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EFFECT OF EXHAUST LAYOUT ON THE INDOOR THERMAL COMFORT UNDER HARSH WEATHER CONDITIONS

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ABSTRACT

The inlet and outlet size, number and location have a significant influence on the thermal environment indoor and air distribution for the temperature and velocity. In this work, numerical and experimental studies were performed to examine the influences of the inlet and outlet distribution, number and location, on the air movement and temperature distribution indoor. Also, the separation of the amount of the extracted air has been investigated in this study. To provide a comfortable environment for the occupants, important factors such as air temperature distribution, thermal sensation and draft rate should be evaluated carefully. Therefore, in this paper the occupant's thermal sensation and the air movement and temperature distribution were used as the main evaluation index. In this investigation, three cases study were used tested. The experimental work was performed under the Iraqi weather conditions which are hot and dry in summer. The finding showed that the indoor thermal environments were significantly influenced by the opening locations of the exhaust. Also it was found that the satisfied human thermal comfort was obtained when the exhaust diffuser installed relatively far away from the supply diffuser. In addition, the best results were found by separate the amount of the exhaust air and extracted from the two exhausts opening. This will give the supplied air the ability to distribute inside room perfectly. Also, in order to prevent the air short circuit, the exhaust opening should not be located at the wall in front of the supply opening.

Keywords: *Thermal Comfort, Exhaust Layout, Displacement Ventilation, CFD, Ventilation*

INTRODUCTION

The main important goal of the Heating, Ventilation, and Air Conditioning (HVAC) system is to provide a comfortable and healthy living environment and reduce the contaminant in the indoor room air. A good air distribution system needs to select the proper locations of the supply and exhaust vent based on the configuration of room, heat sources position indoor and the indoor air condition [1-3]. According to the 1970s energy crisis, many efforts have been taken into account in the field of HVAC for finding a balance between the most debatable criteria's in term of air distribution, quality of the air indoor, human comfort, and energy consumption [4-9].

There were many recorded complaints by occupants about the comfortable air quality and thermal environment using the conventional variable air volume system [11-12]. The complaints were recorded due to the drawbacks of productivity and activity of occupants. In addition, a high rate of energy consumption by HVAC system has been recorded by the Hong Kong Energy End-use Data (HKEEFUD) [13]. The systems of stratified air distribution (STRAD) have been attracted much attention by researches during the last two decades, which have an advantage in term of higher performance and ventilation efficiency comparing to the traditional mixing ventilation system (MV) [1]. For the STRAD system, the air supply is directly located over the thermal sources which are generated in the occupied zone. The generated heat sources are subjected under the buoyancy law by inducing the fresh air inside the zone to circulate from the lower room level to the breathing zone. Furthermore, the STRAD systems have an effective air distribution and higher thermal comfort comparing to the MV systems [1, 14-15]. The STRAD system has also a potential to apply the cooling on the lower level inside the zone while there is no need to cool the upper level inside the same zone. Thus, there is a potential of human thermal comfort improvement and saving energy for air conditioning applications using STRAD systems [10]. The positions of diffusers inside the zone have a significant influence on the STRAD system. Kuo et al. [17] simulated the locations effects of the supply and exhaust outlet in the working zone on the human comfort. They found that the thermal comfort could be improved with the longer path of cold air supply

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in the occupied area. Wan et al. [18] experimentally investigated variant densities of thermal load supply and inlet temperatures of space using a floor return technique based under floor air circulation system. They concluded that there were a significant saving in energy with an acceptable thermal comfort level under a high density of thermal load and inlet temperature of 18℃. Lau et.al. [19] suggested to detach the positions of exhausts and returns inside the zone for saving more energy and providing a good indoor thermal environment using STRAD systems. Xu et al. [20] also summarized that the STRAD systems offers an energy saving and comfortable environment by enlarging the zone and increasing the height of ceiling. Lam et al. [21] studied the temperatures distribution and air flow circulation inside the zone. They numerically found that the position of exhaust has a significant impact on the human comfort of the STRAD system, in turn; it has a considerable impact on the annual load of cooling system. Awad et al. [22] practically investigated the STRAD system based on the level of interface as it was affected by the location of exhaust grilles, which also significantly influenced the cooling effect of HVAC systems. Safer indoor thermal environment can be achieved in terms of air quality when using the exhaust diffuser at the upper level of the room space and the supply vent at the low level [23]. The effects of the outlet diffuser positions were investigated numerically by Khan et al. [24]. They found that an acceptable indoor air quality (IAQ) was obtained when the exhaust opening locating close to the ceiling. Kuo and Chung [25] examined numerically the influence of the air vent locations on the indoor human comfort using different ventilation methods. Their results revealed that the best thermal comfort for the indoor air founded at the longer supply air throw in the working area. He et al. [1] found that the locations of exhaust opening may not affect by the behavior of airflow, however it can greatly influence the exposure level indoor. Lin et al. [27] studied the effect of the location of the air supply vents on the performance of displacement ventilation system. The results revealed that the good indoor environment achieved when the air supply diffuser located close to the room center. Verma et al. [28] have presented that the IAQ in a hospital ward was highly influenced by the amount of the air change and the outlet locations. They found that a high rate of air change will reduced the amount of the contaminant concentration and this will lead to enhance the IAQ. Also, a proper selection of the outlet opening will improve the quality of the air in the inhaled area. Another- study by Verma et al. [29] was performed to provide healthy and a comfortable working environment for the doctors and patients in the Intensive Care Unit (ICU). The results found that the stagnant zone in the ICU room was unhealthy for the occupants, patient and doctor, and careful considerations should be taken in account when designing a ventilation system for such room. Another investigation by Verma et al. [30] found that the contaminant distribution was influenced by the position of the patient's bed and the arrangement of the air ventilation system in a hospital ICU room.

