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# Numerical and Experimental Modeling of Indoor Air Quality Inside a Conditioned Space with Mechanical Ventilation and DX-Air Conditioner

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K E Y W O R D S

Carbon dioxide ($CO_2$); Computational fluid dynamics CFD; Indoor air quality (IAQ); 4-way cassette air-conditioner; ventilation.

A B S T R A C T

*Elevated $CO_2$ rates in a building affect the health of the occupant. This paper deals with an experimental and numerical analysis conducted in a full-scale test room located in the Department of Mechanical Engineering at the University of Technology. The experiments and CFD were conducted for analyzing ventilation performance. It is a study on the effect of the discharge airflow rate of the ceiling type air-conditioner on ventilation performance in the lecture room with the mixing ventilation. Most obtained findings show that database and questionnaires analyzed prefer heights between 0.2 m to 1.2 m in the middle of an occupied zone and breathing zone height of between 0.75 m to 1.8 given in the literature surveyed. It is noticed the mismatch of internal conditions with thermal comfort, and indoor air quality recommended by [ASHRAE Standard 62, ANSI / ASHRAE Standard 55-2010]. CFD simulations have been carried to provide insights on the indoor air quality and comfort conditions throughout the classroom. Particle concentrations, thermal conditions, and modified ventilation system solutions are reported.*

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## 1. Introduction

Humans spent more than half of their lives inside the building and the internal conditions. Improving indoor air quality and health benefits were outweighing physical costs. The importance of experimental and numerical research on indoor air quality and thermal comfort. Investigated numerically and experimentally the indoor air quality (IAQ) that became an important issue and as a result, researchers have developed a large number of different air quality indicators. Air exchange efficiency ($\varepsilon_a$) and contaminant removal effectiveness ($\varepsilon$) are the focuses of this study. Novoselac and Srebric [1] presented that the experimental and numerical study was performed for the comparison of thermal comfort (TC) and indoor air quality (IAQ) in the lecture room for cooling loads when the operating conditions are changed. Noh1[2] Conducted an experimental and numerical study on the comparison of thermal comfort (TC) with the indoor air quality (IAQ) in the lecture room for cooling Loads when the operating conditions are changed. Noha [3] investigated experimentally the indoor air (IA) pollution in lecture rooms of Tottori University in Japan. The air was studied quality by monitoring the CO2 level, an index of IA pollution. The IA environment of lecture rooms was changed, while the lecture was given by using fan ventilation or not and with room doors open or closed. Daisuke Houri 2009[4] studied experimentally the effect of low ventilation rates (1 or 0.5 air change per hour) on the thermal comfort and ventilation effectiveness in a simulated residential room equipped with radiant floor heating/cooling and mixing ventilation systems for various positions of supply and extract air terminals and different winter and summer boundary conditions. Tomasi [5] investigated experimentally how CO2 is distributed within a complex indoor environment of a classroom and how this distribution is affected by different parameters. The main source of indoor carbon dioxide (CO2) due to occupants is the exhalation. Elevated levels of CO2 may serve as an indicator of insufficient ventilation. Mahyuddin, et al. [6] investigated experimentally the airflow distribution systems and ventilation effectiveness to identify and assess the most suitable room air distribution methods for various spaces. The investigation showed that numerous studies have been carried out on ventilation effectiveness, and different ventilation systems are classified according to specific requirements and assessment procedures. Cao [7] Presented that the experimental is a very popular method for the Measurements of carbon dioxide concentration becomes a very popular method for the determination of the air exchange rate in buildings. One sensor was used in a single room this causes that the positioning of the CO2 sensors is crucial for such measurements and attempts to find the representative measuring area of mean CO2 concentration in a particular room with one sleeping person. Anna Bulinska 2014[8] investigated an office room in cooling mode ventilated by a 4-way cassette air conditioner, which is increasingly being installed in Eastern China 4-way cassette ac was compared against a typical wall-mounted mixing ventilation system based on ADPI calculation difference from the air and velocity. Bamodu, and Oluleke [9] study dealt with the effects of carbon dioxide concentration ventilation and indoor air quality and the extent of the impact of high concentration on the building occupants and on ventilation rates. Andrew [10] demonstrated that indoor air quality has a global public concern for many years. Most people spend up to 90% of their time indoors. The findings show that most researchers prefer heights between 1.0 m to 1.2 m in the middle of an occupied zone as a representative location. This compares to the recognized breathing zone height of between 0.75 m to 1.8 given in the literature surveyed [11].

