Anbar Journal of Engineering Science (AJES)
P-ISSN: 1997-9428; E-ISSN: 2705-7440

Vol. 11 , No. 2 (2020) , pp. 170 ~ 183
DOI: 10.37649/aengs.2020.171219

Univiersty of Anbar

Anbar Journal Of Engineering Science©

journal homepage: http://www.uoanbar.edu.iq/Evaluate/

Numerical study of thermal comfort levels in a conference hall

Mhaned A. Mudher [a*], Ahmed A. Najeeb [a]o

[a] Department of Mechanical Engineering, College of Engineering, University of Anbar Ramadi, Anbar, Iraq

PAPER INFO

Paper history:
Received 5 /9/2020
Received in revised form 2/10/2020
Accepted 13/10/2020

Keywords:
Thermal comfort
Air conditioning
Turbulence k-ϵ
predicted mean vote

ABSTRACT

The present study was concerned with the analysis, simulation of the air flow patterns and thermal comfort levels in the University of Anbar at conferences hall (Ibn Al Haitham hall). The study was performed in a hot - dry season. The purpose of the present work was to investigate the level of thermal comfort and the influence of the air flow on the flow patterns at the conferences hall. It has been assumed that the total number of occupying audiences in the hall was approximately 100 persons. The present work simulated and analyzed four hypothetical cases, namely: in the first case, the hall was assumed as an empty place, whereas the other three cases were performed by redistribution for the three units of air conditioning, the hall was assumed as a filled place with persons in September 2019. The study was accomplished using simulation techniques, a CFD code (FLUENT 6.2) v.17, which is commercially available. The CFD modelling techniques were applied to solve the continuity, momentum and the energy conservation equations in addition to the Turbulence k-ϵ (RNG) model equations for a turbulence closure model. Thermal comfort was assessed by finding the values of predicted mean vote (PMV), predicted percentage of dissatisfied (PPD), and ASHRAE standard-55. In conclusion, the second case was the superior in comparison to these other cases. It was noted that the PMV value was 0.17, whereas the PPD value was 6.79 at the breathing level.

Corresponding author: Mhaned A. Mudher ; mohanedsowd@gmail.com ; +9647803694136

1. **Introduction**

The main aim of air conditioning is to improve the interior environments of buildings for providing a comfortable and healthy indoor environment for building systems or facilities. The demand of air conditioning units increasingly raises that pay attaintion the researchers to improve the quality of the indoor environment conditions. Thus it has been necessary to study the thermal conditions of these indoor environments. Numerous studies have been conducted to explore the conditions of educational institutions buildings. The American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE) is a society that focuses on building systems, energy efficiency, indoor air quality, refrigeration and sustainability within the industry. The definition of thermal comfort has been formulated by this society as the condition of mind, which expresses satisfaction with the thermal environment [1]. Iskandriawan [2] studied the influence of the air maintaining on the thermal comfort and the quality of the air in the lecture room. Koskela et al. [3] studied the flow patterns and draft risk in office environment where cooling and air distribution is implemented with active chilled beams. Kampelis [4] exercised HVAC set-point control as a way to reduce the cost of energy and improve thermal comfort in a University building. Skotnicka [5] illustrated some possibilities of applying instruments for computational fluid dynamics to assess thermal comfort on the example of a lecture hall in the building of the Institute of Building Engineering at the University of Warmia and Mazury in Olsztyn. Yuce B. E and Pulat E [6] computed airflow and temperature fields in an office room with a table and a sitting person to predict the thermal comfort by means of percent dissatisfaction (PD). Hayatu [7] investigated the level of thermal comfort in the lecture theatres of the University of Bayero permanent site where data collected by field measurement and survey were analyzed and found that, the indoor air temperature of the theaters was in the range of 31.8 to 36.2 , and the indoor relative humidity was in the range of 36.5 to 50.6 , while the air velocity is between 0.29m/s and 0.05m/s. Cena and Richard [8] discussed the effects of indoor climates on thermal perceptions and adaptive behaviour of office workers during a large field study in Kalgoorlie-Boulder, located in a hot-arid region of Western Australia. Kim and Dear [9] studied the way to better understand thermal comfort perception and related behavioural characteristics of school children. Statistical analyses were performed the thermal comfort survey database consisting of 4866 responses collected from primary- and secondary school classrooms in Australia across two summer seasons. Zaki [10] investigated the comfort temperature and adaptive behaviour of university students in Malaysia and Japan where the classrooms in three universities were set to one of two conditions during the summer season: mechanical cooling (CL) mode, where AC was switched on for cooling purposes, and free-running (FR) mode, where AC was switched off. Tamaraukuro and Japo [11] investigated the relationship between students' perceived thermal discomfort and stress behaviours affecting their learning in lecture theatres in the humid tropics at the Niger Delta University. Ibrahim et al. [12] evaluated thermal comfort in a library constructed using Industrialised Building System (IBS) where the data were analyzed using Corrected Effective Temperature (CET) index. From the data analysis, it shows that thermal comfort in the library could not be achieved most of the time unless when the mechanical cooling is used. Noh et al. [13] performed the expermintal and numerical studies on thermal comfort (TC) and Indoor air quality (IAQ) in the lecture room with cooling loads when the operating conditions are changed predicted mean vote (PMV) value and CO_2 concentration of the lecture room were measured and compared to the numerical results. Indraganti et al [14] analyzed the data from a recent thermal comfort field study of office buildings in two capital cities of Chennai and Hyderabad in India where behavioral adaptation formed a key mechanism contributing to the subject's thermal comfort and user satisfaction in buildings. Haddad et al. [15] investigated the application of the principal methods underlying the adaptive comfort theory for children relating thermal comfort indoors to the prevailing mean outdoor temperature for schoolchildren in classrooms. Abou-deif et al. [16] concerned primarily with the analysis, simulation of the air flow and thermal patterns in a lecture room where it is devoted to investigate the influence of location and number of ventilation and air conditioning supply and extracts openings on air flow properties. Pereira et al. [17] studied the conditions of thermal comfort based on PPD and PMV assessment in a classroom with a split system were analyzed using experimental and numerical results where the air temperature and velocity were measured and compared with the numerical results.