The previous works have examined the effect of the positions of each the supply diffuser and the return diffuser on the energy consumption and indoor thermal comfort. However, limited research or insufficient studies has been performed to study the relationship between the amount of extracted air, depending on the number of the exhaust diffusers, and indoor thermal indoor thermal indoor thermal comfort. The amount of extracted air related to the number of the exhaust diffusers has a major effect on the indoor human comfort. Therefore, in this paper the influences of using a different number of the exhaust diffusers with different amount of the extracted air in an equipped office room on the human thermal comfort, velocity distribution, and indoor air temperature were investigated experimentally and numerically.

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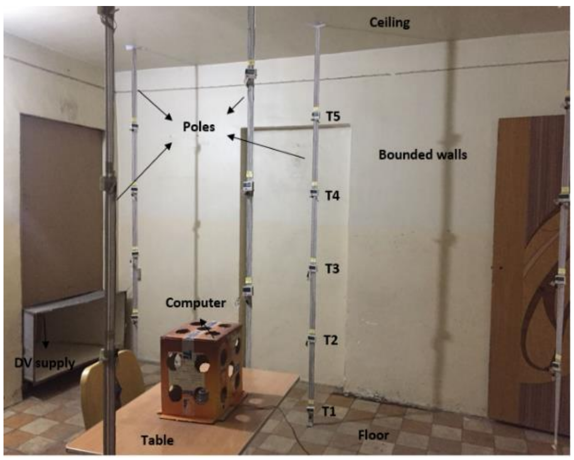
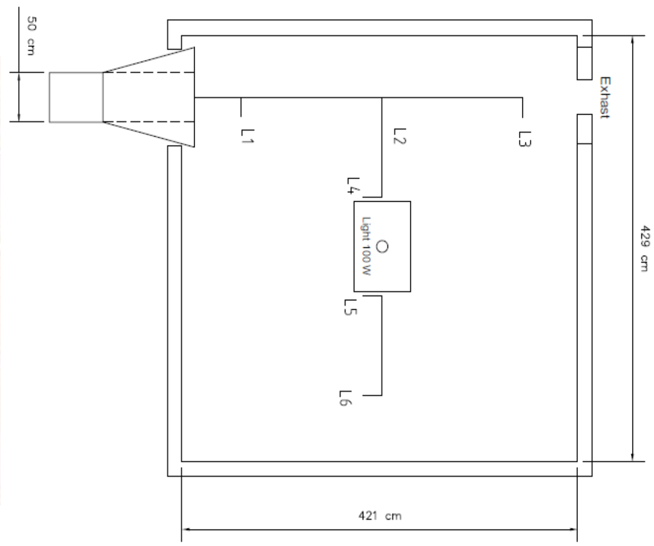
EXPERIMENTAL WORK

An experimental work was performed to study the impact of the exhaust locations and the amount of the extracted air in an office room ventilated by the displacement ventilation (DV) system on both thermal indoor environment and indoor thermal comfort. Figure 1 shows the experimental room and the schematic drawing of the investigated room respectively. The tested room had dimensions of (4.25 m) length, (4.2 m) width and (3 m) height. In this room, the DV system consider the main air distribution system with dimension of (0.6 m × 1 m) which was installed at the level of floor near to room corner. To satisfy the requirements of the indoor thermal comfort, a required ventilation rate for one person was covered in this study. Therefore, in this investigation the supply air velocity and supply air temperature were 0.3 m /sec and 25 °C respectively. The bounded wall, ceiling and floor were insulated carefully to prevent heat transfer through these walls (adiabatic walls). A box with heat generation of 100 W was used and installed in the center of the room to represent the computer. The exhaust opening was 0.35 m×0.35 m. In this experiments, three different exhaust locations were employed to calculate the impact of the exhaust layout on the indoor thermal environment (see Table 1). Six different locations (L) 1, 2,3,4,5 and 6, were employed to measure the air temperature. The temperature was measured for five different heights at each pole. Figure 1 shows the poles locations and the measured point for each case study. The HT-315 thermocouple was employed to measure the air temperature at the supply opening (See Figure 1 c). Digital thermocouples were used to measure the rest of the measuring points (see Figure 1 d). The experiments were performed to examine the impact of using different numbers and locations of the exhaust opening on the indoor human thermal comfort. Table 1 presents the detailed information for each case study in these experiments.



a)



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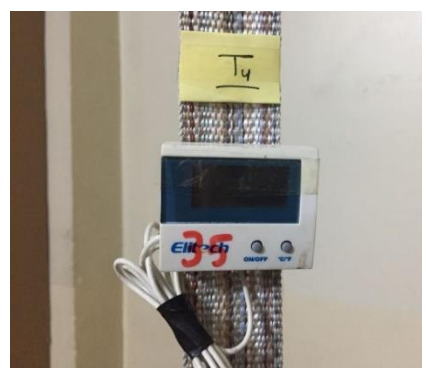

d)

Figure 1. a) experimental chamber b) layout of the tested room c) HT-315 thermocouple d) digital thermocouple

Table 1. Case studies

Case study	Return number	Return location (side wall in front of DV supply opening)
Case - 1	1	In front of supply opening
Case - 2	1	Away from supply opening
Case - 3	2	Both for Case 1 and 2

CFD METHOD

Grid design and grid independent test

In this study the ANSYS ICEM CFD was employed to generate the mesh system. A tetrahedral unstructured grid type was employed to create the required grid for investigated cases. A careful distribution for the mesh density generation was consider in account to cover the interested regions for the simulated room such as region near the indoor heat source and opening outlet. In addition, a required y+, 3<y+<10, was used in this investigation to gain accurate predicted results especially in region near the walls. The air velocity distribution is highly influence by the heat generate from the sources. Therefore, accurate predictions in near wall regions are highly required. For these reasons, the value of y+ should be taken carefully. To check the selected mesh was suitable to achieve a required accuracy of the simulated results, a mesh test was used in this study. 1,750,000 cells were selected to be the best size of the mesh and this was used for the all case studies.

