A COMPARATIVE STUDY OF UNDERFLOOR AIR DISTRIBUTION SYSTEM AND CEILING VENTILATION SYSTEM: CFD SIMULATIONS

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ABSTRACT

The purpose of this study was to fully understand the influence of the underfloor air distribution (UFAD) and the conventional ceiling air distribution systems on the indoor environment in a real open-plan office using Computational Fluid Dynamics (CFD) methods. Based on that, the near optimal ventilation system arrangement and operating state can be estimated, so as to maintain acceptable comfort level and air quality in the office without compromising the energy efficiency of the ventilation systems. The open-plan office investigated was located in a government office building in Ottawa, which is currently using nozzle diffusers integrated with personal environmental control (PEC) capability. The performance of nozzle diffusers integrated with PEC was previously investigated and compared with in-situ measurements. Another CFD model is built by substituting the ceiling system with UFAD. In this study, the simulation results give detailed description of the parameters related to thermal comfort of occupants, ventilation efficiency and energy consumption. For both UFAD and ceiling system, the range of system configuration and parameters investigated includes: (1) supply airflow rate; (2) supply air temperature; (3) location of diffusers with respect to occupant; (4) heat load density. Indoor air temperature, air velocity, mean age of air, Predicted Mean Vote (PMV), Air Change Effectiveness (ACE), and Air Diffusion Performance Index (ADPI) were estimated to investigate the performance of the ventilation system during occupancy.

The main conclusions drawn from this study are as follows, (1) the room temperature stratification produced by the UFAD indicate higher contaminants removal efficiency and higher energy efficiency; (2) both overhead and UFAD systems with the current supply air conditions provide acceptable ACE and comfort level within the workstation; while the ADPI value for the underfloor system is significantly lower than that for the overhead system; (3) compared to the overhead system, the optimum supply temperature for the UFAD system in the present office can be set to a higher value, which in turn can improve the coefficient of performance (COP) of the chiller and thus may result in higher energy efficiency.

INTRODUCTION

Occupants' health, comfort, productivity and increased concerns about the environmental and energy related issues have been the main objectives of the Federal Government building providers, Public Works and Government Services of Canada. Indoor air quality, thermal comfort and building energy consumption are strongly influenced by the ventilation air distribution systems. In the light of the notion behind PEC (Brager and de Dear 2001; Arens et al. 1998), the eighth floor of a PWGSC building is an open plan office that incorporates nozzle diffusers with PEC. That is, the office worker in each workstation is capable of controlling his/her own immediate workspace environment by adjusting individual air supply volume and air delivery angle. Previous work (Liang et al., 2005), by performing CFD simulations based on a validated model, investigated the advantages of such a PEC-equipped ventilation system in terms of enhancement on the local thermal comfort and potential for energy saving.

UFAD system with floor mounted and or partition/desk mounted supply diffuser also provides possibility of PEC, in terms of supply direction, volume, and/or temperature. UFAD is claimed to provide substantial energy saving while improve air quality, thermal comfort and productivity, and thus is gaining considerable attention as a suitable replacement for the conventional delivery methods (Bauman and Webster, 2001; Bauman, 2003; Shute, 1995). Extensive studies on UFAD systems have been performed by taking measurements in either controlled environment chambers or in actual offices (Chao and Wan, 2004; Melikov et al., 2002; Sekhar et al., 2002; Fukao et al., 2001; Bauman et al., 1995). In order to obtain complete information, such experimental works involve extremely high cost and large amount of labor effort.

CFD techniques can provide detailed description about the airflow, temperature distribution, and age of air dispersion, and have been adopted universally in indoor climate analysis. The current study was based on a quasi-validated CFD model (Liang et al.,

2005) using a commercial CFD code (Airpak from Fluent Inc., 2002), in order to investigate the influences of both the mixing system (with ceiling mounted nozzles) and UFAD (with floor mounted grilles) on ventilation performance, thermal comfort, and potential energy costs. The effects of air supply temperature, air supply volume, location of diffusers, and internal heat load density have been considered.

