Numerical simulation and experimental validation of the ventilation system performance in a heated room

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Abstract

The time spent by the occupant indoor the building is significant; therefore, the central objective of the major research was the evaluation of the thermal sensation for the existing people. This study examines the numerical simulation in a room containing a manikin sitting in front of a computer. The computational fluid dynamics (CFD) tools were considered using ANSYS Fluent 16.2 software. This software exploits the finite volume method that is based on the resolution of the Navier-Stokes equations. The distribution of the temperature, velocity, static pressure, turbulent kinetic energy, turbulent viscosity, and turbulent dissipation is tested in different planes and different directions to characterize the airflow indoor a heated room. Equally, the thermal comfort is examined by calculating the predicted mean vote (PMV). The comparison between the numerical results and the experimental data founded from the literature prove that the supply of airflow was affected by the presence of the heat sources and the thermal climate is considered as a slightly hot. The use of the adequate meshes is in a good agreement with the experimental data and confirms the validity of the numerical approach.

Keywords Ventilation · CFD · Manikin · Computer · Airflow · Indoor air quality · PMV

Introduction

The heating, ventilation, and air conditioning (HVAC) systems have important energy consumption. In this way, the decrease of the dependence between fossil fuel energy and the building consumption becomes a significant global objective. Furthermore, the time spent by the occupants is very important. For these reasons, it is necessary to maintain a healthy space with reasonable energy consumption (Guo et al. 2004; Calay and Wang 2013; Guo et al. 2012).

In this context, several researches have been developed in order to maintain a healthy indoor air keeping a reasonable energy consumption of the HVAC systems. The determination of the factors affecting the humanity and the

environmental quality is well studied by many researches (Khan et al. 2020). In fact, ventilation system can be controlled by a deep knowledge of the characterization of the thermal phenomenon around the occupied zone. Generally, the efficiency of this system is profoundly influenced by the climatic conditions and buildings design. In this way, Chen et al. (2015) followed the effect of the heat flux using a heat gain. It generates the heat flux for the manikin and the temperature was measured using thermocouples. They prove that the presence of the heat gain increase the indoor temperature and the walls' temperature.

Furthermore, the use of the thermal manikin is very important as a heat source to generate a thermal plume (Chen et al. 2015). Today, thermal manikins are the most realistic to evaluate the heat and the mass transfer of a human body in the buildings (Fan and Chen 2002). For the thermal comfort in occupied zones and the thermal indices, Zhang et al. (2019) studied the characterization of microclimates using predicted mean vote (PMV) index in order to optimize the thermal sensation indoor classes. In addition to that, Kong et al. (2019) examined a comparison between two methods assisting to calculate an assessment index. The buildings are really equipped with different heat sources such as computer and lighting lamps. Consequently, the presence of these sources cannot be negligible. Cetin (2020) investigated the relation

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between the thermal comfort and the geographical conditions using the physical equivalent temperature (PET) index. This study demonstrates the importance of a correct future urban conception planning taking attention to the climatic conditions. Zhang et al. (2017) were modifying this index. Thus, the PMV will be in function of the indoor temperature and the airflow rate. In other words, Ahmed and Gao (2017) used the predicted mean vote (PMV) and the predicted percentage of dissatisfied (PPD) to examine a novel exhaust ventilation systems performance indoor an office room.

Presently, the propagation of the coronavirus increases significantly the researches in order to ensure a healthy environment in the buildings such as houses, hospital, and offices. In this context, Bashir et al. (2020) tested the relation between the climatic factors, the environment pollutant, and the COVID-19 situation. Their study demonstrates that the temperature presents the most significant indicator to evaluate the current state of this recent virus. In addition, the concentration of indoor pollutants in South America was tested by Zambrano-monseratte Manuel (2020). They prove that the concentration of O_3 is significantly increased in 2020. As well as, Jiang and Xu (2020) examined the correlation between the COVID-19 death and the characteristic of ambient air such as the indoor pollutant concentration rate and the index of indoor air quality using the method of Poisson and Pearson. Based on their study, it has been noted that the PM2.5 pollutant has a positive relation with the COVID-19 death. Sher et al. (2020) were interested on the indoor pollution sources and the effect of the climate change on the indoor air quality in Malaysia.

In this context, the aim of this study is to examine the impact of the presence of two heat sources on the indoor air quality. We are interested in the simulation of the air distribution indoor a ventilated room occupied by a manikin and a computer using a CFD tool to characterize the thermal climate using PMV method and to understand the thermal transfer around the persons.

