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RESEARCH ARTICLE



The role of design parameters on the performance of diffuse ceiling ventilation systems – thermal comfort analyses for indoor environment

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ABSTRACT

Thermal comfort conditions profoundly affect the occupants' health and productivity. A diffuse ceiling ventilation system is an air distribution system in which the air is supplied to the occupied zone with relatively a low velocity through the perforated panels installed in the ceiling. The current study evaluated the impact of diffuse ceiling design parameters, i.e. diffuse panel configurations and heat load distributions, on the thermal comfort condition of the occupants. In this regard, the computational fluid dynamics technique was used to evaluate thermal comfort conditions in a waiting room, meeting room and office. The central and dispersal configuration of active diffuse panels was considered. The PMV-PPD model was applied to evaluate the overall occupants' comfort, while the draft rate was considered to assess local thermal comfort. The model validation was performed by comparing the collected laboratory measurement data. Overall, the results indicated that the central active diffuse panel configuration had a better thermal comfort than the dispersed one. The evaluation of dispersed configuration in realist scenarios, including office and waiting room, had the highest dissatisfaction, with a PPD value of 9%. Local thermal comfort assessment revealed that dispersed configuration had the highest draft rate of 14% in the office.

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1. Introduction

Buildings are expected to provide satisfying indoor environments for the occupants. Thermal comfort is an important design criterion for indoor environmental quality that affects occupants' comfort, productivity, and well-being (Bakhtiari et al., 2020; Humphreys et al., 2007). According to the World Health Organization, thermal comfort is a situation in which occupants are satisfied with the indoor thermal condition that can affect their physical, mental and social health conditions (World Health Organization, 1999). The

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ventilation system of a building is designed to control indoor air quality and provide a thermally comfortable environment (Gomes et al., 2021; Yuan et al., 2022). Thus, careful design and analysis of the factors that affect indoor thermal comfort is a key point to achieving a satisfying thermal condition for the occupants.

Ventilation systems are responsible for providing clean air in an enclosed environment and delivering a satisfying thermal condition for the occupants (Sadrizadeh et al., 2021). A wide range of studies experimentally and numerically evaluated the thermal comfort condition of different ventilation systems (Cheong et al., 2003; Noh et al., 2007; Sadrizadeh et al., 2017; Sadrizadeh & Holmberg, 2016; Stamou et al., 2008). Various air distribution strategies have been used to supply clean air in offices, schools and hospitals. Displacement and mixing strategies are among the common ones in offices and classrooms.

A diffuse ceiling ventilation system is a recent ventilation strategy that supplies cold air with low airflow momentum (Fan et al., 2013; Jacobs et al., 2008). This system distributes air to the occupant area using the space between the diffuse ceiling panels and the ceiling slab, called a plenum. A ceiling with a diffuse ceiling system consists of active and passive perforated panels, in which the cold air is ventilated to the room by passing the active panels. In this system, the thermal plume generated by the heat loads in the room is the driving force for the air distribution. Since the upward thermal plume and downward cold air from the diffuse ceiling drive the air movement, the heat sources' relative location with active diffuse panels plays an important role in airflow behaviour in the room (Zhang, Yu, et al., 2016b).

Diffuse ceiling ventilation systems are proper for cooling indoor environments with high heat loads, such as office buildings and schools. This ventilation system has a high potential for removing heat from a room compared to the conventional air distribution system (Nielsen & Jakubowska, 2009). Due to the large area of the diffuse ceiling, the system performs with a lower pressure drop to distribute the room's air. Jacobs and Knoll (2009) reported a 50% reduction in energy use in the classroom with the diffuse ceiling ventilation system since the fan needs less energy in this case rather than fully ducted systems.

Applying a proper air distribution system to satisfy thermal comfort has always been challenging for designers. Since there is a demand for a high velocity of clean air in mixing ventilation systems, this might increase the dissatisfaction risks of the occupants. Moreover, locating the air suppliers close to the floor level in the displacement system increases the draft risk. However, diffuse ceiling air distribution systems supply air with a low velocity, reducing the draft risk even if the outdoor cold air is directly supplied to the occupied zone (Jacobs & Knoll, 2009; Nielsen & Jakubowska, 2009; C. Zhang et al., 2016b). Moreover, diffuse ceiling ventilation systems have been more energy-efficient in removing indoor heat compared with conventional air distribution systems. Nielsen and Jakubowska (2009) performed an experimental study about the cooling capacity of six different air distribution systems, including diffuse ceiling systems. Their results indicated that the diffuse ceiling system had a high potential to manage the highest heat load condition.

The performance of diffuse ceiling ventilation systems depends on various design parameters, such as the room's height, diffuse ceiling structure, design of the plenum, amount of heat loads and their locations in the occupied space. Various research studies investigated the impact of different design parameters, as listed in Table 1. The

Table 1. Summary of previous studies on diffuse ceiling ventilation.

Reference	Investigated design parameter	Function	Main method	Major findings
Zhang et al. (2016b)	Plenum dimension-location of the supply air	Cooling capacity	Full-scale measurement	Supply air duct location affected the cooling capacity
Nielsen et al. (2015)	Three different configurations of the heat loads	Cooling capacity	Full-scale measurement	Evenly distributed heat loads had the better cooling performance
Zhang, Kristensen, et al. (2016a)	Four different configurations of heat loads	Cooling capacity	Experimental and numerical	Evenly distributed heat loads had lower draft risk
Lestinen et al. (2018)	Asymmetrical and symmetrical heat sources distribution	Cooling capacity	Full-scale measurement	asymmetrical heat load distribution had higher airflow fluctuation
Nocente et al. (2020)	Chess-board and complete cover of diffuse panels configurations	Cooling capacity	Full-scale measurement	both configurations had no different pressure drop

numerical results of C. Zhang et al. (2016b) showed that the dimension of the plenum and the location of the supply air duct had an impact on the cooling capacity of the diffuse ceiling system. Regarding the location of the heat sources, Nielsen et al. (2015) experimentally studied the impact of three different configurations of heat loads in a room. Their measurements revealed that evenly distributed heat loads resulted in the highest cooling capacity of the diffuse ceiling. Zhang, Kristensen, et al. (2016a) evaluated the performance of diffuse ceiling in a classroom with four configurations of the heat loads. They considered even distribution, centred configuration, and locating in front and back of the test room. Their results showed that evenly distributed heat sources generated lower draft risk in the occupied space. Furthermore, Lestinen et al. (2018) studied the distribution of asymmetrical and symmetrical heat sources in a room with a diffuse ceiling system. They reported that asymmetrical heat load distribution resulted in a higher airflow fluctuation compared with the symmetrical one. The distribution of the active diffuse panels (ADP) also affects the cooling capacity of the diffuse ceiling. The numerical study performed by Nocente et al. (2020) investigated the chess-board and complete cover of diffuse ceiling configurations. Their results showed that both configurations had no considerable difference in pressure drop aspects.

The previous literature mainly concentrated on a specific design parameter, excluding other parameters' impact. The relative location of heat loads and active diffuse panels has not been investigated in mentioned studies. As an instance, Nocente et al. (2020) evaluated the role of diffuse ceiling active panels in a room without heat sources. Moreover, the investigated cases in the aforementioned works have been based on a few experimental or numerical cases, which is difficult to reach a general result and conclusion. For instance, Zhang, Kristensen, et al. (2016a) and Nocente et al. (2020) investigated three and two scenarios, respectively. Thus, the authors' previous study (Rahnama et al., 2020) concentrated on the impact of the relative location of diffuse ceiling active panels and heat load distribution on the cooling capacity. In this regard, ten different scenarios were evaluated numerically and experimentally. Our previous results showed that the central configuration of ADPs with about 30% opening area and evenly distributed heat loads had the highest cooling capacity. The current study aimed to investigate further the effect of diffuse ceiling design parameters on the occupants' thermal comfort condition of selected

cases from our previous study. The current work has the following contributions compared with our previous study (Rahnama et al., 2020):

- Our previous study investigated the role of design parameters on the cooling capacity, while the present study evaluated the local thermal comfort and overall comfort.
- The geometry and configuration of investigated cases were modified to a more realistic condition, including considering human manikin models and monitors instead of simplified cylinders.
- The application of the diffuse ceiling has been mainly investigated for classrooms and offices. However, the performance of these systems in meeting and waiting rooms has not been discussed in detail while employees and patients spend long hours in these environments. In this regard, the present study evaluated this system's application in several cases, including meeting and waiting rooms in a clinic building beside an office.

