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Thermal Comfort Analysis in a Room with Displacement Ventilation System Coupled with Passive Chilled Beams

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ABSTRACT

Displacement ventilation (DV) has been widely studied in literature, and is currently used in enclosed environments like office buildings. Although DV has been shown to be able to provide better indoor air quality in the occupied zone than conventional mixing ventilation (MV), its capability of removing heat is usually limited. Since it supplies air directly to occupied zone, the temperature of supply air is inevitably higher than that in MV. Meanwhile, passive chilled beams (PCB), which cool occupied space by running chilled water through beams at ceiling level, has been reported to have a high heat removal capability, and can save energy. Thus, a coupled system that combines both DV and PCB has the potential of keeping the advantages of both systems, and was explored in detailed by using experimental and simulation methods.

This study focused on investigating the thermal performance of a coupled system of DV and PCB, and compared it with a corresponding DV-only system. Experiment was carried out in a full-scale environmental chamber to measure air velocity and temperature at various locations in the chamber. Moreover, a computational fluid dynamics (CFD) model was used to simulate the air velocity and temperature distributions, which were then validated by the measured data. With the validated CFD model, thermal comfort in the occupied zone were analyzed with the coupled system and the DV-only system. Thermal comfort level was presented by the predicted mean vote (PMV), and percentage dissatisfied people due to draft (PD). The results indicated that the PCB can help to cool down the temperature in the room, and the air temperature stratification was still observed in most regions. Meanwhile, due to the cold downward jet generated by the PCB, the region under the PCB had a higher air velocity and lower temperature than that with DV-only system. This resulted in a local zone with less thermal comfort. Based on reported results, this study provided preliminary guidelines for designing of the coupled system with acceptable thermal comfort in occupied zone.

1. INTRODUCTION

Since first being applied in Scandinavian countries in the 1970s, displacement ventilation (DV) has been gaining popularity in both industrial and office buildings during the past four decades (Chen and Glicksman 1999). It was reported that in Nordic countries, half of industrial buildings and 25% of offices were equipped with displacement ventilations by 1989 (Svensson 1989). In the US, there is also an increasing interest for applying displacement ventilation in buildings, especially in computer rooms, classrooms and large offices (Burt 2007). A typical displacement ventilation system supplies clean air from lower part of the room, and exhausts air near the ceiling. Previous research showed that such a system provides a room with higher indoor air quality (Chen and Glicksman 2003), and uses less energy (Hu, Chen and Glicksman 1999), as compared to mixing ventilation. Besides, the low velocity of supply air in displacement ventilation produces a lower level of noise than other ventilation systems. However, because displacement ventilation supplies fresh air to occupied zone directly, its supply air temperature is usually higher than that of mixing ventilation. This feature limits the cooling capability of displacement ventilation.

On the other hand, passive chilled beams (PCB), which are usually installed inside false ceiling or suspended at upper part of a room, were reported to be able to remove large cooling loads and can save energy. For example, Fredriksson and Sandberg (2009) demonstrated by experiments that a cooling efficiency (percentage of heat removed by PCB in all cooling loads) of 80% could be achieved by PCB. Rumsey and Weale (2007) and TIAX LLC (2002) showed that chilled beams can yield 15% to 20% higher energy efficiency than conventional systems, due to the fact that chilled beams use higher chilled water temperature than traditional air-conditioning systems (Roth et al. 2007). Thus, it is possible to combine PCB with DV, to address the low cooling ability issue of a single DV system. Nevertheless, the downward jet produced by PCB, when installed with DV, could influence the thermal plumes built up in the original DV system. As a result, PCB may or may not destroy other benefits of DV system, depending on the percentage of cooling load handled by PCB. Hence, there needs a systematic study of the thermal and ventilation performances of such a coupled system, in order to achieve an optimal combined system design.

Meanwhile, thermal comfort has always been an important indicator for appraising indoor environment design. Studies have shown that a satisfying thermal comfort level is of great significance for maintaining occupants' health and productivity (Akimoto et al. 2010). On the contrary, a low thermal comfort level could adversely affect individuals' performances in thinking, typing and skilled manual work (Wyon 2000). Hence, in the design of a combined system of PCB and DV, it is of great value to investigate the thermal comfort in a room with such a system. Particularly, the comparison of thermal comfort levels at different locations of the room will provide very useful information for design engineers to establish detailed configuration of the room, such as where to place PCB, desks, etc.

