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Effect of opening diffuser and return vent location on air quality, thermal comfort and energy saving in desk displacement ventilation (DDV) system

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ABSTRACT

Many investigations have recently been performed on return vent height and indicate that having 1.3 m distance from the floor is the optimized height for it. In this article, the effect of distance between opening diffuser and return vent on air quality, thermal comfort and energy saving was investigated. According to the results, by increasing the distance between opening and return vent up to 5 m, the return vent could be placed near the floor at height of 0.6 m without any unacceptable consequence in indices. Therefore in this case, energy saving of 15.8% could be achieved rather than 8%, 10.9% and 15.2% in other cases. However, the air quality was lower compared to the other cases. In the case having better air quality and more thermal comfort with acceptable energy saving of 15.2%, the opening and return vent were relocated at maximum distance between them (5 meters) and return vent was placed at the suggested height of 1.3 m from the floor, which was found to be the optimum scenario.

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1. Introduction

In the near future, it is expected that 70% of the world's population residents would live in the cities and will spend 80-90% of their time inside the buildings [1-2]. So, it is essential to employ an efficient method of HVAC technology.

Desk displacement ventilation (DDV) is an air conditioning system which operates like the famous Displacement Ventilation (DV) system with all the same principles [3]. However, the opening diffuser in the DDV system is placed under the worker's desk. The DDV concept, divides the room into two sections, the section which consists the surrounding of the occupant called micro-climate, and the outside of this area called macro-climate (as shown in Fig. 1). In fact, the purpose of this system is to achieve more ventilation efficiency in the breathing zone in micro-climate area with a lower cooling load capacity. Similar to the other STRAD systems, the standard range for the air velocity and air temperature for a DDV system is 0.1-0.2 and 18-20℃, respectively [4]. The cool air which is circulated around the room by an opening diffuser in a DDV system, removes heat and transfers contaminants from the heat sources in the occupied zone (OZ) to the upper zone (UZ) where the warm and polluted air exits through ceiling exhaust vents with a minimal effect on occupant's thermal comfort. By this phenomenon which is caused by buoyancy forces, qualified air would be circulated properly over the breathing zone and increases indoor air quality (IAQ) and

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also leads to ventilation efficiency with a lower draft discomfort compared to MV systems [5-7]. Several studies have been done for comparison between MV and STRAD systems. Shan et al. [8] by comparing between MV and DV systems reported that the draft risk in an MV system was more than a DV system in the room. However, the draft risk in the foot area or lower part of the body was higher in the DV system. Chen et al. [9] analyzed the performance of MV and DV systems on energy saving, air change efficiency and the heat removal by a numerical study. The results showed the improvements on indices by use of the DV system. Schiavon et al. [10] investigated the influence of draft at uncovered ankles for women by a displacement ventilation system. They found that the reaction of draft discomfort around the occupant's ankle in DV systems is more than what is expected. The same conclusion was reported by Kavgic and Mumovic [11].

The research background on optimization or development of the DV system, is not limited to the DDV system.

Hweij and Ghaddar [12] optimized the DV system performance by adding the chair fan. They found that the height of 1.0 m and 0.3 m for the fans from the floor, would respectively decrease 20.6% and 11.6% of energy consumption, compared to a normal DV system. By applying a similar protocol, Fil et al. [13] optimized the ceiling personalized ventilation (CPV) on IAQ by analyzing the effect of height and flow rate of a chair fan.

The optimization of indoor ventilation systems does not only depend on the new ideas in HAVC technology since the location of an opening diffuser, an exhaust or return vent and relocating the heat sources are effective on thermal stratification and on indices. Lin et al. [14] investigated the effect of the opening diffuser and the exhaust location in a DV system. They found that the position of an exhaust vent has a minor influence on the thermal comfort. Furthermore, they suggested that the supply diffuser should be located near the middle of the room for a better thermal stratification. Raftery and Bauman [15] tested the performance of UFAD and DV systems. The result indicated that both UFAD and DV systems were based on a similar principle. However, the UFAD diffusers had two advantages compared to the side wall displacement diffusers. First of all, it was appropriate for an open plan office and they weren't limited by position of the wall like DV diffusers. On the other hand, they could be located nearby the occupants. In a DDV system, the displacement diffuser is close to the occupant and it is not limited to side walls and can be relocated in the center of the room with the aim of ducting inside the desk.

