CFD Analysis for The Effect of Personal Ventilation Combined with Mixing Ventilation on Performance Index (ADPI) and Thermal Human Comfort

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Abstract. Good indoor air quality and ideal thermal comfort are very important indicators for the comfort of occupant’s space. The Iraqi climate is hot dry in summer, so the use of air conditioners is indispensable. This work focuses on solving the IAQ problems in office room environments, and it evaluated a new principle of ventilation which is combined personal and mixing ventilation systems. Personal and mixing ventilation system can provide individual control of indoor climate. The airflow motion and distribution of temperature investigated numerically with the best flow rate from combined system. CFD study is used for simulation the indoor airflow and temperature distribution by using (AIRPAK3.0.16) for solving the turbulence equations, Naiver-Stocks, energy equations, and using (FVM). The Renormalization Group RNG K-ԑ turbulence model was used for simulation this study cases, the airflow rate from ATD was 5 l/s and 10 l/s. For mixing ventilation supply temperature of 17°C and The temperature of the supplied air from the personal ventilation system ranges from (17 to 23) °C, it was found that if using personal ventilation system at airflow rate of 10 l / s it would improve the quality of the inhaled air present in the breathing zone. The personal and mixing ventilation system provides good human thermal comfort based on the values of effectiveness temperature (Et) and air distribution performance index (ADPI) which give (1.241) and (63.226%) respectively. The increased airflow rate from (ATD) leads to a decrease in the temperature in the breathing zone, this increases the human thermal comfort of the occupants inside the room.

Keywords: Thermal comfort, AIRPAK, Computational fluid- dynamics (CFD), Mixing ventilation (MV), Indoor air quality (IAQ), Personalized ventilation (PV).

1. Introduction
Nowadays a lot of people spend about 80% of their time inside an industrial environment, i.e. office, factory, public building, home, transport vehicle, etc. Therefore, it is essential to provide comfortable and healthy indoor environments for their occupants. Both thermal comfort and indoor air quality are important factors affecting the quality of the indoor environment. Recent research has improved the understanding of the influence of the thermal environment on human comfort and performance. The range of temperature of room should be between (20 to 26) °C with relative humidity from (30 to 60%). For the purpose of avoiding drafts, the average air speed is from (0.2 to 0.6) m/s, depending on
temperature of room and turbulence intensity [1]. MV system provides air to the room from the diffuser at a relatively high speed, usually located at or near the roof over the occupied zone, jet of air is provided in the top of the room (roof or wall at a high level) at high speed usually greater than 2 m.s-1 to ensure air circulation in the room, the resulting distribution of temperature and concentration of pollutants in the occupied zone must be uniform [2]. The air conditioning provided from outside the room mixes with the polluted air inside the room before reaching the occupant, which leads to a decrease in the quality of the air inhalation while the PV system provides fresh air directly to the occupant breathing zone, PV systems have the ability to improve inhaled air quality and thermal comfort, and also provide individual control of the temperature of air, air direction, and possibly speed of air as well [3].

PV system aims to provide fresh, cool air directly to occupant's breathing zone. The positive effects of personal ventilation, that is, clean air breathing, are documented for health, comfort and performance [4]. PV system has the ability to protect occupant from infectious agents causing disease, each PV user can control the appropriate local cooling for him and also can achieve a good environment based on individual preferences, [5, 6].

Additionally, since personal ventilation systems target the occupant breathing zone, the interaction between the supplied air and the flow in the delicate environment surrounding the occupant is the determining factor for personal ventilation systems performance [7]. The thermal comfort of the human was greatly improved as well as the perceived indoor air quality was improved and the PAQ were documented, and the symptoms of SBS were reduced when using PV, which was performed in field laboratories. [8, 9, 10]. PV system has the ability to protect space occupants from diseases transmitted through the exhaled air of sick people [11]. Many simulation and experimental works such as elmagh Rabey, (2014) [12]. Hussein et al. [13] have been done to study airflow pattern, temperature, and concentration distribution in mixing and Personal ventilation systems. Chen (1995) [14], studied five k-ε models, conclude The RNG k-ε model is best than the standard k-ε model and is therefore recommended for simulations. The RNG k model is widely used for indoor airflow for different configurations as it gives accurate results and is closer to experimental results, [15].

Kaczmarczyk et al., 2002 [16] studied the impact of PV system on the perceived air quality, the experiment was conducted on an office containing six occupants, through a personal ventilation system, the occupant is allowed to control the amount and direction of air. Four experiments were performed: (I) PV system provides outdoor air at 20 °C; (II) PV system supplies external air at 23 °C; (III) PV system provides recycled room air; And (IV) MV. Temperature of room was kept constant at 23 °C and relative humidity at 30%. The results showed that the best case with PAQ was when PV system introduced outdoor air at 20 °C. PAQ was much better in the Case-1.

Hooff & Blocken, 2019 [17] introduced MV driven by two oppositely located supply jets with a time-periodic supply speed, theoretical analysis using (CFD) program. Temperature of air in the small office, with higher cooling load, so it was chosen the change of air rate carefully to avoid a higher temperature gradient. This study showed that the use of the turbulence model (RNG k-ε) gave very good and accurate results as well as the ability to predict well on jet spreading rate and the behavior of recirculating air.

Al-ssaad et al., 2017 [18] investigated improvement of ventilation by combining personal ventilation with mixing ventilation, the study was carried out numerically and experimentally, at ideal conditions, a thermal comfort about 0.95 and also ventilation effectiveness 77% was obtained, the cost of energy for PV system was reduced by 21.34% compared to a constant personal ventilation system at equal thermal comfort. This study demonstrated that RNG k-ε model with enhanced wall treatment is used to model turbulence it is characterized by accuracy and very good predictability of jet spreading rate and the behavior of recirculated air.

Al-ssaad et al., 2018 [19] studied the best performance of a MV system Compound with PV, Simulated were performed using the CFD technique to find out the airflow the most suitable supply frequency, Fresh air is emitted from (PV) towards the upper part of the person. The purpose of this study was to achieve good thermal comfort and good IAQ in breathing zone. In this study, the test
model was simulated numerically using CFD technology, with a transient 3D test under laboratory conditions at a temperature of 25 °C and PV jet temperature of 22°C. This study showed that when the frequency increases at a constant flow rate, thermal comfort increased by 15.2%. However, when the medium flow rate increased at a fixed frequency, thermal comfort decreased at a low frequency of 0.3 Hz but remained acceptable at a higher frequency of 0.5 Hz.