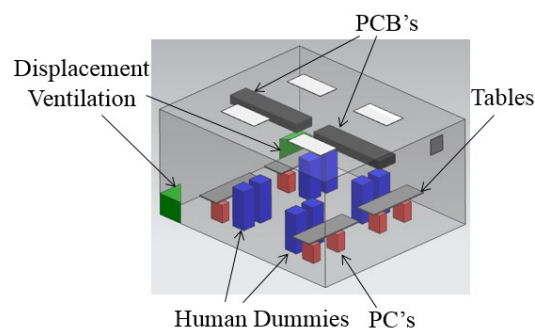
In this study, airflow velocities and temperatures were measured in an environmental chamber which is equipped with both DV and PCB. Moreover, a CFD model was further developed to simulate the airflow under such a coupled system, and was validated by measurement data. The validated CFD model was further used to evaluate the thermal comfort levels at various locations of the room, which were compared with those in a room with only DV system. Indices predicted mean vote (PMV) and percentage dissatisfied people due to draft (PD) were adopted to quantify thermal comfort levels. Finally, preliminary recommendations on designing such a coupled system were proposed based on thermal comfort analysis results.

2. RESEARCH METHODS

This section describes how experiments and CFD simulations were performed to obtain airflow velocity and temperature information in an environmental chamber. Besides, details of indices, PMV and PD, are provided to illustrate how these two parameters are used to quantify thermal comfort.

2.1 Experiments

2.1.1 Room Layout



(a) Schematic of items inside chamber



(b) Realistic picture of environmental chamber

Figure 1: Environmental chamber layout

Measurements were conducted in an environmental chamber which has a dimension of $6.08m$ (length) \times $5.15m$ (width) \times $3.05m$ (height) (Figure 1). In the chamber, fresh air was supplied from diffusers located at floor

level, and exhausted near the ceiling. 4 tables, 8 human dummies, and 8 simulated personal computers (PC's) were placed in the room, to simulate an office layout. Inside each human dummy and simulated PC, light bulbs were installed to generate heat, and mini-fans were used to stir inside air so the heated air could circulate inside these boxes. On the ceiling of the chamber, there were four lights.

A set of two PCB's (Figure 2) were placed on the center plane of the room, near the ceiling. Inside each piece of PCB, chilled water feeds into cooling coils, which are surrounded by metal fins. The fins were designed to enhance the heat transfer coefficient between air and PCB. Figure 3 shows the structure of the plumbing which supplies chilled water to the water pipes inside PCB's, and recycles returned water.

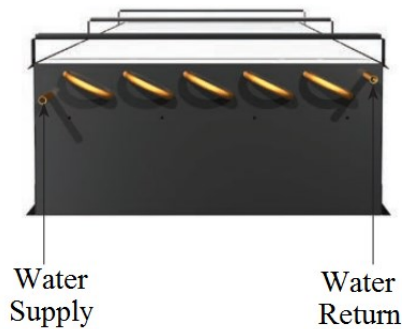


Figure 2: A piece of PCB (Price Industries 2010) used in this study

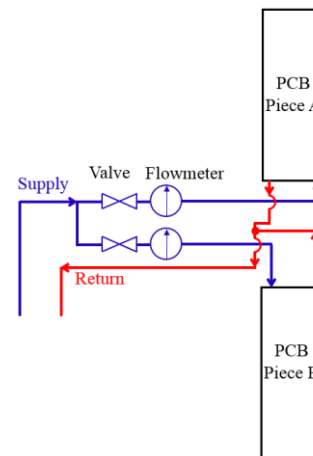


Figure 3: Plumbing structure of PCB's

Table 1 lists the dimensions of the displacement ventilation diffusers, exhaust, human dummies, simulated PC's and PCB's. The total cooling load in this room was 1930W, with breakdowns as shown in Table 2.