Heidarinejad et al. [16] studied the effect of return vent height form the floor. They found that the height of 1.3 m from the floor satisfies the occupant's thermal comfort within an acceptable energy consumption compared to an

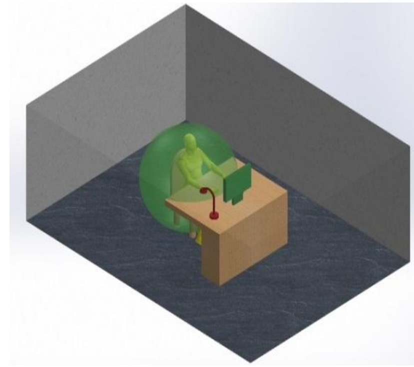

Figure 1. A shematic view of the desk displacement ventilation concept and the defenition of micro/macro-climate area

MV system. Also, they concluded that the lower height of the return vent, would significantly improve the energy saving. However, it caused the thermal discomfort and a lower air quality. Ahmad et al. [17] performed a numerical study on exhaust vent locations on energy saving. Their results indicated a significant improvement achieved by the combination of an exhaust vent and the heat sources such as ceiling lamps. The most amount of energy saving was obtained when the exhaust vent was combined with the light slots. Also, it improved the IAQ in the breathing zone by analyzing the CO2 concentration. Furthermore, they proposed the optimum flow rate of the return vent in another numerical study [18]. They considered three flow rates for the return vent. The case studies included 35%, 50% and 65% of the total indoor air flow. The result showed 61.4% improvement in the IAQ at the breathing zone, when 65% of total air flow went through the return vent. Also, it decreased 30% of energy consumption compared to an MV system.

As previously mentioned, the location of vents or diffusers are critical for achieving an optimum performance of a ventilation system. Also, the significant efficiency requires a new idea for the air conditioning systems. In this article, the performance of a DDV system is investigated with the separate return vent strategy. Previous studies on DDV systems were conducted in absence of an air recovery policy. Also, there is no study to clarify the effect of the relative distance between return vent and opening diffuser. Consequently, it was tried to determine if the air recovery policy is suitable for a DDV system or not. The other purpose was to find the best possible distance between a return vent and an opening diffuser in order to improve the IAQ, ITC and energy saving.

1. Methodology

2.1. Physical model

The model used for the validation was based on the Loomans experiment [3]. This system was tested numerically and experimentally with different temperatures and flow rates. The chamber, as shown in Fig. 2, was 5.16 m long, 3.6 m wide and 2.5 m high. The room heat sources include; 2 PC-simulators, three lamps, one occupant and one light simulator. The cooling load of each heat sources are listed in Table 1.

The geometry of mannequin and the other heat sources were assumed to be as simple as possible in order to generate the structured/hexahedral grids throughout the room. The human body was assumed to produce 76 W/m^2 heat flux in the room.

2.2. Governing Equations

The prediction of the temperature gradient, velocity field and the other parameters in the room was calculated by the aid of computational fluid dynamics (CFD) techniques. The continuity, momentum and energy conservation equations were assumed to be in a three dimensional turbulent and steady state air flow. Reynolds average method (RANS) in Cartesian coordinates was employed to relate the Reynolds stresses to the mean velocity gradients. Calculations are performed for all structural cubic cells in three dimensions.

Continuity equation:

$$\nabla \cdot (\rho \vec{v}) = 0 \tag{1}$$

Where, ρ and $\vec{v}$ are the density and velocity vector of air, respectively.

Momentum equation:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla P + \nabla \cdot (\bar{\bar{\tau}}) + \rho \vec{g} \tag{2}$$

Where P is the static pressure of air, $\rho \vec{g}$ is the gravitational body force and $\bar{\bar{\tau}}$ is the stress tensor.

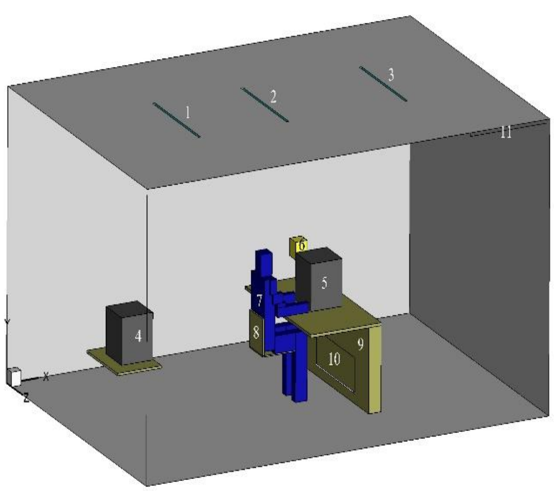

Figure 2. Configuration of the test chamber; 1-3 lamps, 4-5 Pc simulator, 6- light simulator 7- occupant, 8- chair, 9- supply unit, 10- displacement diffuser, 11- exhaust vent.