Geometrical system

The SolidWorks design software is used to prepare the geometry. Figure S1 shows the geometrical parameters of the considered room. Indeed, the dimensions of the room were a length $L = 6$ m, a width $W = 3.9$ m, and a height $H = 2.35$ m. The inlet opening is characterized by a length l equal to 0.4 m and a width w equal to 0.15 m. In addition, the outlet has a length equal to 0.34 m and a width equal to 0.14 m. The presence of the occupants was simplified by a manikin. Hence, this room is occupied by a manikin with 1.6 m at the height sitting in front of a computer which was considered as a second heat source. This configuration was tested experimentally by Xu et al. (2009).

Numerical method

To simulate the indoor airflow and to predict the indoor distribution, a numerical model is needed. Accordingly, in this section, the mathematical formulation, the boundary conditions, and the meshing choice are presented.

Mathematical formulation

For the characterization of the air flow indoor the buildings, the flow governing equations is used. In fact, the computational fluid dynamics (CFD) is a helpful tool to facilitate the study of HVAC system. In this field, three-dimension simulations were developed using the software ANSYS Fluent 16.2 in order to characterize the air flow indoor a ventilated room. The Navier-Stokes equations were resolved based on the finite volume method. Thus, the continuity, the momentum, and the energy equation can be expressed as (Fluent 2012).

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = 0 \quad (1)$$

$$\begin{aligned} \frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = & -\frac{\partial p}{\partial x_i} \\ & + \frac{\partial}{\partial x_i} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_i}{\partial x_i} \right) \right] \\ & + \frac{\partial (-\rho \overline{\mu'_i u'_j})}{\partial x_j} \end{aligned} \quad (2)$$

$$\begin{aligned} \frac{\partial \rho E}{\partial t} + \nabla \cdot (\rho(E + p) \vec{V}) \\ = \nabla \cdot (\lambda_{eff} \nabla T - \sum \lambda_{eff} h_j \vec{J}_j + \tau \vec{V}) + S \end{aligned} \quad (3)$$

where ρ is the density (kg m^{-3}), p is the pressure (Pa), μ is the dynamic viscosity ($\text{kg m}^{-1} \text{s}^{-1}$), x_i is the Cartesian coordinate (m) for $i = 1, 2$, and 3 , u_i is the velocity component in the x_i direction (m s^{-1}), δ_{ij} is the mean strain rate tension, $\rho \overline{\mu'_i u'_j}$ is the Reynolds stresses, E is the total energy, λ_{eff} is the effective heat conduction coefficient, h_j is the enthalpy, $\vec{J}_j$ is the diffusion flux of species, and S is the term source.

For this numerical simulation, the RNG k- ϵ turbulence model is used. This model is adaptable for the study of the airflow characterization with a high precision. Thus, the RNG k- ϵ model equations were obtained by the Navier-Stokes equations using a mathematical method nominate renormalization group. In fact, these equations can be formulated as:

$$\begin{aligned} \frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k u_i) = & \frac{\partial}{\partial x_j} \left(\alpha \mu_{eff} \frac{\partial k}{\partial x_j} \right) + G_k \\ & + G_b - \rho \epsilon - Y_m + S_k \end{aligned} \quad (4)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left(\alpha_\varepsilon \mu_{eff} \frac{\partial \varepsilon}{\partial x_j} \right) + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon \quad (5)$$

where G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients, G_b is the generation of turbulence kinetic energy due to buoyancy, Y_M represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate, and α_k and α_ε are respectively the inverse of the effective Prandtl numbers for k and ε . S_k and S_ε are user-defined source terms. $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are equal respectively to 1.42 and 1.68.

Boundary conditions

The boundary conditions are directly related to the stability of the results and they are applied for the whole domain. In fact, Figure S2 illustrates the considered boundary conditions. As well, based on the experimental study done by Xu et al. (2009), two heat sources are considered; they used a heating panels to generate a heat flux equal to $H_F = 76$ W for the manikin and a lamp to generate a heat flux equal to $H_F = 40$ W for the computer. In the inlet, the air is characterized by a temperature equal to $T = 19$ °C and a flow rate equal to $Q = 43$ m³ h⁻¹. The air properties are presented in Table 1. At the outlet, the pressure is equal to the atmospheric conditions. The temperatures of the different walls are presented as well in this figure.