The objective of the study is the improvement of thermal comfort for audience inside the hall by the redistribution of the air conditioning system to obtain the best thermal comfort.

2. Mathematical formulation

2.1. Physical model and assumptions

To assess thermal comfort, the indoor factors such as air temperature, air velocity and water vapor need to be calculated. These can be calculated by solving the system of coupled governing equations for the conservation of mass, momentum and energy.Assuming Steady-state, incompressible flow of air as a multi-component fluid which includes dry air and water vapor. The fluid properties are taken as constants.Assuming that there is no heat generation, energy fluxes due to inter-diffusion.

2.2 Numerical Methodology and Governing Equations

The equations are given by:

Conservation of mass equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad \textbf{(1)}$$

Momentum equation (x-direction)

$$\rho g_x-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)=\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right) \qquad \textbf{(2)}$$

Momentum equation (y-direction)

$$\rho g_y-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)=\rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right) \qquad \textbf{(3)}$$

Momentum equation (z-direction):

$$\rho g_z - \frac{\partial p}{\partial z} + \mu\left(\frac{\partial^{٢} w}{\partial x^{٢}} + \frac{\partial^{٢} w}{\partial y^{٢}} + \frac{\partial^{٢} w}{\partial z^{٢}}\right) = \rho\left(\frac{\partial w}{\partial t} + u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}\right) \quad \textbf{(4)}$$

Energy equation

$$\rho c_v\left(\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = \rho\dot{q} + K\left(\frac{\partial^{٢} T}{\partial x^{٢}} + \frac{\partial^{٢} T}{\partial y^{٢}} + \frac{\partial^{٢} T}{\partial z^{٢}}\right) + \emptyset$$
(5)

Where Ø is:

$$\emptyset = \mu\left[٢\left(\frac{\partial u}{\partial x}\right)^{٢} + ٢\left(\frac{\partial v}{\partial y}\right)^{٢} + ٢\left(\frac{\partial w}{\partial z}\right)^{٢} + \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)^{٢} + \left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}\right)^{٢} + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)^{٢}\right]$$
(6)

Turbulence k-ϵ (RNG) model:

Kinetic energy equation:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i}\left(\alpha_k \mu_{eff}\frac{\partial k}{\partial x_j}\right) + G_b + G_k - Y_M - \rho\varepsilon + S_k \quad \textbf{(7)}$$

Dissipation equation:

$$\frac{\partial}{\partial t}(\rho\epsilon) + \frac{\partial}{\partial x_i}(\rho\epsilon u_i) = \frac{\partial}{\partial x_j}\left(\alpha_\epsilon \mu_{eff}\frac{\partial \epsilon}{\partial x_j}\right) + C_{١\epsilon}\frac{\epsilon}{k}(C_{٣\epsilon}G_b + G_k) - C_{٢\epsilon}\rho\frac{\epsilon^{٢}}{k} - R_\epsilon + S_\epsilon \quad \textbf{(8)}$$

C1ϵ = 1.24 , C2ϵ = 1.68

Transport equation:

The solution of the conservation equations for chemical species present in the domain of the CFD model, the CFD code should predict the local mass fraction of each species Yi in the control volume. This can be made by solving the convection-diffusion equation for the species i. The general differential form for the species (Transport equation) is:

$$\frac{\partial}{\partial t}(\rho Y_i) + \nabla.(\rho\vec{v}Y_i) = -\nabla.\vec{J_i} + R_i + S_i \quad \textbf{(9)}$$

Fanger's PMV and PPD equations:

The PMV is based on the heat balance of the human body: in thermal balance, the internal heat production in the body is equal to the loss of heat to the environment. The PMV equation combines four physical variables (air temperature, air velocity, mean radiant temperature, and relative humidity), and two personal variables (clothing insulation and activity level) into an index that can be used to predict the average thermal sensation of a large group of people in a space.

