# STUDY AND CFD ANALYSIS OF THE EFFECT OF SOLAR ENERGY ON THE TIMES OF THE FIVE DAILY PRAYERS IN MOSQUES

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**ABSTRACT**

Mosques are one of the most important places in Muslims life. Thus, air conditioning application in them is a highly important issue. Moreover, there are many heat generation water vapor, and carbon dioxide because of occupancy load in such places. This paper present a simulation study of a typical Mosque with a capacity about 800 persons during five prayer times (FAJR, DHUHR, ASER, MAGHREB, and ESHAA). In this work, PMV (Predicted Mean Vote) model and the PPD (Percentage Predicted Dissatisfied) model are used to predict the thermal comfort. FLUENT® version 14, commercial CFD code, by using computational fluid dynamics (CFD) simulation to calculate the thermal comfort. This by solving the Continuity, momentum, energy, and species transport equations in addition four turbulence. The number of elements that are used in this study is about 5,300,000. All the requirements of ASHRAE conditions have been achieved taking into account average temperature, the average air velocity, and average relative humidity.

INTRODUCTION

People satisfaction with surrounding environment is a state of mind which called thermal comfort. The most parameter effect this condition is air temperature as it sensible and ratable. The thermal comfort is studied in this work with the solar energy effect during five prayers times (FAJR, DHUHR, ASER, MAGHREB, and ESHA) in a typical mosque. Although the air temperature is an efficient tool to predict thermal comfort, it should be related to other parameters that include environmental and personal factors. ASHRAE standard 55/2013 and ISO 7730/2005 are used in this article as they have been used widely as international thermal comfort standards.

A steady state condition has been assumed to obtain human energy balance though obvious standards. "Fanger" (1970) made series of experiments to deduce ASHRAE standards using predicted mean vote (PMV) mode and Predicted Percentage Dissatisfied (PPD) mode and used them to estimate the human mean response to the thermal environment from six thermal variables. All of this related to the combination of personal and environment facts which are independent and have a vital role on thermal comfort. These factors are six and they are air temperature, air velocity, radiant temperature, relative humidity, metabolic rate and clothing insulation [1].

The spirit of the people inside any place or building is effected by the thermal comfort due to its psychological nature. Many papers had been conducted to study the environment effect inside different types of building such as (Abanto et al. 2004) [2] computer room, (Chen 2009) [3] buildings, (Pereira et al. 2012) [4] classroom and (Abou-deif et al. 2013) [5] Lecture Rooms, in addition to other applications.

Other articles have been made within similar conditions building. Thermal environments were made within an air conditioned masjid in Malaysia during prayer times by Hussin et al. (2014). Another field study had been conducted inside the main prayer hall in Al-Malik Khalid, Penang, Malaysia. This study investigated and observed the air temperature and the effect of direction on the temperature Bakhlah and Hassan (2012) [6].

Budaiwi and Abdou (2013) conducted a study to investigate two strategies which are discontinuous operational of HVAC system and operational zoning and the effect of them on energy performance. This study was carried out while the thermal comfort was preserved. This study showed that by choosing an appropriate strategy of discontinuous operational and operational zoning, the annual energy was reduced up to 23% and 30% respectively[8].

Many factors have been studied to show their impact on air outlets position, size and orientation in different building by Khalil et al (2013). These factors are the type of work, place arrangement, and the position of the occupants. 1000000 tetrahedral grid had been used to study those factors through a commercial package. Air flow patterns, temperature, and relative humidity contours, as well as the PMV and PPD, were used to characterize the performance of the air conditioning system [9].

The necessity of selecting Mosques to be the target of this research is that Mosque can be defined as the best place for Muslims for achieving their worship God. Furthermore, they have very important roles in the Muslim daily life and these roles can be summarized as bellow.

1. It is the recommended place for the daily five prayers, communication, discussion about different issues, reciting the holy Quran and social consultation between Muslims each other.
2. Mosque is considered as a stand for linguistics disciplines such as oratory, poetry, eloquence and in most times is devoted to holding some learning circles.
3. It is the ideal avenue to collect aids for needy and poor people, so it is the perfect place for Muslims to urge them regarding help others.[10]

CASE STUDY AND NUMERICAL SIMULATION

One of the archaeological mosques located in Cairo city centre has been chosen to be a good example for this study. The capacity of this Mosque is reasonable compared to the modernistic archaeological Mosques. This capacity refers to (800 worshipers) and an estimated area of (820 m2) and a total height (6m). The congregational prayer in the majority of Islamic countries is notified as a very common phenomenon. This event is taken into consideration as the first step in the design of building requirements. Thus, the human body in the vertical form is modeled as a rectangular shape with dimensions 1.75 [m] height x 0.25 [m] width x 0.5 [m] as it is seen in Fig.1.

The air conditioning of this Mosque is done by using (9A/C) commercial units and the efficiency of cooperation for each unit 60,000 BTU/Hr. Moreover, these equipment are simulated accurately with ideal dimensions of 2 [m] height x 0.4 [m] width x 0.6 [m] length, while the dimensions of air outlet are 0.3 [m] height x 0.6 [m] length and return air inlet are 0.9 [m] height x 0.6 [m] length. The illustration of all cases that have been investigated in this work can be shown in Table -1 and also Fig.4,5,6,7 and 8 describe these cases.

**MATHEMATICAL MODELLING**

**Governing Equations**

The main governing equations are continuity, momentum (Navier Stokes equations), energy, and species concentrations which are described as follows:

The mass balance can be stated as follows, Rate of increase of mass in fluid element = Net rate of mass flow into the fluid element.

Then the continuity equation can be easily formulated as:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \left( \rho \cdot \vec{v} \right) = S_m \qquad (1)$$

In the present case, the compressibility effect may be neglected without loss of accuracy due to the relatively slow air velocities, thus Eq1 yields

$$\rho \left( \nabla \cdot \left( \vec{v} \right) \right) = S_m$$

$$\rho \left( \frac{\partial (u)}{\partial x} + \frac{\partial (v)}{\partial y} + \frac{\partial (w)}{\partial z} \right) = S_m$$

(2)

Newton's second law represents the principal form of momentum conservation. Hence, it can be stated that:

Rate of increase of fluid particle momentum = sum of forces exerted on the fluid particle

For the above equation, the rate of increase of momentum in *x*, *y*, *z* direction is defined as

$$\rho \frac{Du}{Dt}, \rho \frac{Dv}{Dt}, \rho \frac{Dw}{Dt} \text{ respectively.}$$

Where

$\frac{D}{Dt}$ is the substantial derivative $= \frac{d}{dt} + \frac{d}{dx} + \frac{d}{dy} + \frac{d}{dz}$

On the other hand, the forces exerted on the fluid particle may be either a surface or body forces. Surface forces are either pressure or viscous induced. Body forces may be gravity, centrifugal, coriolis, or electromagnetic forces.