In this study, the indices used to quantify and compare the influence of air terminal device and air supply parameters include ACE, PMV, APDI, air temperature, velocity, and mean age of air. The ACE is the ratio of nominal ventilation time constant and the average age of air at a particular location.. ASHRAE (1997), defines the ACE based on ages of air as: ACE = t_n / t_{avg}. Where t_n is the nominal ventilation time which is equal to the overall age of air in the room and t_{avg} is the average age of air at the breathing level. PMV represents 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 current CFD code 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. ADPI is an index that integrates the air velocity and effective draft temperature. The ADPI of a region is defined as the percentage of locations in a given space that meets the ASHRAE Standard. ADPI can be extracted directly from the computational results.

SIMULATIONS

A five-workstation area located in PWGSC office 8B1, with the dimension of L x W x H = 9.6m × 6.2m × 2.5m, has been used for case studies and is shown in Figure 1. 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.

Nozzle diffusers serve as air terminals for the ventilation system in use, as shown in Figure 2a. the temperature of air delivered is fixed by the central air handling unit within the range from 13°C to 16°C; while the supply volume from each nozzle can be adjusted from 16L/s to 51L/s. In this study, alternatively, grille type floor mounted diffusers were also integrated into the model so as to facilitate the comparison between the performance of ceiling system and UFAD system. The floor diffusers were pressure independent, constant velocity and variable airflow rate, and they could be orientated by the occupant in any direction, thus providing PEC capability, as shown in Figure 2b. The supply flow rate and supply temperature from such a floor diffuser may vary from 0.0L/s to 71.0L/s and 15.6°C to 18.3°C, respectively. Also, there are re-circulation air induction units along the exterior walls and windows, and air returns through the vents at the ceiling level, which are located in workstations near the windows, as shown in Figure 1.

After tentative calculations, in term of comparing the CPU time and relative accuracy with different turbulence models, CFD simulations were performed by applying the indoor zero-equation turbulence model (Chen and Xu, 1998) to the current problem. Other modeling methods and calculation conditions are summarized in Table 1.

In previous work, the predicted temperatures at measured points were compared with the readings of wireless sensors, which presented good agreement (Liang et al., 2005).

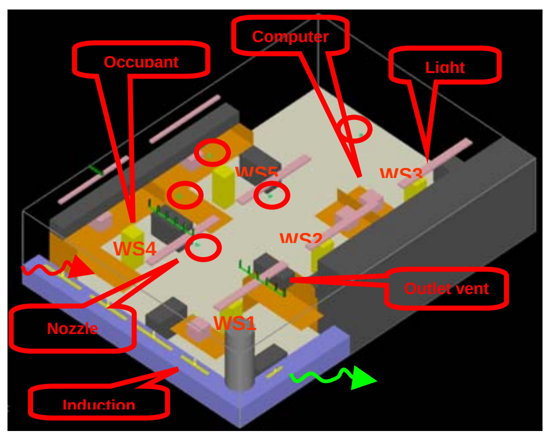

Figure 1 Layout of the office

DISCUSSION AND RESULTS ANALYSIS

The following cases are designed based on the previously validated model in order to simulate and analyze the effects of air distribution system types and air delivery conditions on the office environment and on occupants' thermal sensations. Table 2 lists all the cases studied for the five-workstation layout with overhead ceiling mounted nozzles and Table 3 lists all the cases for the five-workstation with grill type floor diffusers. In these two tables, the supply air temperature and volume flow rate were estimated (within the above mentioned controllable range) based on the given room air temperature and inner surface temperatures at exterior walls. The range of system configuration and parameters investigated include: (1) supply airflow rate; (2) air supply temperature; (3) location of diffusers with respect to occupant; (4) heat load density. Case 1, which is under spring operation condition served as the base case for simulations.