$$PMV = \left(٠.٣٠٣\, e^{(-٠.٠٣٦\cdot M)} + ٠.٠٢٨\right)\{(M-W) - ٣.٠٥\,[٥.٧٣٣ - ٠.٠٠٧(M-W) - ٠.٠٠١P_w] - ٠.٤٢[(M-W) - ٥٨.١٥] - ٠.٠١٧٣M\,(٥.٨٦٧ - ٠.٠٠١P_w) - ٠.٠٠١٤M\,(٣٤ - t_a) - ٣.٩٦ * f_{cl}\left[(t_{cl} + ٢٧٣)^{٤} - (t_{MR} + ٢٧٣)^{٤}\right] - f_{cl}\cdot h_c\cdot(t_{cl} - t_a)\} \quad \textbf{(10)}$$

$$PPD = ١٠٠ - ٩٥\cdot e^{(-٠,٠٣٣٥٣\cdot PMV^{٤} - ٠,٢١٧٩\cdot PMV^{٢})} \quad \textbf{(11)}$$

$$t_{cl} = ٣٥.٧ - ٠.٠٢٧٥(M-W) - R_{cl}\{٣.٩٦ * f_{cl}\left[(t_{cl} + ٢٧٣)^{٤} - (t_{MR} + ٢٧٣)^{٤}\right] - f_{cl}\cdot h_c\cdot(t_{cl} - t_a)\} \quad \textbf{(12)}$$

$$f_{cl} = \begin{cases} ١.٠ + ٠.٢I_{cl} & I_{cl} < ٠.٥ \\ ١.٠٥ + ٠.١I_{cl} & I_{cl} > ٠.٥ \end{cases} \quad \textbf{(13)}$$

$$h_c = \begin{cases} ٢.٣٨(t_{cl} - t_a)^{٠.٢٥} & ٢.٣٨(t_{cl} - t_a)^{٠.٢٥} > ١٢.١\sqrt{V} \\ ١٢.١\sqrt{V} & ٢.٣٨(t_{cl} - t_a)^{٠.٢٥} < ١٢.١\sqrt{V} \end{cases} \quad \textbf{(14)}$$

٣. The case of the present study

The present work is studied a conferences hall (Ibn Al-Haitham hall) as shown in Figures 1and 2, it located in the university of Anbar, city of Ramadi to asses and improve the thermal comfort inside the hall. The city of Anbar located on the longitudinal line 43.26 and latitude 33.43 above of sea level at 45 m. the weather in Anbar is a hot- dry climate in the summer season. The summer months in Iraq generally extend from April to October, with temperatures reaching a peak of more than 50 °C in mid of July. The dimensions of the hall are 12.5m length by 12.5m width, the area of the hall is 156 m² and height is 3m.

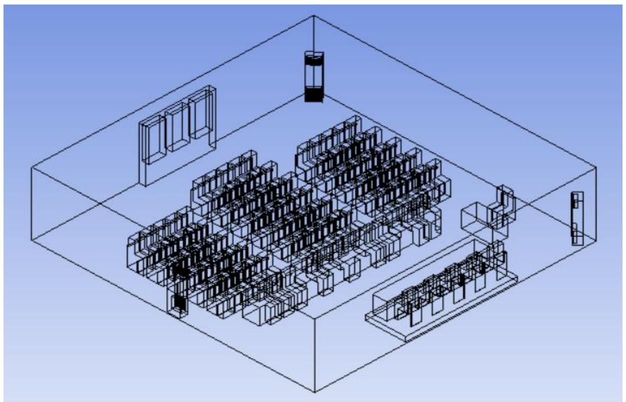

Fig 1 scheme of geometry

Fig 2 Ibn Al Haitham hall

٤. The Numerical procedure

The whole building of the hall is designed modeled by using the package of Soildwork 2016. The representation includes audience and air conditioning units. The area of the hall is 156 m². The height of the walls is up to a roof of 3 m. The wall on the eastern side contains four glass windows with an area of 1.5 m×2 m. The effect of heat transfer from outside to inside the hall will be transfer from the windows to the other walls. The splits are installed over one side with equal distances. The split includes two pieces (indoor, and outdoor units). The indoor unit is ground-based. It is a rectangle with dimensions of 2.19 m high, 0.55 m width, and 0.34 m thickness. Also, it has an upper cold air intake slot with dimension of 0.26 × 0.45 m2 and contains five blades for air steering while the lower air exit slot with a dimension of 0.45 × 0.5 m^2 as shown in Figure3.The occupants are represented as a rectangle shape with dimensions of 1.7 × 0.5 × 0.25 m^3 per person as shown in Figure 4. There is a gap between persons which is equal to 0.1m. There are two processes for human breathing; exhalation (entering the breathing air to the body) and inhalation (return charging the air outside).The source of these types are the mouth and nose.The mouth and nose are represented as a square shape with dimensions of 0.3×0.3m². This is represented at a distance of 0.1m on the upper end of the parallel rectangles as shown in Figure4