Airflow modelling

A suitable turbulence model is selected to simulate the accurate indoor air movement and air temperature distribution. Therefore, in this study, the two equations renormalized group (RNG) k-ε turbulence model was used to predict the indoor air velocity and movement of indoor air temperature. Most recent researches have been used this model to predict the air movement and indoor air temperature distribution [31-33]. This model gives an accurate simulation results and saving more time [34]. Also, this model can predict the viscosity in region near-wall accurately comparing with standard k-ε model [35-38], and can be expressed as follow [39]:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+P_k+\rho\varepsilon \tag{1}$$

$$\frac{\partial(\rho \varepsilon)}{\partial t}+\frac{\partial(\rho \varepsilon u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}P_k+C_{2\varepsilon}^{*}\rho\frac{\varepsilon^2}{k} \tag{2}$$

where,

$$C_{2\varepsilon}^{*}=C_{2\varepsilon}+\left(C_\mu\eta^3(1-\eta/\eta o)\right)/1+\beta\eta^3 \text{ with } \eta=(Sk/\varepsilon) \text{ and } S=\sqrt{2S_{ij}S_{ij}}$$

The constants used in this study are given in Table 2.

Table 2. Model constants

Constants	C_{μ}	σ_k	σ_{ε}	$C_{\varepsilon 1}$	$C_{\varepsilon 2}$	ηo	β
	0.0845	0.7194	0.7194	1.42	1.68	4.38	0.012

For each case study, the indoor thermal environment was evaluated using ANSYS Fluent. The boundary layers in region close to the walls were calculated by using the enhanced wall treatment. The Boussinesq assumption was also employed in this study. In addition, the semi-implicit method for pressure-linked equations SIMPLE algorithm was used to treat the velocity and pressure coupling. The second order was adopted to calculate all terms in previous equations except the pressure which it calculated via PRESTO. Table 3 lists all required details of the turbulence model and the boundary conditions for the all case studies.

Table 3. Detailed information for the investigated room

Simulation details	
Turbulence model	Renormalized group RNG $k-\varepsilon$.
Radiation model	Discrete ordinates (DO) radiation.
Numerical schemes	For pressure, Staggered third order scheme PRESTO; for other terms, upwind second order; SIMPLE algorithm.
Boundary conditions for the simulated room	
Floor, ceiling, tables and bounded walls	Adiabatic wall
Supply air	Velocity inlet (0.14 m/s, 19 °C)
Exhaust vent	Pressure –outlet
Occupants	heat flux 60 W×2
PC case	heat flux 60 W ×2
PC monitor	heat flux 70 W ×2
Lamps	heat flux 24 W ×2

Validation work

In this investigation an experimental study was performed to validate the accuracy and the ability of the turbulence model in prediction of the indoor air temperature distribution and indoor air movement. In this validation, temperature distribution for the six different locations was used to validate the selected turbulence model. The locations of the measured temperatures distribution for the experimental work were shown in Figure 2.

In addition, the comparison between the simulated and the experimental results for the case 1only was shown in Figure 3. Depending on the Figure 3, a good agreement was achieved between the numerical and the experimental results. This will give approve of the validity of the selected turbulence model to simulate the thermal environment correctly with an acceptable accuracy (error about 6%).