In addition to the aforementioned indices for comfort and ventilation effectiveness evaluation, the room air velocity and temperature (near the occupants) at six heights of 0.1m, 0.6m, 1.1m, 1.7m, 2.0m and 2.4m were also recorded. The height of 0.1 m, 0.6m and 1.1m were recommended 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 levels, 2.0m and 2.4m, were used to evaluate the behavior of airflow close to the overhead air distribution terminal.

Representative cases were chosen from the cases listed in Table 2 and Table 3 to be discussed next. The results presented would focus on the thermal conditions near the occupants and the overall performance of the ventilation system. The parameters recorded for analyzing thermal condition near the occupant was taken as the average of four points around the occupant at each required height.

Optimal Supply Air Conditions

Air supply temperature and volume flow rate have critical impacts on the energy consumption of the supply fan and cooling plant, as well as on the subjects' thermal sensation. Cases 3-7 in Table 2 were design to search for the near optimal supply air conditions within the controllable range for nozzle diffusers in the current office, while cases 12-18 were performed for the same purpose for floor diffusers. The five different combinations of supply airflow rate and temperatures for the nozzles are: case 3, 16L/s and 13°C; case 4, 35L/s and 13°C; case 5, 16L/s and 15.8°C; case 6, 35L/s and 15.8°C; and case 7, 51L/s and 15.8°C. The seven different combinations of supply airflow rate and temperatures through the five floor diffusers are: case 12, 71L/s and 18.3°C; case 13, 50L/s and 18.3°C; case 14, 50L/s and 17°C; case 15, 35L/s and 18.3°C; case 16, 35L/s and 17°C; case 17, 35L/s and 15.6°C; and case 18, 16L/s and 15.6°C. All the boundary conditions in these two sets of cases are identical. Table 4a presents the average PMV values at the height of 1.1m near occupant in workstation 2, ACE and APDI values obtained from cases 3-7, while Table 4b gives the same information from cases 12-18. Observations are as follows:

1. With the current heat load density (roughly 45W/m^2) in the workstations, the near optimal delivery airflow rate and temperature for ceiling nozzle diffusers can be concluded at 35L/s and 15.8°C, respectively, by taking the resulting comfort level, supply fan power input, and energy consumption of cooling plant as evaluation criteria. These two values for floor diffusers would be 51L/s and 18.3°C, or alternatively, 35L/s and 17°C.

   Compared to the overhead nozzle, the near optimum supply temperature for the floor diffuser in the current office environment is higher, (e.g. 17°C verses 15.8°C); while the magnitude of the optimum supply airflow rate for both system is about the same (35L/s×5 for both systems). The high supply air temperature can also improve the COP of the chiller and thus may contribute to the energy efficiency. However, such information as the precisely prediction of the supply module performance and quantitatively correlation between the supply air temperature and the energy efficiency of the cooling plant are critical to quantify the energy saving potential of the underfloor system.

2. The comparison of the ACE results obtained from both systems under different air supply conditions evidenced that slight improvement in ACE is achieved by the presently used floor modules, when air is supplied at high and medium airflow rate through the floor diffusers. When airflow rate is minimum, ACE value with underfloor system is almost the same as that with ceiling system.

3. ADPI results with underfloor system, ranging from 45% to 60%, demonstrate dramatic drops when compared to the ADPI results with overhead system with nozzles (within the range from 75% to 81%, shown in Table 4a), and this indicates that the air diffusion performance of underfloor system is unacceptable. However, ADPI may not be the most appropriate index to evaluate the performance of floor based air distribution system, which is also indicated by previous literature. The reason is ADPI does not allow the velocities and temperatures to exceed specific limits within the occupied zone, which is too strict for the underfloor system delivering air in the vicinity to the occupant.