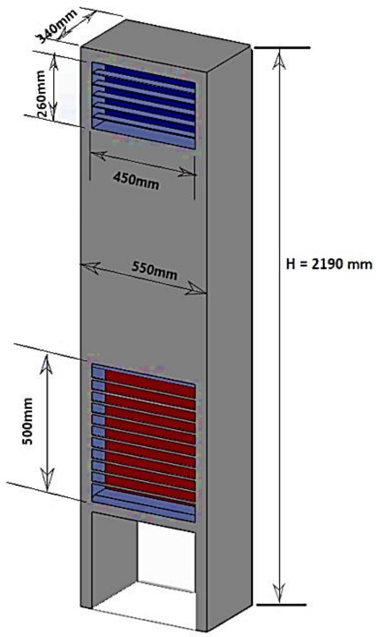

Fig 3 A/C unit

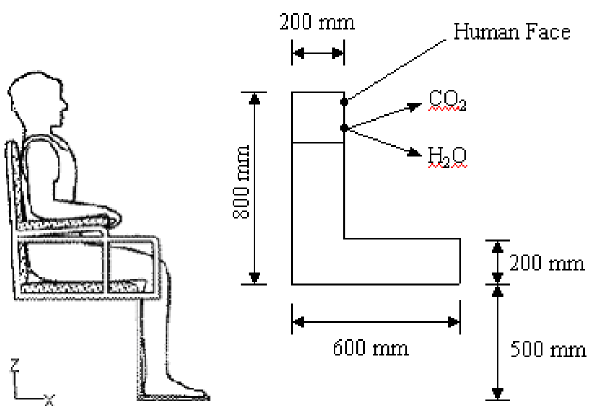

Fig 4 scheme of person

The air velocity and temperature at the outlet from A/C are assumed, which is reported by the supplier in their catalog, present experimental tests, and previous studies that have the same conditions for the same A/C type. The assumption of the temperature of the body surface is taken depending on many studies focused on the effectiveness of the sitting state and rest of the person sitting at the same rate of metabolism described in ASHRAE-55 standard. The temperature of the wall was measured based on the program of Hap 4.9 A/C loads for external conditions to the highest thermal load from hall when is full occupied during the year, which occurs on 15 September at 1 PM.

4.1 Boundary conditions:

The temperature of the inner surface and the roof wall of the hall were calculated by the Hap 4.9 program which design to calculate the thermal loads of the building. The outside conditions were described throughout the summer from April to October. The maximum external temperature of the program is at 50 ˚C according to the weather tables of the Iraqi forecast. The surface temperature of the human body skin is assumed 34 ˚C. This value is derived from the level of the person's effectiveness and food metabolism. The metabolic rate (M) is 1.2 met or 70 w/m^2, by applying these values in the equation T_{sk}= 35.7 - 0.028× (M-W) to find the skin temperature (T_{sk}).All the numerical studies of thermal comfort adopted this value as the temperature of the surface of the skin .Table 1 is shown the assumed boundary conditios.

Table 1 the boundary conditions

Details of Boundary condition	Value of Boundary condition
Activity type	Sitting, Standing
Metabolic rate	1.2 met
Skin temperature	34°C
breathing rate	8 -16 breaths/min
humidity from air breathing	0.042 kg_{water}/ $kg_{d.air}$
Temperature of air breathing	37°C
rate of cooling air	0.5664 m^3/s
The temperature cooling air	13°C
Moisture content of cooling air	0.0048 kg_{water} /$kg_{d..air}$
Inner surface Walls temperature	30°C
Inner surface Roof temperature	34°C
Rate of ventilation air	0.3152 m^3 /s
Moisture content of ventilation air	0.013 kg_{water} /$kg_{d.air}$
temperature of ventilation air	49°C
Clothing insulation	0.073 [m^2·K/w]
Initial air temperature	30°C

4.2 Mesh Independent Study

The choice of the independent mesh by testing the number of elements to obtain the precisest value. It began from 1,500,000 elements until 3,500,000 elements. It is found that the best number of elements is 2,500,000 which provide a negligible error, which it is found by testing the velocity as shown in figure5, temperature as shown in figure 6 and relative humdity as shown in the figure7 .All the cases were taken with 0° angle of blades for splits. The mesh is chosen at a breathing level which is equal to 1.1m where the type of mesh was cutcell and the solution method was second order upwind as shown in figure 8.