In this work, an experimental and numerical analysis will be conducted in a full-scale test room to study the effect of the discharge airflow rate of the ceiling type air-conditioner on ventilation performance in the lecture room with mixing ventilation.

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## 2. Research Method

CFD software "ANSYS-Fluent 17.0" was employed for simulating. The airflow in the Chamber is 3-D, steady-state, incompressible, and turbulent. The experiment rig was made on a full-scale room design concentrated in mechanical of the department of the University of Technology, Iraq. The dimension of (3 m×2.5 m×2.3 m) that was built according to a suitable rang.

The air conditioning system was a Cassette Air Conditioner, the maximum thermal load was calculated using the carrier program and then selecting the suitable power System. It is an Interior unit that may be installed in a ceiling and blows downwards. Depending on the number of air vents, it is called 4-way, 2-way, or 1-way. Physical data are available in Table, 1.

## 3. CFD Simulation

A 3D model of lecture room was generated utilizing (SolidWorks 2016) used CFD software ANSYS (Fluid FLUENT) version 17.0, to analyze the heat transfer and flow, the physical model to verify the numerical method shown in Figure 1. For the computational model [12]. The geometry is generated by using SolidWorks to build the lecture room components (students, lights, president, and table). Then to generate a three-dimensional solid model. Two cases of lecture rooms were studied and modeled are shown in Table, 2 (a, b). The ANSYS software was the solution is made by connecting the penalty to produce the solution [13]. The occupants were modeled as the cylinder each human body was simulated by 0.3 x 1.2 m2 N. Mahyuddin and H. Awbi 2010[6].

*I. Conservation Equations for Scalar property*

$$\frac{\partial(\rho\Phi)}{\partial t}+\frac{\partial}{\partial x_j}(\rho u_j\Phi)=\frac{\partial J_{\Phi,j}}{\partial x_j}+S_\Phi \quad (1)$$

**Table 1: Physical data for the LG inverter Cassette Air Conditioner ModelUT18Q2**

- Indoor unit

| Physical data | | | |
|---|---|---|---|
| Power Supply | | ø/V/Hz | 220-240 / 1 / 50 |
| Capacity | Cooling 5.5 | Min/Nom/Max | kW2.0 ~ 4.7 ~ |
| | Heating 6.1 | Min/Nom/Max | kW2.2 ~ 5.5 ~ |
| Running Current | | | A0.4 |
| Heat Exchanger 1 | | 1_Coil Row x Column x FPI | mm(2 x 10 x 18) x |
| Fan | | Motor Output x Number of Unit | W x No43 x 1 |
| Air Flow Rate (H / M / L) | | m3/min | 25 / 22.8 / 21.6 |
| Sound Level 36 | | (H / M / L) | dB(A) 41 / 39 / |
| Dehumidification Rate | | l/h | 2.1 |
| Refrigerant | | | R410 |

- Outdoor unit

| Physical data | | | |
|---|---|---|---|
| Running Current | Cooling | A | 6.6 |
| | Heating | A | 6.9 |
| Power Supply | | ø/V/Hz | 220-240 / 1 / 50 |
| Power Supply Cable (includes Earth) | | No.xmm² | 3C x 2.5 |
| Dimensions | | WxHxD | mm 870×655×320 |
| Compressor | Oil Type | FVC68D | |
| | Oil Charge | cc x No | 900 x 1 |
| Refrigerant | | Cycle Charge | g1,400 |

**Table 2: (a): First case study parameter (b): Second case study parameter**

**(a)**

| Ventilation flow rate $m^3/h$ | Air cooled flow rate $m^3/h$ | ACH | $T_S$ $^0C$ | $T_{vent}$ $^0C$ | Velocity $_{(s)}$ m/s | Velocity $_{(vent)}$ m/s | $CO_{2(vent)}$ Ppm |
|---|---|---|---|---|---|---|---|
| 135 | 1512 | 7.8 | 20 | 37 | 4.2 | 0.4 | 380 |