Hence the conservation of momentum in an inertial (non-accelerating) reference frame may described by:

$$\frac{\partial \left( \rho \vec{v} \right)}{\partial t} + \nabla \cdot \left( \rho \cdot \vec{v} \vec{v} \right) = -\nabla p + \nabla \cdot \left( \overline{\overline{\tau}} \right) + \rho \vec{g} + \vec{F}$$

(3)

Where : *p* is the static pressure, $\overline{\overline{\tau}}$ is the stress tensor (described below), and $\rho \vec{g}$ and $\vec{F}$ are the gravitational body force and external body forces, respectively. The stress tensor $\overline{\overline{\tau}}$ is given by

$$\overline{\overline{\tau}} = \mu \left[ \left( \nabla \vec{v} + \nabla \vec{v}^T \right) - \frac{2}{3} \nabla \cdot \vec{v} I \right]$$

(4)

Where $\mu$ is the molecular viscosity, $I$ is the unit tensor, and the second term on the right hand side is the effect of volume dilation.

The energy equation may be written in the differential form as:

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + p)) = \nabla \cdot \left( k_{\text{eff}} \nabla T - \sum_j h_j \vec{J} \right)$$

(5)

Where: $k_{\text{eff}}$ is the effective conductivity (keff $= k + k_t$), where $k_t$ is the turbulent thermal conductivity, defined according to the turbulence model being used.

$\vec{J}_j$ is the diffusion flux of species $j$.

$E$ is the total energy of a fluid particle (control volume) defined as:

$$E = h - \frac{p}{\rho} + \frac{v^2}{2} \quad (6)$$

For ideal gases, the sensible enthalpy, $h$, is defined as:

$$h = \sum_j Y_j h_j \quad (7)$$

And for incompressible flows as:

$$h = \sum_j Y_j h_j + \frac{p}{\rho} \quad (8)$$

Where $Y_j$ is the mass fraction of species $j$ and

$h_j = \int_{T_{\text{ref}}}^{T} c_{p,j} dT$ , where $T_{\text{ref}}$ is 298.15 K.

The general differential form for the $i$th species is:

$$\frac{\partial}{\partial t}(\rho Y_i) + \nabla \cdot (\rho \vec{v} Y_i) = -\nabla \cdot \vec{J}_i + R_i + S_i \quad (9)$$

**Turbulence Modeling**

The model of RNG k - ε form as[1]:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i}\left( \alpha_k \mu_{\text{eff}} \frac{\partial k}{\partial x_j} \right) + G_b + G_k - Y_M - \rho \varepsilon$$
$$+ k_k \quad (10)$$

And

$$\frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_i}(\rho \epsilon u_i) =$$

$$\frac{\partial}{\partial x_j}\left( \alpha_\epsilon \mu_{\text{eff}} \frac{\partial \epsilon}{\partial x_j} \right) + C_{1\epsilon} \frac{\epsilon}{k}(C_{3\epsilon} G_b + G_k) - C_{2\epsilon}\rho \frac{\epsilon^2}{k}$$
$$- R_\epsilon + S_\epsilon \quad (11)$$

Where $G_k$ and $G_b$ represent the generation of turbulence kinetic energy because of the gradients of the mean velocity and buoyancy respectively.

$Y_M$: Represents the fluctuating dilatation contribution in compressible turbulence to the general rate of dissipation.

**Thermal Comfort Factors**

PMV-PPD[12] model also named the steady state energy balance; it also has a description of thermal comfort of the human, which based on how energy is equilibrated in the body of human. Equation 12 shows the energy balance:

$$M - W = q_{sk} + q_{res} + S \quad (12)$$

Where

$M$ : is the metabolic heat production rate, $W/m^2$

$W$ : is the mechanical work accomplished rate, $W/m^2$

$q_{sk}$ : is the total rate of heat loss from skin, $W/m^2$

$q_{res}$: is the total rate of heat loss through respiration, $W/m^2$

$S$ : Total rate of heat storage in the body, $W/m^2$

The model gathers the personal and environmental variables, which creates a neutral sensation, is represented such as:

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$$M - W = \quad 3.96 \times 10^{-8} f_{cl}[(t_{cl} + 273)^4 - (\bar{t}_r + 273)^4] + f_{cl}h_c(t_{cl} - t_a) + 3.05[5.73 - 0.007(M - W)P_a] + 0.42[(M - W) - 58.15] + 0.0173M(5.87 - P_a) + 0.0014M(34 - t_a) \quad (13)$$

Where:$t_{cl}$: Represents the clothing temperature, which specified from the following equation via the iteration:

$$t_{cl} = 35.7 - 0.0275(M - W)R_{cl}\{(M - W) \\ - 3.05[5.73 - 0.007(M - W)P_a] \\ - 0.42[(M - W) - 58.15] \\ - 0.0173M(5.87 - P_a) \\ + 0.0014M(34 - t_a)\} \quad (14)$$

Both $h_c$ and $f_{cl}$ values referred above could be determined, from the relationships Fanger utilized as:

$$h_c = \begin{cases} 2.38(t_{cl} - t_a)^{0.25} & 2.38(t_{cl} - t_a)^{0.25} > 12.1\sqrt{V} \\ 12.1\sqrt{V} & 2.38(t_{cl} - t_a)^{0.25} < 12.1\sqrt{V} \end{cases} \quad (15)$$

$$f_{cl} = \begin{cases} 1.0 + 0.2I_{cl} & I_{cl} < 0.5clo \\ 1.05 + 0.1I_{cl} & I_{cl} > 0.5clo \end{cases} \quad (16)$$

Fanger (1970) [11] connected PMV index to this unbalance by the following equation:

$$PMV = \left(0.303 \, exp^{(-0.036M)} + 0.028\right)L \quad (17)$$

Where **L**: Represents the thermal load as specified previously.

$$PPD = 100 - 95 \exp[-(0.03353PMV^4 \\ + 0.2179PMV^2)] \quad (18)$$

## OCCUPANT DATA AND BOUNDARY CONDITIONS

Fanger's model [11] as a prediction tool for thermal comfort assumes a steady state conditions. Thus, the indoor initial and boundary conditions are simply assumed according to moderate summer conditions in Tables 2 & 3 .