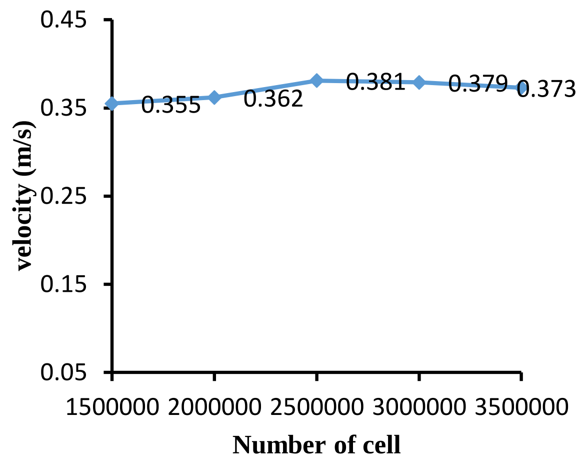

Fig 5 Velocity Test

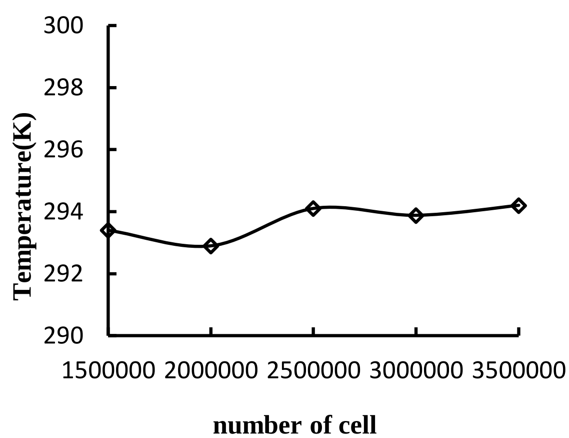

Fig 6 Temperature Test

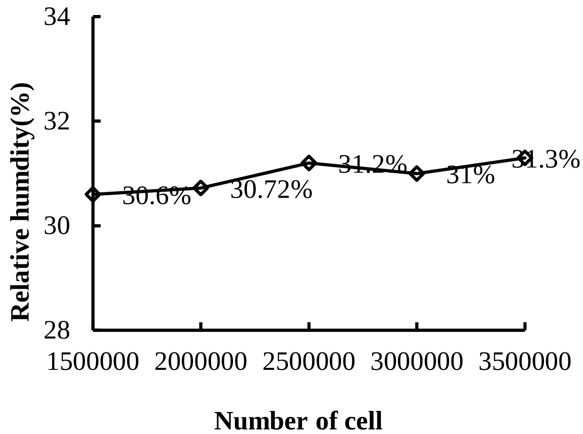

Fig 7 Relative humdity Test

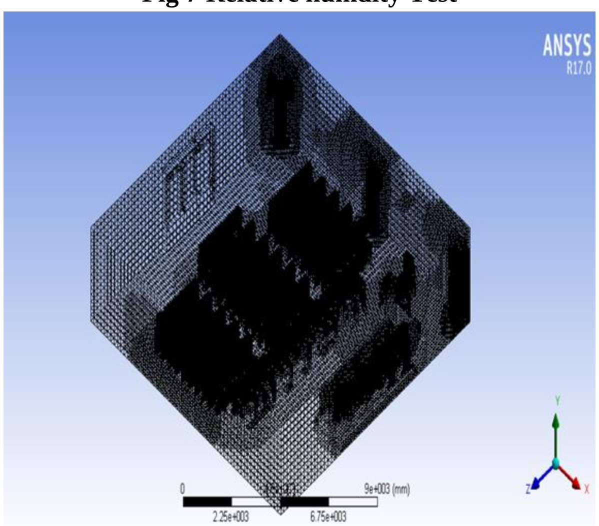

Fig 8 The mesh of hall

4.3 The choice of cases:

In this work, three different cases are drawing for redistribution of air cooling units when the hall was occupied by audience as shown in figures 10, 11 and12 then they compared with the reference case when the hall was empty as shown in figure9. The difference among the cases appeared in the redistribution of air conditioning units with keeping the angle of blade inclination of cooling air supply at three cases 0°. In order to study the effect of A/C redistribution on audience' thermal comfort.