Effects of air supply volume and temperature

The vertical variation of average air velocity and temperature in cases 3-7 and cases 12-18 are ploted in Figure 3a and 3b. Generally speaking, the supply air temperature from floor diffusers is higher than that from the ceiling nozzles, which is constrained by the characteristics parameters of the diffusers. The following comparisons are made based on the velocity and temperature distribution patterns:

1. In both sets of cases, the distribution pattern and magnitude of air speed near the occupants are similar to each other. Except for case 7 that had the

highest supply airflow rate from the nozzles, other cases with both air distribution systems resulted in velocity no higher than 0.25m/s at ankle level, which is within the recommended comfort limit.

2. According to the temperature distribution, the ceiling nozzles mixed the room air very well, as characterized by the uniform temperature distribution up to 2m height. When the space is served by floor diffusers, room air stratification is noticeable. When the supply airflow rate is high, e.g. 71L/s or 50L/s, two zones with different temperature distribution patterns can be observed: the lower mixed zone and the upper stratified zone. The transition plane separating these two zones is located at the height about 1.7m. When the supply air flow rate is at low level, e.g. 35L/s or 16L/s, three zones can be observed: lower mixed zone, middle stratified zone, and upper mixed zone. The cool supply air remains close to the floor, resulting in vertical temperature profiles similar to those with the displacement ventilation. The transition plane separating the upper mixed zone and middle stratified zone is located at the height about 1.3m. Concerning the head-foot temperature difference, out of all these cases, only the combination of minimum supply airflow rate/minimum supply air temperature with floor diffusers (case 18) resulted in unpleasant temperature stratification (more than 3°C).

Asymmetric distributions of PMV value with floor diffusers

When the air supply condition is set to the near optimal values determined in previous section for both systems, the average PMV is at satisfactory level, as specified in Table 4a and 4b. Here, "average" indicates that all the four sides of the occupant, namely left, right, front, and back are taken into consideration. However, by examining the more detailed PMV results for a standing object obtained in case 14 (as shown in Figure 4b), one would find the highly asymmetric distribution of PMV (e.g. in front of and at the back of the occupant) was generated by floor diffusers. This indicates that even though the overall PMV value near the occupant is acceptable, the occupant may feel "slightly cold" on his side close to the diffuser, and may feel "slightly warm" on his side away from the diffuser, and such a thermal condition is obviously not acceptable. In addition, from Figure 4b, the occupant is exposed with different thermal sensation for different body part, e.g. PMV values distribute unevenly from head to ankle. In contrast, with the ceiling nozzles, even the supply air temperature was 1.2°C lower than that from the floor diffusers, evenly distributed PMV values can be observed, as demonstrated by Figure 4a, though the supply air temperature (15.8°C from nozzles) is even lower

Effects of floor diffuser location

Based on the above observations, for the present layout of workstation and the air diffuser characteristics, installation of one diffuser per each workstation may not be a good arrangement in terms of providing uniform and acceptable thermal comfort level to the occupants. Cases 19-23 were designed to examine the performance of three diffusers installed in the five-workstation zone. The arrangement was as follows: WS1 and WS4 shared a diffuser, WS2 and WS5 shared another one, and the third diffuser was located close to but still outside WS3. The resulted temperature stratification and velocity variations in all the five workstations obtained in case 20 are presented in Figures 5a and 5b. To aid the comparison of temperature profiles, temperatures are presented in a non-dimensional form: $(T_{local}-T_{supply})/(T_{exhaust}-T_{supply})$. With the present heat load density and air suppy conditions, the transition plane separating the upper mixed zone and middle stratified zone is located at the height about 1.5 m. Compared to the velocity results obtained in the cases with five floor diffusers, the air velocity in workstations produced by three floor diffusers is lower. PMV variation with height obtained from case 20 are presented in Figure 6. One could find that the PMV (in front of and at the back of the occupant) values vary slightly both in the horizontal and in the vertical direction. Consequently, when the supply floor modules are installed outside the workstations, satisfactory PMV values with symmetry can be obtained near the occupant. This is coherent with the suggestions in previous work that floor modules should be positioned away from the frequently used areas to avoid local cool thermal sensation.