## RESULTS AND DISCUSSION

This section is aimed to explain the numerical results and their discussion. Whereas, Fig.9 refers to the temperature contours of Case1 at 1.6m from ground level in the z direction and XY plane. It is clear from this figure that the lowest temperature distribution is close to the discharge openings. It is clear that the temperature distribution in this plane is almost the same with this case. The acceptable reason for that the difference between the inside temperature and outside is not that an issue because FAJR prayer time is taken for this case. For the same case, Fig.10 provides relative humidity contours which say that the highest level of this distribution occurs near the discharge openings.

Moving into Fig.11 which shows two parameters, first of all, the distribution of air velocity int XY plane and secondly, the air speed between worshippers themselves. PMV contours in the XY plane has illustrated in Fig.12,and Fig.13 explains PPD contour in the same plane. It is noticed that the ratio of worshipers in case1 is between 5% to 15%, that means it is assumed to be in the thermal comfort term.

The coming cases are case 2&3 as it is shown in Fig.14 and 19 and they are for the DHUHR and ASER times. The increasing in rate of temperature distribution, exactly close to the walls and roofs. The rise in temperature is the result of the heat of solar energy which is fixed on walls and roof in both cases. Finally, from the calculated results for cases 1, 4 and 5, is noticed that the low level of temperature is a result of the weakness of the solar energy role and the surrounded temperature of the mosque, as they are demonstrated in Figs.9, 24 and 29.

## CONCLUSIONS

Many factors have been studied numerically in this article to show the effect of these factors on the thermal comfort condition in a typical Mosque. One of the most important factors is the number of people inside the Mosque.

This study shows that the increase of this number causes a significant increase in convection load in the mosque. This load is more than the convection load from walls, roof, floor and lighting. Time of prayer also plays an important role. Day prayers consume more energy than night prayers. Finally, Roof and wall heat effect the relative humidity. Any inccrease in this heat caus a decrease of relative humidity

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**Nomenclature & Subscripts**

| | |
|---|---|
| C | Convective heat loss (W/m2) |
| $C_p$ | Specific heat (J/kg K) |
| $C_{res}$ | Sensible heat loss due to respiration (W/m2) |
| E | Evaporative heat loss (W/m2) |
| $F_{cl}$ | Clothing area factor |
| h | Heat transfer coefficient (W/m2 K) |
| m | Body mass (kg) |
| M | Metabolic heat production (W/m2) |
| p | pressure (kPa) |
| q | heat flow (W/m2) |
| R | radiative heat loss (W/m2) |
| $R_a$ | thermal resistance of air layer (m2 8C/W) |
| $R_{cl}$ | thermal resistance of clothing (m2 K/W) |
| $R_{e,cl}$ | evaporative resistance of clothing (m2 kPa/W) |
| $R_{e,t}$ | total evaporative resistance (m2 kPa/W) |
| $R_t$ | total resistance of a bedding system including the air layer around a covered body (m2 8C/W) |
| S | heat storage (W/m2) |
| t | time (s) |
| T | temperature (K or 8C) |
| v | air velocity (m/s) |
| w | skin wettedness |
| W | external work (W/m2) |

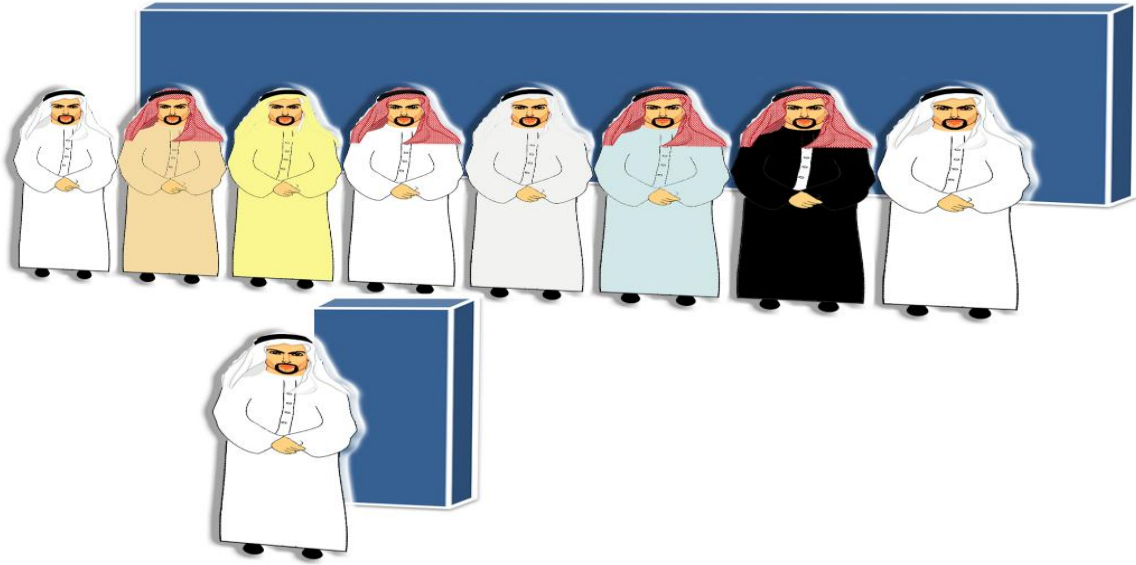| | |
|---|---|
| **Subscripts** | |
| a | water vapour in ambient air |
| a,s | water vapour in saturated air at ambient temperature |
| b | body tissue |
| c | convective |
| cl | clothing |
| dif | moisture diffusion through skin |
| e | evaporative |
| o | operative |
| r | radiant |
| res | respiration |
| rew,req | regulatory sweating required for comfort |
| rsw | regulatory sweating |
| sk | skin compartment |
| sk,req | skin required for comfort |
| sk,s | water vapour in saturated air at skin temperature |
| PMV | Predicted Mean Vote |
| PPD | Percentage Predicted Dissatisfied |