Yakoob et al., 2019 [20] studied personal ventilation with displacement ventilation (DV) for the purpose evaluated the efficiency of air exchange in rooms, the study was practical and theoretical, this study using RNG, K-ε turbulence model for simulating temperature profile. This study concluded, that the arrangements at an office rooms for the air supply diffuser personal and DV are combined gives Acceptable human thermal comfort depending on the value of (Eε) and (ADPI) which gives (71%) and (1.8) respectively which improved the air quality and thermal comfort. (Katramiz et al., 2020) [21] studied the interaction between breathing of human and the (IPV) jet and investigated the effectiveness of IPV in supplying fresh and cold air of breathable. Developed a transient three-dimensional CFD mock-up of a breathing occupant in a chamber equipped with an IPV coupled with an MV system. Two modes of breathing were considered: breathing of nose and mouth. To evaluate air quality of the breathing zone, using the tracer gas method. The same model was practically conducted, and the same experiments were conducted to validate numerical results and compare them with experimental results. The results of this study gave that maximum effectiveness of ventilation ε during inhalation was obtained at an IPV frequency of 0.5 Hz ranging from 36.75% to 86.6 % for breathing through the nose, 49.6% to 87.3% for breathing through the mouth. The use of intermittent personal ventilation system improves the quality of air in the breathing zone.

Most researchers discussed in their studies, whether experimental or theoretical (the location, Geometric shapes for diffusers, number of diffusers, comparing DV with MV system, the position of the inlet and outlet of air and effect of an air change per hour on thermal comfort). The main objectives of this study are:

- This study aims to investigate how air quality in the breathing zone is affected when PV system is added.
- Studying the effect of adding PV on the thermal comfort of the occupant in office environment.
- Calculation the actual values of the input airflow rate and temperature of air supply needed by the testing room depending on the cooling loads of room.
- To investigate Air Diffusion Performance Index indoor air quality and comfort issues by CFD modeling by ventilation the room under the Iraqi climate.
- Study the possibility used PMV system type under the Iraqi climate.

In this work an office room was simulated using a program AIRPAK3.0.16 For the purpose of conducting the numerical study for airflow rates, distribution of temperature around occupants due to multiple sources of heat by used the CFD method to get a better level of IAQ and improved human thermal comfort.

2. Description room configuration and assumptions

An office room was simulated to numerical study for the airflow and distribution of temperature with the MV and PMV system. Steady state three-dimensional computational (CFD) mode is used to simulate the speed and temperature profiles in an office room. Dimensions of the simulation room 3.0 m (length) × 2.5 m (height) × 2.5m (width). Includes different ranges of speeds for personal ventilation with a constant flow rate for (MV) device to evaluate the performance of the various configurations facing local and general thermal comfort. The figures 1 & (2) show the schematic diagram of the ventilation systems (MV) & (PMV), respectively. MV device using to supply air into the office room located at the middle of the room's south wall is below the ceiling. Simulated two persons, two computers and one lump were placed as a heat source. Description of various details in the simulated office room numerically, with the ventilation system, is only MV system(case I) and PMV system (cases II&III) are given in tables (1,2) respectively, it shows us all the contents of the office room.
With the two types of ventilation systems used, a set person is 1.1m height, heat gain (only Sensible Heat) for person is (75W/person), computer (60W/computer) and lump (100W), and ΔT_{mf} is 2 °C. Table (3) shows the required Iraqi conditions inside the office buildings during hot and dry climates. It was used [Cooling Load Temperature Difference Method (CLTD)] to calculate the cooling loads as shown in the equations listed below [22].

\[ Q_{\text{person}} = N_0 \times \frac{q_{s}}{\text{person}} \times \text{CLF} \]  
\[(1)\]

Where, \(Q_{\text{person}}\) = Sensible Cooling Load, \(N_0\)= number of persons, \(q_s\) = sensible heat for one person, CLF= person cooling load factor [22]. While The luminescent load is calculated by the following equation

\[ Q_{L} = N_0 \times W \times F_b \]  
\[(2)\]

Where, \(Q_L\) = Sensible Cooling Load, \(N_0\)= number of lump, W= lump power, \(F_b\)= factor equal to1.2, [22].

Air is the supposed working fluid and the flow assumed to be steady state, 3D flow and incompressible, [22]. The test room at indoor temperature (25°C). It was measured before the start of the test. The tested room was thermally insulated, so the values of \((q_l\& q_{ex})\) are equal to zero in this work. (PV) system at different flow rate 5 l/s and 10 l/s. The air is supplied with different speeds and flow rates from (PV) is (0.6 m/s) at flow rate 5 l/s and (1.2 m/s) at flow rate 10 l/s. The following two
steps are used to estimate the supply ventilation rate for mixing ventilation and total heat applications [23]. The supplied air temperature from mixing air terminal is 17 °C [23]. The indoor room temperature before air is supplied is 25 °C.

The necessary air volume is calculated (airflow rate). The heat transfer due to the ventilation process is calculated by the following equations which are illustrated as,

$$ Q_t = m \times cp \times \Delta T $$  \hspace{1cm} (3)

$$ Q_s = \frac{m^*}{\rho} = 0.046/1.2 = 0.04 m^3/s $$ \hspace{1cm} (4)

The effective draft temperature, is given as, [24]:

$$ EDT = (T_x - Tr) - (8Vx - 1.2) $$  \hspace{1cm} (5)

Where: $T_x$ : local temperature\(^{(K)}\), $Tr$ : mean temperature of room\(^{(K)}\), and $Vx$ : local speed of air m.s\(^{-1}\). Because it represents local cooling not suitable for the human body due to movement of air and low temperature of air in space [25]. The draft temperature limit was taken as -1.7 °C < EDT <1.1 °C and the higher value for speed of air was taken as 0.35 m.s\(^{-1}\) [25&26]. The measured values of temperature and speed of air at uniformly spaced points throughout the occupied zone or through a vertical, centerline plane through the air supply [24].

(ADPI) used to satisfy the comfort limits, a percentage of the points number of draft temperature measured in the occupied zone in which (-1.7 °C < EDT <1.1 °C) to the total number of draft temperature points measured in the occupied zone, [12]. Kansas test data showed that ADPI depends on the type of ATD, load of room, airflow rate, and geometry of room. ADPI is useful for turning on the cooling mode; a value of 80% is usually considered an acceptable minimum, [24].

$$ ADPI = \frac{N_{\theta}^{'}}{N} \times 100 $$ \hspace{1cm} (6)

Where $N_{\theta}$ in which -1.7 °C < EDT < 1.1 °C

Air change per hour (ACH) It can be calculated by the following equation. (7), [24].

$$ ACH = \frac{Q_{s}}{V_{room}} \times 3600 $$ \hspace{1cm} (7)

In MV system there is a set of goals that must be achieved represented by clam operation, thermal comfort, suitable speed, low energy consumption, and low noise diffuser. In this present work rectangular diffusers were used [27], by assumed the air supply speed 2.5m/s (According to the recommendations of ASHRAE), the $Q_s$ It was calculated by the equation (4) and found its value (0.04) m³/s. Eqn. (8) used to calculate the area of the diffuser and found the dimensions of air supply unit are (0.016) m².

$$ Q_{s} = U_s \times A_s \quad \text{where} \quad U_s = 2.5 \quad [23]. $$  \hspace{1cm} (8)

Then $A_s = 0.04/2.5 = 0.016$ m²

Where: $U_s$: local air speed (m/s), $A_s$: Surface area of supply air diffuser (m²), the area of exhaust grills is about 75% of the air supply unit area, [28].