(b)

| Ventilation flow rate $m^3/h$ | Air cooled flow rate $m^3/h$ | ACH | $T_S$ $^0C$ | $T_{vent}$ $^0C$ | Velocity $_{(s)}$ m/s | Velocity $_{(vent)}$ m/s | $CO_{2(vent)}$ ppm |
|---|---|---|---|---|---|---|---|
| 171 | 1512 | 9.9 | 19 | 37 | 4.2 | 0.52 | 380 |

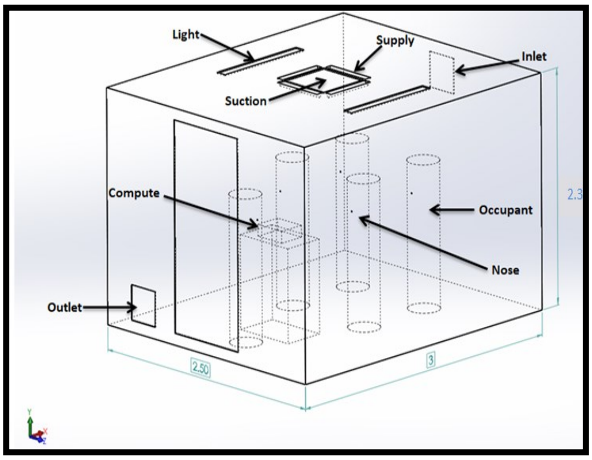


**Figure 1: Basic dimensions of the lecture room model geometry**

*II. Governing Equations*

The three physical laws are mass (Continuity), momentum, and energy conservation equations [15].

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad (٢)$$

$$\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho vu)}{\partial y}+\frac{\partial(\rho wu)}{\partial z}=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right] \quad (٣)$$

$$\frac{\partial(\rho v)}{\partial t}+\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho wv)}{\partial z}=-\frac{\partial p}{\partial y}+\mu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right]+S_{bj} \quad (٤)$$

$$\frac{\partial(\rho w)}{\partial t}+\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}=-\frac{\partial p}{\partial z}+\mu\left[\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right] \quad (٥)$$

$$\rho\frac{\partial}{\partial_x}(uT)+\rho\frac{\partial}{\partial_y}(vT)+\rho\frac{\partial}{\partial_z}(wT)=\frac{\partial}{\partial_x}(\Gamma_{eff.h}\frac{\partial T}{\partial x})+\frac{\partial}{\partial_y}(\Gamma_{eff.h}\frac{\partial T}{\partial y})+\frac{\partial}{\partial_z}(\Gamma_{eff.h}\frac{\partial T}{\partial Z})+S_T \quad (٦)$$

*III. Turbulence Modeling: standard k-epsilon*

$$\frac{\partial}{\partial_x}(\rho\, uK)+\frac{\partial}{\partial_{y(}}\rho\, v\, K)+\frac{\partial}{\partial_z}(\rho wK)=\frac{\partial}{\partial_x}(\frac{\mu t}{\sigma k}\frac{\partial K}{\partial x})+\frac{\partial}{\partial_y}(\frac{\mu t}{\sigma k}\frac{\partial K}{\partial y})+\frac{\partial}{\partial_z}(\frac{\mu t}{\sigma k}\frac{\partial K}{\partial Z})+G_{K-}\rho\varepsilon \quad (٧)$$

σk, σε, C1, C2, C3, Cμ are the constants of the turbulent model Are given in Table 3.

*IV. Convergence criteria*

Residuals of scaling for Convergence Criterion of continuity, momentum, Energy are presented in Table 4.

*V. Boundary Conditions*

Boundary conditions specify the flow and thermal variables on the boundaries of the physical model are presented in Table 5.

*VI. Volume Mesh Generation*

After all the parameters mentioned previously have meshed, now volume mesh can be created. Building the mesh requires fine cells in the outlet sections, students, light, president, and inlet section. On the other hand, using this element size in the whole domain would lead to an enormous number of elements, that is why it was decided to use a fine mesh in the region near to the surfaces mentioned, and to use coarse meshes as the distance from the surface grows as shown in Table 6, [17].

*VII. Mesh independence*

To adequately predict the situation, several grid options were considered. Grid independence test was performed to make sure that the accuracy of the simulation is not influenced by the grid choice as shown in Table 7, [18].

