Effects of supply Diffuser Shape on Air Age and Human Comfort in an Office with Displacement Ventilation and Chilled Ceiling System in Hot and Dry climate

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Abstract In recent years, the need to provide a clean environment in enclosed places has drawn increasing levels of attention due to the spread of epidemics and viruses across the world. Chilled ceilings with displacement ventilation represent promising technology. The combined ventilation system has not yet been used in Iraqi buildings, then it’s a good starting point for study the performance of this system in Iraq-Hilla city climate (hot and dry climate). An office room with a combined ventilation system was simulated using AIRPAK software. Indoor air age, air temperature distribution, CO₂ concentration and thermal efficiency were estimated numerically. During the simulation process, the effects of two different supply diffuser shapes (one way rectangular diffuser and semicircle diffuser) were compared to investigated the effect of supply diffuser shape on air age and thermal environment in hot and dry climate (passed on peak summer temperatures in Hilla city). Three tests were performed for each case based on changing the load treated by the chilled ceiling as (25%, 50% and 80%) respect to total cooling load and represented (26, 53, 85W/m²) respectively based on floor area. The temperature of chilled ceiling surface varied as (21.5, 19 and 16°C) and the air supply temperature similarly varied (19, 22, and 24.5 °C) respectively at constant air flowrate of (0.045m³/s). The results of the tests found that the effect of diffuser shape on the local age of air decreased with height, the maximum difference between two cases at the 0.1m level was about 49s while at 2.25m level it was only about 11s. The air exchange efficiency for the CC/DV system using a semicircle diffuser in the predetermined office space was higher by an average of 8.5% as compared with a similar system using a rectangular diffuser; this differential reduced about 3% at each increase in portion of cooling load treated by the chilled ceiling, however.

Keywords: chilled ceiling; displacement ventilation; age of air; air exchange efficiency; indoor air quality

1. Introduction
Humanity suffers in most countries of the world from the spread of viruses that cause many diseases. In addition, a healthy environment and thermal comfort improve efficiency of the work of its occupants
[1]. Achieving this can impose great challenges on researchers, however, who must find new systems to ventilate enclosed spaces at minimal cost.

One of these new ventilated system is combined between displacement ventilation and chilled ceiling system. This system has the ability to save energy and improve the quality of indoor air due to energy saving feature in displacement system [2 and 3] and the ability of chilled ceiling to provide thermal comfort [4].

Most of the results of several previous studies examining both chilled ceiling and displacement system found that the system improved the cooling efficiency [5 and 6]. Chilled ceiling offers many advantages over a traditional air conditioning systems. It’s better in energy consumption, thermal comfort, and no noise due mainly energy transfer by radiation and convection [7]. The cooled water temperature used by the chilled ceiling is relatively high, this gave advantage to system in used natural or free water cooling, that’s led to saving in energy. [8 and 9]. Because of all these advantages, the chilled ceiling system had become widespread in the last years [10 and 11]. One of the problems associated with the chilled ceiling was inability to control the humidity level [1]. Fresh air can be supplied to do this by mixing ventilation or displacement ventilation with chilled ceiling systems, though many researchers favoured displacement ventilation systems due to their improved IAQ [12]. The chilled ceiling is thus treated part of the sensible cooling load and the displacement system treats only the remaining sensible load and the latent cooling loads [13].

Many researchers had studied the (CC/DV) system as Hao. et. al. [1], Kim and Leibundguta [14], Muslmani. et.al[15], Tian. et. al. [16] and Jin. Et. al. [17] were studied the several methods to avoid condensation problem on chilled ceiling. Bahman et. al. [18], Chakroun et. al. [19], Itani. et. al. [20], and Seblany. et. al. [21] investigated the saving energy in the combined system. Ayoub. et. al. [22], Rees and Haves. [23], Mateus and Grac. [24], and Krajcik. et. al. [25] studied the effect of used chilled with displacement ventilation ceiling on the air temperature and velocity discerption. Yang. et. al. [26] investigated the best location of the cold ceiling in an office room and found that the best location at ceil.

Most of the above studies don’t focus directly on the shape of air supply diffuser and air age in hot and dry ventilated zone, although the study of age of air inside closed rooms can provide us with many data as evaluating air quality in an indoor environment, refers to the effectiveness of replacing the air in the room by fresh air from the ventilation system and assessing air pollution control [27 and 28].