### 3. Computational Fluid Modeling

#### 3.1 Governing Equations:

The field of fluid flow can be described by a set of equations including continuity equation, momentum equation and energy equation in addition to other equations, [28]

$$ \frac{\partial}{\partial x} (\rho u) + \frac{\partial}{\partial y} (\rho v) + \frac{\partial}{\partial z} (\rho w) = 0 $$ \hspace{1cm} (9)

$$ \frac{\partial}{\partial x} (\rho uu) + \frac{\partial}{\partial y} (\rho uv) + \frac{\partial}{\partial z} (\rho uw) = - \frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left( \mu \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left( \mu \frac{\partial u}{\partial y} \right) + \frac{\partial}{\partial z} \left( \mu \frac{\partial u}{\partial z} \right) + \frac{\partial}{\partial x} (-\rho uu') + \frac{\partial}{\partial y} (-\rho uv') + \frac{\partial}{\partial z} (-\rho uw') + \rho g_x $$  \hspace{1cm} (10)
\[
\frac{\partial}{\partial x}(\rho u v) + \frac{\partial}{\partial y}(\rho w v) + \frac{\partial}{\partial z}(\rho w w) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}(\mu \frac{\partial v}{\partial x}) + \frac{\partial}{\partial y}(\mu \frac{\partial v}{\partial y}) + \frac{\partial}{\partial z}(\mu \frac{\partial v}{\partial z}) + \frac{\partial}{\partial y}
\]
\[
\left(\nu \frac{\partial}{\partial z} \left[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right] \right) + \frac{\partial}{\partial z}(-\rho u' v') + \frac{\partial}{\partial y}(-\rho v' w') + \frac{\partial}{\partial z}(-\rho w' w') + \rho g_y
\]
\[
\frac{\partial}{\partial x}(-\rho u' w') - \frac{\partial}{\partial y}(-\rho v' w') + \frac{\partial}{\partial z}(-\rho w' w') + \rho g_z
\]
\[
\frac{\partial}{\partial x}(-\rho u' t') + S_t
\]
Where:

\( (\Gamma) \) is the coefficient of diffusion, which is given by: \( \Gamma = \frac{\mu}{\alpha} \), \((\sigma = \frac{\mu \times C_p}{\gamma})\) is the Prandtl or Schmidt number for the fluid. The terms \(-\rho u' v', -\rho v' w'\) and \(-\rho w' w'\) are the turbulent heat fluxes, \( S_t \) is a source term allowing for the rate of thermal energy production.

### 3.2 Models of Turbulent:

In this numerical work, the turbulent model RNG K-\( \varepsilon \) was chosen for use. Chen, [29] investigated eight K-\( \varepsilon \) models for mixed convection flow and found that the Renormalization Group (RNG K-\( \varepsilon \)) model was the best, closest to reality and acceptable results. Yakhot [30]. It is the best type of turbulence models has tested. Yuan, [31]. Studied of predicting the distribution of indoor pollutants in the displacement ventilation chamber using the model (RNG K-\( \varepsilon \)), the equation for model of turbulent used are [32]:

\[
(\rho U_i) \frac{\partial k}{\partial x_i} = \mu_t S^2 + \frac{\partial}{\partial x_i} \left[ a_i \mu_{eff} \frac{\partial k}{\partial x_i} \right] - \rho \epsilon
\]

Where \( S \equiv \sqrt{2S_{ij} S_{ij}} \)

\[
S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right)
\]

\[
\rho u_i \frac{\partial \epsilon}{\partial x_i} = C_{\varepsilon} \frac{\varepsilon}{k} \mu_t S^2 + \frac{\partial}{\partial x_i} \left[ a_i \mu_{eff} \frac{\partial \epsilon}{\partial x_i} \right] - C_2 \varepsilon \frac{\varepsilon^2}{k} - R
\]

The model constants are the following values: \( C_1 \varepsilon = 1.42 \) and \( C_2 \varepsilon = 1.68 \)

### 3.3 Numerical solution and mesh generation:

In addition to the measurement of the Heat removal effectiveness (\( \varepsilon_i \)), a simulation was performed to calculate the ADPI of the ventilation of office room While using the software system (CFD), ANSYS Airpak 3.0.16. Airpak simulation software has been widely applied to numerical simulation based on the limited size method, Airpak uses the FLUENT CFD engine to solve thermal calculations and flow of fluid to solve equation to conserve mass, energy, and air momentum. RNG, K-\( \varepsilon \) turbulence model was chosen to solve flow equations. In this study, the cells number was found (440145) for MV and (482985) for PMV, while using Hexa unstructured geometry to discretize. For this function, all types of element were used to fit the grid to the geometry. Simulations were repeated to a level of approximation \( 10^3 \) until the solutions were stable as shown in the figure. (3). In addition, a network
improvement study was conducted to identify and reduce the error due to the estimate. Four different mesh systems were coarser type, course type, medium type, and fine were generated, to perform the test. The AIRPAK3.0.16 software was used to create the model and meshed study cases based on several testing meshes as shown in Figures 4&5, the dimensions of room edge in (x, y, z) meshed as interval size of (0.04), mesh parameter normal, with Max side ratio (2). The selection was made after several attempts. After applying the meshing strategy.

Figure 3. Residuals Plot

Figure 4. A part from meshed model for only MV

Figure 5. A part from meshed model for PMV

3.4. Boundary conditions

Boundary conditions have a major impact on the success of a CFD simulation in giving a reliable result of a problem solved. In this work, boundary conditions are assumed in a CFD simulation consisting of one type speed, inlet and outflow as tabulated in the table (4) wall boundary included isothermal walls, When the problem is solving numerically it is impossible to get very precise solution, therefore accepted scaled residuals of error must be specified for different terms such as continuity, components of speed and energy.

Finally, it is necessary to verify the validity of the use of the turbulent model RNG K-with another experimental study, The comparison is based on the vertical air temperature, which is measured by five points in the vertical pole as shown in Figure 6. The comparison provides a good fit between the numerical data for this study and experimental data [33]. The temperature of air was simulated using
the turbulent model RNG K-epsilon. The error rate between experimental and numerical data was calculated and found 1.5%.