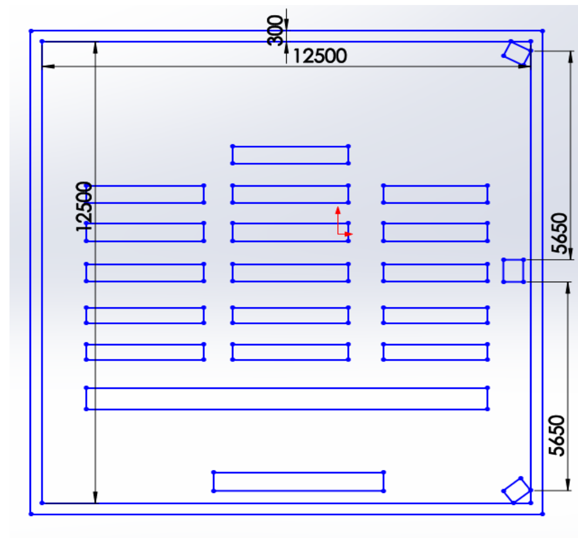

Fig 9 Reference case empty hall

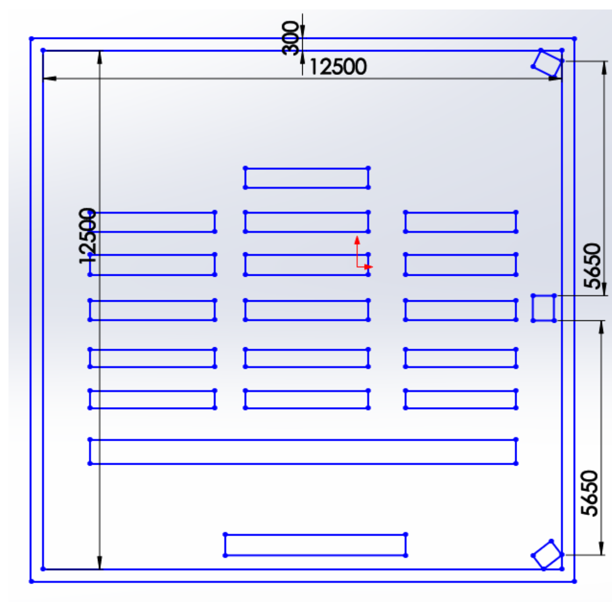

Fig 10 case1 occupied hall

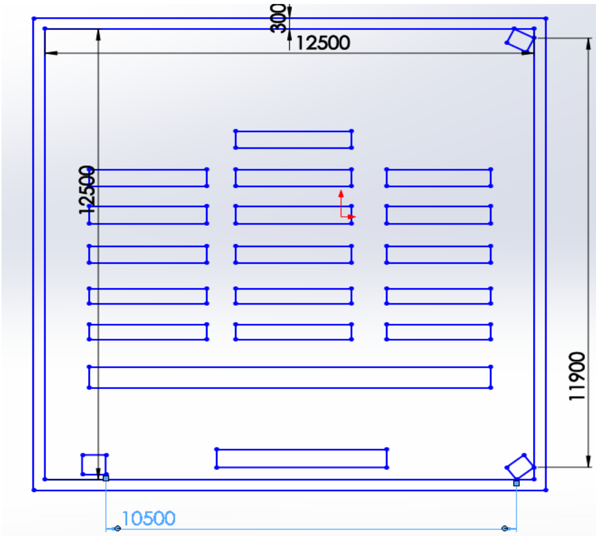

Fig 11case2 occupied hall

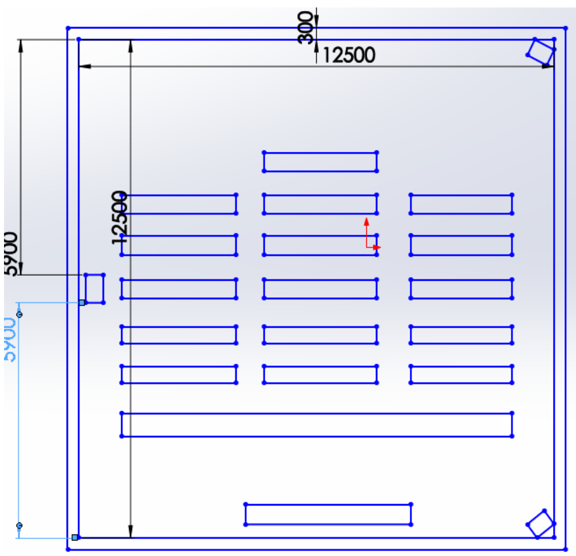

Fig 12 case3 occupied hall

٥. Results and Discussion

This section presents the results of four different cases of re-distribution of air conditioners within the hall, in each case, the angle of air entry was 0° and numerically simulated by Ansys-Fluent v.17. The air velocity, average air temperature and relative humidity in the space at the breath level for humans at 1.1 m were also measured. The results of these factors will calculate the thermal comfort equations approved by (ASHRAE) the PMV and PPD equations. The results comprises the three cases with each other and comparison between these cases and reference case in terms of the air intake angle of 0° the space. The results are discussed and the best case is identified to be the case that provides the best thermal comfort for the audience. The strategy of redistribution of air conditioning within the space is effective as they help improve the thermal comfort of the place with the same load of heat and without an increase in the number of electrical appliances required to condition the space. Also effects of the thermal comfort in the space were observed a general behavior is reported to take place and this behavior applies for all four cases. The values of PMV and PPD are in their best values when the inclination angle is 0°. The reference case the values of PMV and PPD at the angle 0° were -1.05 and 29.7 respectively which were the biggest value that made the thermal comfort very low. The first case values of PMV and PPD in the level 1.1m was 0.18 and 7.33 respectively. The values of PMV and PPD were 0.17 and 6.79, respectively at the level 1.1m for the second case.In the third case it was made the values of PMV and PPD to decrease in the level of breathing 1.1m to be -0.63 and 14.83 respectively. When comparing the reference case with the first case, it is noticed that the improvement rates in PMV values 8% and their percentage at PPD was 5%. The reference case compared with the second case, the improvement rates in PMV and PPD are 14% and 10 % respectively. The improvement rate was the best. When the reference case is compared with the third case, it is found that the values of improvement in PMV is 11 %, and the value of PPD is 8%.As result of all the cases ,it can obtained that the second case was the best.Figures 13 and 14 shows the countor of temperature and velocity.

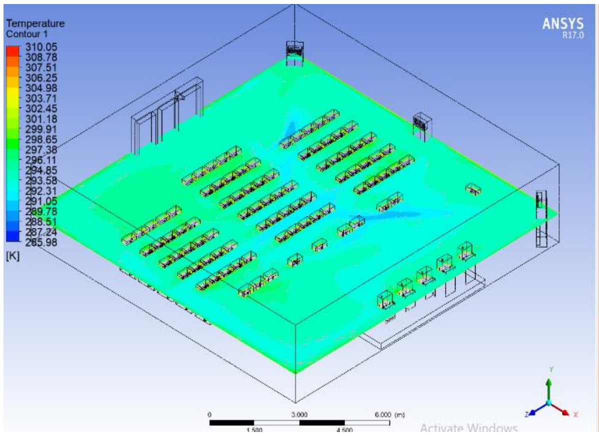

Fig 13 contour of temperature at level 1.1m

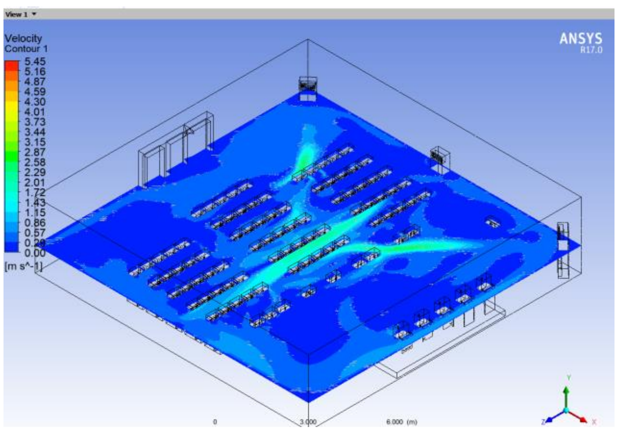

Fig 14 contour of velocity at level 1.1m

٦. Validation

The aim of validation using of (CFD) results by comparing it with previous experimental and numerical results. The work of Mohammed Saeed [18] is used to validate the numerical results of (Fluent 6.2) code with the results of the present work. The results which obtained will discuss and compare by Mohammed Saeed [18] with the used CFD program. The results which were chosen for the line in the centre of the mosque explain there was good agreement in these results .The difference in temperature with Mohammed Saeed [18] was 0.8 ˚C with error percent of 2.58 % as shown in Fig 15, while the difference in relative humdity was 1.7% with error percent was 4.52% as shown in Fig 16.