Table 1: Dimensions of items placed in environmental chamber

	Number	Dimensions
Diffuser	2	1.08m × 0.61m
Exhaust	1	0.44m × 0.44m
Human dummy	8	0.41m × 0.41m × 1.13m
PC	8	0.41m × 0.25m × 0.51m
PCB	8	2.42m × 0.58m × 0.25m
Light	4	1.22m × 0.61m
Table	4	1.83m × 0.61m × 0.03m

Table 2: Cooling loads in environmental chamber

	Cooling Load
Light	96 W each
Human dummy	~84 W each
PC	~109 W each
Total heat	1930 W
Heat per unit area	61W/m ²

2.1.2 Measurements of airflow velocity and temperature in two cases

This experiment used hot-sphere anemometer probes to measure air temperatures and airflow velocities. Figure 4 shows one anemometer probe, which has a temperature measurement accuracy of 0.3 °C and velocity measurement accuracy of 0.01 m/s. Measurements were conducted at 5 pole locations that were evenly distributed in the chamber, with 7 heights on each pole. Figure 5 is a plane view of the chamber, which illustrates where the poles were placed.

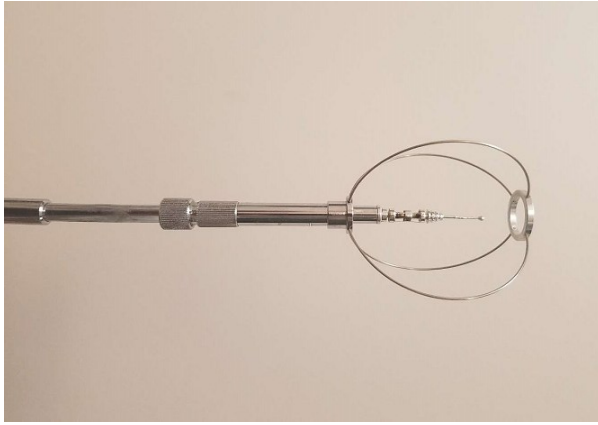


Figure 4: Hot-sphere anemometers used in this experiment

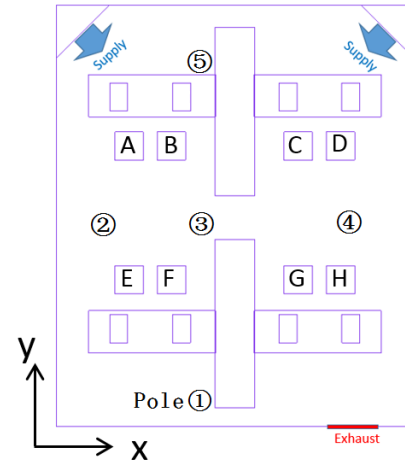


Figure 5: Measurement locations (top view)

This study also used an infrared thermometer to measure surface temperatures, which were documented to be used for boundary conditions in CFD simulations. The repeatability of reading from this infrared thermometer is $\pm 0.02\%$ of absolute temperature.

Measurements were performed in two cases: (1) “DV-only” case, where only displacement ventilation was used to remove cooling load in the chamber, and (2) “DV+PCB” case, where both DV and PCB were used, and PCB removed 60% of the cooling load in chamber.

2.2 CFD Modeling

This study used numerical methods to solve the following equations that governed the airflow inside the chamber:

$$\text{Continuity equation: } \frac{\partial \rho}{\partial t} + \frac{\partial \rho}{\partial x_i} (\rho u_i) = 0 \quad (1)$$

$$\text{Momentum equation: } \frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_i} \quad (2)$$

$$\text{Energy equation: } \frac{\partial}{\partial t} (\rho h_{tot}) + \frac{\partial}{\partial x_j} (\rho h_{tot} u_j) = \frac{\partial P}{\partial t} + \frac{\partial}{\partial x_i} \left(u_i \tau_{ij} + \lambda \frac{\partial T}{\partial x_j} \right) \quad (3)$$

where ρ , t , u_i , P , h_{tot} , τ_{ij} and T represents density, time, velocity at direction i , pressure, energy, stress tensor and temperature, respectively. RNG $k-\varepsilon$ model has been proved to be able to model turbulence effects quite well for indoor airflow problems in a lot of previous literatures, such as Chen (1995). Therefore, the current research adopted this turbulence model to close Reynolds-Average Navier-Stokes (RANS) equations that needed to be solved.