![Validation between the numerical data of this study and an experimental data.](image)

**Figure 6.** Validation between the numerical data of this study and an experimental data. [33].

| Item                | Location (m) | Size (m)   | Heat (W) |
|---------------------|--------------|------------|----------|
|                      | X-start Y-start Z-start | X-end Y-end Z-end |          |
| Office Room         | 0 0 0       | 3 2.5 2.5  |          |
| Mixing device       | 0 2.15 1.17 | 0 2.25 1.33|          |
| Exhaust grille      | 0 0.3 1.19  | 0 0.4 1.31 |          |
| Person no.1         | 0.2 0 0.4   | 0.4 1.1 0.6 | 75       |
| Person no.2         | 2.7 0 1.7   | 2.8 1.1 2 75 |         |
| Computer no.1       | 0.65 0.76 0.3 | 0.95 1.06 0.6 | 60       |
| Computer no.2       | 2.1 0.76 1.6 | 2.4 1.9 2.2 | 60       |
| Light               | 1.45 2.4 2.3 | 1.55 2.45 2.5 | 100      |
| Computer table1     | 0.6 0.76 0.1 | 1 0 1.1   |          |
| Computer table2     | 2 0.76 1.4  | 2.5 0 2.4 |          |

**Table 1:** room configuration for MV system only (Case-I)

| Item                | Location (m) | Size (m)   | Heat (W) |
|---------------------|--------------|------------|----------|
|                      | X-start Y-start Z-start | X-end Y-end Z-end |          |
| Office room         | 0 0 0       | 3 2.5 2.5  |          |
| Mixing device       | 0 2.15 1.17 | 0 2.25 1.33|          |
| Exhaust grille      | 0 0.3 1.19  | 0 0.4 1.31 |          |
| Person no.1         | 0.2 0 0.4   | 0.4 1.1 0.6 | 75       |
| Person no.2         | 2.7 0 1.7   | 2.8 1.1 2 75 |         |
| Computer no.1       | 0.65 0.76 0.3 | 0.95 1.06 0.6 | 60       |
| Computer no.2       | 2.1 0.76 1.6 | 2.4 1.9 2.2 | 60       |
| Light               | 1.45 2.4 2.3 | 1.55 2.45 2.5 | 100      |
| Computer table1     | 0.6 0.76 0.1 | 1 0 1.1   |          |
| Computer table2     | 2 0.76 1.4  | 2.5 0 2.4 |          |
| Personal device1    | 0.95 1.15 0.3 | 1.05 1.2 0.54 |         |
| Personal device 2   | 2.1 1.15 1.6 | 2.2 1.2 1.84 |         |

**Table 2:** room configuration for PMV system (Case-I & Case-II)
Table 3: The required conditions inside the office buildings in the Iraqi climate during the summer season, [21].

| Property | DBT | RH | Recommended air velocity inside the office room |
|----------|-----|----|-----------------------------------------------|
| The conditions required within the office buildings | 23 ºC – 26 ºC | 40% – 50% | 0.25m/s_0.13m/s |

Table 4: Boundary Conditions

| part | types | Condition of momentum | Motion of wall | Shear condition |
|------|-------|-----------------------|----------------|-----------------|
| Person | wall | Stationary | Not Slipping | |
| computer | wall | Stationary | Not Slipping | |
| side walls, floor cell, | wall | Stationary | Not Slipping | |
| tables and lights | wall | Stationary | Not Slipping | |
| Supply air diffuser | Speed inlet | magnitude, normal to boundary | |
| Extract Grill | Pressure outlet | -Gauge Pressure = (0 Pa), [constant].  
-Backflow direction specification Method: (Normal to Boundary). | |

Table 5: Different temperature values for Cases (I, II and III)

| Types of Ventilation system type | No-case | Setup temperature ºC | Average temperature ºC | Different in temperature ºC |
|---------------------------------|---------|-----------------------|------------------------|---------------------------|
| Mixing ventilation system (MV) only | I | 25 | 24.1327 | 0.8673 |
| MV and PV combined system(5L/s per ATD) | II | 25 | 23.8516 | 1.1484 |
| MV and PV combined system(10L/s per ATD) | III | 25 | 22.803 | 2.197 |

Table 6: numerical values of ADPI and effectiveness (εt) temp. for each ventilation type

| Ventilation devices used | Only mixing | MV combined with PV |
|-------------------------|-------------|---------------------|
| CASE | Case II | Case II | Case III |
| Air flow rate from PV device | ------ | 5 l/s | 10 l/s |
| ADPI% | 53.0277% | 62.125% | 63.226 |
| Effectiveness | 0.9224 | 1.0955 | 1.241 |

4. Results and discussion

This section of the current work is dedicated to presenting and discussing the numerical results obtained from the room tested under the Iraqi climate. CFD technique was used to study the temperature behavior and speed fields in addition to other parameters in the simulated test room. The numerical details that are obtained are more than those obtained from experimental studies. Also, to check the appropriateness of the numerical method and to see whether the data obtained from the calculations can be used in developing the specific model.

- **Air speed distribution (only MV) (Case. I)**

The contour of air speed magnitude in (m/s) in the (z-plane through center=1.25m), In the middle of the room wall, toward the z axis, there is an opening to supply air to the room below the ceiling in addition to the exhaust grille on the same side which is slightly above the floor of the room, is shown in Figure 7. The average speed of air at these locations was about (0.153769 m/s) The air supply speed is 2.5m.s⁻¹, the maximum speed was (4.5217 m/s) at the center of exhaust grille because of it less area than the opening supply air. Therefore, noted the speed of the air leaving the office room higher than the speed entering the room.

Figure 8 shows the location in the x-plane through center x = 1.5m of the office room. The average air speed about 0.153769 m/s and the maximum speed was about 1.47523 m/s in front of the air supply hatch. figure 9 shows the location in the z-plane, z = 0.4m pass-through person no.1 and computer
no. 1. The average air speed was about 0.153769 m/s and the maximum speed was about 0.516450 m/s (above the person’s head and shoulders, as well as the computer) located above the occupant’s head and shoulders and in a thin layer near to the room’s floor. Figure 10 shows the location in the z-plane, z=1.75m pass-through person no. 2 and computer no. 2. The average air speed was about 0.153769 m/s and the maximum speed was about 0.494293 m/s (above the person’s head and shoulders, as well as the computer) located above the occupant’s head and shoulders. Figure 11 shows the location at the breathing zone at 1.1 m in the y-plane through 1.1m. The average speed near the person’s nose is close to 0.368 and is suitable for breathing and does not cause heat discomfort.

Figures 7&11 show the air speed field around the simulated occupant. A buoyancy-driven thermal plume develops in the vicinity of the body because of gradient of temperature between ambient air and the body surface. Vertical air movement between layers is caused by strong convection forces associated with heat sources (occupant & computer). The movement of a fluid element noted that it consists of a translation, a rotation, and rate of deformation inside the room, then passes vertically to a high level in the office room where it is exhausted from the exhaust grills during the first airflow cycle the speed of air will be high, but as the quantity of room air that becomes mixed increases, the speed of air will decrease.

**Figure 7.** Contour of air speed magnitude (m/s) at the centers of the supply vents in case 1

**Figure 8.** Contour of air speed magnitude (m/s) in plane (x=1.5) within the office room in case

**Figure 9.** Contour of air speed magnitude (m/s) in Z-plane (pass through person1 and computer1) within the office room (only mixing ventilation)

**Figure 10.** Contour of air speed magnitude (m/s) in Z-plane (pass through person2 and computer2) within the office room (only mixing ventilation) Temperature distribution contours

**Figure 11.** Contour of air speed magnitude (m/s) in y-plane through 1.1m within the office room (only mixing ventilation) Temperature distribution contours
- **Air temperature distribution (case only mixing ventilation) (Case I)**

The temperature contours in the z-plane at z = 0.4m through person1 and computer of the room is shown in figure 12. The average temperature of air around the occupant was 24.235 °C while the minimum air temperature in the room was 22.9475 °C in most parts of it. The mean temperature in the room (except for the (34to 36) °C exhalation temperature out of the occupant’s mouth and also near the simulated human body) was about 31.1 °C found closely around the computer, vertically above his head and above the computer’s monitor.