The present study applied the computational fluid dynamics (CFD) technique to evaluate the thermal comfort of occupants in selected scenarios from our previous study. The numerical models were validated with measurement results by simulating the CFD model of the experimental study. The individuals' overall thermal sensation and local thermal comfort were assessed in all investigated scenarios.

2. Experimental study

Full-scale experimental studies were carried out in a guarded hot box in a laboratory with the internal dimension of 4.2 m length, 3.6 m width and 2.5 m height for the occupied zone. [Figure 1](#) shows the outside and inside configurations of the hot box (Zhang et al., 2015).

A diffuse ceiling made of wood wool cement panels separated the occupied zone from the plenum area. The heat load was generated with six cylinders, each attached to an electric lamp with a power regulator to have different heat loads ranging from 370 to 1070 W in different experimental scenarios, as listed in [Table 2](#).

The height of the cylinders was 1.1 m representing a seated person. Fifteen poles, each equipped with three Dantec hot-sphere anemometer probes, were located in the room to measure and log air velocity and temperature every 0.2 s at 45 points of the occupied

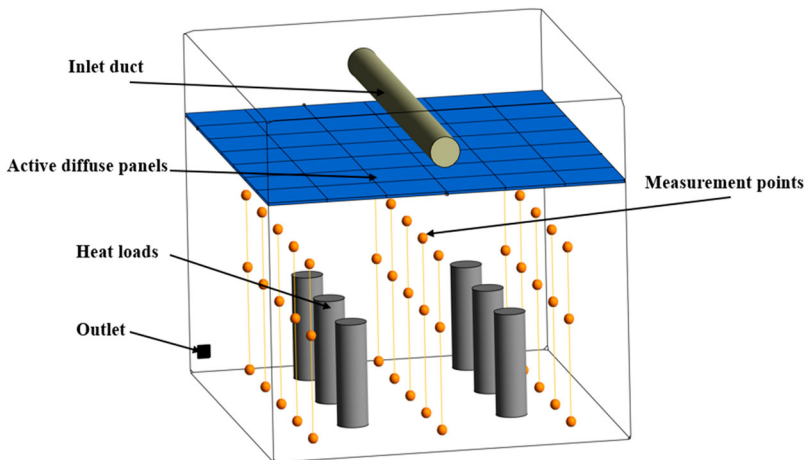


Figure 1. Inside (Rahnama et al., 2020) and outside view of the guarded hot box located at Aalborg University, Department of the Build Environment.

Table 2. The applied heat loads, supply airflow rate and temperature in experimental scenarios.

Experimental cases	Flow rate [l/s]	Temperature [°C]	Q[W]
Case 1	40	9.7	399
Case 2	40	9.7	576
Case 3	80	13.0	732
Case 4	80	9.4	1024
Case 5	100	14.9	723
Case 6	100	11.3	1028

zone. These poles were 0.5 m distant from each wall for covering the occupied zone based on DS/EN 13779 (DS/EN 13779 Dansk Standard, Ventilation for Non-Residential Buildings – Performance Requirements for Ventilation and Room-Conditioning Systems, 2007). Each measurement was performed over 3 min according to DS/EN 13182 (DS/EN 13182 Dansk Standard, Ventilation For Buildings - Instrumentation Requirements For Air Velocity Measurements In Ventilated Spaces, 2002) was calculated and considered for each anemometer probe. The uncertainty of velocity measurement is ± 0.02 m/s for the range of 0–1 m/s. The poles were located at a 0.5 m distance from the walls to cover the occupied zone based on recommendations from DS/EN 13779 (DS/EN 13779 Dansk Standard, Ventilation for Non-Residential Buildings – Performance Requirements for Ventilation and Room-Conditioning Systems, 2007). The anemometer probes were at the height of 0.1, 1.1, and 1.8 m above the floor, according to DS/EN 15726 standard for measurements in the occupied zone for evaluation of thermal conditions (DS/EN 15726 Dansk Standard, Ventilation for Buildings – Air Diffusion – Measurements in the Occupied Zone of Air Conditioned/Ventilated Rooms to Evaluate Thermal and Acoustic Conditions, 2012), as shown in Figure 2. The air was evenly supplied through a cylindrical textile duct in the plenum and was extracted through a 0.13 m \times 0.13 m exhaust duct placed in the lower zone of the room. The supply airflow rate was regulated with a frequency-controlled fan and measured with an orifice plate to provide three different supply airflow rates equal to 40, 80 and 100 l/s in different experimental scenarios. The exhaust airflow rate was controlled to be equal to the supply airflow rate, providing a balanced ventilation system. Thermocouples with the uncertainty measurement of \pm

**Figure 2.** The configuration of the experimental room and location of the measurement points.

0.15 K were used to measure the supply and extract air temperature, as well as wall surface temperature for validation of the CFD model. The details of the experimental study have been explained in (Rahnama et al., 2019).

3. Numerical study

3.1. Modelling the airflow

The current research study simulated the steady-state temperature and airflow field by applying the Reynolds-Averaged Navier-Stokes (RANS) model. The RANS approach needs less computational resources in comparison with the large-eddy and direct numerical simulations. The turbulent behaviour of the indoor airflow was modelled by adopting the k- ϵ turbulence model. In the current study, the two-equation Realizable k- ϵ model was used. This model computes the eddy viscosity and turbulent dissipation rate with two different equations (Shih et al., 1994). Moreover, the realizability of this model results in a more accurate prediction of the airflow field (Chen, 1995; Van Maele & Merci, 2006), besides having less calculation time (Nielsen, 2015; Zhang et al., 2007). This turbulence model has been adopted in previous studies and provided accurate results for the current application (Romano et al., 2015; Sadrizadeh et al., 2016). The conservation equations of mass, momentum and energy were considered. The conservation equations can be written in the general form of Eq. 1,

$$\frac{\partial}{\partial t}(\rho\phi) + \nabla \cdot (\rho\phi\vec{V} - \Gamma_\phi\nabla\phi) = S_\phi \quad (1)$$

where ϕ represents the corresponding transported quantity, is \vec{V} the air velocity vector, ρ is the air density, Γ_ϕ is the (effective) diffusion coefficient and S_ϕ is the source term.

The airflow field was simulated in all cases by using ANSYS Fluent 19.1. The Coupled Scheme was used to couple the pressure and velocity. The Second Order Upwind Scheme was adopted to discretise the convective part of all equations, while the PRESTO! Scheme was applied for the pressure. The root-mean-square residual value of 1×10^{-6} was adopted as convergence criteria for considered equations. The Discrete Ordinates (DO) radiation model was used to simulate the heat transfer by radiation. All solid walls were modelled by considering the Enhanced Wall Treatment model and Non-slip velocity boundary condition. The Enhanced Wall Treatment model simulates the air turbulent behaviour at the boundary layer. This model considers a two-layer model with enhanced wall functions (Chen & Patel, 1988).