Meshing choice

The domain mesh is unstructured formed by a tetrahedral element. The mesh is locally refined in the important gradient area. Indeed, Figure S3 shows the five meshing used for this study. In fact, the element number ranges from 1571279 to 24666798. Figures 1 and 2 show the distribution of the temperature and the velocity along the directions defined in Figure S4. From these results, it has been seen that the temperature increases as a function of the height. Around the manikin, it can attain 299 K at a height equal to $H = 1.8$ m. The meshing with the cell number is equal to 2166447 and 2466198 are the nearest to the experimental data. For the velocity distribution, the comparison between the numerical results and the experimental values

proves that the curves of 2123020 cells, 2166447 cells, and 2466198 cells are the nearest to the experimental curve. In these conditions, the velocity is lower than $V = 0.1$ m s⁻¹ for the zone occupied by the manikin. Based on these results, it concludes that the refinement of 3D domain with 2166447 cells is in good agreement with the experimental study.

Results and discussions

The temperature, the velocity fields, the static pressure, the turbulent kinetic energy, the eddy viscosity, and the turbulence eddy dissipation are studied in this numerical simulation using ANSYS Fluent 16.2. Indeed, Fig. 4 shows the visualization planes.

Temperature

Figure 3 shows the temperature distribution in the plane defined by $x = 0.2$ m, $x = 1.9$ m, $y = -0.1$ m, and $y = 0.65$ m. According to these results, the thermal plume was formed above the heat sources consisting of the manikin and the computer. This is explained by the temperature of the human body for a normal activity being equal to 37 °C; therefore, the heat transfer between the human body skin and the indoor air in the building causes the formation of this plume (Bjørn and Nielsen 2002; Johnson et al. 1996). In this area, the temperature reaches $T = 303$ K. The coldest zone was formed below the table and closes the floor where it is equal to supply air temperature $T = 292$ K. In the rest of the room, the temperature range from $T = 297$ K to $T = 299$ K. Indeed, it is interesting to mention that the temperature difference between the floor and the head of the manikin is significant. This difference ranges from $\Delta T = 5$ K to $\Delta T = 7$ K. It was confirmed by different researches for example Causone et al. (2010) and Krajčik et al. (2013). This stratification of high temperature makes the thermal indoor environment discomfort (ISO 2005). Consequently, it is necessary to decline this difference to maintain a healthy homogenous indoor environment.

Velocity

Figure 4 presents the velocity fields in the planes defined by $x = 0.2$ m, $x = 1.9$ m, $y = -0.1$ m, and $y = 0.65$ m. According to these results, it has been observed that the velocity is the most important in the inlet and outlet opening as presented in the planes $x = 0.2$ m and $x = 1.9$ m and it is equal to $V = 0.2$ m s⁻¹. In addition, the planes $x = 0.2$ m, $y = -0.1$ m, and $y = 0.65$ m prove that the velocity remains highest where it ranges from $V = 0.11$ m s⁻¹ to $V = 0.17$ m s⁻¹. The presence of the heat sources causes the perturbation of the air flow. Therefore, the recirculation zones were formed around the

Table 1 Air properties in numerical simulation

Properties	Values
Density	Constant = 1.225 kg m ⁻³
Cp (specific heat)	Constant = 1006.43 J kg ⁻¹ K ⁻¹
Thermal conductivity	Constant = 0.0242 W m ⁻¹ K ⁻¹
Viscosity	Constant = 1.7894 × 10 ⁻⁵ kg m ⁻¹ s ⁻¹

Fig. 1 Temperature profiles along different directions in the medium plane $x = 0.2$ m, (a) $y = 0.29$ m, (b) $y = -0.1$ m, (c) $y = -0.45$ m

manikin and the computer. The non-homogeneity of the velocity indoor the room produces the human thermal discomfort.

Static pressure

Figure S6 shows the static pressure distribution in the plane defined by $x = 0.2$ m, $x = 1.9$ m, $y = -0.1$ m, and $y = 0.65$ m. Based on these results, the atmospheric pressure in the inlet

and outlet opening is governed by the boundary conditions and it is equal to $P = 0$ Pa. In the rest of the room, the pressure increases progressively. Near to the walls, especially the west, it is maximized and it attains 3 Pa.

Turbulent kinetic energy

Physically, the dissipation of energy is made by the fluctuating viscosity stress. Thus, the production of the

Fig. 2 Velocity profiles along the different directions in the medium plane $x = 0.2$ m, (a) $y = 0.29$ m, (b) $y = -0.1$ m, (c) $y = -0.45$ m

turbulent kinetic energy is defined in the characteristic equation of the turbulence models by the term G_k (Fluent 2012). Indeed, Figure S7 shows the distribution of the turbulent kinetic energy in the different planes defined by $x = 0.2$ m, $x = 1.9$ m, $y = -0.1$ m, and $y = 0.65$ m. According to these results, the kinetic energy ranges from $k = 8 \times 10^{-5} \text{ J kg}^{-1}$ to $k = 1.1 \times 10^{-4} \text{ J kg}^{-1}$. Around the computer and the manikin, it cannot exceed $k = 9.62 \text{ J kg}^{-1}$ as it is presented in the planes defined by $x = 0.2$ m and $y = 0.1$ m.