The temperature contour in the x-plane through the person are shown in figure 13. The average temperature of air around the occupant was 24.134°C with minimum temperature of 17.42 °C and maximum temperature of 36.6 °C above and around the occupant’s head and shoulders.

The temperature contour in (z-plane, z=1.75m) (z plane through person2 and computer)of the room is shown in figure 14. The average temperature of air around the occupant was (24.236) °C while the minimum temperature of air in the room was (23.4124) °C in most parts of it. The mean temperature in the room (except for the (34to 36) °C exhalation temperature out of the occupant’s mouth and also near the simulated human body was about 33.134 °C found closely around the computer, vertically above his head and above the computer’s monitor.

The temperature contour in the x-plane through the person is shown in figure 15. The average temperature of air around the occupant was 24.134°C with a minimum temperature of 22.54°C and a maximum temperature of 36.6°C above and around the occupant’s head and shoulders.

Figures 12 to 15 display numerically the contours of distribution of air temperature at different planes in the tested room for MV only, an increase in temperature is observed from 17 ° C closely at the supply air terminal and reaches about 36 ° C near the person and (PC) simulator.

MV is distinguished by the fact that the air is transported at high speed outside the occupied zone, the high speed of air in the air jet will generate extra pressure, resulting in airflow from the room and inserting it in the air direction. The air is supplied to the office room at high speed and high momentum from the (air diffuser)(MV device) located at below the ceiling, which leads to mixing the room air with fresh air, and thus the temperature of room decreases and reduces the percentage of pollutants inside the room , an air stream is formed. The air becomes hot due to the heat exchange between the air and the heat sources inside the room, then it vertically rises upward to mix again with the air coming from MV device ,thus carries with it the heat produced by (occupants & computers) and put it out of the room with air outside (exhaust grille). Thermal plumes are generated by convection due to the differences in temperature between the heat sources and the surrounding air. It found that the zone below the ceiling region near the supplied diffuser zone shows the lowest temperature values, due to the cooling effect of the supply entering air. Numerical results showed a decrease in temperature in and near the floor of the room, a gradual increase in temperature was observed, with an increase in the height above the floor of the room, and then it started to decrease again due to the cold air coming from MV device Installed to the top of the wall just below the ceiling.

The increase in the indoor temperature of room can be observed near the persons, computers and lighting, this increase in temperature leads to excessive kinetic energy in the stagnation region.
12

- The Effect of Personalized Ventilation Flow Rate (L/s)
The effect of (PV) system conditioned airflow rate in (L/s) on the thermal comfort conditions and indices (air speed, air temperature) is investigated through the case studies (5&10 L/s). The same fresh air supply temperature, the same speed of air supplied from (MV) device. figures 16 &17 shows path lines of air coming from the PV ATD set on the desk and directed towards the occupant’s face at 5 L/s per ATD and 10L/s per ATD colored by speed magnitude

Figure 16. the PV ATDs set on the desk and directed towards the occupant’s face at 5 L/s per ATD(CASEII)
Figure 17. the PV ATDs set on the desk and directed towards the occupant’s face at 10 L/s per ATD(CASEIII)

- Air speed distribution, Case II (5L/s per ATD)

The contour of air speed magnitude in (m/s) in the Z-plane through $Z=0.4\text{m}$ of the office room (through person1) are shown in figure 18. The average air speed around the occupant’s body at this plane was 0.45 m/s and the maximum speed was 1.19542 m/s exactly at the lower part of the occupant’s face and decreases gradually over his head.

The contour of air speed magnitude in m/s in the Z-plane through $Z=1.75\text{m}$ (through person2) of the office room are shown in figure 19. The average speed of air around the occupant’s body at this plane was 0.3125 m/s, the maximum speed was 1.311 m/s exactly at the lower part of the occupant’s face and decreases gradually over his head.

The contour of air speed magnitude in (m/s) in the (x-plane through 0.25m) are shown in Figure20. The average speed of air around the occupant’s body was 0.435 m/s around the occupant’s head then speed decreases gradually above that level in an air vortex up to the ceiling, while the maximum speed at that plane was 2.3 m/s found it is located in front of the air supply opening.

The contour of air speed magnitude in (m/s) in the (x-plane through 2.75m) are shown in figure 21. The average speed of air around the occupant’s body was 0.213m/s around the occupant’s head then speed decreases gradually above that level in an air vortex up to the ceiling, while the maximum speed at that plane was 0.376 m/s.

The contour of air speed magnitude in (m/s), in the y-plane through y=1.1m of the office room is shown in Figure22. The speed near to the simulated human body about 0.1243 and max speed 0.241m/s.

Figures 18 to 22 display the calculated air speed distribution of the cases studied through airflow rate (5 l/s per ATD) for (PV) system. The flow will be studied in two zones. Firstly, high values of air speed obtained from supplied air terminal at the below the ceiling of the room, due to location of MV device in this zone. Secondly, when the air flow pulls by PV system at level 0.4 m (Air intake) to level 1.1m, as shown in Figures.18&19, then it spreads over a large zone lead to the jet momentum will reduce, but still has a sufficient force to reach long distances. Generally, the air speed was increased at level 1.1 m (Figure 22). The maximum speed region where MV exists, remains at a value of 2.5 m/s for MV as stipulated in ASHRAE standard. Then it gradually decreases as the air blowing away from the mixing device.
Figure 18. Contour of air speed in the Z-plane through z=0.4m of the office room through person no.1 at 5l/s per ATD.

Figure 19. Contour of air speed in the Z-plane through z=1.75m of the office room through person no.2 at 5l/s per ATD.

Figure 20. Contour of air speed in the X-plane through X=0.25m of the office room through person no.1 at 5l/s per ATD.

Figure 21. Contour of air speed in the X-plane through X=2.75m of the office room through person no.2 at 5l/s per ATD.

Figure 22. Contour of air speed magnitude in (m/s) in the Y-plane through Y=1.1m of the office room (5L/s per ATD).

- **Air temperature distribution, Case II (5L/s per ATD)**

The temperature contour in the (x-plane through x=0.25m) sectional plane of the room is shown in figure 23. While the minimum temperature of air in the room was 24.145 °C in most parts it due to the heating effect caused by the walls and exactly at the occupant’s face front. The maximum air temperature was 34.3315 °C found very closely around the occupant’s body and above his head in the thermal plume generated from the occupant’s body and blown away by the PV air to a further distance than that of the cases.