|  |  |
|---|---|
|  |  |



**Figure 1:** A worshiper body is modelled in a standing position and consequently a row of worshipers
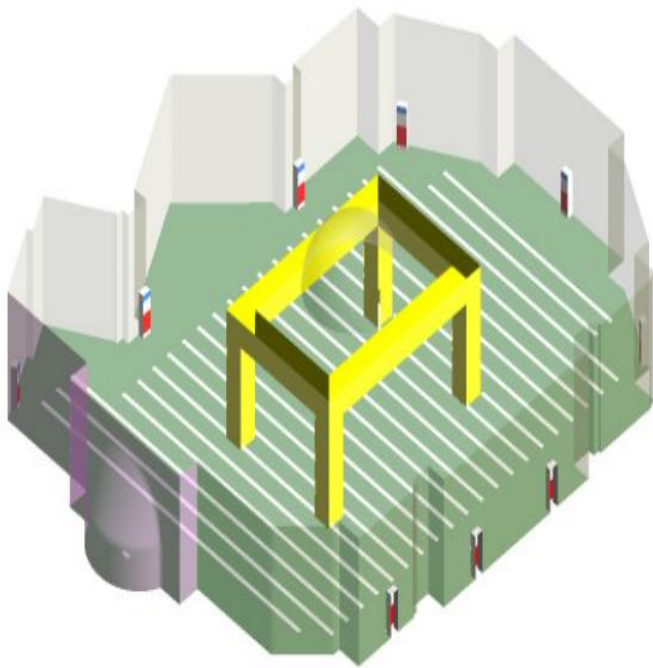
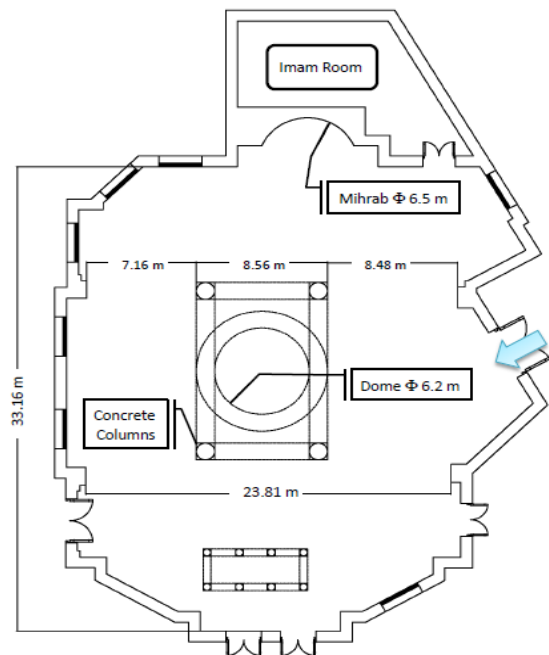

**Figure 2:** Illustrating the distribution of 9A/C units
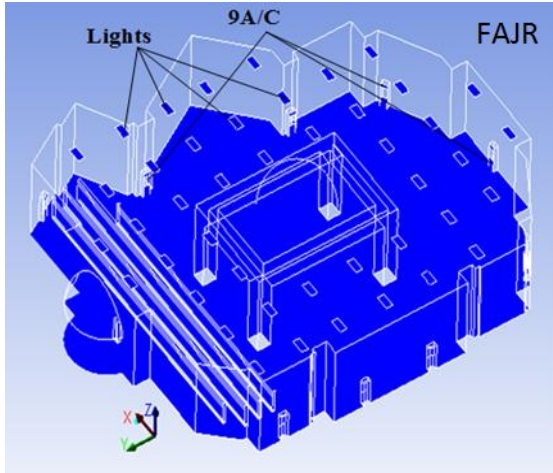


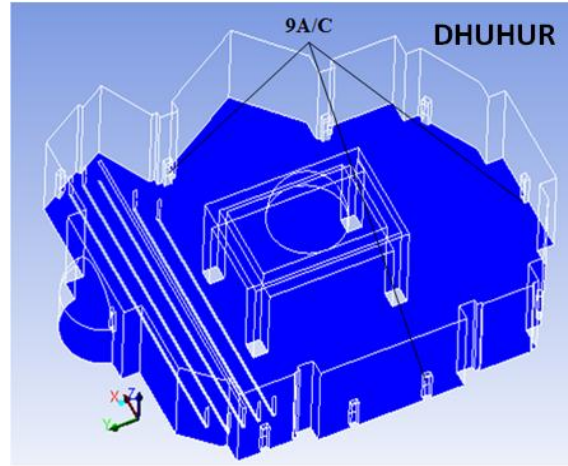**Figure 3:** horizontal plan view of mosque

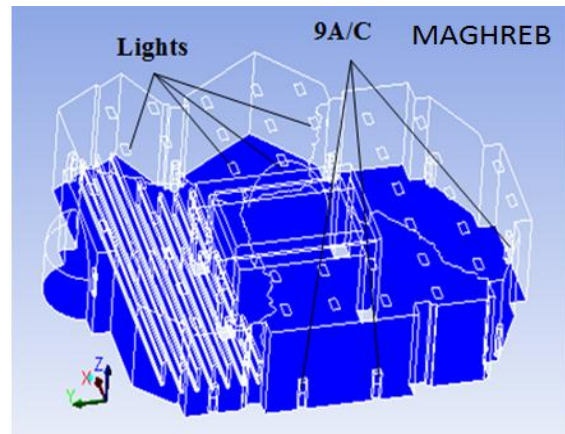**Figure 4 : Case 1**



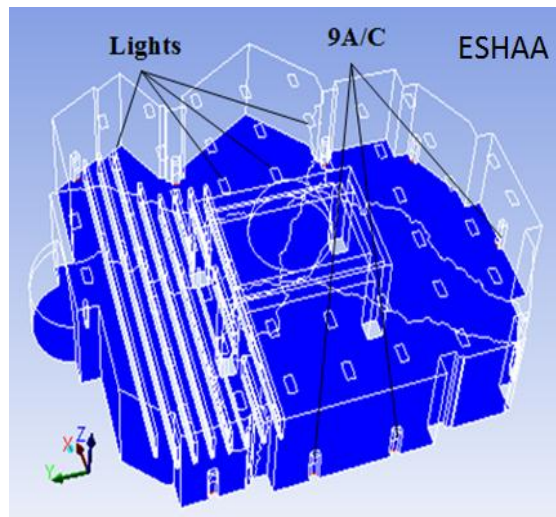**Figure 5 : Case 2**
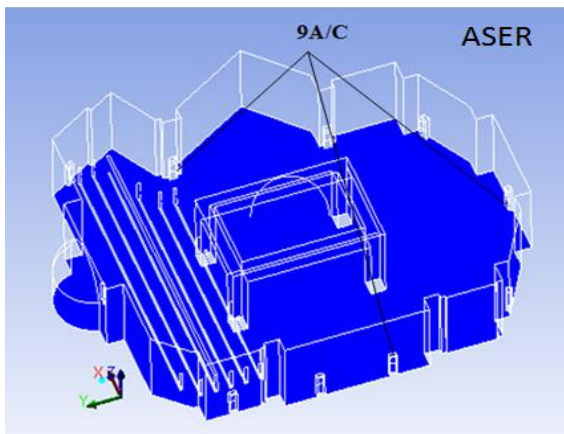