**Table 3: Constants of the standard k-ε model**

| σk | σε | C1ε | C2ε | Cμ |
|---|---|---|---|---|
| 1.0 | 1.3 | 1.44 | 1.92 | 0.09 |

**Table 4: Scaling residual**

| Continuity | X-Velocity | Y-Velocity | Z-Velocity | Energy |
|---|---|---|---|---|
| $10^{-3}$ | $10^{-3}$ | $10^{-3}$ | $10^{-3}$ | $10^{-6}$ |

**Table 5: Boundary Conditions**

| Part | Type | Boundary Conditions |
|---|---|---|
| Wall | Wall | $T_O$ =54 ° C<br>Thermal Convection (h) =8.2(W/m$^2$.K)[16] |
| Door | Wall | $T_O$ =54 ° C<br>Thermal Convection (h) =8.2 (W/m$^2$.K)[16] |
| Ceiling air condition (4way) | Inlet velocity | inlet Velocity = (4.2) m/s<br>out flow/ Gage pressure=0 pa<br>T= (19-20) ° C |
| Outlet sections | Flow out | Gage pressure=0 pa |
| Input section(ventilation) | Inlet velocity | -Velocity = (0.4-0.52) m/s<br>-T= 37 ° C<br>- $CO_2$ concentration =380 ppm |
| Human body | Wall | T= 34 ° C<br>- heat flux(q) = 60 (W/m$^2$.K) [6] |
| Exhalation | Velocity-inlet | $CO_2$ concentration = 350000 ppm equal to , mass flow rate = 0.01 (g/s) [6]<br>Volume Flow rate = (0.005 L/s),<br>( 8.5 L/m )<br>-Velocity inlet = 0.77 m/s<br>-T= 34 ° C |
| External environment conditions | Wall | -T = 45 $^0$ C |

*VIII. The Specific Plan Location*

However, in the experimental work, one has main sections that show the conditions of the space in which different cases are studied through different internal conditions. (Location and height) is shown in the Figures 2 and 3, and Tables 8, and 9.

## 4. Experimental work

Without ventilation, a building's occupants will initially be boor indoor air. The purpose of ventilation is to eliminate airborne contaminants, which are generated both by human activity and by the building itself [19].

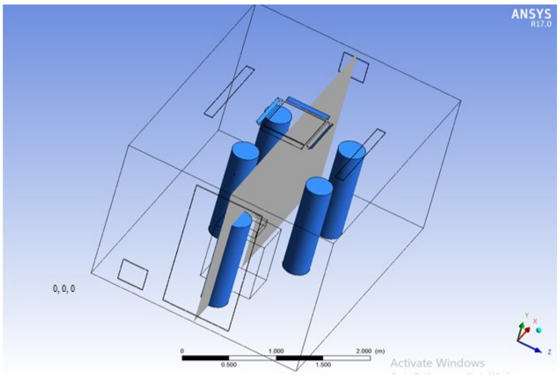


**Figure 2: $CO_2$ section in the test room**

*I. System Description*

The experiment rig was made in a full-scale room. To simulate the distribution of the velocity profile, and $CO_2$ concentration by fixing (33) measuring devices according to the requirement in the array scheme in one zone and heights. Air velocity and $CO_2$ concentration can reach stability taken over a 3hour period the instrument readings of $CO_2$ concentration for these strategies, within did reach constant readings. It neglects the infiltration of air. The effect heat load is due to nature and forced convection currents after calculating the no- parameter Richardson number[20]. Rig characteristics were chosen per Table 10, and Figure 4.