The temperature Contour in the (x–plane through) (x=2.75m) sectional plane is shown in figure 24. The average temperature of air around the occupant was 24.368 °C with a minimum temperature of 22.541 °C and a maximum temperature of 35.8426 °C above the occupant’s head and shoulders. The
temperature Contour in the (z –plane through 1.75m) sectional plane is shown in figure 25. The minimum temperature of 23.2142 ºC and maximum temperature of 35.3033 ºC above the occupant’s head and shoulders and continue upwards in a thermal plume with about 18ºC air vortex up to the ceiling. The temperature Contour in the (z –plane through 0.4m) sectional plane is shown in figure 26. The minimum temperature of 17.2837 ºC and maximum temperature of 34.1980 ºC above the occupant’s head and shoulders and continue upwards in a thermal plume with about 18ºC air vortex up to the ceiling. The temperature Contour in the (y–plane through 1.1) shown in figure 27 the temperature in the breathing area is approximately 23 º C or slightly higher the lowest temperature is approximately 22.4136 º C.

These figure 23 to 27 also display numerically the contours of air temperature distribution at planes in the tested room where the temperature of air increases from 17 º C closely to MV device and reaches about 34º C near the person and PC-simulator. It was observed in this case (II) that the temperature of air surrounding the simulated person is lower than the air temperature in the case (I). A decrease in temperature of air was observed near the ceiling of the room due to the presence (MV device) near the ceiling of the room on the side wall of the room there a gradual increase in temperature obtained as elevation increase inside the room, then the air temperature decreases due to the cold air coming from(MV)device. Also, an increase in indoor temperature can be noted near the person, PC-simulator and lighting. Thermal plumes are generated by natural convection due to the temperature differences between the sources of heat and the surrounding air. Also, the increase in indoor temperature can be noted near the person, PC-simulator and lighting, and noted the existence of personal ventilation in level 1.1m as shown [see figure 27] reduce the heat generated from person and PC-simulator, besides that refreshing air breathing at this zone due to the continuous pumping of air from the layers of cold air in the bottom coming and produce excessive kinetic energy at the stagnation region and distribute approximately in the longitudinal and lateral directions of room, that found from (ADPI) and effectiveness temperature the supply air device of combined (MV and PV) ventilation gives good ventilation.

Figure 23. Contour of air temperature (°C) in (x plane through person1)(x=0.25) within the office room in case (5L/s per ATD)

Figure 24. Contour of air temperature (°C) in (x plane through person2)(x=2.75m) within the office room in case (5L/s per ATD)

Figure 25. Contour of air temperature (°C) in plane(z=1.75m through person2) within the office room in case (5L/s per ATD)

Figure 26. Contour of air temperature (°C) in plane(z=0.4m through person1) within the office room in case (5L/s per ATD)
Contour of air temperature (°C) in plane (y=1.1) within the office room in case (5L/s per ATD)

- **Air speed distribution, Case III (10L/s per ATD)**

  The contour of air speed magnitude in (m/s) in the Z- plane through Z=0.4m of the office room is shown in figure.28 The average speed of air around the occupant’s body at this plane was 0.45 m/s and the maximum speed was 1.4 m/s (at intake air of personal device) exactly at the lower part of the occupant’s face and decreases gradually over his head.

  The contour of air speed magnitude in (m/s) in the Z- plane through Z=1.75m of the office room is shown in Figure.29 The average speed of air around the occupant’s body at this plane was 0.45 m/s and the maximum speed was 1.5m/s (at intake air personal device) exactly at the lower part of the occupant’s face and decreases gradually over his head.

  The contour of air speed magnitude in (m/s) in the (x-plane through 0.25m) is shown in figure .30, The average speed of air around the occupant’s body was 0.435 m/s around the occupant’s head then speed decreases gradually above that level in an air vortex up to the ceiling, while the maximum speed at that plane was 2.3 m/s found It is located in front of the air supply opening.

  The contour of air speed magnitude in (m/s) in the (x-plane through 2.75m) is shown in figure.31, The average speed of air around the occupant’s body was 0.213m/s around the occupant’s head then speed decreases gradually above that level in an air vortex up to the ceiling, while the maximum speed at that plane was 0.376 m/s.

  Contour of air speed magnitude in m/s, in the y- plane through y=1.1m of the office room are shown in figure 32 the speed near to person about 0.48m/s and max speed 1.1m/s.

  Figures 28 to 32 display the calculated air speed distribution through mass flow rate 10 l/s and for two situations. The flow can be considered in two zones. Firstly, high values of air speed obtained from supplied air terminal from the air diffuser located at below the ceiling of the room due to location of supply air for MV system in this zone, Secondly, when the air flow is pulled by PV system at level 0.4 m (Air intake level) as shown (28&29) to level 1.1m (level of breathing zone for The person sitting) as shown in figure. 32 and the resulting temperature mix should be also higher than the surrounding air temperature. The effect of the PV airflow will be only on the plume air temperature and flow rate, then it spreads over a large area causing the jet momentum to recede but still has a sufficient force to reach long distances. Generally, the air speed was increased at level 1.1 m. Air speed distribution at a mass flow rate of 10 l/s is more acceptable than mass flow rate 5 l/s. The PV airflow speed is assumed to be low enough when reaching the thermal plume so it will not destroy its rising profile.
• Air temperature distribution, Case III (10L/s per ATD)

The temperature contour in the (x-plane through x=0.25) sectional plane of the room is shown in figure.30 the minimum temperature of air in the room was 17.8 °C in most parts due to the heating effect caused by the walls and exactly at the occupant’s face front. The maximum temperature of air was 32.21 °C found very closely around the occupant’s body and above his head in the thermal plume generated from the occupant’s body and blown away by the PV air to a further distance than that of the cases.

The temperature contour in the (x-plane through x=2.75) sectional plane is shown in Figure 34. The average temperature of air around the occupant was 24.368 °C with minimum temperature of 22.75 °C and a maximum temperature of 34.8688 °C above the occupant’s head and shoulders. The temperature contour in the (z-plane through 0.4m) sectional plane is shown in figure 35. Minimum temperature of 22.45 °C and a maximum temperature of 31.1°C above the occupant’s head and shoulders.
The temperature contour in the \((z–plane\) through \(1.75m)\) sectional plane is shown in Figure 36. The minimum temperature of \(22.45\ ^\circ C\) and a maximum temperature of \(31.24\ ^\circ C\) above the occupant’s head and shoulders. The temperature contour in the \((y–plane\) through \(1.1)\) shown in Figure 37. The temperature in the breathing area is approximately \(24\ ^\circ C\) or slightly higher. The lowest temperature is approximately \(22.456\ ^\circ C\). The maximum temperature \(31.246\ ^\circ C\). Figures 33 to 37 show the air temperature distribution contours for the vertical and the horizontal planes in the tested room for the turbulence model used for the mass flow rate of PV at \(10\ l/s\). These figures also display numerically the contours of air temperature distribution at planes in the tested room, where the air temperature increases from \(17\ ^\circ C\) closely to the supply terminal and reaches about \(31\ ^\circ C\) near the person and PC-simulator. It found that PV can improve the inhaled air quality in the tested room and each occupant has delegated the authority to optimize and control temperature, flow rate (local air speed) and direction of the locally supplied personalized air according to his/her preference, and thus to improve his/her thermal comfort conditions.