**Figure 6 : Case 3**



**Figure 7 : Case 4**



**Figure 8 : Case 5**

- **Case 1 ( FAJR )**
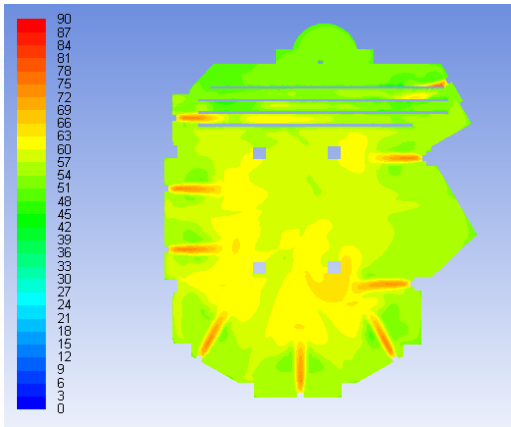


**Figure10 :Case1 Relative Humidity contours(%)**
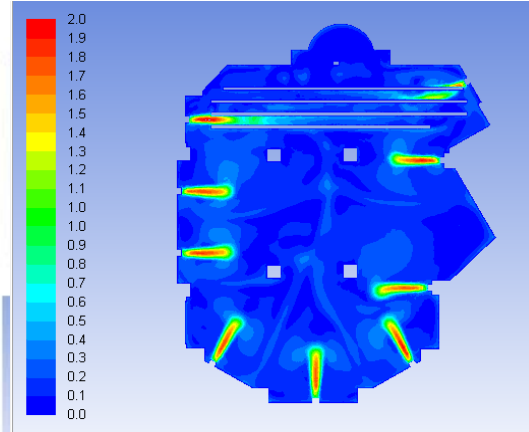


**Figure11:Case1 Velocity magnitude contours (m/s)**

**Horizontal plane (XY) at Z= 1.6m from floor level**
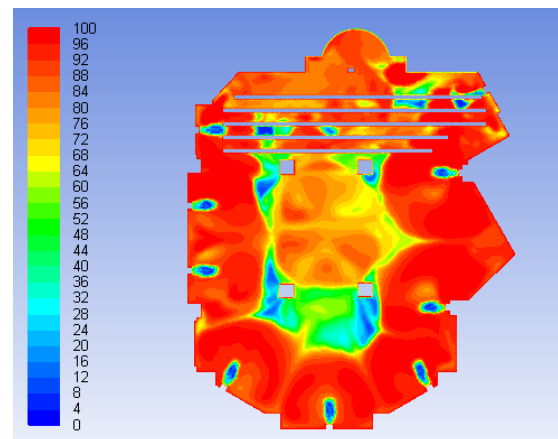
Figure 9: Case 1 Static Temperature contours(K)  a- Horizontal plane (XY) at Z= 1.6m from floor level . b- Vertical plane (XZ) at Y=9.5m  from Qebla. c- Vertical plane (YZ) at X=12.5m from lift side
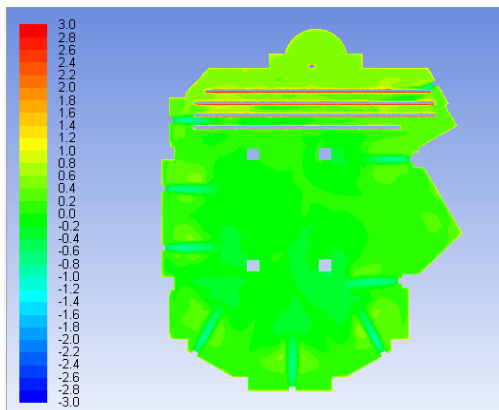