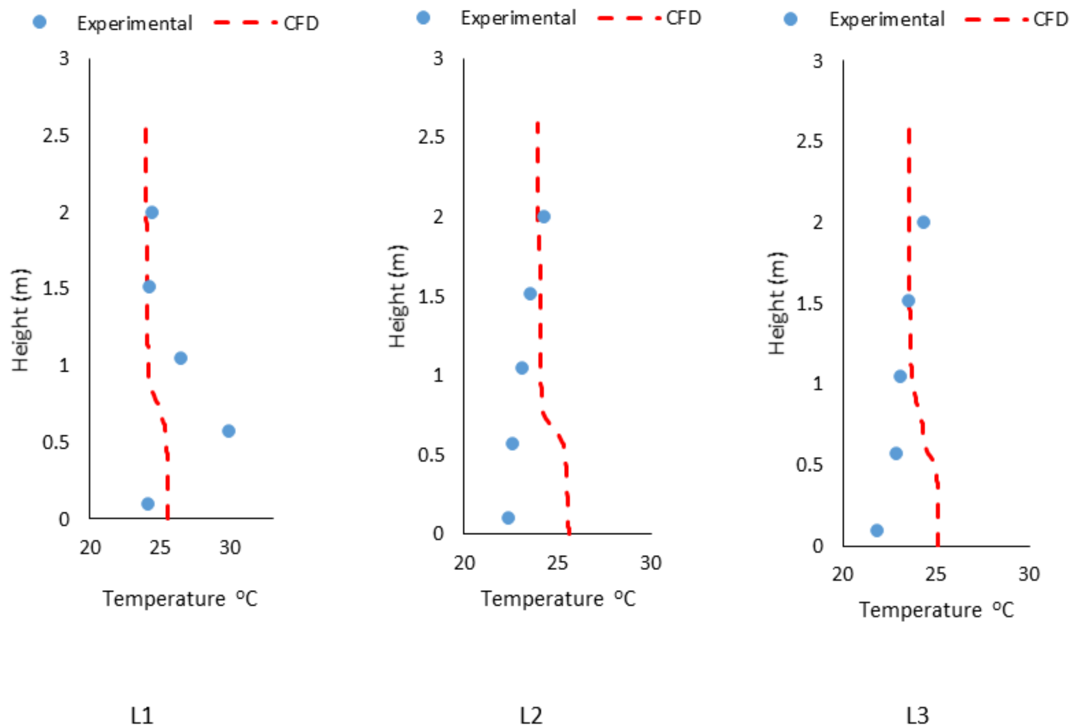

Figure 2. The positions of the measured temperatures

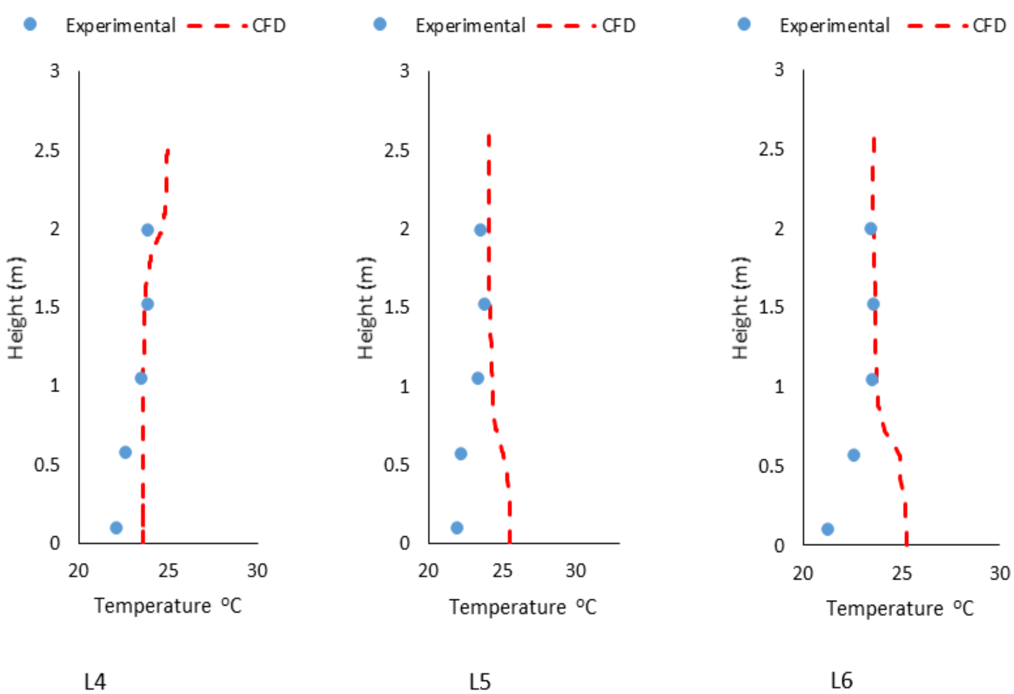

Figure 3. The comparison of the temperature distribution between the experimental and predicted results for case-1

RESULTS AND DISCUSSION

Indoor air temperature distribution

It is very important to create a healthy and comfortable area for the occupants. In order to satisfy this requirement, the temperature distribution indoor especially in occupied zone are very important [31-34]. This will create a thermally comfortable environment. For this reason, the temperature distribution was evaluated in each case study. In this investigation different plane section was used to display the temperature distribution in the investigated room domain.

Figures 4, 5 and 6 show the temperature distribution in different section for the case 1, 2 and 3 respectively. From Figure 4 it is clear to show that there is a noticeable difference between the lower part and upper part of the tested room in both section planes. This was due to that the large amount of the supply air was extracted directly from the exhaust air outlet before mixed with the rest air inside the room. For case 2 when the exhaust opening relatively far away from the supplied opening, the temperature variation between the lower part and upper part was not large compared with case 1. This was because that the supplied fresh air has enough time to circulate inside the room domain before reached to the exhaust opening. Figure 6 shows the temperature distribution when separate the exhaust opening into two opening with the same amount of the extracted air as in case 1 and 2. By comparing with case 1 and 2, a homogenous temperature distribution was found in the room domain. This was due to that the multiple exhaust opening gave the supplied air ability to distribute in all room domains perfectly. This process will create a good air temperature distribution an also provide a better thermal comfort for the occupants compared with case 1 and 2.