Table 1. The cooling load of heat sources

Heat sources	Light simulator	Lamps	PC simulators	Sum
Cooling load (W)	25	36×3	100×2	333

Energy conservation:

$$\frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\rho h \vec{v}) = \nabla \cdot [(k + k_t)\nabla T] + S_h \tag{3}$$

Where h is the sensible enthalpy ($h = \int_{T_{ref}}^{T} C_p dT, T_{ref} = 298.15\,K$), k is the molecular conductivity, k_t is the conductivity due to turbulent transport ($k_t = \frac{C_p \mu_t}{Pr_t}$), and S_h is the source term which includes all defined volumetric heat sources.

The indoor zero equation was selected for calculating the turbulent viscosity in the room. The following equation was generated by Chen et al. [19] and it was successfully examined for modeling the indoor environments:

$$\mu_t = 0.03874\ \rho \cdot v \cdot l \tag{4}$$

Where ρ is the fluid density, v is the local velocity, l is the distance from the nearest wall and 0.03874 is an empirical constant.

For simulating the radiation, the Discrete Ordinates (DO) radiation model was utilized to compute the radiation between heat sources and objects in the room. The discrete ordinates (DO) radiation model solves the radiative transfer equation. The SIMPLE algorithm with finite-volume was used with a second order upwind scheme for the convective terms. The calculation was done by the precise and fast responding software, Airpak-Fluent [20].

2.3. Boundary Conditions

The boundary conditions which were applied for the external walls were dirichlet boundary conditions. The temperature and inlet supply flow rate were 19.8 ºC and 0.047 m^3/s, respectively. The mentioned condition was the highest tested air speed compared to the other cases [3].

The walls boundary conditions, listed in Table 2 and Table 3, briefly present the governing conditions and numerical methods for the solving the problem.

2. Validation of CFD results

The precision of the numerical simulation is an important issue that must be checked. One of the basic checkpoints is grid independency. After this process, the results should be validated with an experimental reference. The important factors for a precise simulation are the geometry of the cells, number of cells and the structured/unstructured grids. Three dimensional structured hexahedron grid, as shown in Fig. 3, was considered for generating the mesh throughout the room.

In order to achieve better results, the number of cells were risen near the heat sources, near the occupant and higher or lower points of the wall. As mentioned, the experimental study performed by Loomans [3] on the DDV system was selected to validate the numerical model. The results of the temperature validation are shown in Fig. 4 along 4 poles. The maximum calculated error between numerical and experimental data of the temperature profile was under 2.8%.

The number of cells tested as grid independent solution were 923909, 1384759 and 1717253. There was no significant change in temperature profile by increasing from 1384759 cells to 1717253 cells as shown in Fig 5 along 2 poles.

3. Result and discussion

3.1. Case studies description

The aim of this article is to optimize the distance between return vent and the opening diffuser. Also, the performance of the DDV system along a separate return vent strategy was evaluated. The position of the return vent and the opening diffuser in the room are listed in Table 4 for each case study. Furthermore, a schematic view of all case studies are shown in Fig. 6. In order to determine the optimum case, Fanger's thermal comfort criteria, local thermal discomfort, air quality and energy consumption were evaluated. According to Table 4, four scenarios were prepared for this research. In case 1, the distance between return vent and the opening diffuser was considered to be close to each other, with the height of 1.9 m from the floor for the return vent.

In case 2 and case 3, the height of the return vent was 1.3 m from the floor and the only difference between them, was the place of the opening diffuser. Actually, case 2 and case 3 were defined for analysing the effect of the opening diffuser position on the indices. Case 4 presented the low height return vent strategy for the room. The main reason of testing this scenario was find if it is possible to use the low height return vent for an office while the opening diffuser is placed with the maximum distance of the return vent.

3.2. Fanger's criteria evaluation

In the case of indoor thermal comfort for the CFD simulations, the classic steady-state model by Fanger was used [21]. Air flow Temperature/velocity, mean radiant of the surrounding surface and the relative humidity are the variables of the Fanger's model. Furthermore, two personal variables of clothing insulation and metabolic rate are involved. The result is known as the Predicted Mean Vote (PMV). Predicated Percentage of Dissatisfaction (PPD) developed by Fanger only evaluates the individual vote of the occupants.

According to ISO7730 [22], the acceptable value of a PMV is between -0.5 and +0.5 and the proper value of a PPD is between 0 and 15%. If the PMV and the PPD values are placed between the mentioned ranges, 90% of the residents will be thermally satisfied [23]. In this article, Cloth isolation was considered to be 0.7 for calculating the PMV and the PPD criteria. Fig. 7 shows the area which is considered for calculation of the thermal comfort indices.