In each layer, the temperature inside the rising plumes is different than the homogenous surrounding air temperature zone and from the wall plume air temperature. The airflow of the PV could disturb the buoyant plume airflow, and it might create some mixing with the surrounding air zone, especially if the PV air is injected at high speed of air or if the resulting mix temperature between the thermal plumes and the surrounding air is lower than the surrounding air temperature forcing the rising air plume to descend. Also, the increase in indoor temperature can be noted near the person, PC-simulator and lighting, and noted the existence of personal ventilation in level \(1.1m\) as shown [see figure. 37] reduce the heat generated from the person and PC-simulator, besides that refreshing air-breathing at this zone due to the continuous pumping of air from the layers of in the bottom and produce excessive kinetic energy at the stagnation region and distribute approximately in the longitudinal and lateral directions over the floor, that found from \((ADPI)\) and \(\varepsilon\), the supply air device of combined (MV & PV) ventilation gives good ventilation, this is consistent with the research results Elmaghraby, (2014) [12].

**Figure 33.** Contour of air temperature \((^\circ C)\) in \(x\) plane through person1 \((x=0.25m)\) within the office room in case \((10L/s\ per\ ATD)\)

**Figure 34.** Contour of air temperature \((^\circ C)\) in \(x\) plane through person2 \((x=2.75m)\) within the office room in case \((10L/s\ per\ ATD)\)

**Figure 35.** Contour of air temperature \((^\circ C)\) in plane \((z=1.75m,\) through person2\) within the office room in case \(10L/s\ per\ ATD\)

**Figure 36.** Contour of air temperature \((^\circ C)\) in plane \((z=0.4m,\) through person1\) within the office room in case \(10L/s\ per\ ATD\)
The effect of changing the PV system air flow rate in (L/s) on the thermal comfort conditions and indices was investigated through case studies (II and III). As a predictable result, increasing the PV ATDs flow rate raised the overall average speed of air in the room and around the occupant’s body and decrease the temperature near the person in the breathing zone, these results were compared with the experimental results of the experimental study (Li et al., 2010) [34], and it was noticed that there is a great deal of accuracy and convergence in the results. The effect of altering the MV and desktop PV systems air flow rates concurrently on the thermal comfort conditions and indices was investigated through case studies (II and III). From one case to the other, increasing the PV ATDs flow rate simultaneously made the overall average speed of air in the room and around the occupant’s body increasing and maximum speed changing with no specific trend. The increase in the speed and the amount of air flowing from the PV ATDs also led to a decrease in the temperature near the computers and near the head and shoulders of the simulated human body.

The speed of the air flowing from the PV ATDs should not increase beyond the permissible limit in order not to cause discomfort, as well as cause dry eyes and dry nose and mouth.

### Relative humidity

In all the cases studied (I) (II) and (III), it was found that the relative humidity was within the range from 40% to 50% and the effect of personal ventilation systems on it was very little. And as shown in Figures 38, 39 and 40. Where it is noticed that with an increase in the rate of airflow from (ATD), the relative humidity will increase slightly, which does not clearly affect the thermal comfort of the human.
5. Conclusions

The present work focuses on the numerical analysis of air quality and human thermal comfort in the office room by adopting MV and PV system under the Iraqi climate. Where conclusions of this study can be summarized:

1- Using the mixing ventilation combined with personal ventilation systems are more efficient than used mixing ventilation only.
2- Numerical results at combined MV with PV devices give acceptable result at the air volume flow rate of 10 l/s.
3- It has been proven that the supply of fresh cold and clean air to the occupants breathing zone by using PMV system will improve greatly human thermal comfort, PAQ, and inhaled air quality.
4- The turbulence model (RNG k-ε) gives the best compatibility between experimental and numerical results in verification cases by predicting thermal behavior and air flow patterns for MV and PV devices in simulated office rooms.
5- Through numerical results comparisons, ADPI values were calculated for all cases. It was found that the highest value (63.226%) and with respect to (ε_t), the maximum value (1.241) was obtained from-case III.

| Nomenclature | Description                                      | Unit       |
|--------------|-------------------------------------------------|------------|
| A            | Surface Area for Wall                          | (m²)       |
| C_p          | Specific heat of the air at pressure constant   | (kJ/kg.K)  |
| C1ε, C2ε     | Coefficient in the specific dissipation rate    |            |
| D            | Diameter                                        | (m)        |
| dx dy dz     | Control volume                                  | (m)        |
| G            | Gravitational acceleration                      | (m/s²)     |
| N            | Total number of draft temperature points measured in occupied zone |            |
| N_θ          | Number of points of draft temperature measured in occupied zone |            |
| P            | Pressure                                        | (N/m²)     |
| Q            | Heat transfer through the wall.                 | (W)        |
| Qt           | Source term for the rate of thermal energy Production | (J/kg)     |
| T            | Temperature                                     | (°C)       |
| u, v, w      | component of speed in x, y and z-directions     | (m/s)      |
| u, v, w      | Speed at cell (i,j,k)                           | (m/s)      |
| x, y, z      | Cartesian coordinates                           | (m)        |
| Vx           | Local air speed                                 | (m³/s)     |
| Vroom        | Volume of room (m³)                             |            |
| Q_s          | design flow rate of the supply air             | (l/s)      |
| Ψ             | Cooling load for the heat conduction through the walls and transmitted solar radiation | (W)        |
| q             | Cooling load for the overhead lighting          | (W)        |
| Ψ_w          | Cooling load for occupant, desk lamp and equipments | (W)        |
| Qt           | Total heat transfer                             | (W)        |
| T_e          | average room temperature                        | (°C)       |
| T_e          | Exhaust Air Temperature                         | (°C)       |
| T_o          | outlet design temperature                       | (°C)       |
| T_o          | outlet temperature                              | (°C)       |
| T_l          | local temperature                              | (°C)       |
| T_r          | Setup (design) temperature.                    | (°C)       |

Greek Symbols

| Symbol | Description                                      | Unit       |
|--------|-------------------------------------------------|------------|
| ρ      | Air density                                     | (kg/m³)    |
| δ      | Turbulent energy dissipation rate.              | (J/kg s)   |
| Ω      | Prandtl or Schmidt number                       | -          |
| Σk     | Model constant                                  | -          |
| Σε     | Model constant                                  | -          |
| Ω      | Specific dissipation rate                        | (1/s)      |
| F      | Diffusion coefficient                           | (m²/s)     |
| μ      | Turbulent viscosity                             | (N.s/m²)   |
| Ω      | Rotation speed                                  | (rad/s)    |
| ΔThf   | The difference in temperature from head to foot level. | (°C)     |
| αk, ax | Coefficient in the specific dissipation rate     | -          |
| ε_t    | Effectiveness temperature                       | -          |
| Τ_1    | Turbulent Reynolds stress                       | -          |
Sub-Scripts

| Sub-Scripts | Meaning                  |
|-------------|--------------------------|
| av          | Average                  |
| c           | Correct                  |
| e           | Exhaust Air              |
| ex          | External                 |
| f           | Floor                    |
| hf          | Head to floor level      |
| H           | Hydraulic                |
| i           | Inside                   |
| i,j,k       | Location of point in a Cartesian grid |
| t           | Overhead light           |