To simulate the airflow field, the computational domain was subdivided into 3.1 million unstructured tetrahedral cells. The simulated environment was a replica of the experimental room. The grid was locally refined in the areas with high gradients, such as inlet and adjacent to the walls. The three inflation boundary layers were used at all solid walls, including manikin's surfaces, monitors and room walls. As a result of these refinements, the y^+ was obtained below 5.

Moreover, the grid independence study was accomplished to ensure that the simulated results have no dependency on the grid resolution. To achieve this purpose, the simulation results of the coarse (1.5 million cells), medium (3.1 million cells) and fine (5.9 million cells) grids were compared. The values of velocity and temperature were

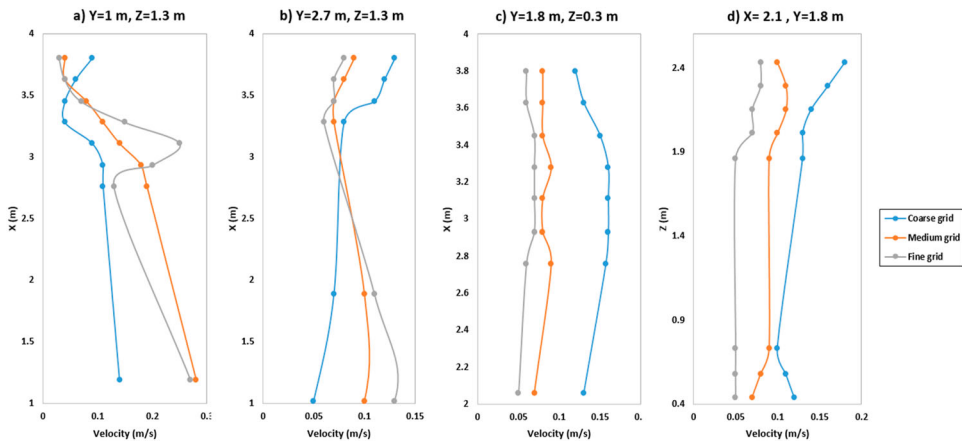


Figure 3. Comparison between velocity values at different locations between coarse, medium and fine grids.

compared at 80 different points in the simulated room for all three grid resolutions. Figure 3 shows the velocity distribution comparison for coarse, medium and fine grids at various locations. Since there was deviation between medium and fine grid along the Line 4 (Fig3. (d)), the grid convergence index (GCI) was computed as introduced by Roache (1994). The GCI values were 2.54% for Line 4 and below 1% for the rest of the lines in the medium grid, indicating the selected grid resolution was sufficiently fine (Kumar et al., 2022). Thus, the medium grid was adopted to save calculation time and resources.

The cold air was supplied with a flow rate of 40 l/s and a temperature of 10 °C for all cases. The supplied air was introduced to the ceiling plenum from the external surface of a cylindrical duct (Figure 2). The velocity inlet and pressure outlet boundary conditions were used for the room inlet and extract grill, respectively. The room walls had a constant wall temperature of 21 °C in the validation case as an experimental condition. The adiabatic condition was assigned to all room walls in parametric studies. A constant heat flux boundary condition was assigned to all heat sources in the rooms, as listed in Table 3. The generated heat by each heat source was defined based on the ASHRAE standard (Mora et al., 2021). The heat transfer from the heat sources was considered through both convection and radiation. The adiabatic wall boundary condition was assigned to the plenum walls.

The Species Transport model was used to simulate humidity distribution in the rooms. This model solves the convection–diffusion equation for predicting each species’s local mass fraction. In all thermal comfort simulations, the maximum relative humidity of 54% was considered for the supplied air, in which the diffuse ceiling system works without condensation, based on the Mollier diagram. An available ANSYS UDF code was applied to calculate the mean age of air.

Table 3. Emitted heat by heat sources in the simulated cases.

Heat source	Emitted heat (W/m ²)
Manikin: seated and reading	55
Manikin: seated and typing	65
Monitor	65

3.2. Modelling the diffuse ceiling panels

The diffuse ceiling panels were made of wood-cement material with a porous media structure. Thus, the porous media model was used that is available in ANSYS Fluent. This model has a momentum source term in the governing equation, and the fluid flow needs to overcome both the viscous and inertial resistance in the porous zone. At a low velocity of the fluid, the viscous term has a higher effect than the inertial term. However, the inertial impact is dominant at higher velocity ranges of the fluid. The momentum source term is as below

$$S_M = -\left(\frac{\mu}{\alpha} + C_2 \frac{1}{2} \rho |v|v\right) \quad (2)$$

The inertial resistance C_2 and $1/\alpha$ viscous resistance coefficient were obtained from the measurement results. These two coefficients were calculated by collecting the pressure drop data as a function of the velocity while the fluid flow passed the diffuse panels; consequently, $C_2 = 50635 \text{ m}^{-1}$ and $1/\alpha = 7.6 \times 10^7 \text{ m}^{-2}$ (Christodoulou et al., 2018). The current study used diffuse panels with thermal conductivity of 0.085 W/m K, a density of 360 kg/m³ and a porosity of 65%

3.3. Model validation

The CFD simulation results were compared with the experimental data to ensure the applied numerical model precisely predicted the airflow behaviour in the simulated rooms. In this regard, the experimental data from the experimental step with a supply flow rate of 40 l/s and a temperature of 9.74 °C was considered. The experimental room had 100% ADPs with evenly distributed headloads, as presented in Figure 2. Moreover, the overall heat load was 532 W during the measurements. The air velocity distribution is compared between the experimental and CFD simulation results in Figure 4a. This comparison showed that the numerical model could accurately predict the air velocity distribution in the experimental room. There was no deviation between simulated velocity values and experimental results in 29 out of 30 points. We had a maximum relative error of 2% at one point for velocity comparison. Figure 4b displays the air temperature values for the measured data and simulation results at five locations in the centre of the room. Overall, a good agreement was obtained between the CFD and experimental results. The temperature results showed a maximum relative error of 3% in 27 out of 30 points between CFD and experimental studies. The maximum relative error was 5–7% acceptable for engineering application in the rest points and had no considerable impact on the conclusion. More details about the validation study are available in previous authors' research work (Rahnama et al., 2020).

3.4. Parametric scenarios

All simulated scenarios are presented in Figure 5. The thermal comfort of the occupants was investigated in three different indoor environments, including a meeting room (Case 1 and 2), an office (Case 3 and 4) and a clinic waiting room (Case 5 and 6) with different diffuse ceiling configurations. In this regard, two different configurations of ADP with