According to the results listed in Table 5, all of the cases were in the defined range of ISO7730. The most important results were obtained from case 1 and 4. Case 1 revealed if the supply diffuser was placed very close to the return vent, and the return vent was placed at the height of 1.9 m from the floor, occupants were still thermally satisfied. Consequently, the height of 1.9 m of the return vent was the safe position without considering the position of the supply diffuser. Also, by increasing the distance between return vent and opening diffuser with the maximum value (5 meters), the low height return vent strategy prepared the thermal comfort in case 4.

Table 2. The boundary condition of the walls

Wall	East	South	North	Floor	Ceiling	West
T (ºC)	22.3	22.8	23.2	22.2	22.3	22.3

Table 3. Summary of the governing conditions and numerical methods

Turbulent model	Indoor zero equation
Radiation model	Discrete ordinates (DO) radiation model
Numerical schemes	upwind second order; SIMPLE algorithm
Supply diffuser	$\dot{Q}$Supply = 0.047 m^3/s, T = 19.8℃
Return Vent	$\dot{Q}$Return = 0.03055 m^3/s ($\dot{Q}$Return = 65% · $\dot{Q}$Supply)
Exhaust	Pressure-outlet

Table 4. Deails of considered case studies

Case study	Distance between western wall and opening diffuser (m)	The height of return vent (m)
1	2.60	1.9
2	3.32	1.3
3	5.00	1.3
4	5.00	0.6

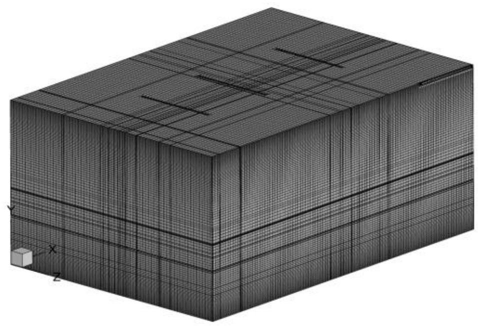

Figure 3. The grids schematic view of 1384759 meshes

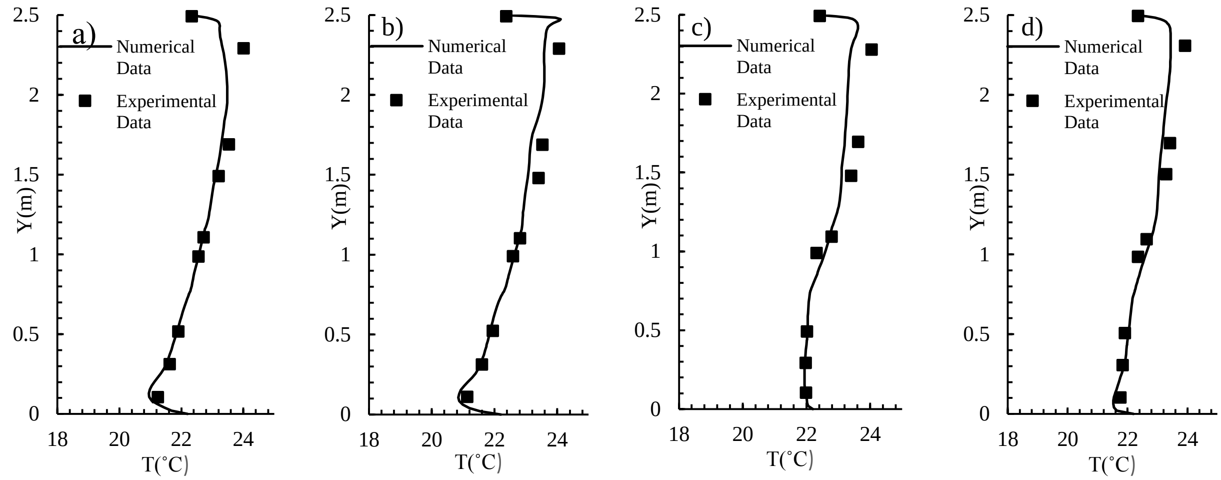

Figure 4. Comparison between CFD simulation and experimental data of temperature gradient along; (a) x=1.5m and z=0.68m, (b) x=1.5m and z=2.43m, (c) x=3.75m and z=1.93m, (d) x=3.75m and z=2.93m