CONCLUSION

1. The near optimal supply temperature from the floor diffuser studied here is higher than that from the overhead nozzles; the overall supply airflow rate for such a UFAD system is slightly higher than that for the ceiling system. The high supply air temperature can improve the COP of the chiller and thus may contribute to the energy efficiency. Further investigations on the energy efficiency of cooling plant and supply fan are necessary to completely address such an issue.

2. Unlike the ceiling nozzles that perfectly mix the room air, floor mounted grille diffusers studied here result in stratification of room air, which may be an indication of high energy saving potential and high contaminants removal efficiency. The overall supply airflow rate and supply air temperature from the floor modules

and the heat load density within the space have impacts on the degree of stratification and on the location of transition plane. When the supply airflow rate is at low level, the performance of underfloor system resembles that of the displacement system. Only when minimum supply airflow rate is combined with low supply temperature, the 3°C head-foot vertical temperature difference limit is exceeded.

3. In the present office, the floor supply modules should not be positioned within the frequently used region, not because of risk of draft discomfort, but due to the undesirable axisymmetric PMV value distribution in the adjacent of the supply modules.

4. Relatively higher ACE values are produced by UFAD system, when compared to the overhead ceiling system. While the ADPI values with underfloor system is significantly lower than those with ceiling system. However, ADPI may not be the most appropriate index to evaluate the performance of floor based air distribution system, which applies strict limitation to the velocities and temperatures within the occupied zone, and UFAD system delivery air in the vicinity to the occupant

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Figure 2a Nozzle diffuser

Figure2b Floor diffuser

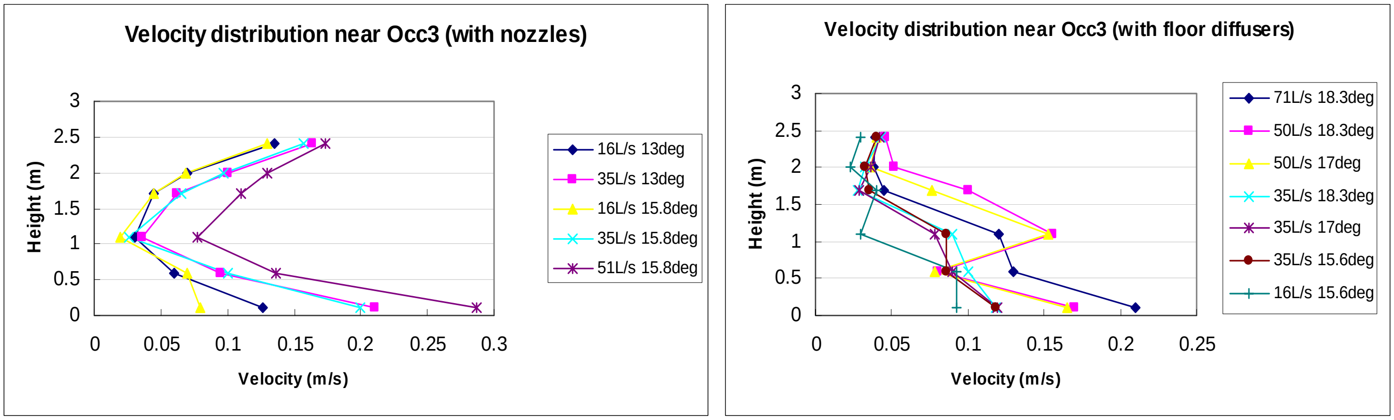

Figure3a Velocity variations near occupant 2 (effects of supply volume and temperature)

