1)	(ws2)	(ws3)	(ws4)	(ws5)	
1(base case)	21	23	15.8	23	51	28	28	16	downward
2	21	23	15.8	23	51	28	28	16	toward
3	21	22	13	16(low)					downward
4	21	21	13	35 (medium)					downward
5	21	23	15.8	16(low)					downward
6	21	22	15.8	35 (medium)					downward
7	21	22	15.8	51(high)					downward
8	27	23	14	35	36	19	16	16	downward
9a	21	23	15.8	Sam as case 5 (occupant under nozzle)					downward
9b				Sam as case 6 (occupant under nozzle)					
9c				Sam as case 7 (occupant under nozzle)					
10	27	23	14	16(low)					toward
11	27	22.5	15	Same as case 9					toward
12 (slot)	21	23	15.8	58 (WS2)		58(WS5)			45 ° from horizontal
13(slot)	21	23	15.8	58(WS2)		58(WS5)			downward
14(slot)	21	23	18	40		40			45 ° from horizontal
15(slot)	21	23	15.8	40		40			45 ° from horizontal

‖ Supply temperature is the temperature of the supply air from the Air Handling Unit, but usually the temperature of air delivered from the diffuser is higher due to the heat exchange through the air duct, also see Table 1 for air supply T

Table 4 PMV and ACE: summer and spring

WS No.	WS1	WS2	WS3	WS4	WS5
Summer condition Air Change Effectiveness = 1.03☆					
PMV(summer)	-0.275	-0.19	-0.13	-0.137	-0.142
Spring condition Air Change Effectiveness = 1.02					
PMV(spring)	-0.03	-0.12	0.05	0.00	0.04

☆ To compute ACE, mean age of air throughout the room is obtained based on the average value age of air at 120 sampling points, while mean age of air at breathing zone is the average value at the height of 1.1m.

Table 5 PMV values near occupant 2 (effects of supply volume and temperature)

Case No.	3	4	5	6	7
PMV	0.115	-0.23	0.316	0.065	-0.046

Table 6 PMV values near occupant 2 (effects of diffuser location)

Case No.	Case 5	Case 9a	Case 6	Case 9b	Case 7	Case 9c
PMV	0.316	0.019	0.065	-0.51	-0.0457	-0.75
ACE	1.01	1.08	1.01	1.11	1.01	1.125

Table 7 PMV values near occupant 2, 3 and 5 (effects of air throw angle)

		Occupant 2	Under noz 2	Occupant 3	Under noz 3	Occupant 5	Under noz 5
PMV	Case 1 (downward)	-0.12	-0.79	0.05	-0.5	0.04	-0.3
	Case2 (toward)	-0.73	-0.31	-0.12	-0.38	-0.117	-0.2