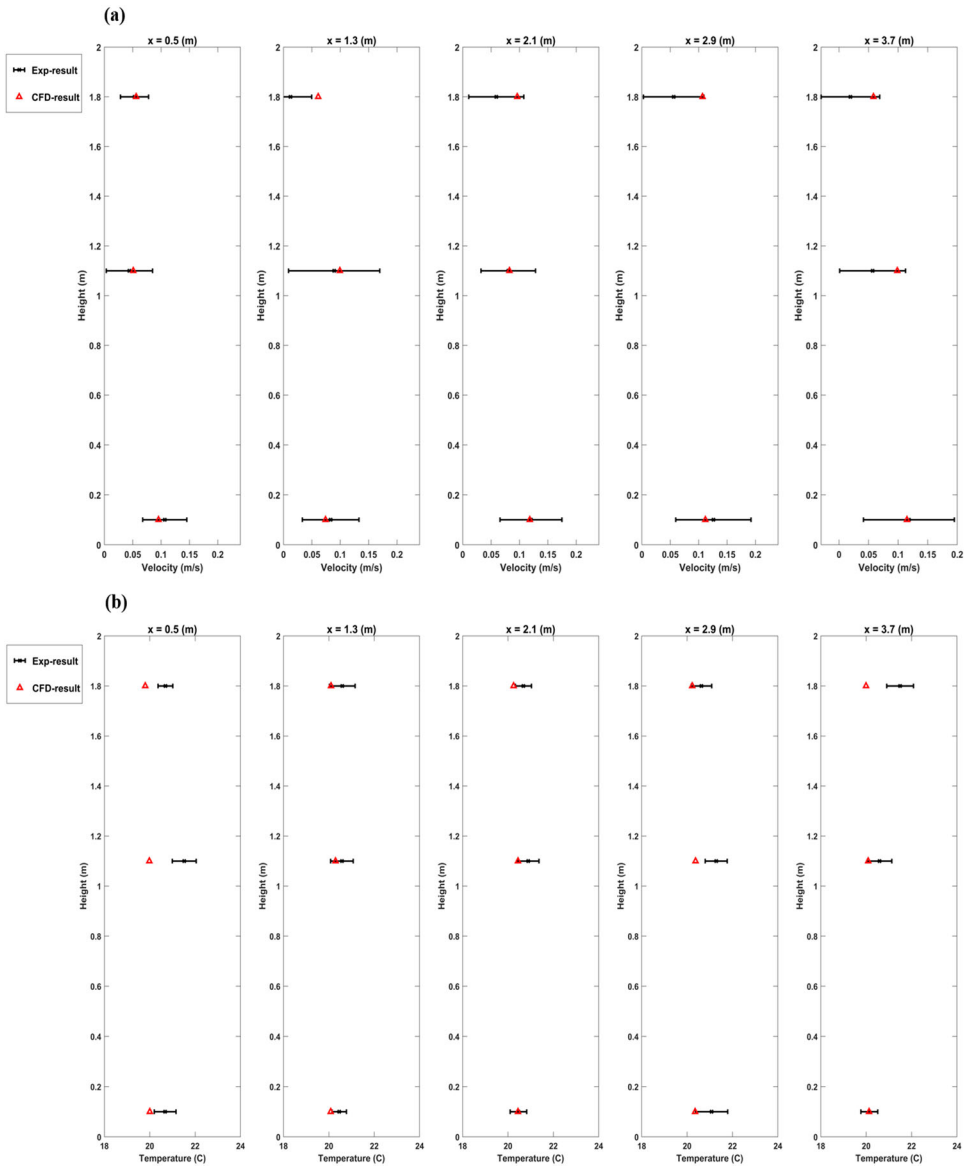


Figure 4. Comparison of the CFD simulation with experimental results for the (a) velocity distribution; and (b) temperature distribution.

about 30% opening area were selected. Cases 1, 3 and 5 had a central configuration of ADPs, while Cases 2, 4 and 6 used dispersed configurations, as shown in Figure 5. The details of the simulated cases are listed in Table 4. The overall headload was about 525 W in all simulated cases. The emitted heat from the occupants was calculated based on the ASHRAE Standard (Mora et al., 2021). In all simulated cases, the average values of temperature and velocity were calculated from the results of the steady-state simulations at a 0.20 m distance from the head of two occupants located close to the centre of the rooms. The selected occupants are presented in Figure 5 with red circles.

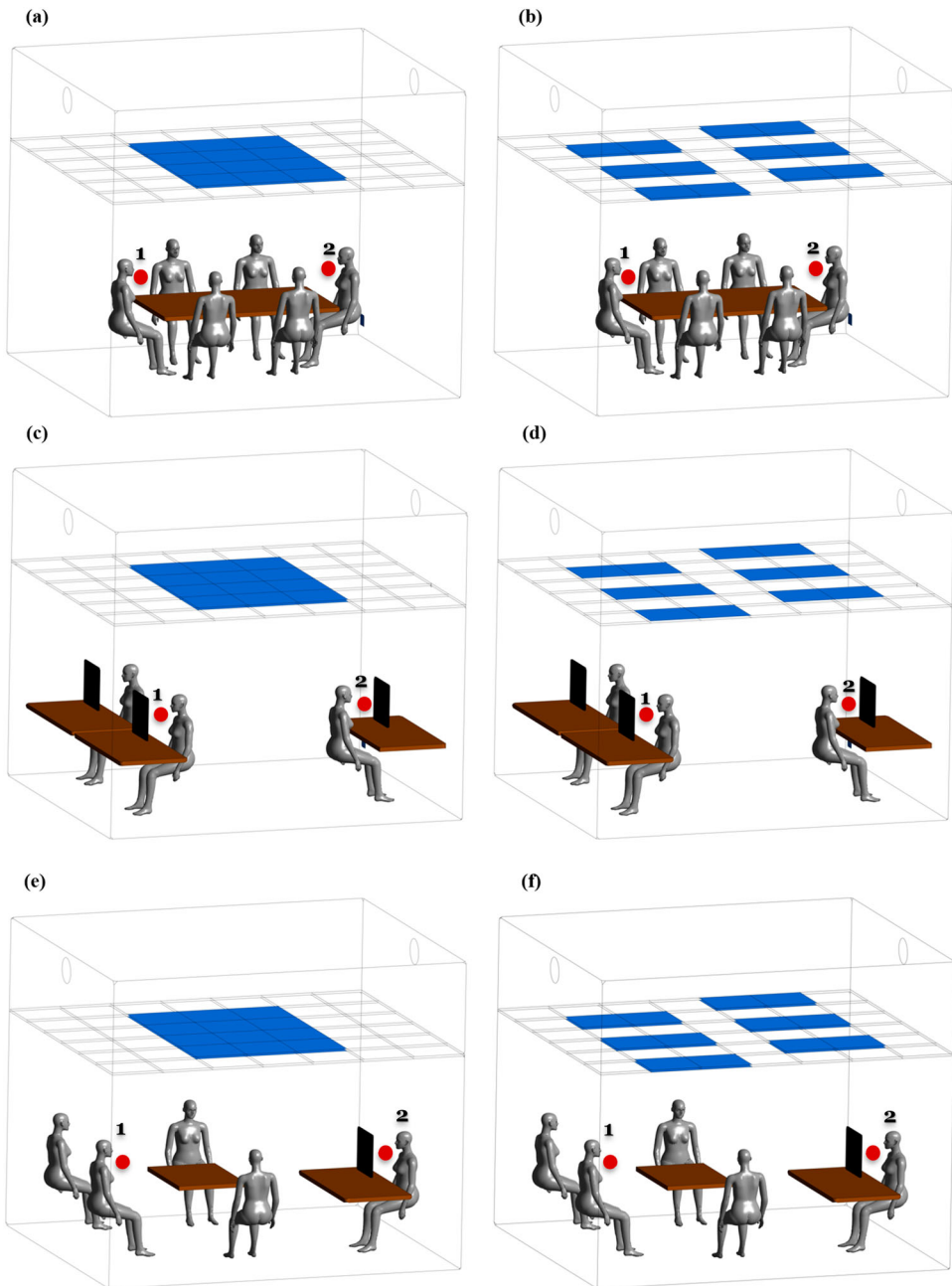


Figure 5. The configuration of the simulated cases: (a) and (b) meeting room; (c) and (d) office; and (e) and (f) waiting room.

4. PMV-PPD method

The thermal comfort condition was evaluated using PMV-PPD indices (Fanger, 1970). According to ISO-7730, thermal satisfaction can be assessed by considering PMV, PPD and draft rate (ISO 7730:2005, 2005). PMV represents the perception of occupants

Table 4. The simulated case studies.