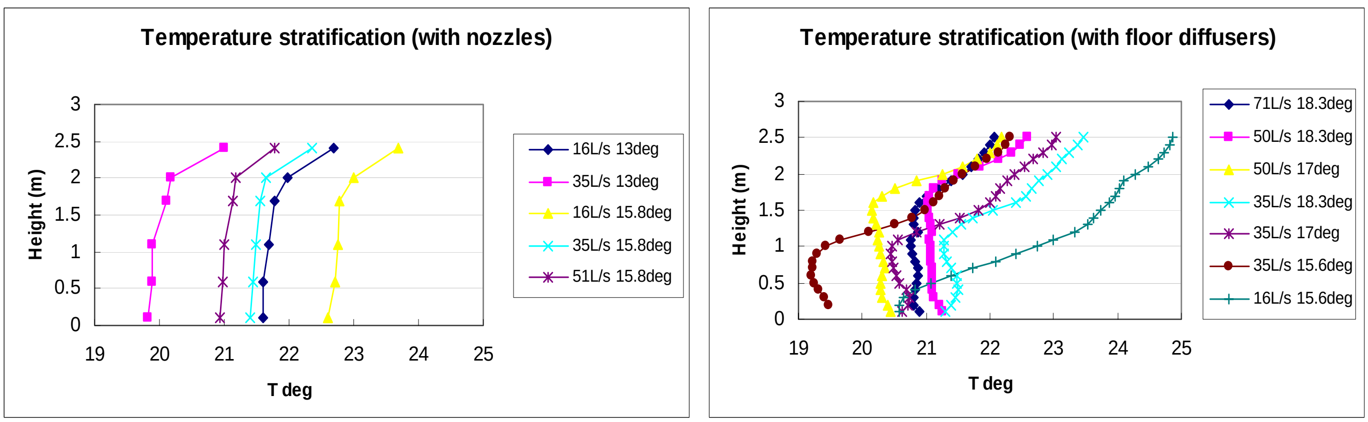

Figure3b Temperature stratifications near occupant 2 (effects of supply volume and temperature)

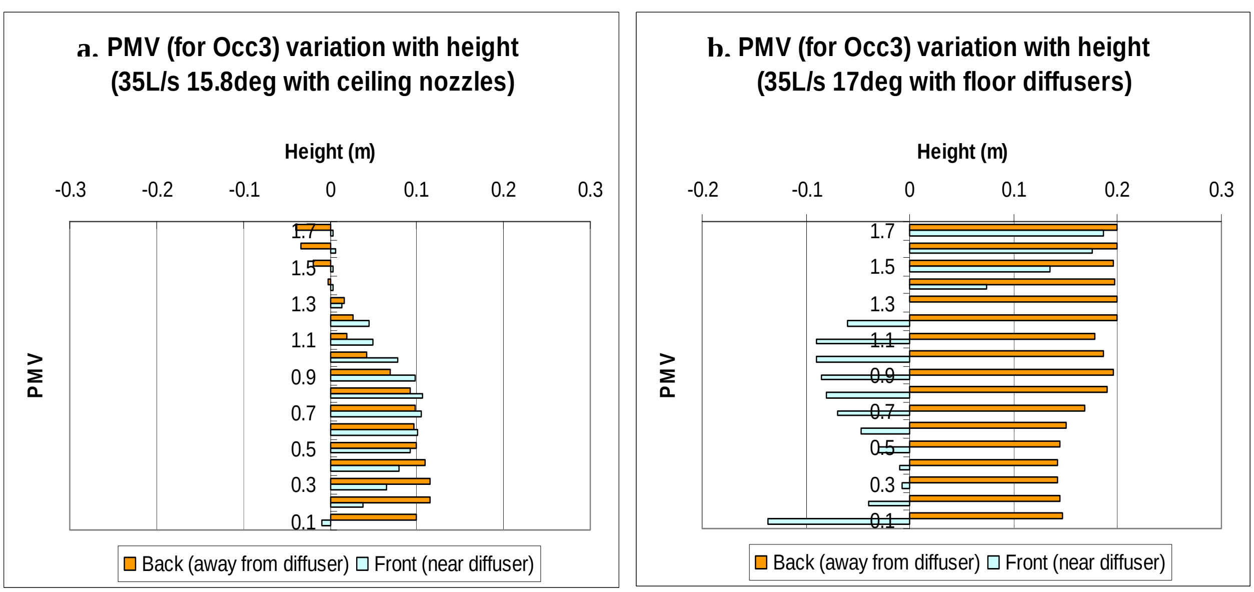

Figure 4 PMV variation with height near Occ3: a with nozzles; b with floor diffusers