Proper specifications of boundary conditions are important for CFD model to accurately capture the physics of the simulation problem. In this investigation, by using measured surface temperature values, temperature boundary conditions were prescribed for the walls and heated surfaces in the model. When temperature differences exist between two surfaces, the colder surface is heated up by the warmer one through radiation (Lau and Chen 2007). These surfaces also exchange heat with ambient air through convection. Thus, using temperature boundary conditions takes both convection and radiation effects into account. For simplifying the problem and saving computational time, each piece of PCB was modeled as a box. In the simulation case where PCB was used, velocity boundary condition was prescribed on PCB bottom. The value of the velocity was adjusted so that the downward jet velocity profile matched with measurements. To satisfy the conservation of mass, velocity outlet boundary condition was prescribed for PCB top surface, with the same velocity magnitude as defined on bottom.

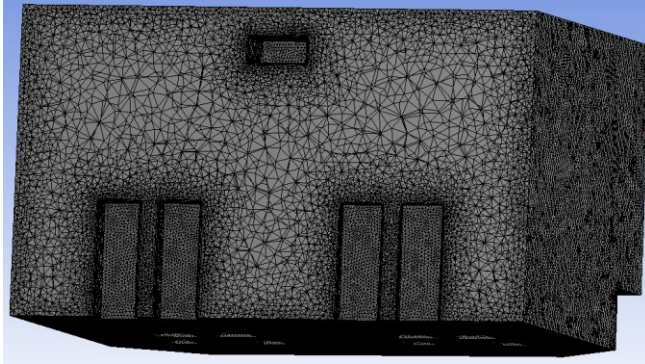


Figure 6: A cross-section view of global mesh

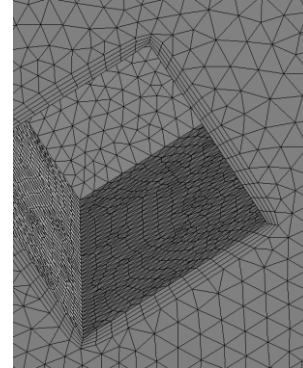


Figure 7: Mesh structure around heated surfaces

A combination of structured and unstructured meshes was used to discretize the computational domain. Figure 6 demonstrates the mesh structures used in this study. 4 boundary layers (with a growth ratio 1.1) of structured meshes were employed to capture the relatively larger temperature gradients around these surfaces, while unstructured meshes were used in the interior of the domain to handle more complicated geometries. The total number of meshes used in this study was 2.8 million. This study employed commercial solver ANSYS Fluent 15.0 (ANSYS Inc. 2013) to perform numerical simulations.

2.3 Indices for Evaluating Thermal Comfort

In order to quantify the thermal comfort levels, this study employed two indices, PD and PMV, to evaluate and compare the thermal comfort at different cases and different locations.

PD defines dissatisfaction of thermal comfort due to unwanted air movement (ASHRAE 2013). It takes into account of air temperature, velocity and turbulence intensity, and is expressed as:

$$PD = (34 - T)(u - 0.05)^{0.62} (3.14 + 0.37uT_u), \quad (4)$$

where u is mean velocity, T is temperature and T_u is the local turbulence intensity. The unit of PD is %.

PMV evaluates a 7 point scale of thermal comfort as the energy balance of body (Fanger 1970). It is calculated as:

$$PMV = [0.303e^{-0.036M} + 0.028]L, \quad (5)$$

where M is metabolic rate, and L is the thermal load on body, which can be calculated as:

$$L = M - W - [(C_{sk} + R_{sk} + E_{sk}) + (C_{res} + E_{res})]$$

In other words, L represents the difference between internal heat production and heat loss to the actual environment. The calculated PMV values can be interpreted to thermal sensations using Table 3.

Table 3: PMV values and thermal sensations (Fanger 1970)

PMV	Thermal Sensation
+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
-1	Slightly Cool
-2	Cool
-3	Cold

With validated CFD models and the equations above, both PD and PMV values can be evaluated in the computational domain.

3. RESULTS

3.1 Validation of CFD model

The velocity and temperature simulations using the CFD were validated by the experimental data. Figures 8 and 9 compare the velocity and temperature, in both “DV-only” and “DV+PCB” cases. Due to the limit of space in this manuscript, comparative results on three poles were presented, although measurements were conducted at five poles in each case.