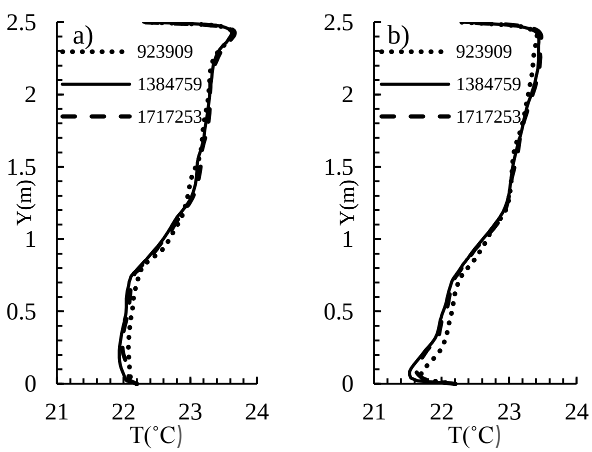

Figure 5. the grid independent test along; (a) x=3.75m and z=1.93m, (b) x=3.75m and z=2.93m

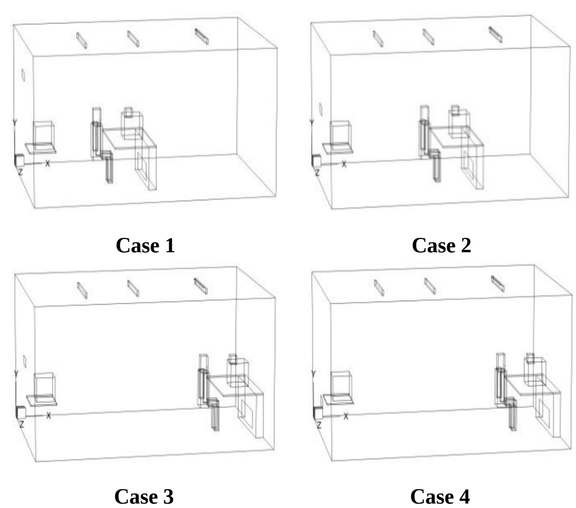

Figure 6. Case studies configuration

3.3. Analysis of the temperature stratification

Basic thermal comfort criteria is a necessary condition for thermal comfort, but it is not sufficient. Since it is possible that the value of both criteria are placed in the defined range while occupants still feel dissatisfaction. The main reason for this problem is the vertical air temperature difference in the space. According to ISO7730 [4], the difference between occupant's head and feet temperature should not exceed by 3ºC. Otherwise, occupants will experience local thermal discomfort. The temperature gradient in front of the occupant is shown in Fig. 8 for all of the case studies. For investigating the local thermal discomfort, the heights of 0.1 m and 1.1 m from the floor were considered as the temperature of the occupants feet and head, respectively. The difference between two values was considered as the local thermal discomfort. According to the results listed in Table 6, the unacceptable value was occurred in case 2. Ín order to reduce the temperature around the occupant's head, the return vent had to be placed at a higher level or the distance between return and opening diffuser had to be increased.

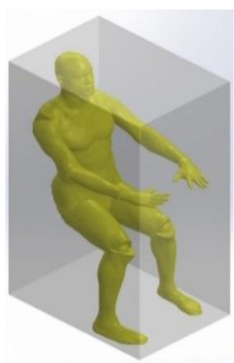

Figure 7. The area which is considered for calculation of the PPD and PMV indices

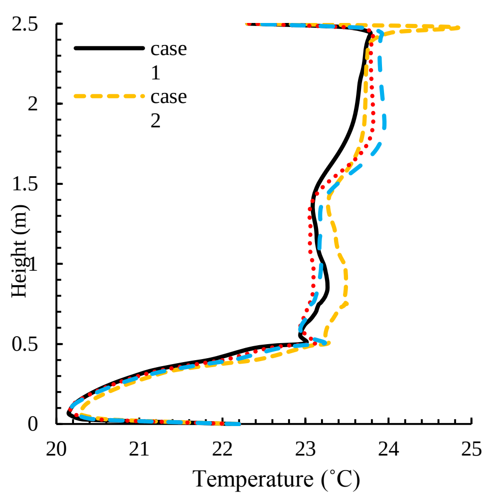

Figure 8. The temperature gradient in front of the occupant

Table 5. PMV-PPD values of case studies

Case study	PMV	PPD
1	-0.18	12.30
2	-0.16	12.34
3	-0.19	12.10
4	-0.16	11.90

Table 6. The result of local thermal discomfort

Case study	Tfeet(ºC)	THead(ºC)	ΔT(ºC)
1	20.21	23.17	2.96
2	20.35	23.40	3.05
3	20.22	23.06	2.84
4	20.23	23.17	2.94