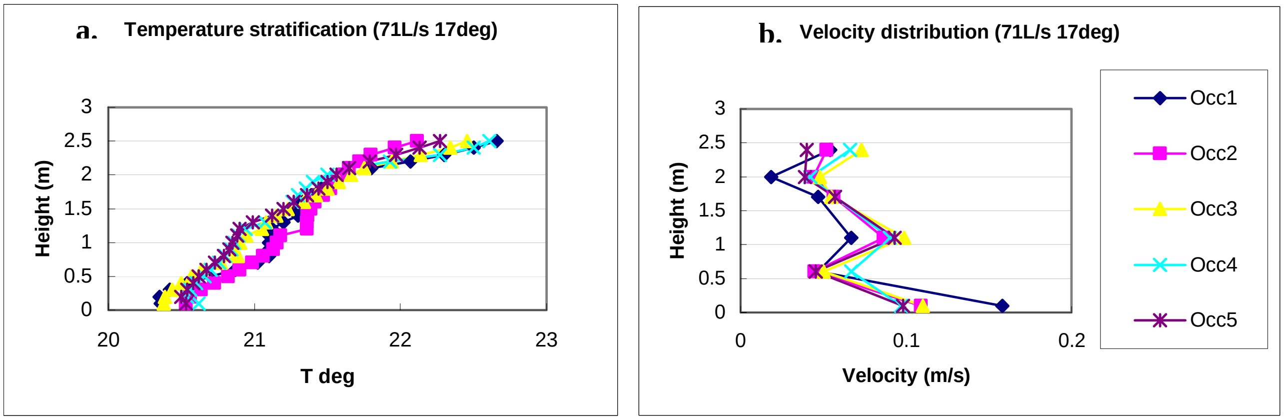

Figure 5: a temperature stratification; b velocity variation with height, near all the five occupants in case20

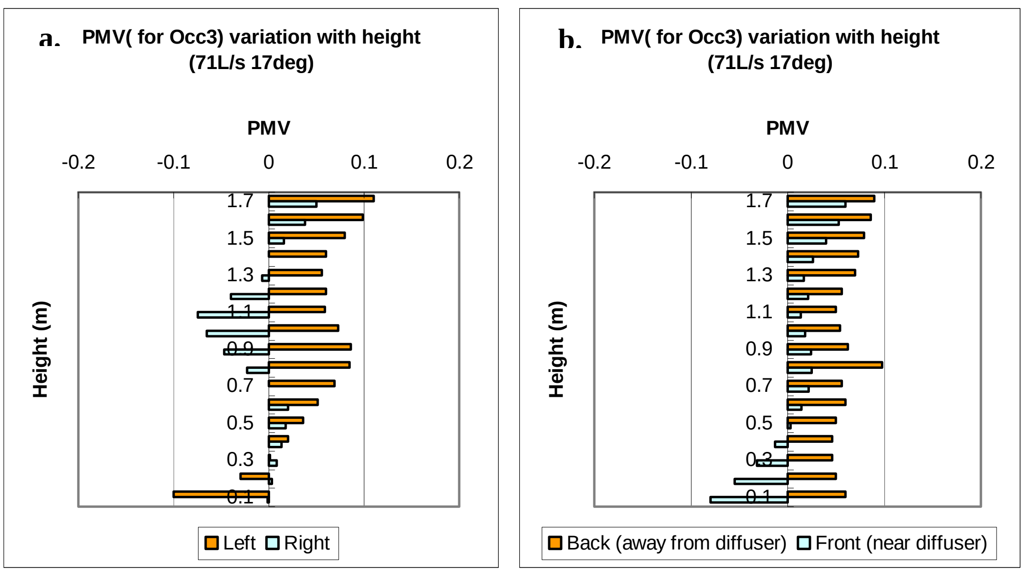

Figure 6 PMV variation with height near Occ3 when floor diffuser outside workstation: a on the left side and on the right side; b at the back of and in front of the occupant