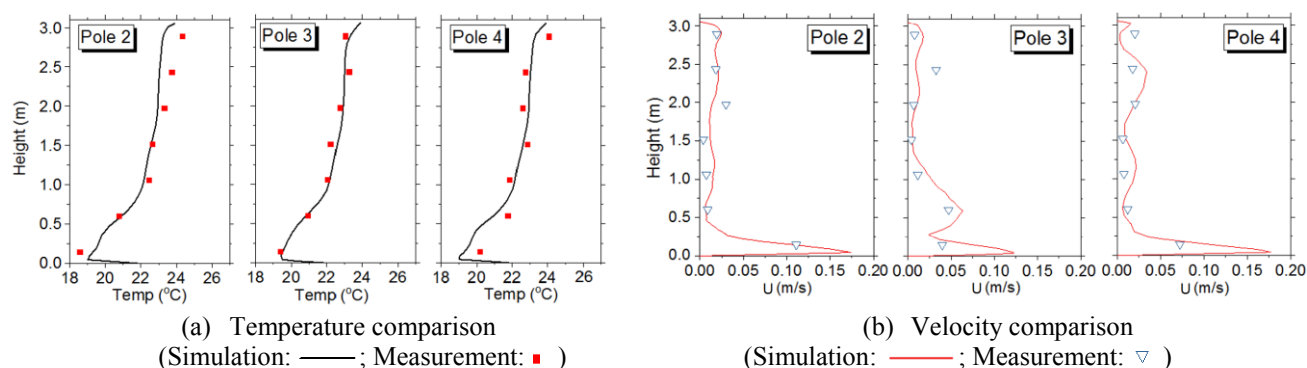


Figure 8: Measurement and simulation results in “DV-only” case

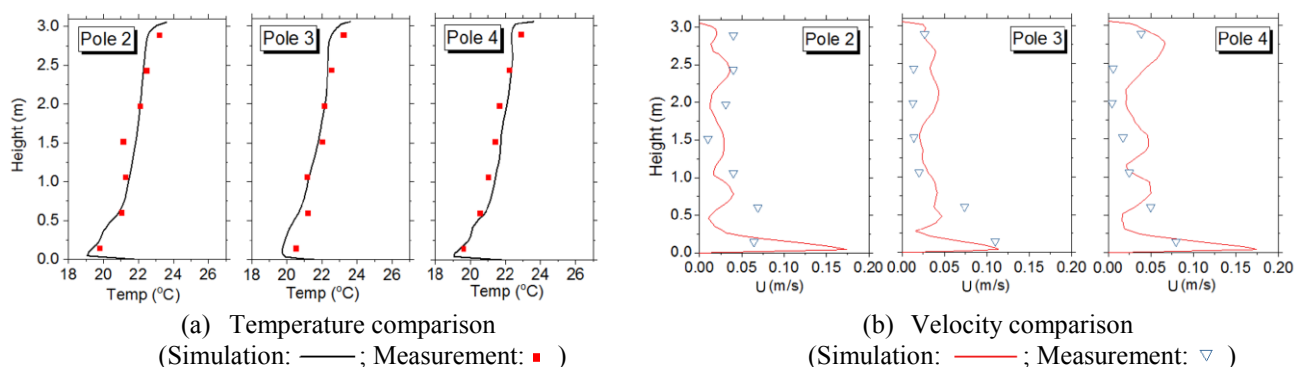


Figure 9: Measurement and simulation results in “DV+PCB” case

It can be observed from Figures 8 and 9 that in both “DV-only” and “DV+PCB” cases, the simulated temperature and velocity matched quite well with experimental data. The simulated results on poles 1 and 5, which are not shown above, aligned well with measurements too.

Both the experimental results and simulation results suggest that temperature gradient could be observed in the measured locations, although PCB seemed to have lowered the temperature gradient in “DV+PCB” case, which could be told by comparing the temperature slopes in the two cases. As for air velocity, Figure 8 (b) illustrates a typical velocity distribution in displacement ventilation, in which air velocity magnitude is largest near floor but becomes lower when height increases. While PCB was used, similar trend of air velocity distribution still existed, but the overall velocity magnitude became larger as compared to “DV-only” case, which can be seen from Figure 9 (b). This could be because the downward jet that was discharged from PCB increased the intensity of air movement in occupied zone.