**Figure12 :Case1 PMV contours**
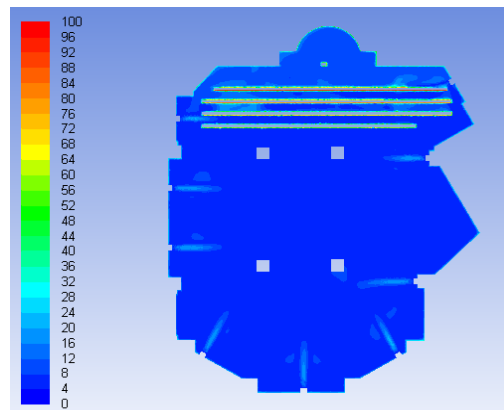


**Figure13:Case1 PPD contours (%)**

**Horizontal plane (XY) at Z= 1.6m from floor level**

- **Case 2 ( DHUHUR )**
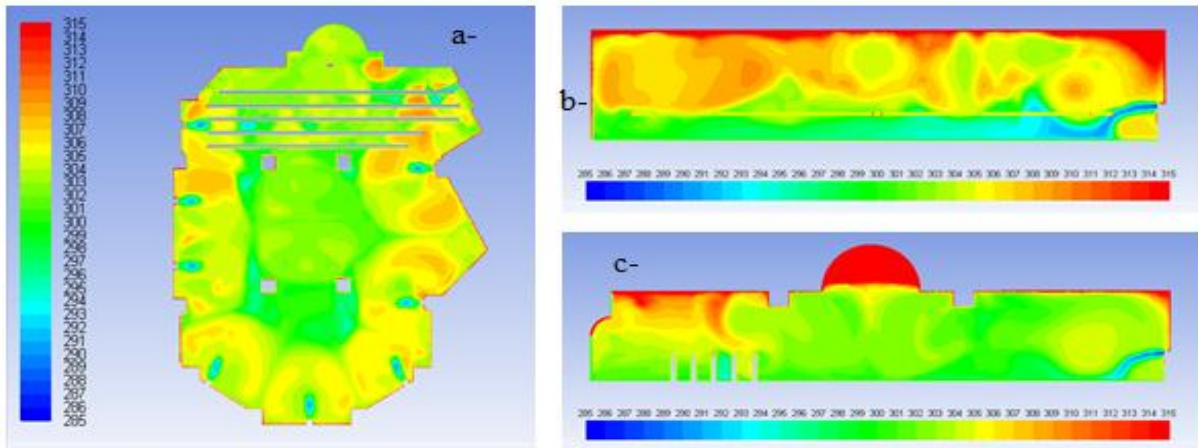


Figure 14: Static Temperature contours (K). a- Horizontal plane (XY) at 1.6 m from floor level.
b- Vertical plane (XZ) at Y=9.5m from Qebla. c- Vertical plane (YZ) at X=12.5m from lift side
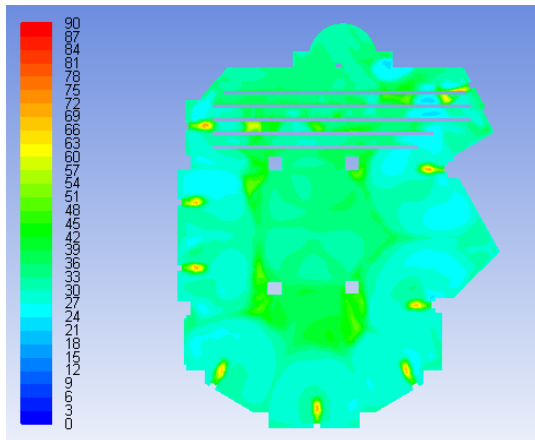


**Figure15 :Case2 Relative Humidity contours(%)**      **Figure16:Case2 Velocity magnitude contours (m/s)**
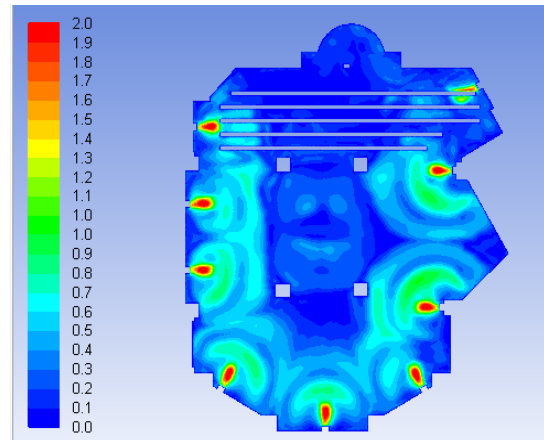
**Horizontal plane (XY) at Z= 1.6m from floor level**
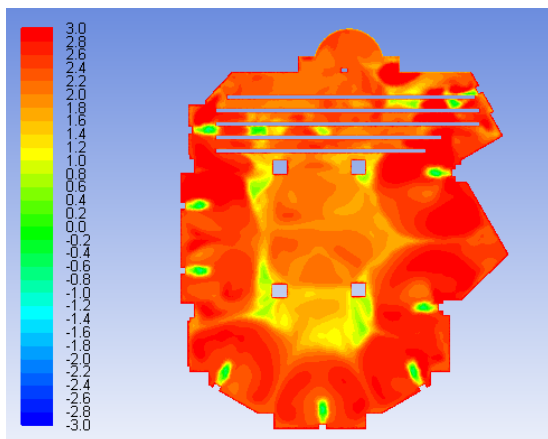


**Figure17 :Case2 PMV contours**      193      **Figure18:Case2 PPD contours (%)**

**Horizontal plane (XY) at Z= 1.6m from floor level**

- **Case 3 ( ASER )**
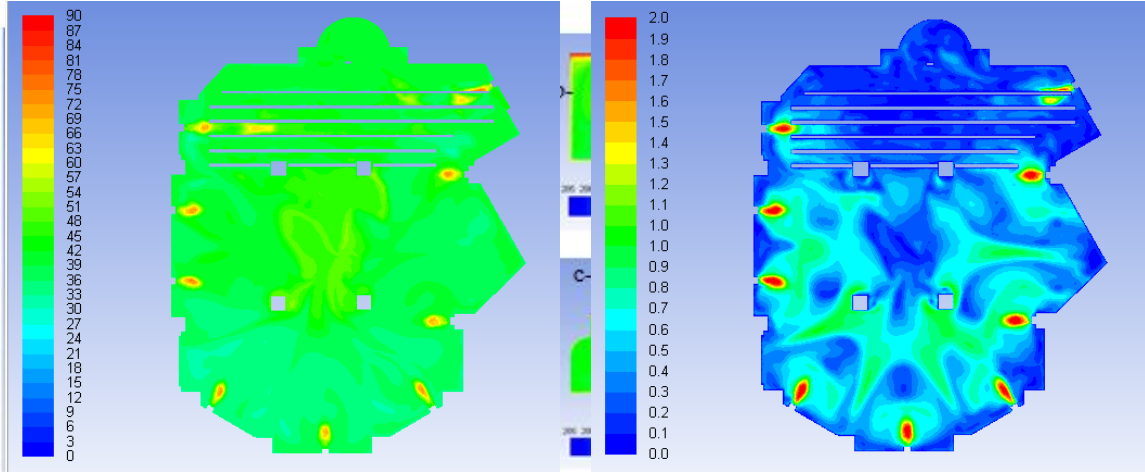


**Figure20 :Case3 Relative Humidity contours(%)**     **Figure21:Case3 Velocity magnitude contours (m/s)**

Figuer 19: Case3 Static Temperature contours(K).a- Horizontal plane (XY) at Z=2.6m from floor level. b- Vertical plane (XZ) at Y=9.5m from Qebla.c- Vertical plane (YZ) at X=12.5m from lift side