Table 1 Summary of numerical aspects in CFD simulation

NUMERICAL ASPECTS	METHOD USED IN THE CURRENT STUDY
Turbulence model	Indoor zero-equation model
Diffuser modeling method	Momentum method
Governing equation discretization	Control volume based technique
Scheme for differencing convection term	Up-wind scheme
Grid type / meshed cell number	Hexahedral / 0.4-0.6 million
Solution algorithm	SIMPLE
Convergence criteria	Residue < 0.001 for mass, velocity, species concentration, pressure, Residue < 10-6 for energy
Under-relaxation factors	0.3 for pressure, 0.7 for velocity, 1.0 for temperature and species concentration
Multigrid scheme controller	Termination criteria at 0.1 for all variables

Table 2 Case studied for system with ceiling mounted nozzle diffusers

CASE NO.	WINDOW INNER FACE T (°C)	TEMPERATURE (°C)		SUPPLY AIRFLOW RATE★ (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

‖ Supply temperature listed here is chosen from the recommended range from 13 to 16°C, as controlled by air handling unit.

★ Supply airflow rate is estimated by: a. the controllable range (16-51L/s); b. inner surface temperature.

Table 3 Case studied for system with grille type floor diffusers

CASE NO.	WINDOW INNER FACE T (°C)	TEMPERATURE (°C)		SUPPLY AIRFLOW RATE★ (L/S)					FLOOR DIFFUSER ALLOCATION
		Room	Supply‖	(ws1)	(ws2)	(ws3)	(ws4)	(ws5)	
12	21	22	18.3	71					1 diffuser per Ws
13	21	23	18.3	50					1 diffuser per Ws
14	21	22	17	50					1 diffuser per Ws
15	21	22	18.3	35					1 diffuser per Ws
16	21	22	17	35					1 diffuser per Ws
17	21	21	15.6	35					1 diffuser per Ws
18	21	23	15.6	16					1 diffuser per Ws
19	21	23	18.3	71		71		71	3 diffusers for 5 Ws☆
20	21	22	17	71		71		71	3 diffusers for 5 Ws☆
21	21	21	15.6	71		71		71	3 diffusers for 5 Ws☆
22	21	22	17	50		50		50	3 diffusers for 5 Ws☆
23	21	21	15.6	50		50		50	3 diffusers for 5 Ws☆

‖ Supply temperature listed here is chosen from the recommended range (15.6 to 18.3°C) by air terminal supplier.

★ Supply airflow rate is estimated by: a. the controllable range (0-71L/s); b. inner surface temperature; c. internal heat load.

☆ Workstations 1 and 4 share a diffuser, workstations 2 and 5 share another one, and the third one is outside workstation 3.

Table 4a Average PMV values, ACE, and APDI under different supply air conditions (with nozzles)

CASE NO.	3	4	5	6	7
Supply air conditions	16L/s 13°C	35L/s 13°C	16L/s 15.8°C	35L/s 15.8°C	51L/s 15.8°C
Average PMV near occ2	0.115	-0.23	0.316	0.065	-0.046
ACE	1.02	1.08	1.03	1.05	1.06
ADPI	77%	78%	75%	81%	76%

Table 4b Average PMV values, ACE, and APDI under different supply air conditions (with floor diffusers)

CASE NO.	12	13	14	15	16	17	18
Supply air conditions	71L/s 18.3°C	50L/s 18.3°C	50L/s 17°C	35L/s 18.3°C	35L/s 17°C	35L/s 15.6°C	16L/s 15.6°C
Average PMV near occ3	-0.05	0.01	-0.1	0.18	0.05	-0.15	0.42
ACE	1.15	1.155	1.16	1.2	1.21	1.206	1.09
ADPI	61%	58%	57%	53%	47%	46%	47%