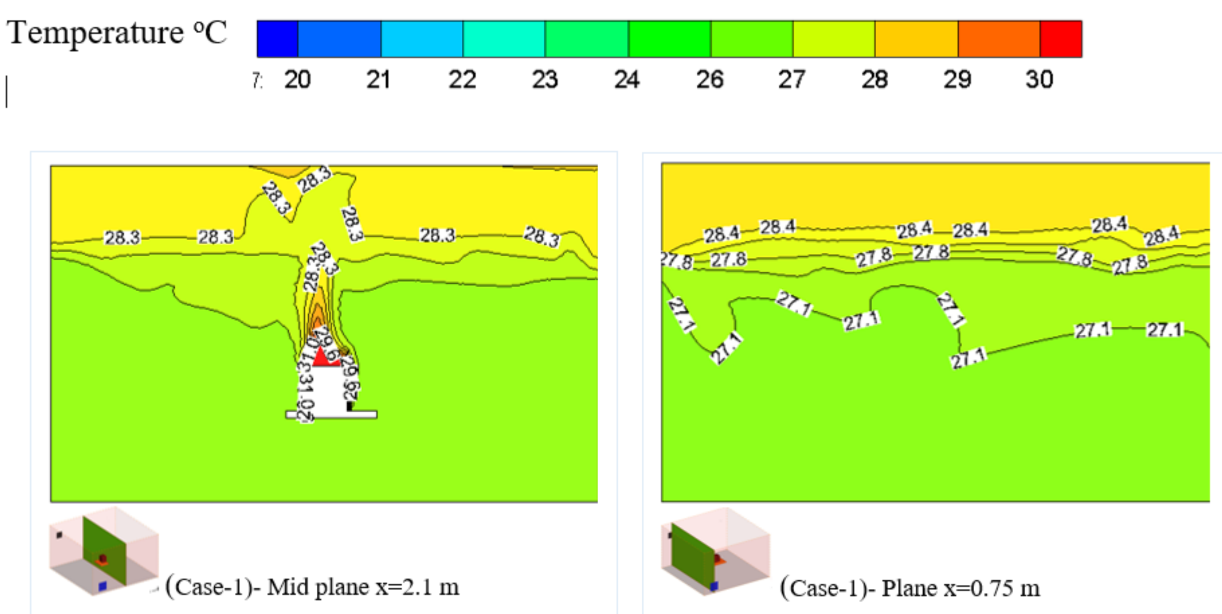

Figure 4. Air temperature distribution at x=2.1 m and x=0.75 m for the case-1

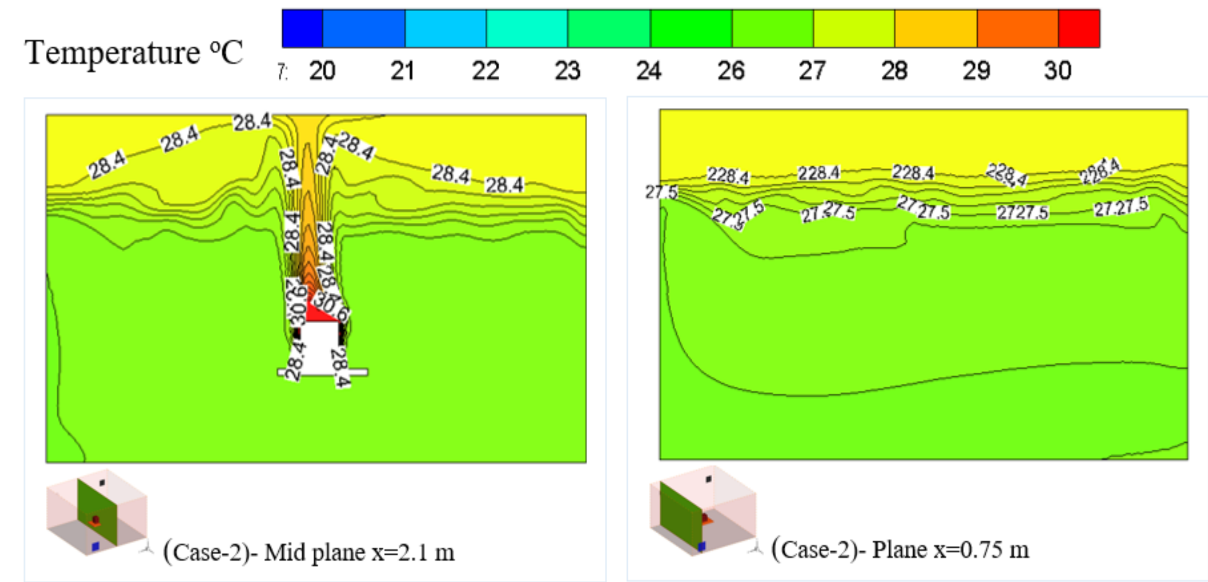

Figure 5. Air temperature distribution at x=2.1 m and x=0.75 m for the case-2

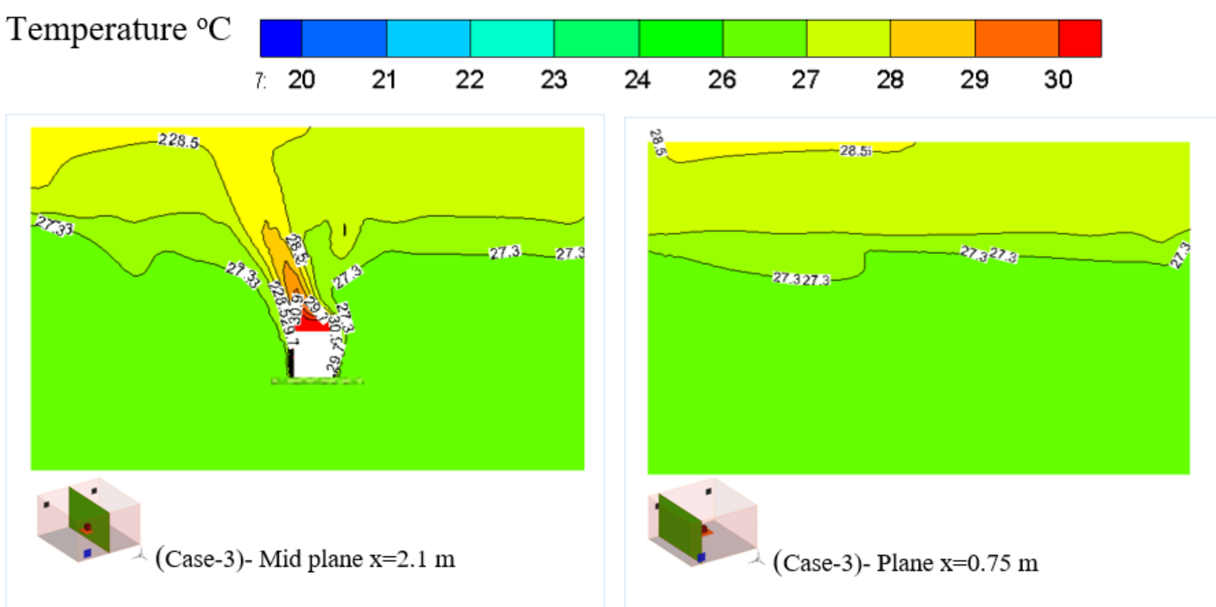

Figure 6. Air temperature distribution at x=2.1 m and x=0.75 m for the case-3

Indoor air movement distribution

The main important parameter for the human thermal comfort evaluations is the air movement distribution. In this work, the distribution of the air velocity was evaluated for the three different case study at different section planes (see Figures 7, 8 and 9). As shown in Figures 7, 8 and 9, the room air velocity distribution in plane x=0.75 m are approximately the same for all case studies. This was because that the supply air velocity for all case study was same and there is no significant difference for the air velocity distribution in this sections plane especially in region near the DV supply opening (see plane x=0.75 m for Figures 7, 8 and 9). For the mid plane, there are a slight difference in air velocity distribution for the case 1 compared with case 2 and 3. In case 1 when the exhaust opening located in front of the supply, the velocity in region of the occupied zone (see mid plan in Figure 7) is higher than other of the room domain. While for case 2 and 3 the air velocity in occupied zone (see mid plan in Figures 8 and 9 respectively) was lower compared with the case 1. Therefore, a good thermal comfort was found in cases 2 and 3.