Chilled ceiling accompanied with displacement ventilation system not used in Iraqi building, then this study would be a good starting point for study the performance of this system in (hot and dry climate such as that found in Hilla in Iraq).

The main objective of this work to investigate numerically the effect of air supply diffuser shape on the air age and the performance of CC/DV system under hot and dry climate (at maximum temperature in Iraqi-Hilla city summer) at deference ratio of cooling load treated by chilled ceiling.

2. Cases Description

Two cases are studied for the office room to satisfy thermal comfort under Iraqi-Hilla climate for different types of supply air diffuser (Case-I under rectangular diffuser and Case-II under semicircle diffuser) each case have three tests at deferent ratio of cooling load treated by chilled ceiling (η) as (25%, 50% and 80%) . The results is compared with ASHREOA standards data.
Where \( \eta \) represented the ratio of the cooling load which treated by chilled ceiling (CL\text{cc}) to the total office room load, Equation (1) [29].

\[
\eta = \frac{CL_{cc}}{CL_{DV} + CL_{cc}}
\]  

(1)

The age of air, air temperature and velocity distribution and concentration of carbon dioxide (CO\textsubscript{2}) are predicted in an office room with a dimensions of (3m x 2.5 m, and 2.5 m height). Figure 1 shown the office room equipped with displacement ventilation device and cooling surfaces. The office room is occupants by two persons with two Personal computers and overhead light as heat sources. The loading of the occupants and equipment in the office room summarize in table.1. In current study assumption used that the wall with window at the north side is exposed outside conditions (at maximum summer temperature in Hilla city 47°C and 30% relative humidity). West wall with door oriented to inside corridor. The other walls are assumed to be adiabatic. The room ceiling area cover by 80 % chilled ceiling surface.

![Figure 1. Schematic diagram for the tested room](image)

**Table 1. Occupants and equipment heat summary.**

| Items           | No. | Power [W]       | Power per floor area [W/m\textsuperscript{2}] |
|-----------------|-----|-----------------|-----------------------------------------------|
| Person          | 2   | 75 W per person | 20                                            |
| Lights          | 1   | 100             | 13.3                                          |
| Personal Computer | 2  | 60              | 16                                            |

The walls components in Iraqi buildings were formed of multiple materials. The wall is 30 cm thickness consist of four layers from inside to outside (gypsum plaster 2 cm, cement plaster 2 cm, common Brick 24 cm, cement plaster 2 cm as show in figure 2. The details of these walls materials are listed in table2.
Table 2. Details of materials for office room walls

| Material | Thickness mm | K [W/m.°C] |
|----------|--------------|-------------|
| Brick    | 240          | 0.69        |
| Gypsum   | 20           | 0.48        |
| Cement   | 20           | 1.16        |
| Wood     | 30           | 0.28        |
| Glass    | 6            | 0.78        |

3. Theoretical Analyses

3.1 Cooling Load

ASHRAE Handbook, 2013, [31] explained the method used to calculate the cooling loads. The heat gain is divided into two parts, the first due to occupants heating, lamps, computer and other heat sources in the room zone as shown in Table 1. The second part is heat gain from the outside due to heat transfer through the windows, walls, and door.

A cooling load temperature difference method (equations (2 to 9)) can be used to calculate heat transfer from the walls, door, and window for this purpose:

\[
Q = U \times A \times CLTD_c
\]  
\[U = \frac{1}{R_{th}}\]  
\[R_{th} = \frac{1}{h_f} + \frac{x_1}{k_1} + \ldots + \frac{1}{h_o} + \frac{x_n}{k_n}\]

CLTDc is calculated as below:

For the walls:

\[
CLTD_c = (CLTD + LM) \times K + (25.5.5 - T_i) + (T_m - 29.4)
\]
For the door and window:

\[ \text{CLTD}_C = \text{CLTD} + (25.5 - T_d) + (T_m - 29.4) \]  \hspace{1cm} (7)

The heat gain due to radiation heat transfer through the window that generated by solar, obtained as follows:

\[ Q_s = A_s \times S \times SHG \times CLF \] \hspace{1cm} (8)

Equation (9) can then be used to determine the corridor temperature [32]:

\[ T_{\text{corridor}} = T_i + \frac{2}{3} (\Delta T - T_i) \] \hspace{1cm} (9)