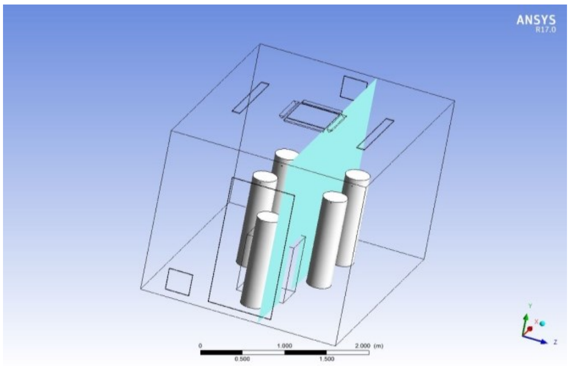


**Figure 3: Velocity section in the test room**

**Table 6: Three-dimensional volumes of meshes parameters**

| Volume | Element Shape | Mesh Type | Interval Size (m) | |
|---|---|---|---|---|
| Air | Tetrahedral | T-Grid | 0.038 | 1.005 |

**Table 7: Different element size with an average of air parameters and $CO_2$**

| Type of mesh | Number of elements | $CO_2$ concentration ppm | Air velocity m/s |
|---|---|---|---|
| Corse mesh | 3952551 | 751 | 0.33 |
| Fine mesh | 4642898 | 760 | 0.34 |
| Percentage difference | | 1.1% | 2.9% |

**Table 8: Coordinated of $CO_2$ sectioned in the test room**

| Contour | X-axis | Y-axis | Z-axis |
|---|---|---|---|
| $CO_2$ sectioned | 3m | 2.3m | 1.25m |

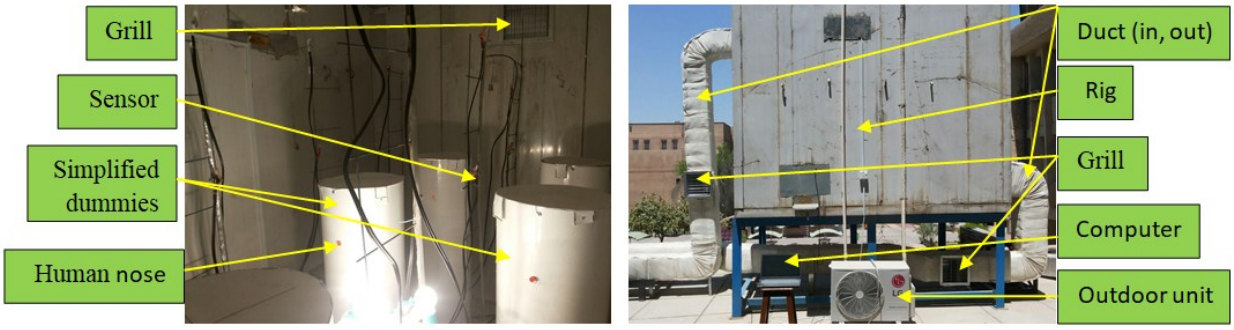


**Figure 4: Full-scale lecture Room geometry**

*II. Occupants model*

Occupants were modeled as the cylinder (0.3D x 1.2 H) $m^2$ [21]. Thermal load was 100 watts, $60W/m^2$ (ASHRAE, 2004) [14] . The $CO_2$ generation rate is 0.0052 L/s from a person in office work [22]. Air exhalation breath has a temperature of 34 $C^0$ and relative humidity 95% suggested by ASHRAE standard 62-1-2013 [23]. The proposed method is used to estimate $CO_2$ concentrations for the spaces where activity levels are relatively stable and occupants are present for long enough time, such as office room, lecture room, conference rooms [24].

$$V_O = N/ (C_s - C_o) \tag{9}$$

$$C_s - C_o = N/V_o = 0.31/(7.5 \times 60 \text{ s/min}) = 0.000689 \text{ L of } CO_2 \text{ per L of air} \approx 700 \text{ ppm}$$

Therefore, recommends that the average $CO_2$ concentration does not exceed 700 ppm inside space Figure 5, [22].

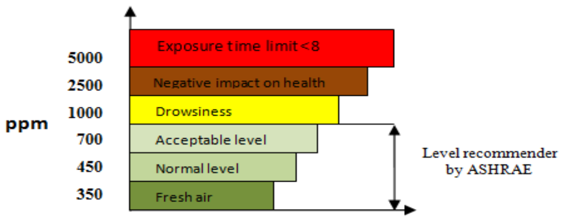


**Figure 5: $CO_2$ level published by American society refrigerating and air conditioning**

*III. Exhaled velocity*

The student's nostril is represented by a 12 mm diameter hole at a height of 1.1 m from the body to simulate real human and from which the expiratory flow rate can be calculated by using the following equation[25].

$$V_{exit}=Q/S=Q / ((\pi D2)/4$$

*IV. Experimental Processing*

The ceiling cassette radiator starts to cool the lecture room in four directions to ensure a good distribution of air inside the space shown in Figure 6.