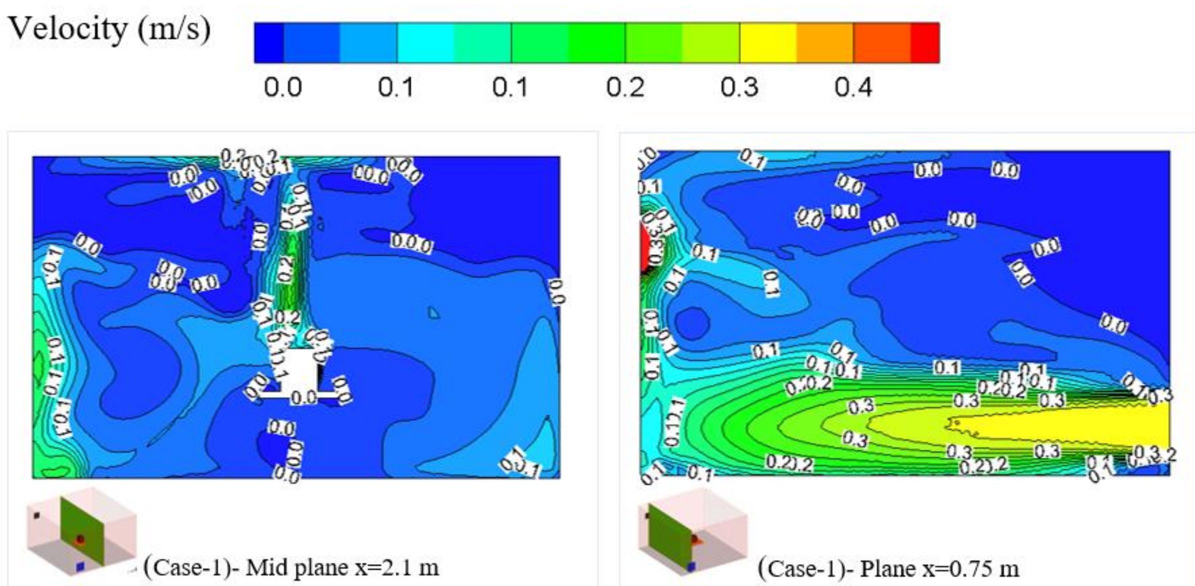

Figure 7. Velocity contour at x=2.1 m and x=0.75 m for the case-1

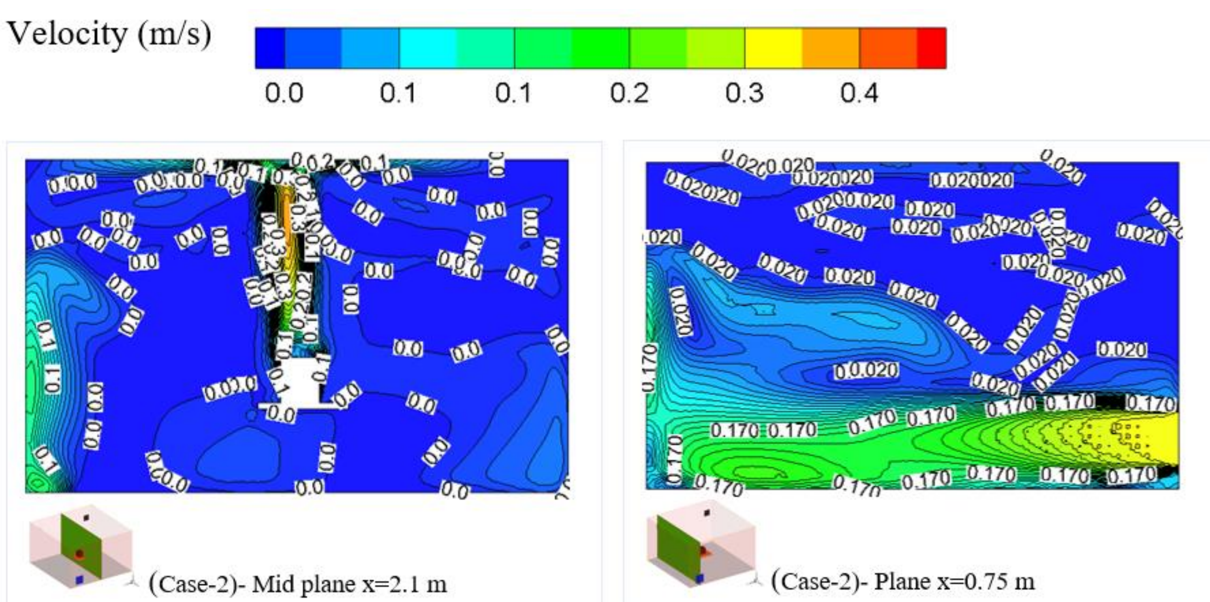

Figure 8. Velocity contour at x=2.1 m and x=0.75 m for the case-2

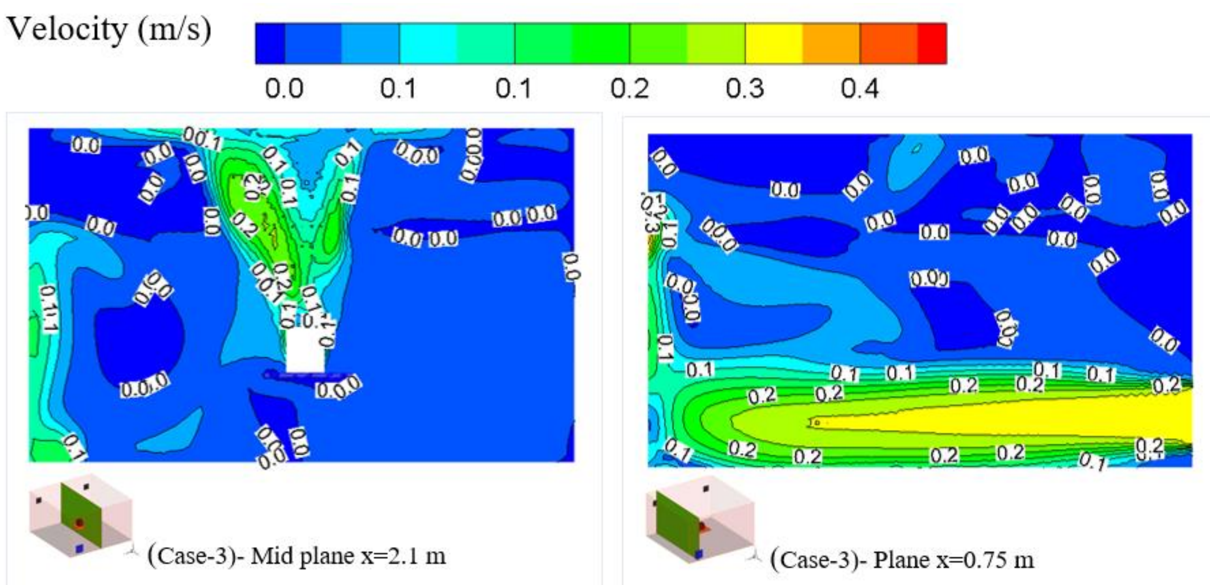

Figure 9. Velocity contour at x=2.1 m and x=0.75 m for the case-3

PMV and PDD evaluation

The indoor thermal comfort is one of the important evaluations index for any ventilation system. For each case study in this investigation, Fanger's comfort equations [40] were employed to evaluate the indoor human comfort. Two indices predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) employed to assess the human thermal balance for the whole body of the occupants. For the acceptable thermal comfort, the required PMV and PPD should be in range of -0.5 to 0.5 for PMV and no more than 10 for PPD [40-41]. The PMV are represented by the seven sensation point. Table 4. lists the PMV and occupant's thermal sensation [40]

Table 4. The PMV and thermal sensation scale

PMV	Thermal sensation
+3	Hot
+2	warm
+1	Slightly warm
0	Neutral
1	Slightly cool
2	Cool
3	Cold

The PMV and PDD evaluations index are influenced by many factors such as metabolic rate, external work, air quality, clothes, temperature, air velocity and mean radiant temperatures. All these factors are used to calculate the PMV and PDD as follow [40]:

$$
\begin{aligned}
\mathrm{PMV} = (0.03e^{-0.036M} + 0.028)\{(M - W) - 3.05 \times 10^{-3} \times [5733 - 6.99(M - W) - p_a] \\
- 0.42 \times [(M - W) - 58.15] - 1.7 \times 10^{-5} M(5867 - p_a) - 0.0014M(34 - t_a) \\
- 3.96 \times 10^{-8} f_{cl} \times [(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] - f_{cl} h_c (t_{cl} - t_a)\}
\end{aligned} \quad (3)
$$

$$t_{cl} = 35.7 - 0.028\,(M - W) - c_{cl}\,\{3.96 \times 10^{8} f_{cl} \times [(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] + f_{cl} h_c\,(t_{cl} - t_a)\} \tag{4}$$