The total cooling load for office room was calculated using the HAP program and found it’s equal to (800W). Table 3 shows the outside heat gain.

| Surface       | West wall | North wall | Window | Door |
|---------------|-----------|------------|--------|------|
| Heat (W)      | 101       | 105        | 157    | 67   |

### 3.2 Air Flowrate and Supply Air Temperature

The procedures that developed by Chen and Glicksma (2003) [33] and Skistad et al. (2002) [34] were acceptable method to calculated supply air flowrate \( (V_{DV}) \) and air temperature supply \( (T_s) \) for displacement ventilation systems as shown in equations (10 and 11) [33]. Different temperature between indoor air and air supply by diffuser should equal or less than 5.5 °C according to ASHRAE Standard 55 [35] to avoid draughts.

\[ V_{DV} = \frac{0.295q_{\text{in}} + 0.132q_{\text{in}} + 0.185q_{\text{in}}}{\rho C_p \Delta T_m} \] \hspace{1cm} (10)

\[ T_s = T_{in} - \Delta T_{hf} = \frac{A_t CL_{DV}}{0.584V_{DV}^2 + 1.2A_t V_{DV}} \] \hspace{1cm} (11)

\[ CL_{DV} = q_{\text{in}} + q_{\text{in}} + q_{\text{in}} \] \hspace{1cm} (12)

\[ CL_{DV} = pQ_{DV} C_p (T_c - T_i) \] \hspace{1cm} (13)

By fixed supply air flowrate at (45L/s) for each tests (to notice the change in air age for each test) and the supply air temperature \( (T_s) \) change with varied the value of cooling load treated by displacement ventilation \( (CL_{DV}) \).

The indoor air dry bulb temperature design \( (T_{id}) \) is (25°C) due to ASHRAE standard 2007. Initial value of CO2 concentration inside test room is (1000 ppm) to estimate the required time to reach to the CO2 concentration in supply air (400 ppm) for each test.

According to ASHRAE Standard 55 [35] the difference in temperature between head and foot \( (\Delta T_{hf}) \) is 2 °C for seated person.
3.3 **Chilled Ceiling Surface Temperature**

Equation 13 is an empirical equation used to calculate the chilled ceiling surface temperature [36].

\[
T_{CC} = T_{sr} \left( \frac{CL_{CC}}{8.92 f A_f} \right)^{0.9}
\]  

(14)

Where \( f \) is the ratio of chilled ceiling area \( (A_{CC}) \) to floor area \( (A_f) \), which was 0.8 in this study. At 50% relative humidity and 25°C dry bulb temperature, the dew point temperature for office room is (14°C), then the chilled ceiling surface temperature must be higher than 15°C to avoid condensation risk. The values of \( T_s \) and \( T_{CC} \) at various \( \eta \) for two cases shown in Table (4).

| Cases | Tests   | \( \eta \) (%) | \( CL_{DV} \) (W) | \( CL_{CC} \) (W) | \( Q_{DV} \) (m³/s⁻¹) | \( T_s \) (°C) | \( \phi_s \) (%) | \( T_{CC} \) (°C) |
|-------|---------|----------------|------------------|------------------|---------------------|---------------|----------------|-----------------|
| Case-1| Test.1-1| 25             | 600              | 200              | 0.045               | 19            | 75             | 21.5            |
|       | Test.1-2| 50             | 400              | 400              | 0.045               | 22            | 60             | 19              |
|       | Test.1-3| 80             | 160              | 640              | 0.045               | 24.5          | 55             | 16              |
|       | Test.2-1| 25             | 600              | 200              | 0.045               | 19            | 75             | 21.5            |
| Case-2| Test.2-2| 50             | 400              | 400              | 0.045               | 22            | 60             | 19              |
|       | Test.2-3| 80             | 160              | 640              | 0.045               | 24.5          | 55             | 16              |

4. **Air Supply Diffuser**

According to the principle of displacement ventilation system, the air supply should be at low speed, for this the air supply speed (0.2m / s) was chosen to give constant air flowrate (0.045m³/s, ACH= 8.5 1/h), then the area of supply diffuser equal to (0.225 m²) with dimensions 55cm width at 40cm height for one way rectangular diffuser for case-I and 36cm diameter and 40cm height for semicircle diffuser for case-II. The supply diffuser was placed at the midpoint of the north wall side near the floor of booth cases. ‘figure 3’ shows the diffuser models used in this study.