Case study		Each occupant's emitted heat [W/m ²]	Emitted heat of a monitor [W/m ²]
Case 1	Meeting room + central configuration of ACDs	65	–
Case 2	Meeting room + dispersed configuration of ACDs	65	–
Case 3	Office + central configuration of ACDs	65	65
Case 4	Office + dispersed configuration of ACDs	65	65
Case 5	Waiting room + central configuration of ACDs	Patient: 55 Medical staff: 65	65
Case 6	Waiting room + dispersed configuration of ACDs	Patient: 55 Medical staff: 65	65

about the thermal condition in the indoor environment. In this regard, the proposed equation by ISO-7730 was considered for the calculation of PMV, as below

$$\begin{aligned}
 PMV = & (0.352e^{-0.042(M/A_{Du})} + 0.032) \\
 & \times \left\{ \frac{M}{A_{Du}}(1 - \eta) - 0.35 \times \left(43 - 0.061 \left(\frac{M}{A_{Du}}(1 - \eta) \right) - P_a \right) \right. \\
 & - 0.42 \times \left(\left(\frac{M}{A_{Du}}(1 - \eta) \right) - 50 \right) - 0.023 \frac{M}{A_{Du}}(44 - P_a) - 0.0014 \frac{M}{A_{Du}}(34 - t_a) \\
 & \left. - 3.96 \times 10^{-8} f_{cl} \times [(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] - f_{cl} h_c (t_{cl} - t_a) \right\}
 \end{aligned}$$

where M is the metabolic rate of the occupant, A_{Du} is the nude body surface area, η is mechanical efficiency, I_{cl} is defined as clothing insulation, f_{cl} is the clothing surface area factor, t_a shows the air temperature, \bar{t}_r represents the mean radiant temperature, P_a is water vapour partial pressure, h_c is the coefficient of convective heat transfer and t_{cl} is the surface temperature of the clothing. The selected metabolic rates are presented in Table 3. Moreover, the clothing factor was 0.5 clo for all simulated cases and was chosen based on the recommended values from ASHRAE Standard 55 (Mora et al., 2021). For calculation of the PMV, the mean values of the air temperature and velocity were computed over a volume with a relative humidity of 54%.

The PPD index shows the percentage of the occupants that feel too cold or too warm in the studied environment. The PPD computation is based on the PMV value as the given equation

$$PPD = 100 - 95 \times \exp(-0.03353 \times PMV^4 - 0.2179 \times PMV^2) \quad (4)$$

Based on this model, thermal comfort is achieved when the PMV value falls between -0.5 and 0.5 . In this range of PMV, about 90% of occupants are satisfied with the thermal condition in the room. More detailed explanations about the PMV-PPD model and thermal comfort are available in ISO-7730 (ISO 7730:2005, 2005).

5. Results and discussion

Previous authors' research evaluated the cooling capacity of the various configuration configurations of the diffuse ceiling opening area and the location of the heat loads in

the room (Rahnama et al., 2020). Two different configurations of active diffuse ceiling panels were considered to evaluate the impact of diffuse ceiling system design parameters on overall and local thermal comfort.

5.1. Local thermal comfort

The velocity distribution at a cut plan passing the occupants in all simulated rooms is presented in Figure 6. The velocity distribution in the meeting room showed an average value of 0.13 m/s close to the seated occupants in Case 1 (central configuration of ADPs). In comparison, this value was around 0.05 m/s for Case 2 (dispersed configuration of ADPs). In the office, the generated thermal plume from the occupants was dominant for both Cases 3 and 4. Consequently, the maximum velocity of 0.25 m/s was obtained in the rooms. However, the generated thermal plume from the manikin located exactly below the diffuse ceiling opening was damped in Case 4 (Figure 6). In Case 6, the thermal plume from the occupant close to the diffuse ceiling was degraded due to the cold airflow from the ceiling, while this phenomenon was not observed in Case 5, as presented in Figure 6.

The temperature distribution at a cut plan is presented in Figure 7 for all simulated rooms. In all investigated cases, the average air temperature was between 18.5 and 19.5 °C with the central opening of diffuse panels (Case 1, 3 & 5 in Figure 7). This value was about 20–21 °C in the rooms with the dispersed configuration of the diffuse ceiling (Case 2, 4 & 6 in Figure 7). The relative distance of the occupants to the diffuse ceiling opening panels had a strong impact on the damping of the generated thermal plume, as shown in Figure 7. In all simulated cases, the temperature gradient between the occupants' ankles and heads was less than 3 °C, which is in the thermal comfort range based on the ASHRAE standard (Mora et al., 2021).

The draft rate has been the main concern in indoor environments using ventilation systems. Figure 8 displays the draft rate at the neck level of the two selected occupants (Figure 5). In all simulated cases, the draft rate values were below the recommended value of 20% (Mora et al., 2021). The draft rates were 13.06% in Case 3 and 14% in Case 4, higher than other cases. These two cases evaluated the performance of both diffuse ceiling configurations in an office. Thus, the application of a diffuse ceiling in the office showed the highest draft rate, consequently more thermal comfort dissatisfaction than other cases.

5.2. Overall comfort

5.2.1. Age of air

The age of the air is a parameter that calculates the mean time that a particle travels from one point, such as an inlet, to a measurement point (Federspiel, 1999). The higher age of air results in the increase of air contamination concentration in the room, consequently negatively affecting the comfort condition of the occupants. In the current work, the age of air was investigated to evaluate the efficiency of the applied diffuse ceiling systems. The average age of the air in all cases is listed in Table 5. Overall, the age of air values had a uniform distribution in all simulated cases. The highest age of air was 328 s in waiting rooms with both diffuse ceiling configurations (Case 5 & 6), while this