3.2 Temperature Distributions in “DV-only” Case and “DV+PCB” Case

Using the validated CFD model, this research also compared the temperature distributions between “DV-only” case and “DV+PCB” case, to study how PCB influences the air temperature in the chamber.

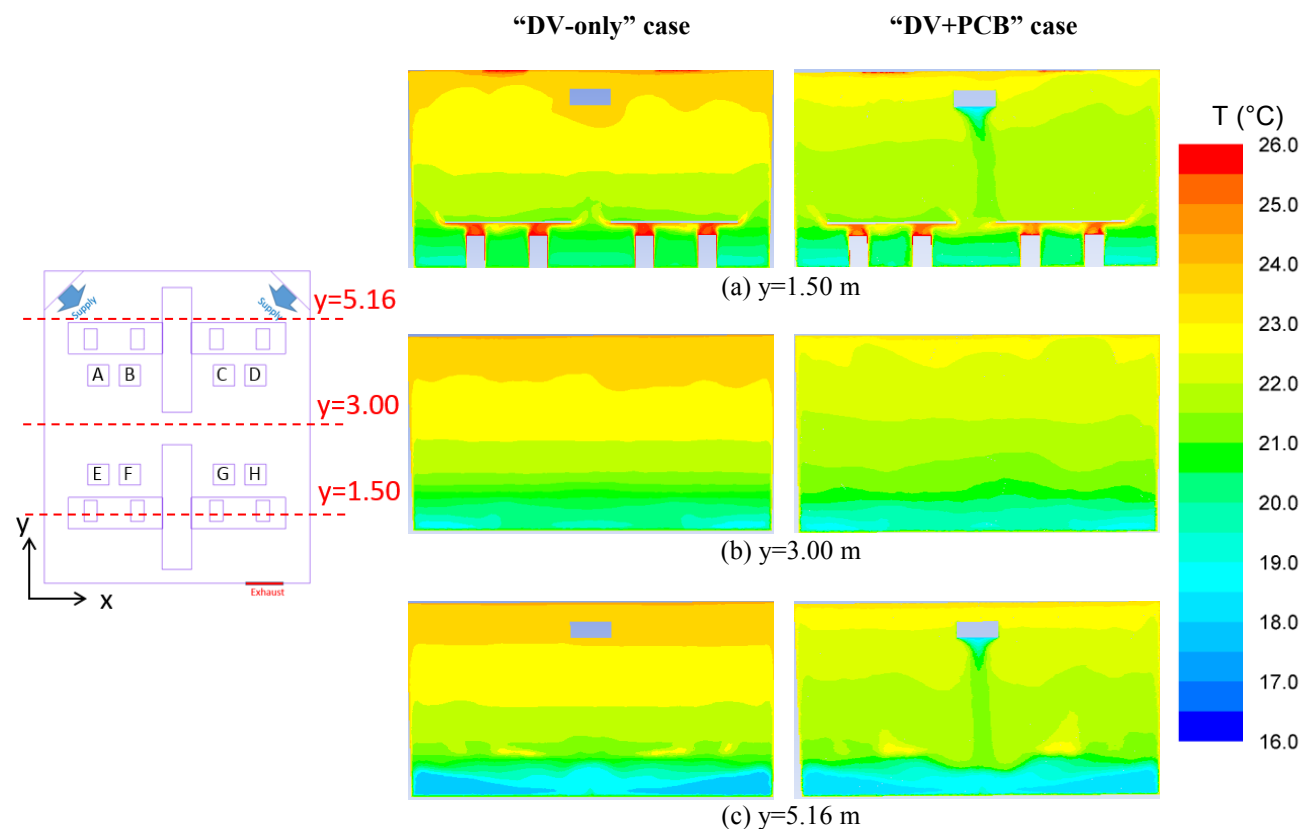


Figure 10: Temperature contours in three cross-sectional planes from CFD models

Figure 10 plots temperature contours on three cross-sectional planes in the validated CFD models. Two of the planes ($y=1.50$ m and $y=5.16$ m) cut PCB directly, while the third plane is between the two PCB's. The results suggest that when PCB was used, temperature gradient still existed in most of the locations in the room, which can be easily noticed from the contour plots in “DV+PCB” case. On the other hand, PCB created a downward cool jet under its bottom surface, which cooled down the ambient air while the jet traveled towards the floor.