In case 1 and 4, the local thermal discomfort was 2.96 and 2.94, respectively. It is clear that these two scenarios were in the critical risk of local thermal discomfort. However, the important thing was that by creating the maximum distance between opening diffuser and return vent (5 metres), the local thermal discomfort was still in an allowable value range and the return vent could be placed nearby the floor at the height of 0.6 m from the floor. The lowest temperature difference between feet and head level of the occupant was achieved in case 3 with 2.84 (ºC). In order to determine the opening position effects on the local thermal discomfort, cases 2 and 3 were analysed. Regardless of the height of the return vent, by increasing the distance between opening and return vent, the local thermal discomfort was improved. The temperature distributions of all cases are shown in Fig. 9. It is clear that the heat loads from the heat sources, in case 3, were removed more effectively compared to the other cases.

3.4. Air Quality

In this article, the local Mean Age of Air (MAA) was evaluated as the determination of the air quality in the room. MAA is the average age of the air in a certain position in space, when the air is sent into space for the first time. MAA could be calculated via the following equation [24].

$$\frac{\partial}{\partial t}(\rho\tau) + \frac{\partial}{\partial x_j}(\rho u_j \tau) = \frac{\partial}{\partial x_j}(\Gamma \frac{\partial \tau}{\partial x_j}) + \rho \quad (5)$$

In which, $\tau = 0$ and $\partial\tau/\partial x_j = 0$ represent the boundary conditions of the air inlet and the exhaust vent, respectively. The result of the MAA in inhaled zone are listed in Table 7.By increasing the distance between opening diffuser and the return vent, the local MAA in micro-climate was increased. Case 1 was the best scenario in air quality in the breathing zone sincethe airflow first passed the 2.60 m distance between opening and the western wall and then came back

As a result, it passed the least possible distance and the MAA was younger compared to the other cases. Therefore, the longer the distance between the opening diffuser and the western wall, the older the MAA was circulated over the breathing zone.

However, by considering the air quality of the entire room, case 3 was the optimum case. The distribution of the MAA is shown in Fig 10. In case 3, the airflow was circulated more easily throughout the room rather than the other cases. It was the reason that the mean age of air behind the occupant's desk significantly increased in cases 1 and 2. However, even by creating a maximum distance between opening diffuser and the return vent, the low height return vent strategy in into the inhaled zone.

case 4 caused the fast exit of the fresh air throughout the room, and it increased the MAA in both the inhaled zone and the entire area.

3.5. Energy saving

The main purpose of this section is to explain the method of calculatiing coil-cooling load in a DV system by comparing it with an MV system.

For mixing ventilation, the cooling coil load could be written as:

$$Q_{coil} = Q_{space} + Q_{vent} \tag{6}$$

Where Q_{space} (W) and Q_{vent} (W) are the space cooling load and the ventilation load, respectively. The following equation evaluates the space cooling load in a DV system which is suggested by Cheng et al. [25].

$$Q_{space} = C_p \times \dot{m}_r \times (T_r - T_s) + C_p \times \dot{m}_e \times (T_e - T_s) \tag{7}$$

Where, $\dot{m}_r$ and $\dot{m}_e$ (kg/s) are the return and exhaust flow rate, respectively. T_r is the outlet air temperature of return vent, T_e is the exhaust air temperature and T_s is the supply air temperature. The coil-cooling load is calculated in terms of the following equation:

$$Q_{coil} = Q_{vent} + Q_{space} - C_p \times \dot{m}_e \times (T_e - T_{set}) \tag{8}$$

Table 7. Local mean age of air (s) at micro-climate area

Case study	**1**	**2**	**3**	**4**
τ (s)	133	233	305	427

Table 8. The reduction of the energy consumption

Case study	**Tr** (℃)	**Te** (℃)	**(W)** ΔQ_{coil}	$\Delta Q_{coil}/Q_{space}$
1	23.11	23.06	13.85	8.0%
2	22.85	23.16	18.00	10.9%
3	22.81	23.61	26.10	15.2%
4	22.38	23.66	24.60	15.8%

Where T_{set} is the set-point temperature at the specified height from the floor. By Comparing Eq (6) with Eq (8), the term $C_p \times \dot{m}_e \times (T_e - T_{set})$ is the saving energy of coil capacity of a DV system compared to an MV system [25].

Table 8 illustrates the energy reduction by a DDV system compared to an MV system. It is clear that by reducing the height of the return vent, the value of T_r in Eq (7) has been reduced. Therefore, Q_{space} has decreased.