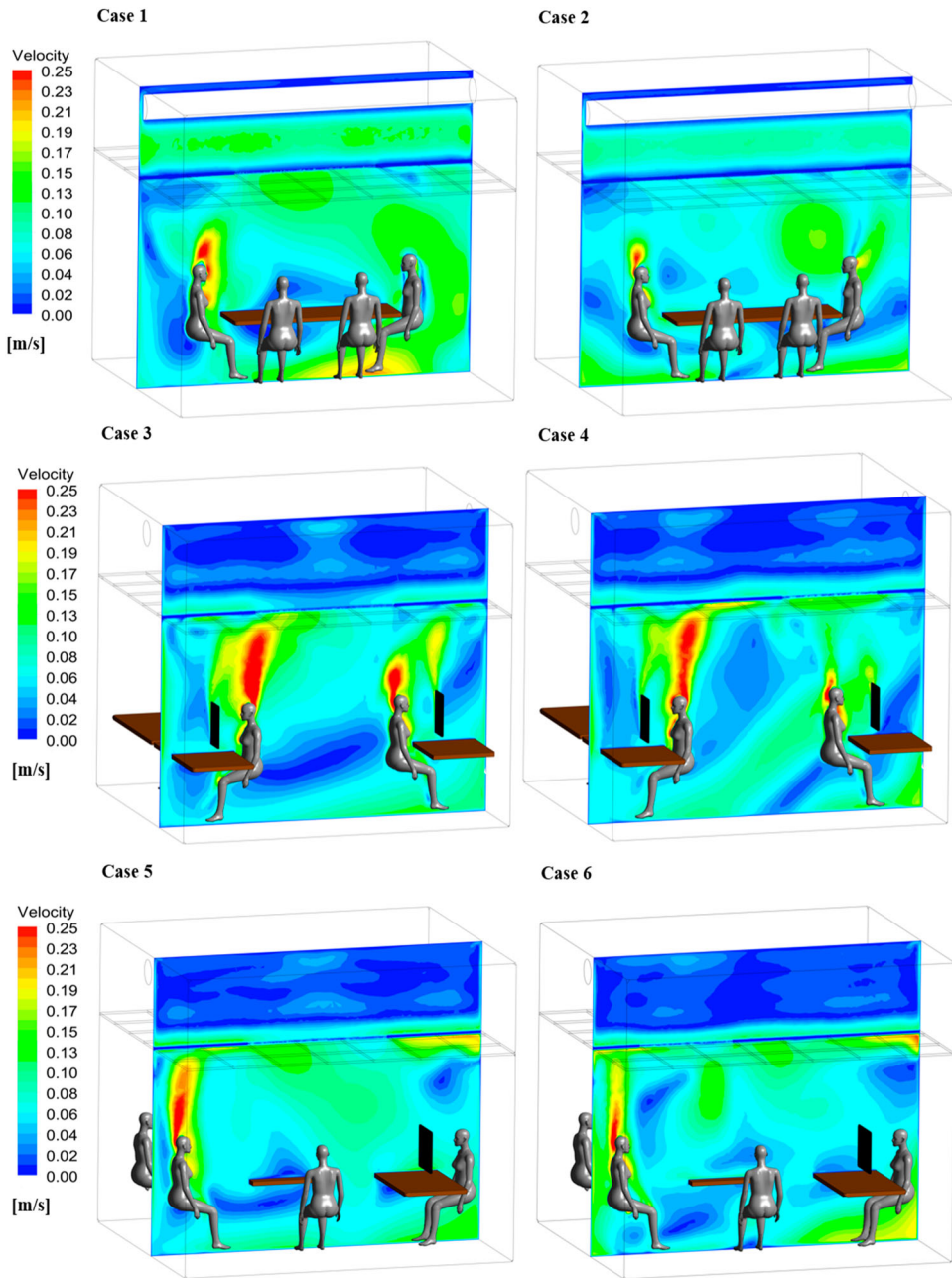


Figure 6. Velocity distribution contour plots for all the simulated cases.

value was between 290 and 294 s in the meeting rooms. This difference might be due to the fact that all the manikins were located at the centre of the meeting room; however, in the waiting room, all the occupants and heat loads were dispersed in the room. Thus, locating the occupants and heat sources can increase the age of air, consequently reduction of the indoor air quality and comfort.

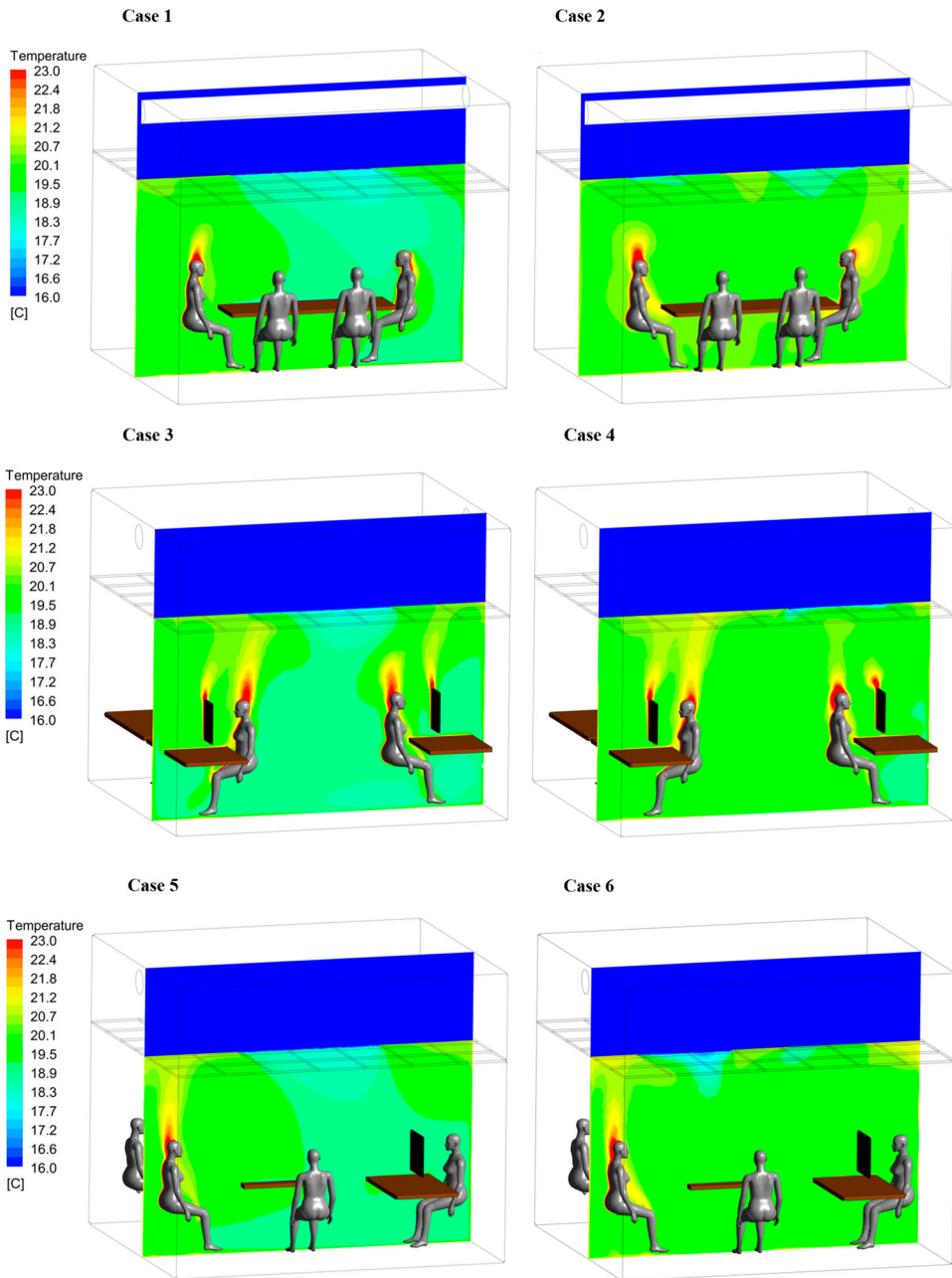


Figure 7. Temperature contour plots for all the simulated cases.

5.2.2. Thermal sensation

The PMV and PPD values were computed to evaluate the occupants' thermal sensation in all rooms, as shown in Figure 9. Overall the PMV and PPD values were in the acceptable range of $-0.5 < PMV < 0.5$ for a comfortable thermal condition of the occupants based on the standards (ISO 7730:2005, 2005; Mora et al., 2021). Thus, changing the design

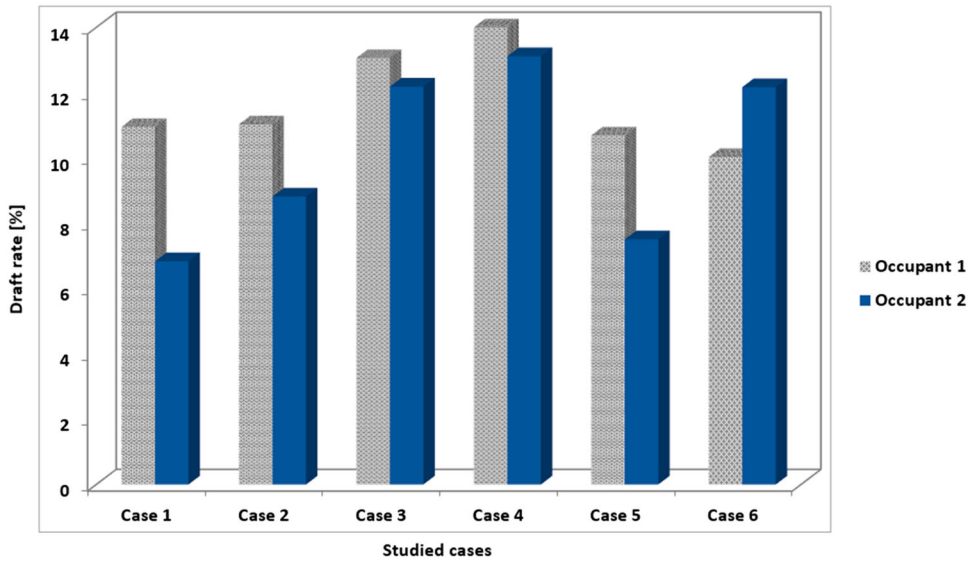


Figure 8. Draft rate plots in all simulated rooms.