It can also be noticed that the downward jet from PCB bottom did not always point to the floor in a perfectly vertical direction. For example, the downward cold jet on plane $y=1.50$ m bends toward one side while it is discharged from PCB bottom surface. This could be due to the horizontal airflow that drives downward jet to bend.

The validated CFD models are able to provide a comprehensive picture of how the temperature distributed in the room, and particularly in the “DV+PCB” case, how the downward cold jet interplayed with ambient airflow.

3.3 Thermal Comfort Analysis

With the validated CFD models and the thermal comfort indices introduced in section 2.3, this research further compared the thermal comfort levels between “DV-only” case and “DV+PCB” case. Within each case, CFD model was also used to provide detailed information as to how thermal comfort distribute from location to location.

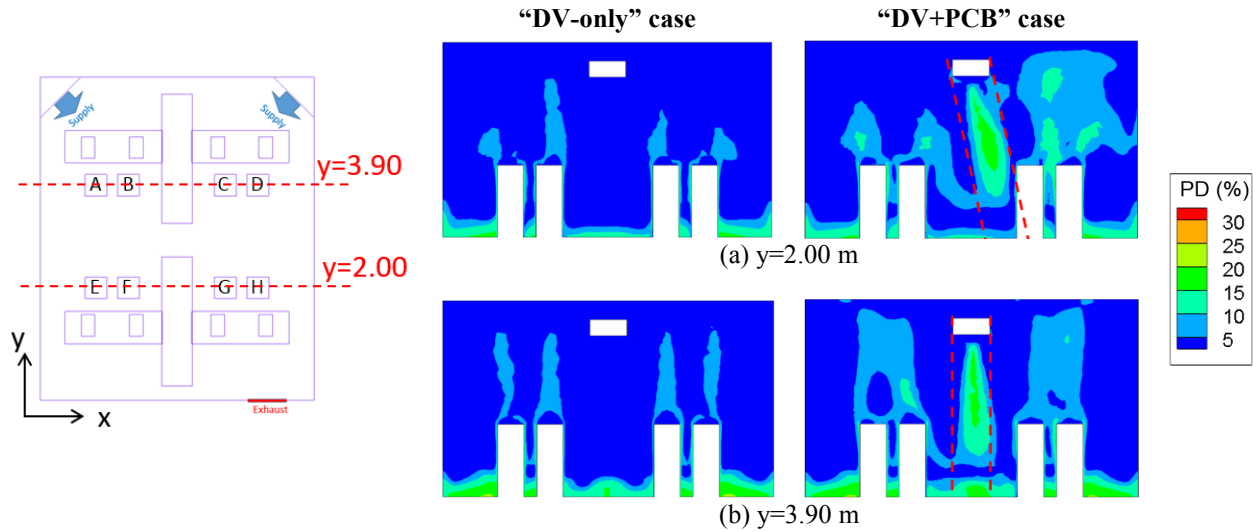


Figure 11: Thermal comfort evaluated by PD in two cases

Figure 11 demonstrates the PD distribution on 2 cross-sectional planes where human dummies were located: $y=2.00$ m and $y=3.90$ m. In "DV-only" case, PD was very low (<5%) in most of the regions, except for areas close to floor, where airflow velocity was relatively larger. The low PD value in occupied zone could be explained by its suitable air temperature, and low air velocity (<0.07m/s) in most part of the domain. In contrast, in "DV+PCB" case, a band of "high PD region" (PD>10%) was found under PCB. The formation of "high PD region" is well explained by the relatively lower temperature and higher air velocity of downward jet generated from PCB. Since the downward jet bended in some locations, the "high PD region" could deviate from PCB centerline too, as can be observed in the plane $y=2.00$ m. However, it looks like the width of "high PD region" kept more or less the same even if it bended. Meanwhile, the results suggest that in regions other than those under PCB, the overall PD was below 5%.