Abbreviations

| Abbreviations | Meaning                                      |
|---------------|----------------------------------------------|
| ACH           | Change of air per hour                       |
| ADPI          | Air Distribution Performance Index           |
| ASHRAE        | American Society for Heating, Refrigeration, and Air Conditioning Engineers |
| CFD           | Computational Fluid Dynamics                 |
| PV            | Personalized ventilation                     |
| PMV           | Personal-mixing Ventilation                 |
| FVM           | Finite Volume Method                         |
| EDT           | Effective Draft Temperature                  |
| IAQ           | Indoor air quality                          |
| MV            | Mixing Ventilation                          |
| RNG           | Re-Normalization Group                       |
| SBS           | Sick Building Syndrome                      |
| ATD           | Air terminal device                          |
| PAQ           | Perceived air quality                        |
| DV            | Displacement Ventilation                    |

References

[1] Zeng, Qingfan, and Rongyi Zhao. "Prediction of perceived air quality for personalized ventilation systems." Tsinghua Science and Technology 10.2, 227-232, 2005.
[2] Awbi, Hazim B. "Ventilation for good indoor air quality and energy efficiency." Energy Procedia 112 (2017): 277-286.
[3] Melikov, Arsen Krikor. "Personalized ventilation." Indoor air 14, 157-167, 2004.
[4] Kaczmarczyk, Jan, A. Melikov, and Povl Ole Fanger. "Abstract." Indoor Air 14, 17-29, 2004.
[5] Cermak, Radim, et al. "Performance of personalized ventilation in conjunction with mixing and displacement ventilation." Hvac&R Research 12.2, 295-311, 2006.
[6] Nielsen, Peter V., et al. "Chair with integrated personalized ventilation for minimizing cross infection." The International Conference on Air Distribution in Rooms, Roomvent. FINVAC ry, 2007.
[7] Melikov, Arsen K. "Human body micro-environment: The benefits of controlling airflow interaction." Building and Environment 91, 70-77, 2015.
[8] Kroner, Walter M. "Environmentally responsive workstations and office-worker productivity." ASHRAE Transactions 100.2, 750-755, 1994.
[9] Chen, Huijuan. Experimental and numerical investigations of a ventilation strategy–impinging jet ventilation for an office environment. Diss. Linköping University Electronic Press, 2014.
[10] Olesen, Bjarne W. "The philosophy behind EN15251: Indoor environmental criteria for design and calculation of energy performance of buildings." Energy and Buildings 39.7, 740-749, 2007.
[11] Wouters, Peter, and Christophe Delmotte. "Ventilation, good indoor air quality and rational use of energy." Pollution atmosphérique 1, 65, 2005.
[12] ElMaghraby, Hossam A., "Numerical Simulation for Thermal Comfort using Conditioned Air through Mixing and Personalized Ventilation Systems in Field Environmental Chamber (FEC)." 2014.
[13] Mohammed, Hyder. A theoretical study of a cold air distribution system with different supply patterns. Diss. M. Sc Thesis, University of Technology, Iraq, 2013.
[14] Chen, Q. "Comparison of different k-ε models for indoor air flow computations." Numerical Heat Transfer, Part B Fundamentals 28.3, 353-369, 1995.
[15] Bunn, R. "Finding the right mix." Building Services Journal 18.8, 43-4, 1996.
[16] Kaczmarczyk, Jan, et al. "The effect of a personalized ventilation system on perceived air quality and SBS symptoms." 9th International Conference on Indoor Air Quality and Climate, 2002.
[17] Van Hooff, T., and B. Blocken. "Mixing ventilation driven by two oppositely located supply jets with a time-periodic supply velocity: A numerical analysis using computational fluid dynamics." Indoor and Built Environment: 1420326X19884667, 2019.
[18] Al Assaad, Douaa, et al. "Mixing ventilation coupled with personalized sinusoidal ventilation: Optimal frequency and flow rate for acceptable air quality." Energy and Buildings 154, 569-580, 2017.
[19] Al-Assaad, Douaa, Nesreen Ghaddar, and Kamel Ghali. "Performance of Mixing Ventilation System Coupled with Dynamic Personalized Ventilator for Thermal Comfort." ASME 2017 Heat Transfer Summer Conference. American Society of Mechanical Engineers Digital Collection, 2017.
[20] Yakoob, Mahdi, Alamir, “Evaluation of Air Exchange Efficiency in Rooms with Personal Ventilation in Conjunction with Displacement Ventilation Systems”, International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS Vol:19 No:06, 2020.
[21] Iraqi cooling code, 2012.
[22] Katramiz, E., Al Assaad, D., Ghaddar, N., & Ghali, K. “The effect of human breathing on the effectiveness of intermittent personalized ventilation coupled with mixing ventilation”. Building and Environment, 106755, 2020.
[23] Awbi, Hazim B. Ventilation systems: design and performance. Routledge, 2007.
[24] Awbi, Hazim B. Ventilation of buildings. Taylor & Francis, 2003.
[25] Cho, Sooyeon, Piljae Im, and Jeff S. Haberl. Literature review of displacement ventilation. Energy Systems Laboratory, Texas A&M University, 2005.
[26] John, David A. "Designing air-distribution systems to maximize comfort." ASHRAE Journal 54.9, 20, 2012.
[27] Al-Shemmeri, Tarik. Engineering fluid mechanics. Bookboon, 2012.
[28] Halton Co. Care for Indoor Air, AHD - Exhaust Grille guide, 2012.
[29] Melikov, Arsen Krikor, et al. "Impact of airflow interaction on inhaled air quality and transport of contaminants in rooms with personalized and total volume ventilation." 7th International Conference on Healthy Buildings 2003. Natiol University of Singapore, Department of Buildings, 2003.
[30] Versteeg, H. K., and W. Malalasekera. "An introduction to computational fluid dynamics." Finite Volume Method, Essex, Longman Scientific & Technical, 1995.
[31] Yuan, Xiaoxiong, et al. "Measurements and computations of room airflow with displacement ventilation." Ashrae Transactions 105, 340, 1999.
[32] Yakhot, V. S. A. S. T. B. C. G., et al. "Development of turbulence models for shear flows by a double expansion technique." Physics of Fluids A: Fluid Dynamics 4.7, 1510-1520, 1992.
[33] YAN, Shuai; LI, Xianting. Analytical expression of indoor temperature distribution in generally ventilated room with arbitrary boundary conditions. Energy and Buildings, 208: 109640, 2020.
[34] Li, R., Sekhar, S. C., & Melikov, A. K. "Thermal comfort and IAQ assessment of under-floor air distribution system integrated with personalized ventilation in hot and humid climate. Building and Environment", 45(9), 1906-1913. 2010.