Table 5. The average age of air in the rooms.

Simulated case	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Average age of air	290	294	298	298	327	328

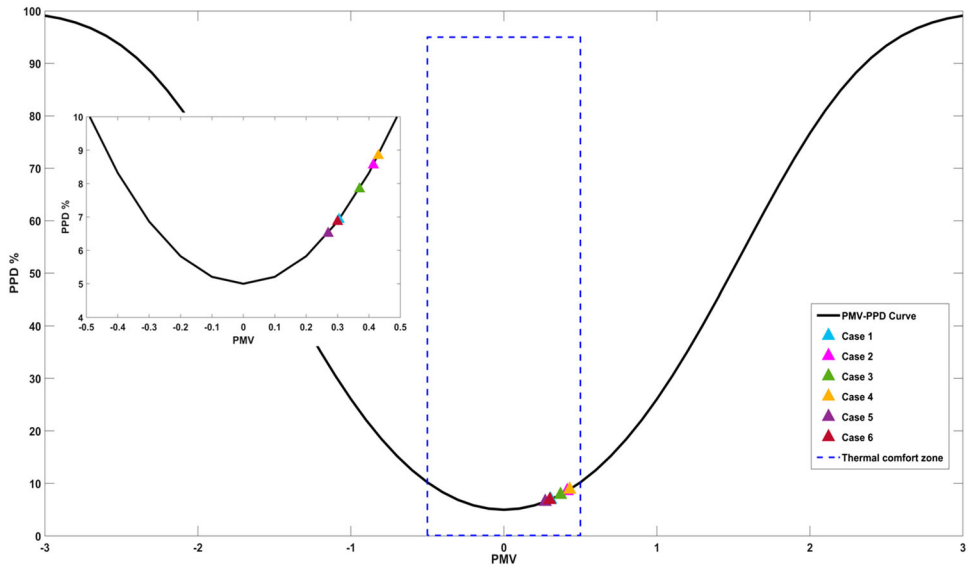


Figure 9. Comparison of PMV and PPD values for all simulated cases.

parameters of the diffuse ceiling system had no considerable impact on the occupants' thermal sensation according to PMV-PPD calculations.

In all cases, the PPD values were below 10%, proving the applied diffuse ceiling systems provided a satisfying condition for the occupants. The maximum PPD values obtained in Cases 2 and 4 were 8.5% and 9% (Figure 9), respectively. As a result, diffuse ceiling configuration with the dispersed configuration of ADPs results in higher dissatisfaction in the waiting rooms and offices. Moreover, the application of the diffuse ceiling for the office had the maximum dissatisfaction. Consequently, the thermal discomfort might increase in rooms occupied with more headloads than in the investigated case.

Overall, Cases 1, 5 and 6 showed the highest thermal comfort satisfaction for the occupants. Moreover, the lowest dissatisfaction of the occupants was reported in Case 5 (Figure 9). Thus, applying the diffuse ceiling configuration with the central opening of the diffuse panel is highly recommended for waiting and meeting rooms. Moreover, our previous study (Rahnama et al., 2020) showed that the central opening of ADPs had the highest cooling capacity compared to the dispersed configuration. The diffuse ceiling system with the central opening of diffuse panels had the highest cooling capacity and thermal comfort satisfaction in the rooms with higher heat loads than other configurations. In contrast, the dispersed configuration resulted in less thermal satisfaction and the lowest cooling capacity.

6. Conclusion

The current research investigated the role of diffuse ceiling design parameters on thermal comfort conditions in the office, meetings, and waiting rooms. In this regard, two different diffuse ceiling configurations were adopted. Based on our previous study (Rahnama et al., 2020), we have considered two configurations with about 30% opening area, including the central and dispersed configuration of ADPs. The experimental results were used to validate the applied numerical models. Moreover, CFD simulations were adopted for thermal comfort analysis.

The simulated results showed that selected configurations of diffuse ceiling systems provided satisfying thermal comfort conditions in all simulated case studies. Thus, changing the design parameters of the diffuse ceiling showed a limited impact on the thermal comfort in the rooms with the selected configurations of diffuse panels. Furthermore, the applied ADP configurations provided thermally comfort conditions in realistic scenarios, including the waiting room and meeting room. The diffuse ceiling system with the dispersed ADPs configuration had the highest PPD value of 9% in the office and waiting room. Furthermore, this configuration showed the highest draft rate of 14% in the office case. The highest age of air of 328s was obtained in the waiting room, consequently reducing occupants' comfort. Overall, the thermal comfort condition was more satisfying in the meeting and waiting rooms compared to the office.

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References

- Bakhtiari, H., Akander, J., & Cehlin, M. (2020). Evaluation of thermal comfort in a historic building refurbished to an office building with modernized HVAC systems. *Advances in Building Energy Research*, 14(2), 218–237. <https://doi.org/10.1080/17512549.2019.1604428>
- Chen, H. C., & Patel, V. C. (1988). Near-wall turbulence models for complex flows including separation. *AIAA Journal*, 26(6), 641–648. <https://doi.org/10.2514/3.9948>
- Chen, Q. (1995). Comparison of different k- ϵ models for indoor air flow computations. *Numerical Heat Transfer, Part B: Fundamentals*, 28(3), 353–369. <https://doi.org/10.1080/10407799508928838>
- Cheong, K. W. D., Djunaedy, E., Chua, Y. L., Tham, K. W., Sekhar, S. C., Wong, N. H., & Ullah, M. B. (2003). Thermal comfort study of an air-conditioned lecture theatre in the tropics. *Building and Environment*, 38(1), 63–73. [https://doi.org/10.1016/S0360-1323\(02\)00020-3](https://doi.org/10.1016/S0360-1323(02)00020-3)
- Christodoulou, G., Kubilius, M., & Andersen, R. B. K. (2018, June). *Analysis of the indoor environment in a room with diffuse ceiling ventilation* [Master thesis]. Aalborg University.
- DS/EN 13182 Dansk standard, Ventilation For Buildings - Instrumentation Requirements For Air Velocity Measurements In Ventilated Spaces. (2002).
- DS/EN 13779 Dansk standard, Ventilation for non-residential buildings – Performance requirements for ventilation and room-conditioning systems. (2007).
- DS/EN 15726 Dansk standard, Ventilation for buildings – Air diffusion – Measurements in the occupied zone of air conditioned/ventilated rooms to evaluate thermal and acoustic conditions. (2012).
- Fan, J., Hviid, C. A., & Yang, H. (2013). Performance analysis of a new design of office diffuse ceiling ventilation system. *Energy and Buildings*, 59, 73–81. <https://doi.org/10.1016/j.enbuild.2013.01.001>
- Fanger, P. O. (1970). *Thermal comfort: Analysis and applications in environmental engineering*. McGraw-Hill.
- Federspiel, C. C. (1999). Air-change effectiveness: Theory and calculation methods. *Indoor Air*, 9(1), 47–56. <https://doi.org/10.1111/j.1600-0668.1999.t01-3-00008.x>
- Gomes, R., Ferreira, A., Azevedo, L., Costa Neto, R., Aelenei, L., & Silva, C. (2021). Retrofit measures evaluation considering thermal comfort using building energy simulation: Two Lisbon households. *Advances in Building Energy Research*, 15(3), 291–314. <https://doi.org/10.1080/17512549.2018.1520646>
- Humphreys, M. A., Nicol, J. F., & Raja, I. A. (2007). Field studies of indoor thermal comfort and the progress of the adaptive approach. *Advances in Building Energy Research*, 1(1), 55–88. <https://doi.org/10.1080/17512549.2007.9687269>
- ISO 7730:2005. (2005). Ergonomics of the thermal environment — Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria.
- Jacobs, P., & Knoll, B. (2009). Diffuse ceiling ventilation for fresh classrooms. *4 Th Intern. Symposium on Building and Ductwork Air Tightness*, 1–7.