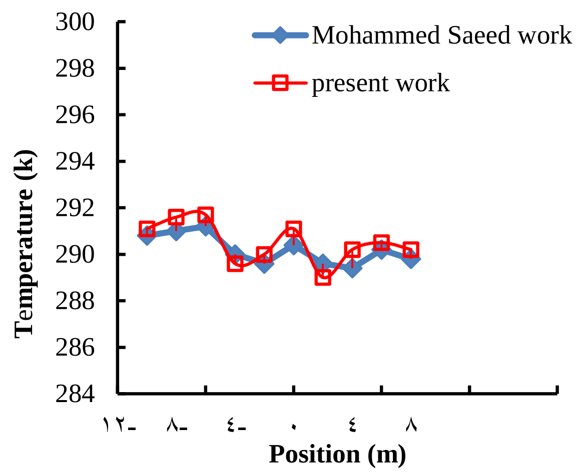

Fig 15 Validation of temperature for present work with Mohammed work(centre line)

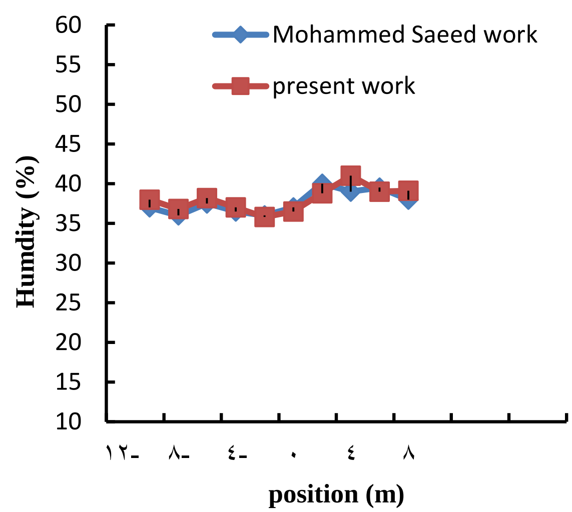

Fig16 validation of relative humdity for present work with Mohammed work(centre line)

7. Conclusion

This study aims to asses and improve the thermal comfort conditions for audience inside the hall in case the hall is filled with people, and on one of the hottest days of the year for the hot - dry climate in Iraq, which falls on 15 Septemper at 1:00 pm at the time of Sunday. Numerical methods of solution were used by Ansys – fluent v.17. The thermal comfort conditions inside the hall are improved in a way that redistributes the air conditioners inside the hall and the inclination angle is 0° to inlet air to provide the best air quality and thus the best distribution of temperature, velocity and relative humidity. There was an identical behavior for the flow of air flow circulation the space from the A/C, at the entry angle of 0°, the direction of the current is parallel to the breathing level and almost horizontal. It caused an air velocity in the rate and decreasing the temperature at the breathing. It means that the angle 0° provides a low mixing of the air between the layers at the breathing level and the higher layers. This behavior was applicable to the four cases of air conditioning distribution. Due to this behavior, the values of PMV and PPD at angle of 0º are the best for all cases [18]. It made a unified current that ascended causing to greater mixing of air between the lower and upper air layers. This behavior can be utilized in the process of improving the level of thermal comfort in the upper layers of the breathing level. Three cases of non-original air conditioning were carried out in the hall. When comparing the reference case with the first case, it is noticed that the improvement rates in PMV values 8% and their percentage at PPD was 5%. The reference case compared with the second case, the improvement rates in PMV and PPD are 14% and 10 % respectively. The improvement rate was the best. When the reference case is compared with the third case, it is found that the values of PMV improvement was 11 %, and the value of PPD is 8%. There is a good improvement in the level of thermal comfort for all cases, but the best was the second case. Air ventilation of the hall was provided by the A/C units themselves to decrease the concentration of CO2 at the breathing level to 0.0015 Kg CO_2/ Kgd.a. All air streams coming from the A/C at the angle of 0° were horizontal and parallel to the level of breathing. In some cases, these streams reach the middle of the hall to meet the stream on the other side, as in the first and third cases. The numerical results of turbulent models were depended the (k-ε RNG) which was the best model and the closest one to the experimental results [18]. Assuming a steady-state and incompressible flow of air as a multi-component fluid which includes dry air, water vapor. The fluid properties are taken as constants. Considering that there is no heat generation and energy fluxes .