**Horizontal plane (XY) at Z= 1.6m from floor level**
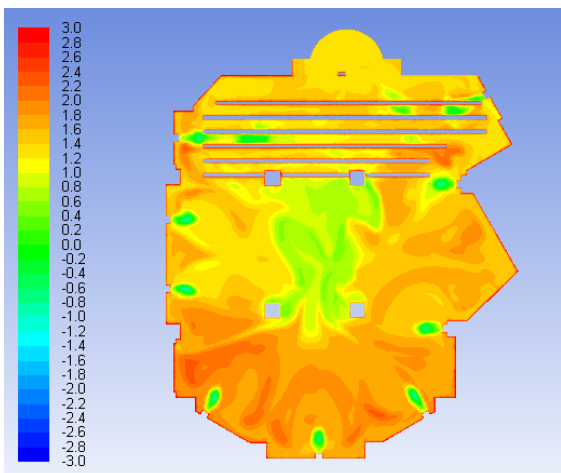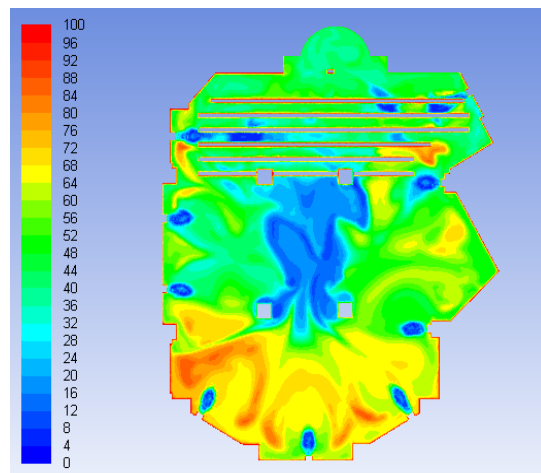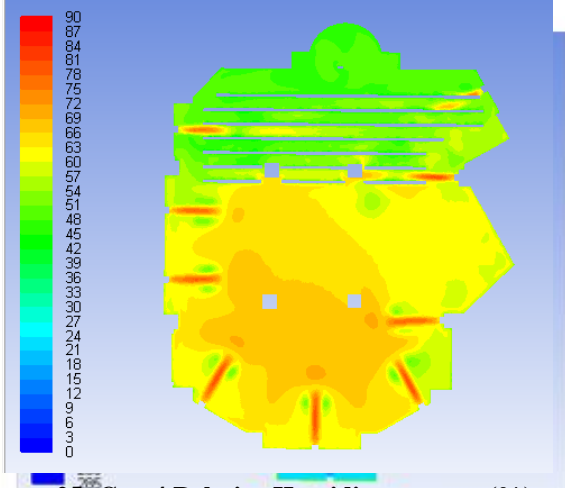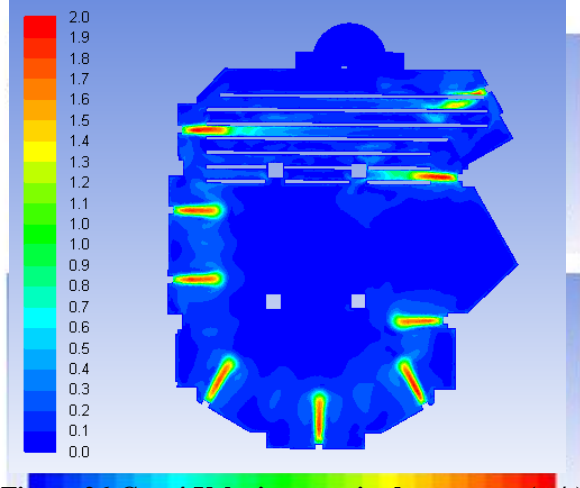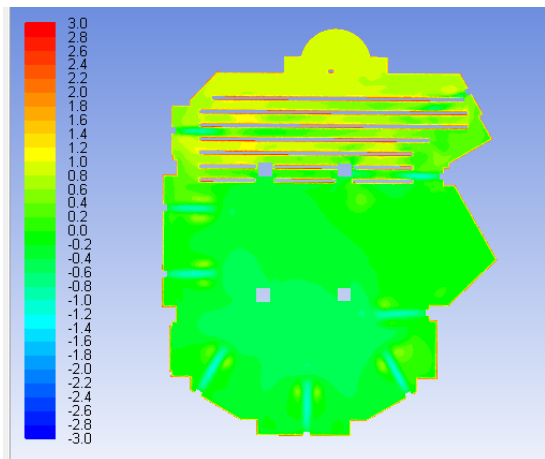


**Figure22 :Case3 PMV contours**                              **Figure23:Case3 PPD contours (%)**

**Horizontal plane (XY) at Z= 1.6m from floor level**

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- **Case 4 ( MAGHREB )**



**Figure 25 :Case4 Relative Humidity contours(%)**



**Figure 26:Case4 Velocity magnitude contours (m/s)**

Figure24 :Case4 Static.Temperature contours (K), a- Horizontal plane (XY) at Z= 1.6m from floor level. b- Vertical plane (XZ) at Y=9.5m from Qebla. c- Vertical plane (YZ) at X=12.5m from lift side
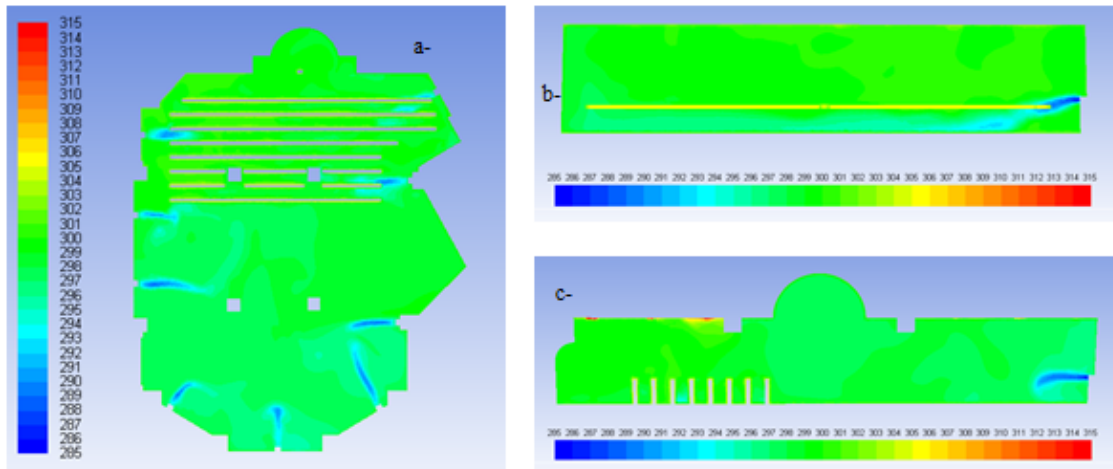
**Horizontal plane (XY) at Z= 1.6m from floor level**