The term $\Delta Q_{coil}/Q_{space}$ is the energy saving achieved by each case studies. So, by decreasing the Q_{space}, the energy saving of the system increased. Case 4 had the highest energy savings among the other cases. Also, by comparing cases 2 and 3, in coul be decraed that the more gap between the opening diffuser and the return vent, the more reduction in the energy consumption. The reason was an increase in the outlet air temperature of the exhaust vent. Case 1 had the lowest potential for the energy savings. Cases 3 and 4 with 15.2% and 15.8%, respectively, were the optimal energy saving scenarios.

4. Conclusion

In this study, the effect of the distance between the opening and return vent in a disk displacement ventilation system was simulated numerically. The following concluding marks summarizes the results:

- By analyzing the Fanger's thermal comfort and local thermal discomfort, case 2 with a spacing of 3.32 m between the opening and the return vent was not sufficiently spaced, and if it was located at this distance, the return vent had to be at a higher height than 1.3 m. Furthermore, against the previous studies, it was possible to create a 5-meter gap between the return and the opening diffuser and put the return vent at the height of 0.6 m without any thermal discomfort.

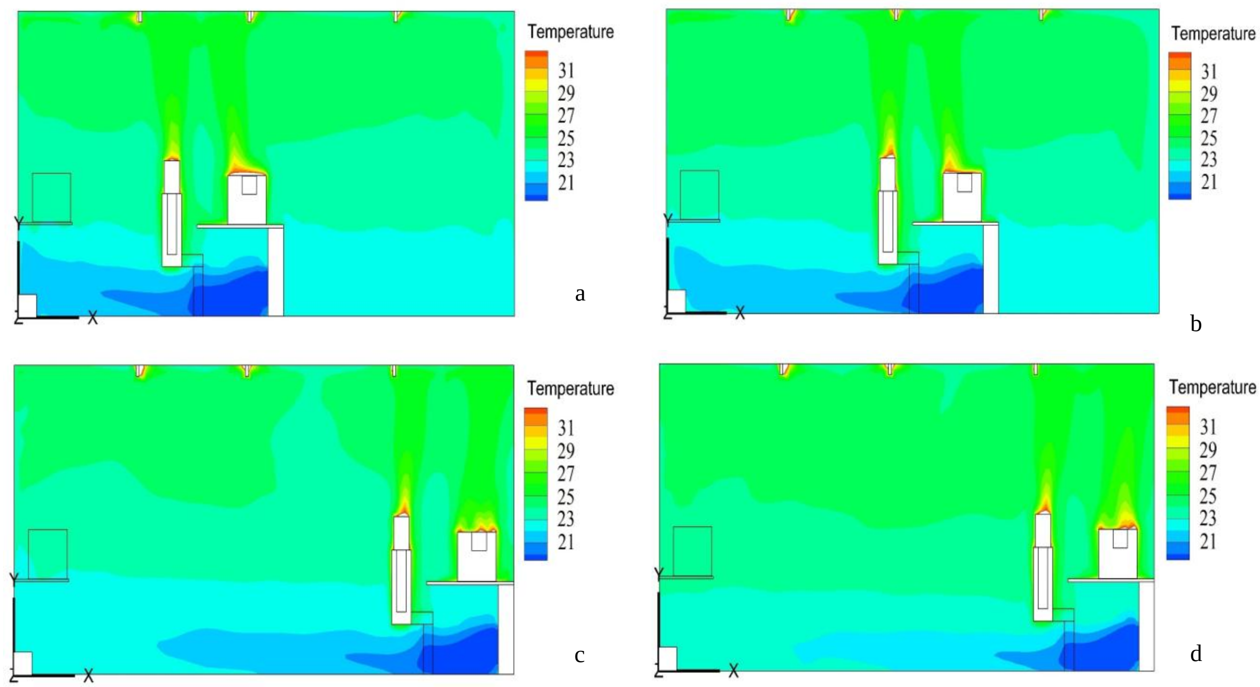

Figure 9. the temperature Distribution (°C); (a) case 1, (b) case 2, (c) case 3, (d) case 4