**Figure 3.** dimensions for one way rectangular and semicircle diffusers

5. **Simulation method**
AIRPAK.3.0.16 software used to predict and analysis all type of ventilation systems [37]. Many of researchers who have studied DV/CC system was used (AIRPAK) software in their numerical analyses as Gao S. et. al.[38], Ayoub. et. al.[39], Yang. et. al.[40].

Two equations Renormalization Group model (RNG k- ε) was a good choose for used in numerical analyses for displacement ventilation system [41and 42]. Inlet velocity set as boundary for air supply, the walls as no slip condition. Constant heat flux for walls, door and windows. Occupants, T.V and lights simulated as constant heat. Zero pressure condition used for exit air. Ambient temperature is 47°C at 30% relative humidity (due to Hilla climate) and Initial value of Co2 concentration inside test room is (1000 PPM) to estimate the time required to reach to the supply concentration (400 PPM).

Governing equations for steady incompressible and three dimension flow represented as follow [43 and 44]:

- Continuity equation

\[ \nabla \cdot \vec{v} = 0 \]  \hspace{2cm} (15)

- Momentum equation

\[ \frac{\partial}{\partial t} (\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla P + \nabla (\rho g) \] \hspace{2cm} (16)

- Energy equation

\[ \frac{\partial}{\partial t} (\rho h) + \nabla \cdot (\rho h \vec{v}) = \nabla \left( [K + K_r \nabla T] \right) + S_h \] \hspace{2cm} (17)

The working fluid are three species. The main is air and the others are carbon dioxide and H2o as pollutant and humidity respectively. Flow was assumed steady and incompressible. The second order upwind scheme used to describe energy, temperature, velocity, CO2 and H2o. PREssure Staggering Option (PRESTO) describe for Pressure is used with body force weighted. The SIMPLEx (Semi Implicit) scheme is used for the coupling between pressure and velocity. Figure 4 shows the Residual error monitor for numerical solution.

![Residual error monitor](image)

**Figure 4.** Monitor of residual error

5.1 **AIRPAK Software validation**
Experimental study conducted by (Simon J. Rees) [45] is used to validate AIRPAK software by compared the numerical results done by AIRPAK software with experimental data obtained from office room with DV/CC system. The comparison was depended on air temperature distribution with height on vertical line at point (0.7, 1.54) on XZ plane. The RNG k-ε turbulent model gave a good agreement between experimental and simulate results at 3.27% average error as shown in figure 5. The maximum error occurred at the height level between (0.2-1.2m) approximately, This level is within the zone of heat sources, and for this there is a temperature disturbance between the heat sources and the supply cold air, which leads to an error rate in some reading between the numerical and experimental results, addition to the accuracy of measuring devices and humans errors.

![Figure 5](image_url)

**Figure 5.** AIRPAK software validation with another experimental data.

5.2 **Mesh Strategy**

Mesh independent test depend on five mesh strategy (221315, 29805, 303319, 360530 and 425725 nodes). Four positions were chosen to test the air age and temperature simulating results as P1 (0.5, 0.5, 1.25), P2 (0.5, 1.8, 1.25), P3 (0.75, 1.1, 0.5) and P4 (2.75, 1.1, 0.5). Figure 6 shows the air age and temperature profiles for the five different node volumes. Based on this, the most acceptable number of nodes was 303,319 for case I and 339,600 nodes for case II. Figure 7 shows parts of the meshed models for these two cases.

![Figure 6](image_url)

**Figure 6.** profiles for five different node numbers per thousand
Figure 7. part of the meshed model for two cases

6. Parameters of human comfort

6.1 Age of the air
Age of air define as (ventilation system ability to remove the old air in a room and exchange it by new fresh air) [46]. If assume \( f(\tau) \) is the age of air particles that arriving at a given location is between \( \tau \) and \( (\tau + d\tau) \) and \( F(\tau) \) that this age is larger than \( \tau \). The two functions related by [47]:

\[
f(\tau) = \frac{dF(\tau)}{dt}
\]

(18)

\[
F(\tau) = 1 - \int_0^\tau f(t)dt
\]

(19)

The local mean air age at \((r)\) point was defined by average age of all the air particles arriving at that point [47].

\[
dF(\tau) = \int_0^\tau f(t)dt = \int_0^\tau F(t)dt
\]