**Figure 27 :Case4 PMV contours**
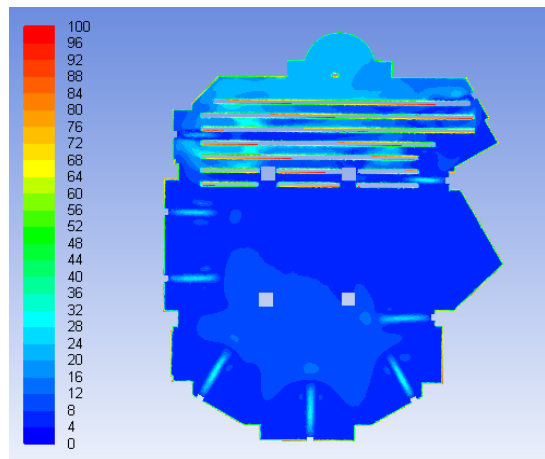


**Figure 28:Case4 PPD contours (%)**

**Horizontal plane (XY) at Z= 1.6m from floor level**

- **Case 5（ESHAA）**



Figure 29 :Case5 Static Temperature contours (K). a-Horizontal plane (XY) at Z= 1.6m from floor level. b- Vertical plane (XZ) at Y=9.5m from Oebla. c- Vertical plane (YZ) at X=12.5m from lift side
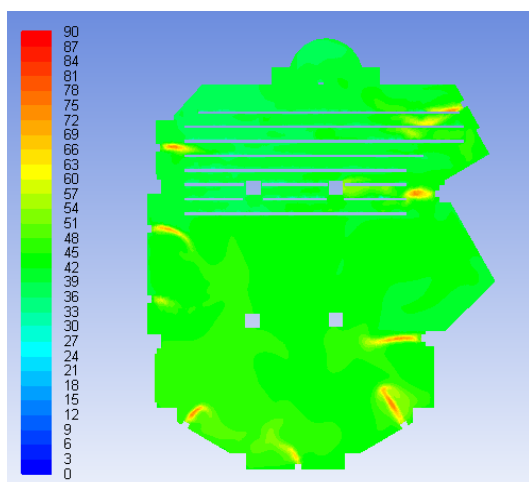


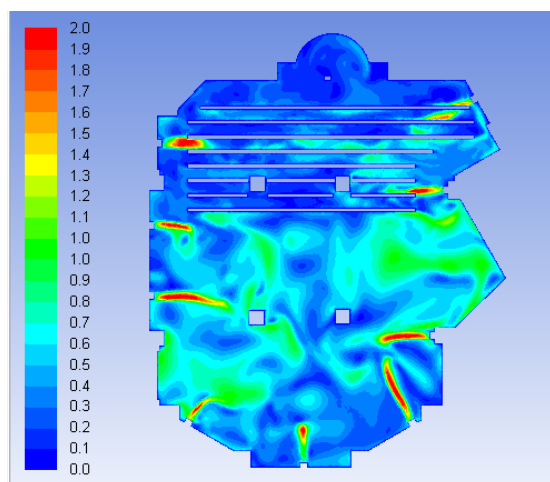**Figure 30 :Case5 Relative Humidity contours(%)**    **Figure 31:Case5 Velocity magnitude contours (m/s)**
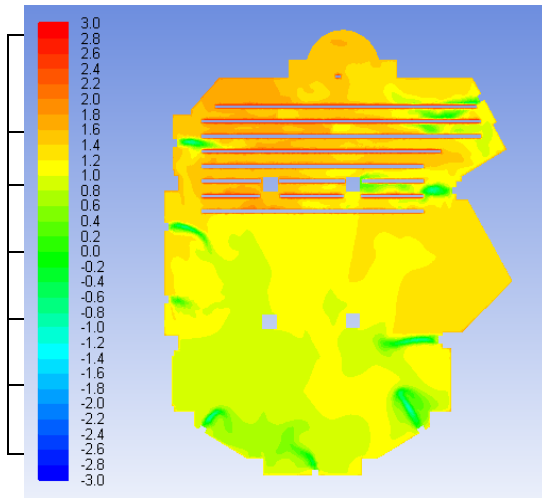
**Horizontal plane (XY) at Z= 1.6m from floor level**
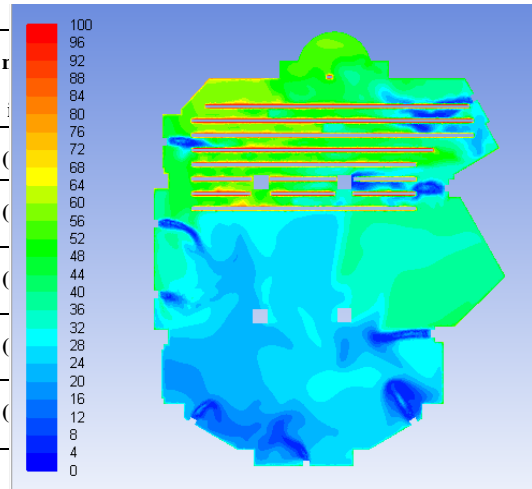
**Figure 32 :Case5 PMV contours**

**Figure 33:Case5 PPD contours (%)**

**Horizontal plane (XY) at Z= 1.6m from floor level**

**Table 1:** Rows and worshipers number in each case study

**Table 2 :** Occupant data details.

| Occupant Data | value |
|---|---|
| Gender | Male |
| Activity type | Standing, relaxed |
| Metabolic rate | 1.2 [met]. |
| Skin temperature | 307 [K] |
| breathing rate | 8 L/min |
| Species mass fraction from mouth | 0.042 $kg_w$/ $kg_{d.a}$ |
| $CO_2$ concentration (volume or mole fraction) in the human respiration | 0.03715 $L_{co2}/L_a$ |

**Table 3 :** Boundary Conditions

| Boundary | value |
|---|---|
| Roof temperature | 306 [K] |
| Walls temperature | 305 [K] |
| Outlet air temperature from A/C | 286 [K] |
| Outlet air velocity from A/C | 3.15 [m/s] |
| Initial air temperature | 304 [K] |

| Lights | 10.6 W/m$^2$ |
|--------|--------------|