Turbulent viscosity

Figure S8 shows the distribution of the turbulent viscosity in the different planes defined by $x = 0.2$ m, $x = 1.9$ m, $y = -0.1$ m, and $y = 0.65$ m. For the $k-\epsilon$ model, the turbulent viscosity is a function of the turbulent kinetic energy. Indeed, it is expressed by the friction of the internal fluid (Fluent 2012). As a result, there is a linear combination between the turbulent viscosity and the turbulent kinetic energy which explains the similarity between their distributions. Therefore, the

Fig. 3 Temperature distribution in different planes, (a) $x = 0.2$ m, (b) $x = 1.9$ m, (c) $y = -0.1$ m, (d) $y = 0.65$ m

maximum values of the turbulent viscosity equal to $\mu_t = 1.7 \times 10^{-4}$ Pa s are placed near the walls.

Dissipation rate of the turbulent kinetic energy

The turbulent kinetic energy and the turbulent viscosity are the turbulence characteristics as well as the dissipation which presents the speed of the conversion of the turbulent kinetic energy in the internal thermal energy (Fluent 2012). Indeed, Figure S9 shows the distribution of the dissipation rate of the turbulent kinetic energy for the four planes defined by $x = 0.2$ m, $x = 1.9$ m, $y = -0.1$ m, and $y = 0.65$ m. Based on these results, it

has been observed that the dissipation rate of the turbulent kinetic energy is important in the inlet and outlet opening where it recorded its maximum $2 \times 10^{-4} \text{ m}^2 \text{ s}^{-3}$. In the rest of the room, the dissipation is lower than $3.5 \times 10^{-5} \text{ m}^2 \text{ s}^{-3}$.

Thermal comfort index: predicted mean vote

The thermal comfort is an important human necessity. It is a state of balance desired from the cold and warm sensation feeling. In this way, the popular thermal index used to test the thermal sensation is the thermal comfort index PMV (predicted mean vote). It was determinate by Fanger (ASHRAE

Fig. 4 Velocity fields in different planes, (a) $x=0.2$ m, (b) $x=1.9$ m, (c) $y=-0.1$ m, (d) $y=0.65$ m

2017). This index presents the combination between the human metabolism, the ambient conditions, the heat transfer coefficient, and the clothes' coefficient. The PMV varies according to the variation of the human metabolism (M , W), the air pressure p_a (bar), and the air temperature t_a ($^{\circ}F$). Indeed, Fig. 5 shows the evaluation of PMV in function of the

temperature in occupied area. According to these results, the PMV presents a linear variation and it can be expressed by this correlation equation:

$$PMV = 0.015 T - 2.951 \quad (6)$$

Fig. 5 Profiles of PMV in function of the temperature along the different directions, (a) $y=0.29$ m, (b) $y=-0.1$ m, (c) $y=-0.45$ m

Indeed, based on the profiles shown in Fig. 5, it has been observed that the PMV increases progressively in function of the augmentation of the indoor temperature. In addition, for the direction which is located between the two heat sources, this index ranges from 1.5 to 1.8. While above and behind the manikin, it does not exceed 1.7. Therefore, for the three directions, the PMV is more than 1. Consensually, based on the PMV scale shown in the Table 2, the climate is slightly warm. The PMV comfort index is directly linked to the temperature. Therefore, it presents the most significant environmental

Table 2 PMV scale (ASHRAE 2017)

Sensation	Value
Much too hot	+ 3
Too hot	+ 2
Hot but comfortable	+ 1
Neutral	0
Cold but comfortable	- 1
Too cold	- 2
Much too cold	- 3

factor. Especially, for the current state of the COVID-19, it is necessary to maintain an indoor environment which does not help the development of this virus.

Conclusion

In this paper, the characterization of the air ventilated indoor a building was developed. The distribution of the temperature, the velocity, the static pressure, and the turbulence characteristics is presented to examine the ventilation system indoor a space containing different heat sources. The numerical results prove that the thermal plume and the recirculation zone were formed around the person and the computer. The comparison between the results given by the different meshing demonstrates that the results which are in good deal with the experimental data were given by the cells number equal to 2166447 and to 2466198. So as to gain the space and the time of the simulation, the 2166447 cells can be the most consistent meshes. Involving the thermal sensation, the thermal environment would be tested by calculating the PMV index. This index shows that the climate is warm but acceptable. In future work, it is necessary to optimize this air conditioning system in order to attain the thermal comfort and to decrease the PMV lower than 1.

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