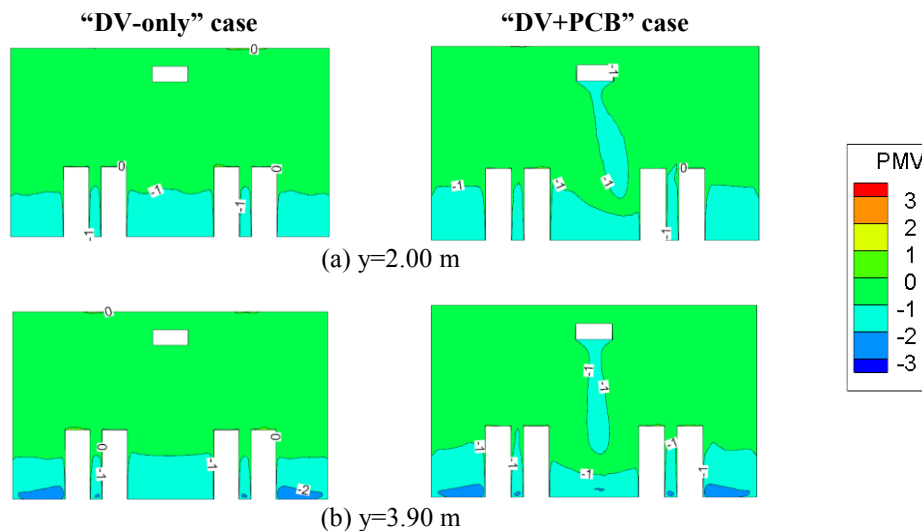


Figure 12: Thermal comfort evaluated by PMV in two cases

Figure 12 demonstrates the PMV distributions in "DV-only" and "DV+PCB" cases, on the same two planes studied in Figure 11. In "DV-only" case, the occupied zone has a PMV value of around 0, which means the occupants in the room would perceive the thermal environment as neutral. By contrast, in "DV+PCB" case, although the PMV level in most part of the room was still around 0, a band of "low PMV region" was found under PCB in both cross-

sectional planes. Thus, from the perspective of thermal comfort, when the overall thermal environment in the room is in a neutral level, occupants who sit or stand under PCB are more likely to feel cool than in other regions of the room.

4. DISCUSSIONS

This research yields preliminary design directions for a room that is equipped with a combined system of DV and PCB. It is clear from thermal comfort analysis that when PCB is in operation, the region below PCB has a low thermal comfort level, as evaluated by PD and PMV calculations. The thermally less comfortable region is also likely to deviate slightly from PCB centerline, if the ambient horizontal airflow is strong enough. Therefore, in order to create optimal thermal comfort levels for occupants, it is suggested to keep seats at some distance away from PCB. Moreover, since the thermally less comfortable region is likely to hit on a table and propagate to the surroundings, it is further recommended to also keep any tables away from areas where a PCB is installed right above.

In this study, thermal comfort analysis was performed by employing two indices, PD and PMV. Although these two indices are well recognized in the research community for evaluating thermal comfort level, they are empirical models which might not apply perfectly well to individuals of every age, gender, race, etc. Hence, it would be valuable to have real occupants in a thermal environment with such a combined system, and to get feedbacks from them, in the evaluation of thermal comfort. These investigations will be included in our future study.

5. CONCLUSIONS

The current investigation leads to following conclusions:

- (1) A CFD model was used to simulate the air flows in a “DV-only” case and a “DV+PCB” case in this study. The simulated air velocity and temperature results matched very well with measurement data.
- (2) When PCB was in operation, a downward cool jet was generated under PCB. The jet was also liked to bend away from PCB centerline, provided that the ambient horizontal air flow was strong enough. However, in regions other than the ones under PCB, temperature gradient was still observed.
- (3) The downward cool jet created a thermally less comfortable zone under PCB, as indicated by the increased PD and less satisfying PMV values. In an indoor environment which had an overall PD level below 5%, the PD value under PCB was found to be as high as 20% in the investigated “DV+PCB” case. Meanwhile, the PMV value under PCB was found to have been brought down by a magnitude of 1 by PCB, when the overall thermal sensation in the room was neutral.

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