$$\text{and } h_c \begin{cases} 2.38(t_{cl} - t_a)^{0.25} \\ 12.2\sqrt{v_{ar}} \end{cases} \quad \text{for} \quad \begin{cases} 2.38(t_{cl} - t_a)^{0.25} > 12.1\sqrt{v_{ar}} \\ 2.38(t_{cl} - t_a)^{0.25} < 12.1\sqrt{v_{ar}} \end{cases} \tag{5}$$

$$\text{and } f_{cl} \begin{cases} 1 + 1.290\, I_{cl} \\ 1.05 + 0.645\, I_{cl} \end{cases} \quad \text{for} \quad \begin{cases} I_{cl} \le 0.078 \quad (m^2 {}^\circ C/W) \\ I_{cl} > 0.078 \quad (m^2 {}^\circ C/W) \end{cases} \tag{6}$$

Then PPD index can be calculated

$$PPD = 100 - 95 \times e^{-(0.03353 \times PMV^4 + 0.2179 \times PMV^2)} \tag{7}$$

In this investigation the PMV and PPD was evaluated for the working zone as well as for all room domain. Figures 10 and 11 show the calculated values for of the PMV and PPD for all room domain respectively. From these figures, it is clear to see that the PMV and PPD values for the case 1 are higher than in case 2 and 3. This was because that the supplied air in case 1 was extracted directly from the front exhaust opening and this causes short air circuit and may impact on the indoor human comfort. For the case 2 and 3, the well distribution of the air creates an acceptable thermal environment comparing with case 1. In addition, the higher velocity distribution in case 1 (see Figure 7) has a great influence on the indoor air. For the same reasons, the PMV and PPD (for the occupied zone) for the case 1 were higher than in case 2 and 3 as shown in Figures 12 and 13 respectively. A slight difference of the PMV and PPD values were found between case 2 and 3. Where the case 3 consider the best among all case studies. This was due to that the separate exhaust opening provides a very well indoor air and temperature distribution and creates a good indoor thermal environment.