(20)

The nominal time constant calculated by used Equation (18)

\[
\tau_a = \frac{V}{Q_{ov}}
\]

(21)

\( \langle \tau \rangle \) is a room mean air age, it’s can’t be less than half a nominal time constant, then the air exchange efficiency represented as:

\[
\eta_a = \frac{\tau_a}{2\langle \tau \rangle}
\]

(22)

6.2 Predicted Mean Vote PMV
Its index used to predicted thermal comfort and the human conditions as shown in Equation.20 [48]:

\[
PMV = [0.303\exp(-0.036M) + 0.028]L
\]

(23)

The PMV value classified to seven points as (from +3 at cold indoor air to -3 at hot indoor air). Depending on (ASHRAE standards) the best rang of (PMV) for acceptable internal comfort is (-0.5<PMV< 0.5) [35 and 49].
6.3 Predicted Percentage Dissatisfied PPD

(PPD) predicts percentage of people thermal discomfort who feel very hot or very cold. PPD calculated from the predicted mean vote (PMV) as show in Equation 21. [49]. For general, the limit of PPD for good thermal environment is 10%, according to (-0.5≤PMV≤ 0.5) [35 and 49].

\[
PPD = 100 - \exp\left[-0.0353(PMV)^2 + 0.2179(PMV)^3\right]
\]  

6.4 Air Diffusion Performance Index. ADPI

ADPI used to satisfy the comfort level. It’s estimated as a ratio of the number of points in occupied zone that have EDT value between -1.7 and 1.1°C to the total number of points in the same zone [50].

7. Results and discussion

Displacement ventilation with chilled ceiling system was studied numerically in an office room at peak summer temperature under Iraqi- Hilla city climate. The study was divided in to two cases based on the air supply diffuser shape (rectangular diffuser for case-I and semicircular diffuser for case-II). Each case have three tests at different cooling load ratio treated by chilled ceiling (η) as (25%, 50%, and 80%).

The air supply temperature for displacement ventilation varied as (19°C, 22°C and 24.5°C) depending on cooling load removed by displacement ventilation and chilled ceiling temperature varied as (22, 20 and 17.5 °C) at (η) equal to 25 %, 50 % and 80 % respectively. The airflow rate supply in two cases was fixed at 0.045 m³/s (8.6 air changes per hour). Table 5. listed the main results parameters obtained in the numerical analyses using AIRPAK software.

| Cases | Tests | η % | QDV m³/s | Tsv °C | Tr °C | Mean room temp. °C | ADPI % | PMV | PPD % | ∆Tmf °C | ηa % |
|-------|-------|----|--------|-------|-----|-----------------|------|-----|------|--------|------|
|       | Test.1-1 | 25 | 0.045 | 19 | 21.5 | 24 | 42 | 0.98 | 28.15 | 3.2 | 63.4 |
| Case-1 | Test.1-2 | 50 | 0.045 | 22 | 19 | 25.1 | 45 | 0.9 | 24.8 | 2.3 | 60.1 |
|       | Test.1-3 | 80 | 0.045 | 24.5 | 16 | 26.14 | 61 | 0.763 | 20.8 | 1.5 | 55 |
|       | Test.2-1 | 25 | 0.045 | 19 | 21.5 | 23.25 | 48.3 | 0.763 | 20.8 | 2.7 | 71 |
| Case-2 | Test.2-2 | 50 | 0.045 | 22 | 19 | 24.5 | 56.6 | 0.64 | 15 | 2.1 | 68.4 |
|       | Test.2-3 | 80 | 0.045 | 24.5 | 16 | 25.4 | 72.6 | 0.6 | 14 | 1.1 | 66.4 |

7.1 Mean Air age and air exchange efficiency

Figure 8  shown the mean age of air profiles at different levels in the office room for the two cases (Case-I and Case-II). Notes that the mean air age increase with height regardless of the shape of diffuser. Under 1.2 height level notes that the mean age of air with used semicircle diffuser is less than from rectangular diffuser although the airflow is constant. Above 1.2 m the mean air age is approximately equal regardless of the shape of the diffuser. This result gives a clear idea that the effect of diffuser shape decreases with height. Increase in supply air temperature at (η=80%) lead to decrease in effect of diffuser shape on air age at low room level because of decrease in air density and it’s tend to becomes faster diffusion.
Figure 8. mean age of air profiles with height for two cases at different (η)