Nomenclature

C_p	Air specific heat capacity (J/kg·K)
Clo	Clothing Insulation
C	constant
Di,m	diffusion coefficient for species i
F	force (N)
fcl	clothing surface area factorthermal conduc-

g	tivity, W/m. K°
h_{conv}	Gravitational acceleration (m/s2)
Icl	Convection Heat Transfer Coefficient [W/(m2·K)
i	clothing insulation, (m2·K/W)
j	victor in x –direction
Ji	victor in y –direction
	species diffusion flux i
k	victor in z –direction
K	Kinetic energy (m2/s3)
k	Thermal conductivity (W/m.k)
Le	Lewis number
met	metabolic rate (clo)
m	mass (kg)
M	Metabolic rate, (W/m2)
Pw	water vapor partial pressure (N/m2)
Pws	saturated water vapor pressure (N/m2)
P	pressure (N/m2)
Q	Heat energy (W)
q	Heat flux (W/m2)
AMV	Actual mean vote
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning
A/C	Air condition
PMV	predicted mean vote
PPD	predicted percentage of dissatisfied
T_a	air temperature, (°C)
T_{sk}	average skin temperature
T_n	Predicated neutral temperature (°C)

T_o	outdoor temperature (°C)
T_{cl}	Clothing surface temperature (°C)
U	velocity in x- axis (m/s)
U	Over all heat transfer coefficient (W/m²K)
V	velocity in y- axis (m/s)
Va	air velocity (m/s)
w	velocity in z- axis (m/s)
W	effective mechanical power , (W/m2)
W_v	Energy per unit volume (W/m3)
μ	viscosity ((N.s/m2)
μt	turbulent dynamic viscosity (N.s/m2)
$^{\upsilon}t$	turbulent kinematic viscosity (m2/s)
ε	Dissipation energy (m^2/s^3)

References

[1] Fanger, "Thermal comfort: Analysis and applications in environmental engineering: Fanger, P.O. Danish Technical Press, Copenhagen, Denmark, 1970, 244 pp.: abstr. In World Textile Abstracts," Appl. Ergon., vol. 3, no.3, p. 181, 1972.

[2] Bambang Iskandriawan, "Study of Air Velocity and Temperature Gradient in Lecture Room Through Mixed and Displacement Ventilation Systems to Improve the Thermal Comfort" IPTEK, The Journal for Technology and Science, Vol. 21, No. 2, May 2010.

[3] Hannu Koskela et al, "Flow Pattern and Thermal Comfort in Office Environment with Active chilled Beams," American Society of Heating, Refrigerating and Air-Conditioning Engineers, 2010.

[4] N. Kampelisa et al, "Thermal comfort evaluation in HVAC Demand Response control," 9th International Conference on Sustainability in Energy and Buildings, SEB-17, 5-7 July 2017 .

[5] Aldona Skotnicka, "Assessment of thermal comfort in a lecture hall with the application of instruments for computational fluid dynamics," Technical Sciences, 2016, 19(1), 41–58.

[6] Yuce B. E and Pulat E, "Numerical simulation of thermal flow and comfort in an office room," 12th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, 2016.

[7] I. Hayatu, I. Mukhtar, N. M. Mu, and J. S. Enaburekhan, "An Assessment of Thermal Comfort in Hot and Dry Season (A Case Study of 4 Theaters at Bayero University Kano)," vol. 3, pp. 1117– 1121, 2015.

[8] K. Cena and R. De Dear, "Thermal comfort and behavioral strategies in office buildings located in a hot-arid climate," vol. 26, pp. 409–414, 2001.

[9] Jungsoo Kim and Richard de Dear, "Thermal comfort expectations and adaptive behavioural characteristics of primary and secondary school students," Building and Environment 127 (2018) 13–22.

[10] Sheikh Ahmad Zaki et al,"Adaptive Thermal Comfort in University Classrooms 1 in Malaysia and Japan," Building and Environment, S0360-1323(17)30250, 2017.

[11] Tamaraukuro Tammy Amasuomo and Japo Oweikeye Amasuomo"Perceived Thermal Discomfort and Stress Behaviours Affecting Students' Learning in Lecture Theatres in the Humid Tropics," Buildings 2016, 6, 18.

[12] S.H. Ibrahim et al,"Assessment of thermal comfort in the mosque in Sarawak, Malaysia" ENERGY AND ENVIRONMENT, Vol5, Issue 3 (2014), pp 327-334.

[13] Kwang-Chul Noh et al, "Thermal Comfort and Indoor Air Quality in the lecture room with 4-way cassette air conditioned and mixing ventilation system," Building and Environment 42(2007) 689–698.

[14] Madhavi Indraganti and Djamel Boussaa, "An adaptive relationship of thermal comfort for the Gulf Cooperation Council Countries: The case of Qatari offices," ENERGY AND BUILDING, S0378-7788(17)32901-8, 2017.

[15] Shamila Haddad et al., "Application of adaptive thermal comfort methods for Iranian schoolchildren," Building Research & Information, ISSN: 0961-3218 (Print) 1466-4321, 2016.

[16] Taher M. Abou-deif et al, "Numerical Investigation of Flow Patterns and Thermal Comfort in Air-Conditioned Lecture Rooms," World Academy of Science, Engineering and Technology Vol: 7 2013-05-21.

[17] Marcelo Luiz Pereira"Thermal comfort conditions in a room ventilated with split system-numerical and experimental analysis," 14th Brazilian Congress of Thermal Sciences and Engineering November 18-22, 2012, Rio de Janeiro, RJ, Brazil.

[18] Ahmed Ali and Mohammed Saeed, "Investigation of human thermal comfort and improvement in public places adapted to a hot climate in Iraq," Anbar Journal of Engineering Science (2018).