- Jacobs, P., Oeffelen, E. C. M. V., & Knoll, B. (2008). Diffuse ceiling ventilation, a new concept for healthy and productive classrooms. *Indoor Air*, 17–22.
- Kumar, N., Bardhan, R., Kubota, T., Tominaga, Y., & Shirzadi, M. (2022). Parametric study on vertical void configurations for improving ventilation performance in the mid-rise apartment building. *Building and Environment*, 215, 108969. <https://doi.org/10.1016/j.buildenv.2022.108969>
- Lestinen, S., Kilpeläinen, S., Kosonen, R., Jokisalo, J., & Koskela, H. (2018). Experimental study on airflow characteristics with asymmetrical heat load distribution and low-momentum diffuse ceiling ventilation. *Building and Environment*, 134(November 2017), 168–180. <https://doi.org/10.1016/j.buildenv.2018.02.029>
- Mora, R., Rose, D. M., Eddy, J., Alspach, P. F., Smith, A. R., Williams, J. G., Barnaby, C. S., Burkhead, R. B., & Hanson, S. S. (2021). ASHRAE Standard 55-2020, thermal environmental conditions for human occupancy. *ASHRAE*. www.ashrae.org
- Nielsen, P. V. (2015). Fifty years of CFD for room air distribution. *Building and Environment*, 91, 78–90. <https://doi.org/10.1016/j.buildenv.2015.02.035>
- Nielsen, P. V., & Jakubowska, E. (2009). The performance of diffuse ceiling inlet and other room air distribution systems. In *Cold Climate HVAC 2009: Sisimiut Greenland*, 16–19 March 2009.
- Nielsen, P. V., Vilsbøll, R. W., Liu, L., & Jensen, R. L. (2015). Diffuse ceiling ventilation and the influence of room height and heat load distribution. *Proceedings of Healthy Buildings*, 2015.
- Nocente, A., Arslan, T., Grynning, S., & Goia, F. (2020). CFD study of diffuse ceiling ventilation through perforated ceiling panels. *Energies*, 13(8), 1–14. <https://doi.org/10.3390/en13081995>
- Noh, K. C., Jang, J. S., & Oh, M. D. (2007). Thermal comfort and indoor air quality in the lecture room with 4-way cassette air-conditioner and mixing ventilation system. *Building and Environment*, 42(2), 689–698. <https://doi.org/10.1016/j.buildenv.2005.10.033>
- Rahnama, S., Nielsen, P. V., Afshari, A., Bergsøe, N. C., Johra, H., & Jensen, R. L. (2019). Evaluating the cooling capacity of diffuse ceiling ventilation system - full-scale experimental study. *E3S Web of Conferences*, 111(2019), 4–9. <https://doi.org/10.1051/e3sconf/201911120206>
- Rahnama, S., Sadeghian, P., Nielsen, P. V., Zhang, C., Sadrizadeh, S., & Afshari, A. (2020). Cooling capacity of diffuse ceiling ventilation system and the impact of heat load and diffuse panel distribution. *Building and Environment*, 185, 107290. <https://doi.org/10.1016/j.buildenv.2020.107290>
- Roache, P. J. (1994). Perspective: A method for uniform reporting of grid refinement studies. *Journal of Fluids Engineering*, 116(3), 405–413. <https://doi.org/10.1115/1.2910291>
- Romano, F., Marocco, L., Gustén, J., & Joppolo, C. M. (2015). Numerical and experimental analysis of airborne particles control in an operating theater. *Building and Environment*, 89, 369–379. <https://doi.org/10.1016/j.buildenv.2015.03.003>
- Sadrizadeh, S., Afshari, A., & Iqbal, A. (2017). A numerical analysis of diffuse ceiling ventilation performance in a school classroom and auditorium under different operating conditions. *Healthy Buildings Europe*, 2017(1), 2–5.
- Sadrizadeh, S., Afshari, A., Karimipannah, T., Håkansson, U., & Nielsen, P. V. (2016). Numerical simulation of the impact of surgeon posture on airborne particle distribution in a turbulent mixing operating theatre. *Building and Environment*, 110, 140–147. <https://doi.org/10.1016/j.buildenv.2016.10.005>
- Sadrizadeh, S., Aganovic, A., Bogdan, A., Wang, C., Afshari, A., Hartmann, A., Croitoru, C., Khan, A., Kriegel, M., Lind, M., Liu, Z., Melikov, A., Mo, J., Rotheudt, H., Yao, R., Zhang, Y., Abouali, O., Langvatn, H., Sköldenberg, O., & Cao, G. (2021). A systematic review of operating room ventilation. *Journal of Building Engineering*, 40, 102693. <https://doi.org/10.1016/j.jobe.2021.102693>
- Sadrizadeh, S., & Holmberg, S. (2016). Thermal comfort of the surgical staff in an operating theatre: a numerical study on laminar and mixing ventilation systems. *14th International Conference of Indoor Air Quality and Climate*, 1–6.
- Shih, T., Liou, W., Shabbir, A., Yang, Z., & Zhu, J. (1994). A new k-ε eddy viscosity model for high Reynolds number turbulent flows-model development and validation. *NASA Technical Memorandum*.
- Stamou, A. I., Katsiris, I., & Schaelin, A. (2008). Evaluation of thermal comfort in Galatsi Arena of the Olympics "Athens 2004" using a CFD model. *Applied Thermal Engineering*, 28(10), 1206–1215. <https://doi.org/10.1016/j.applthermaleng.2007.07.020>

- Van Maele, K., & Merci, B. (2006). Application of two buoyancy-modified k- ϵ turbulence models to different types of buoyant plumes. *Fire Safety Journal*, 41(2), 122–138. <https://doi.org/10.1016/j.firesaf.2005.11.003>
- World Health Organization. (1999). Health is a state of complete physical, mental and social well being and not merely the absence of disease or infirmity. *WHO Basic Documents*.
- Yuan, F., Yao, R., Sadrizadeh, S., Li, B., Cao, G., Zhang, S., Zhou, S., Liu, H., Bogdan, A., Croitoru, C., Melikov, A., Short, C. A., & Li, B. (2022). Thermal comfort in hospital buildings – A literature review. *Journal of Building Engineering*, 45, 103463. <https://doi.org/10.1016/j.jobbe.2021.103463>
- Zhang, C., Heiselberg, P. K., Pomianowski, M., Yu, T., & Jensen, R. L. (2015). Experimental study of diffuse ceiling ventilation coupled with a thermally activated building construction in an office room. *Energy and Buildings*, 105, 60–70. <https://doi.org/10.1016/j.enbuild.2015.07.048>
- Zhang, C., Kristensen, M. H., Jensen, J. S., Heiselberg, P. K., Jensen, R. L., & Pomianowski, M. (2016a). Parametrical analysis on the diffuse ceiling ventilation by experimental and numerical studies. *Energy and Buildings*, 111, 87–97. <https://doi.org/10.1016/j.enbuild.2015.11.041>
- Zhang, C., Yu, T., Heiselberg, P. K., Pomianowski, M. Z., & Nielsen, P. V. (2016b). *Diffuse Ceiling Ventilation: Design Guide*, Technical Re-Port, Aalborg University, Department of Civil Engineering, No. 217.
- Zhang, Z., Zhang, W., Zhai, Z. J., & Chen, Q. Y. (2007). Evaluation of various turbulence models in predicting airflow and turbulence in enclosed environments by CFD: Part 2—comparison with experimental data from literature. *HVAC and R Research*, 13(6), 871–886. <https://doi.org/10.1080/10789669.2007.10391460>