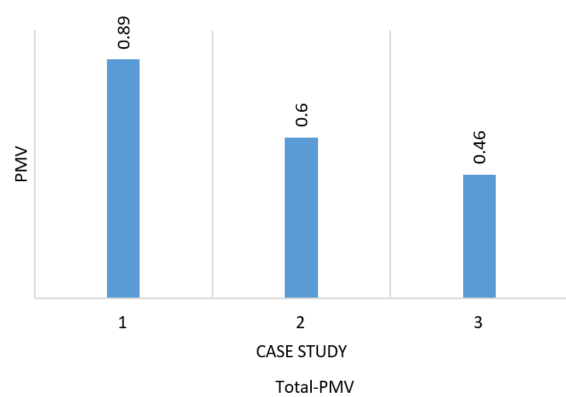

Figure 10. The total PMV evaluation for each case study

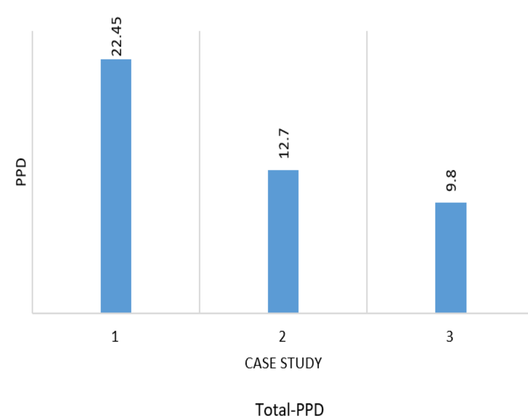

Figure 11. The total PPD evaluation for each case study

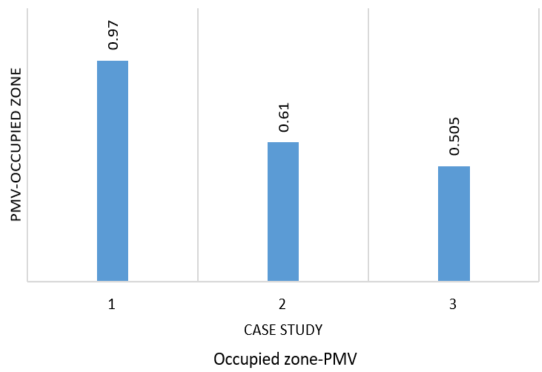

Figure 12. The PMV evaluation at occupied zone for each case study

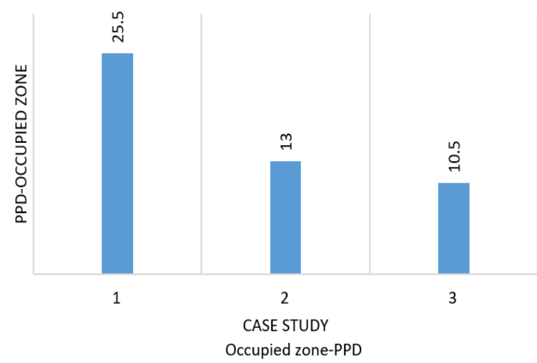

Figure 13. The PPD evaluation at occupied zone for each case study

CONCLUSION

In this research the influence of the exhaust diffuser layout and the amount of the extracted air on the human thermal comfort and thermal environment indoor (velocity and temperature distribution) were studied experimentally and numerically. The finding can be concluded:

- The indoor thermal comfort and the temperature and the velocity distribution were greatly influenced by the exhaust opening locations and the amount of the extracted air.
- A good indoor thermal comfort was found when the exhaust vent located far away the supply opening.
- The best results regarding thermal comfort and indoor thermal environment were found by separate the amount of the extracted air in to two exhaust opening as presented in case 3. This will give the supplied air the ability to distribute inside room perfectly.
- The total PMV and PPD for the case 3 were 0.46 and 9.8 respectively while the PMV and PPD for the occupied zone were 0.505 and 10.5 respectively. Therefore, where the case 3 consider the best among all case studies.
- In order to prevent the air short circuit, the exhaust opening should not locate at the wall in front of the supply opening.

NOMENCLATURE

C_μ	Turbulence model constant
c_p	Specific heat, J/kg-K
p_a	Partial water vapor pressure, pa
f_{cl}	Surface area for the clothed to surface area for the naked body
t_a	Air Temperature, ºC
t_{cl}	Surface temperature for clothing, ºC
$\bar{t}_r$	Radiant mean temperature, ºC
V_{ar}	Relative air velocity
h_c	Convective heat transfer coefficient, W/(m^2. ºC)
t	Clothing thermal resistance, (m^2.ºC) /W
S	Mean strain rate tensor
S_{ij}	Strain rate tensor
V_j	Associated volume with i trajectory and cell j
i, j	Trajectories

Greek symbols

β	Thermal expansion coefficient, 1/K
ε	Turbulent dissipation rate, m^2/s^3
μ	Dynamic viscosity, kg/(m.s)
ρ	Fluid density, kg/m^3
σ_k	Constant for *k* equation of the turbulence model

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