Figure 9 shown the mean age of air profiles with height at different (η) for two cases. At level above (1m) notes that the effect of chilled ceiling temperature is clearly. Mean air age increase with increase portion of cooling load treated by chilled ceiling although the airflow was constant. This result explains the effect of cooled air that move down due to convection from the chilled ceiling and impact with hot air rising, and lead to reduce the air speed. Increase in air age with increase (η) lead to decrease the air exchange efficiency (η_a) as show in Figure 10. Reduces in air exchange efficiency (η_a) means increase in time required for fresh air to replace the old air in a room zone and reduce in occupants performance. The air exchange efficiency for CC/DV system in an office room by used semicircle diffuser is higher compared with used rectangular diffuser at each value of (η), and the increase in (η) have slightly effect on the air exchange efficiency when the semicircle diffuser are used, that’s mean the semicircle diffuser distributes fresh air more uniformly and faster than the rectangular diffuser.

Figure 9. mean age of air profiles with height at different (η) for two cases
Figure 10. air exchange efficiency at different (η) for two cases

Figure 11 a-c shows mean air age distribution contours of the office room for two cases at z=1.25m at different (η). The age of air at air supply diffuser start from zero and increasing in indoor zone. The age of air is increasing with room height and notes the maximum value of air age at zone up to diffuser location, that’s mean low air movement in this zone lead to increase in pollution concentration and occupants discomfort in this zone. Increase value of (η) lead to increase value of age of air due to interchange the cool air coming from the cooled ceiling and the hot air rise. Notes that the increasing in air temperature near the heat sources (person and computer) by convection lead to decrease the age of air in this zone due to increase air velocity around the human body.
7.2 Thermal comfort and Air quality

(PPD), (PMV) and (ADPI) are parameters which gives a good idea about air quality and thermal comfort in occupied zoon [35]. Figure 12 and figure 13 shown the profiles of (PMV) and (PPD) respectively for two cases at different value of (η). Notes that the PMV and PPD values are improving with increase value of (η) in the two cases and converge to the comfort values that specified by ASHRAE standard 2017 (-0.5 ≤ PMV ≤ 0.5 and PPD =10% ). At value of (η) change from (25% to 80%) the PMV decrease from (0.98 to 0.76) when using rectangular supply diffuser (case-I) and from (0.76 to 0.6) when using semicircle diffuser (case-II). PPd percentage values decreases with increase (η) also. That’s mean the increase in portion of load treated by chilled ceiling lead to improve thermal comfort. The values of PMV and PPD for the combined system (CC/DV) for case-II at different (η) is less than from its values
for case-I and more converge from standard values. This result indicates that using a circular diffuser gives a better thermal comfort than a rectangular diffuser when used with (CC/DV) system.

Figure 12. PMV profile for two cases at different (η)

Figure 13. PPD profile for two cases at different (η)

ADPI is a factor used to predict air circulation system of the indoor, increase in ADPI percentage gives a good indication of occupants comfort. ‘figure 14’ shows the percentage values of ADPI for two cases at different (η), notes that the ADPI increase with increase (η) in two cases, this mean that the increasing in portion load treated by chilled ceiling lead to improve thermal comfort due to increase ADPI value. The value of ADPI for (CC/DV) system by used semicircle supply diffuser is higher than from its value when used rectangular supply diffuser at different value of (η), this result shown that the semicircle diffuser has advantage of improving indoor thermal comfort more than rectangular diffuser.

Figure 14. ADPI profile for two cases at different (η)

Figure 15 a-c shows decrease in indoor carbon dioxide concentration with time to estimate the needed time for carbon dioxide concentration to drop from (1000 ppm) to (400 ppm) at different (η) for the two cases. The results shows that changes in the load treated by the chilled ceiling and the shape of diffuser have a little effect on the removal speed of carbon dioxide in both cases. Therefore, the speed at which carbon is removed depends on the amount of supply air flowrate, whatever the supply temperature.
7.3 Air Temperature distribution