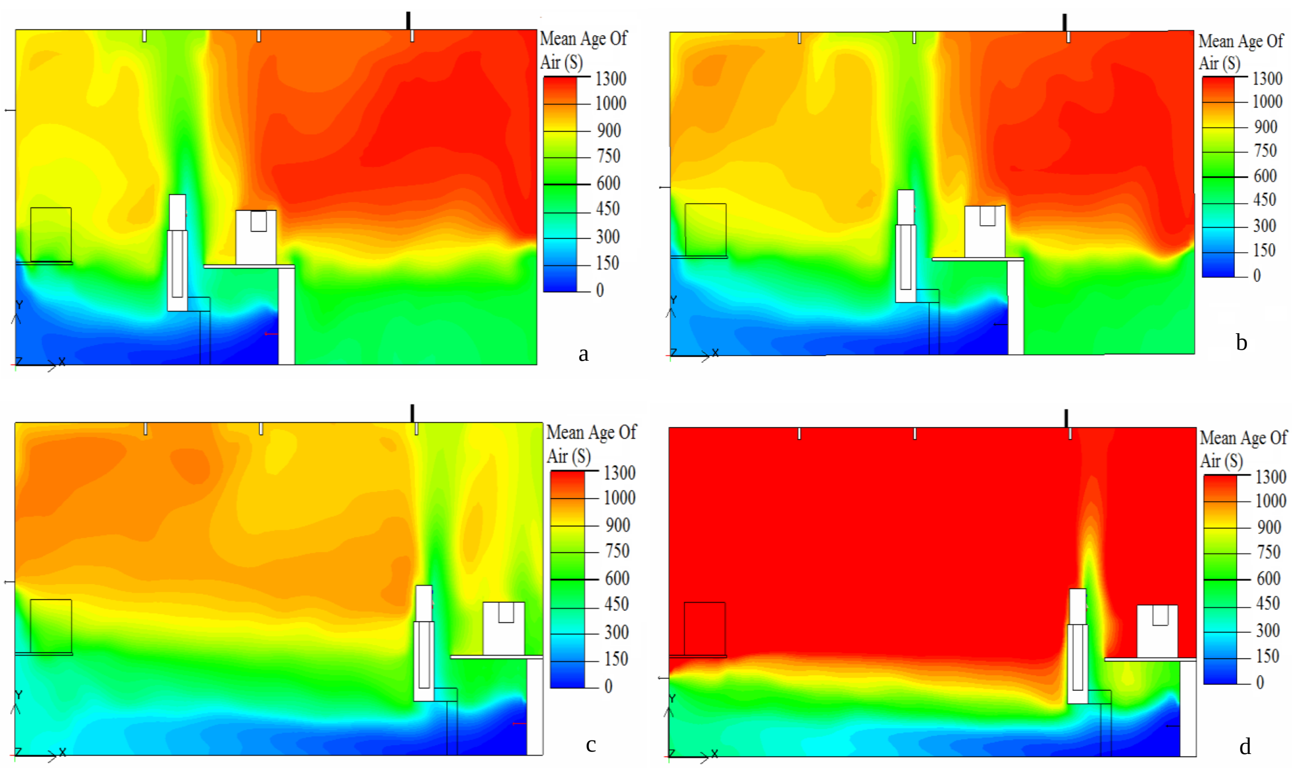

Figure 10. The Mean age of air of the room; (a) case 1, (b) case 2, (c) case 3, (d) case 4

- By evaluating the air quality, it was found that case 1, which was 2.60 m away from the return vent, was more favorable than the other cases. However, considering the entire room, case 3 was the optimum case. Case 4, had the lowest air quality in both inhaled zone and entire area.

- By examining the amount of energy savings, it was found that by increasing the distance between opening and return vent, system worked more economically. The highest energy savings was related to case 4 with a 15.8% reduction in energy consumption.

- By aggregating all of the indicators, case 3 had more acceptable results than the other cases. Consequently, in DDV systems, opening had to be relocated with the maximum distance from the return vent and the return vent had to be placed at the height of 1.3 m from the floor to preserve all of the qualities of a DDV system in presence of a separate return vent.

Nomenclature

ρ	Density of air (Kg/m^3)
P	Static pressure of air (Pa)
$\vec{g}$	Gravitational acceleration (m/s^2)
$\bar{\bar{\tau}}$	Stress tensor (Pa)
h	Sensible enthalpy (J)
c_p	Specific heat capacity at constant pressure(J/K)
Γ	effective diffusion coefficient
τ	Mean age of air (s)
Q_{Vent}	Ventilation load (W)
Q_{space}	Space cooling load (W)
Q_{Coil}	Cooling coil load (W)
$\dot{m}_r$	Return mass flow rate (Kg/s)
$\dot{m}_e$	Exhaust mass flow rate (Kg/s)
T_r	The return temperature (°C)
T_e	The exhaust temperature (°C)
T_s	The supply flow temperature (°C)
T_{set}	Set point temperature (°C)

Abbreviations

DDV	Desk Displacement Ventilation
BZ	Breathing Zone
HAVC	Heating, Ventilation and Air Conditioning
PMV	Predicted Mean Vote
PPD	Predicted Percentage Of Dissatisfied

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