Stratified temperature with height is one of the parameter that differentiate the displacement ventilation system. Figure 16 a-c shows the effect of chilled ceiling on the temperature distribution with height at different value of (η) for two cases. For two cases notes that the temperature increase with height. The temperature gradient with height for (CC/DV) system by used semicircle diffuser (case-II) is lower than from used rectangular diffuser (case-I) at constant airflow rate regardless of (η) value. The stratification of temperatures decreases with increase (η) for both cases as shown in ‘figure 17’ due to effects of the low temperature of the chilled ceiling surface, consequently the air temperature near the ceil decreases by convection and falls to cool the air at lower levels. These results are consistent with the practical results obtained by Rees, S. J.[45] and Schiavon. et.al.[51] they studied performance of displacement ventilation and chilled ceiling systems at different heat source distribution.
The value of temperature different between foot and head for a seated person about 2°C as specified by ASHRAE Standard, [35]. Figure 18 shown the relation between temperature difference between foot and head (ΔT_{hf}) and the value of (η) for two cases. Notes that the increase (η) value from (25% to 80%) lead to decreases in the temperature difference between head and foot by 1.7°C for case-I and 1.9°C for case-II. This shows that the increase in the heat load treated by chilled ceiling has more effect on the (ΔT_{hf}) when the CC/DV system used semicircle difference. Decreases in the stratified temperature (ΔT_{hf}) with increase (η) agree with results found by tan et. al. [52] and Ghaddar. [53] who predicted the stratified temperature in a room depending on the displacement air flowrate and load removed by the chilled ceiling.
Figure 18. Relationship between temperature difference between foot and head ($\Delta T_{hf}$) and the value of ($\eta$) for two cases.

Figure 19a–c shows air temperature distribution contours for two cases with different ($\eta$) at $z=1.25m$. Notes that the air temperature increasing with increase height of the office room and creating a stratified layers due to convection between heat sources and cooled air. Stratified layers is one of the principles of displacement ventilation. The temperature stratification for two cases decrease with increase value of ($\eta$) and becomes unclear at $\eta=80\%$. The supply air at low temperature flows along the floor, after that the air is heated when basses through the human body and move upward due to effects of buoyancy. The effect of gravity is very clear, especially near diffuser supply when air is supplied with a low temperature due to high air density. At same value of ($\eta$), notes that the air at high room level with semicircle diffuser is colder than air with use rectangular diffuser.
AIRPAK software was used to numerically study the effects of supply diffuser shape on indoor air age and comfort level in an office room using displacement ventilation combined with chilled ceiling system under Iraqi-Hilla city climate (hot and dry climate). The study examining the ratio of cooling load treated by chilled ceiling with respect to the total office room cooling load at constant air flowrate supplied by displacement ventilation. Air supply temperature thus varied from (19-24.5°C) at cooling load ratio treated by chilled ceiling varies from (25% to 80%). The main conclusions are found as following:

1. In CC/DV system, the effect of the diffuser shape on the indoor age of air decreases with height from floor area. For two type of diffuser, the mean air age increase with increase the portion of cooling load treated by chilled ceiling lead.
2- Increases in supply air temperature lead to decreases in the effect of diffuser shape on air age.
3- The air exchange efficiency for (CC/DV) system in an office room by used semicircle diffuser is higher compared with used rectangular diffuser and it’s reduce with increase the portion of cooling load treated by chilled ceiling.
4- For two type of supply diffuser, the mean air age increase with increase the portion of cooling load treated by chilled ceil lead to reduce air exchange efficiency.
5- Semicircle diffuser tend to improve (PMV), (PPD) and (ADPI) and gives a better thermal comfort than a rectangular diffuser when used with (CC/DV) system.
6- The shape of diffuser and the amount of load treated by the chilled ceiling have a little effect on the indoor removal speed of carbon dioxide.
Nomenclature

| Nomenclature   | Abbreviations          |
|----------------|------------------------|
| A              | AV                     |
| C              | Amp                    |
| CL             | CLTD                   |
| DR             | CLF                    |
| dx, dy, dz     | CFD                    |
| E              | ADPI                   |
| f              | DV                     |
| G              | EDT                    |
| H              | LAQ                    |
| K              | RNG                    |
| LM             | PMV                    |
| N              | PPD                    |
| P              | SHG                    |
| Q              | Sub – Scripts          |
| R              | U                      |
| T              | V                      |
| U              |                     |
| x,y,z          |                         |
| V              |                         |
| H              |                         |
| Ψ              |                         |
| P              |                         |
| R              |                         |
| Γ              |                         |
| T              |                         |
| εε             |                         |
| ΔT             |                         |

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