**Figure 6: Inverter 4-way cassette air conditioner**

This study focuses that there is some evidence of variations in $CO_2$ concentration in a classroom space, the spatial distribution of $CO_2$ level in a classroom carried out in the previous field. In a typical building, the constant emission of $CO_2$ by occupants is diluted by outside air introduced by mechanical ventilation. The desire for thermal comfort and acceptable indoor air quality usually provides conflicting constraints on the ventilation system. While a high ventilation rate may increase the removal of gaseous contaminants, ANSI / ASHRAE Standard 55-2010 [23]. may increase the removal of gaseous contaminants, ANSI / ASHRAE Standard 55-2010 [23].

*V. Calibration of the tools*

A Digital-output sensor was calibrated before using $CO_2$ concentration (MQ-135 and MG811 GAS sensor), as shown in Figure 7.

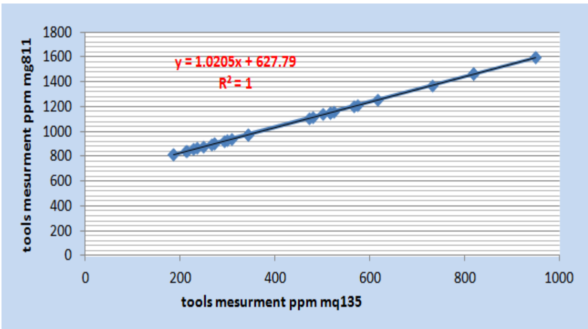


**Figure 7: Calibration between MQ-135 and MG811 GAS sensor**

*VI. Hardware: a measurement device Data Acquisition DAQ*

The National Instrument Two Arduino MEGA 2560 is one of the main instruments that are used in this project, as shown in Figure 8.

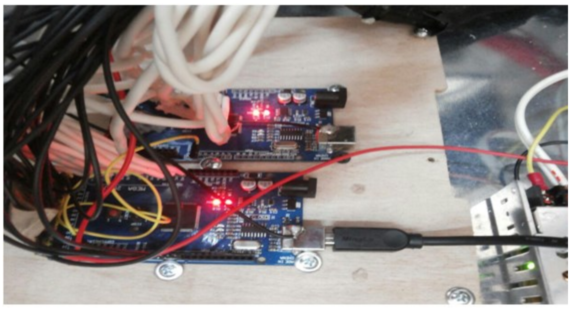

**Figure 8: Pre-configuration component Data Acquisition**

Air Quality Control sensor has good sensitivity to carbon dioxide when humidity and temperature are low, as shown in Figures 9, and 10 which show respectively Mg-811- MQ-135 gas sensors and $CO_2$ tank steel gas cylinder. The air velocity measurement was done by using the hot wire Anemometer as shown in Figure 11. The thermal Anemometer, Model No. Yk-2005AH. It works excellent for conducting commissioning work and troubleshooting HVAC systems.

**Table 9: Coordinated of air velocity sectioned in the test room**

| Contour | X-axis | Y-axis | Z-axis |
|---|---|---|---|
| air velocity sectioned | 3m | 2.3m | 1m |

**Table 10: laboratory chamber characteristic**

| Parameter rang | value |
|---|---|
| Grill dimensions | 0.3×0.3 m |
| Ach | ٧.٨ |
| Nominal face velocity | 0.25 m/s |
| Room air temperature | 23 $^0$C |
| Supply air temperature | 20 $^0$C |

**Table 11: Characteristics of the measurement system**

| sensor -model | Function | Range | Resolution | Accuracy |
|---|---|---|---|---|
| MQ-135 GAS | $CO_2$ | 350 to 10000 ppm | 10 ppm | +-1 ppm |
| Yk-2005AH | Air velocity | 0.2 to 20 m/s | 0.01 m/s | ±0.025 m/s |
| GM700 | Temperature | -50~700 Celsius | 0.1 °C | ±1.5 Celsius |
| Mg-811 GAS | $CO_2$ | 350 to 10000 ppm | 10 ppm | +-1 ppm |

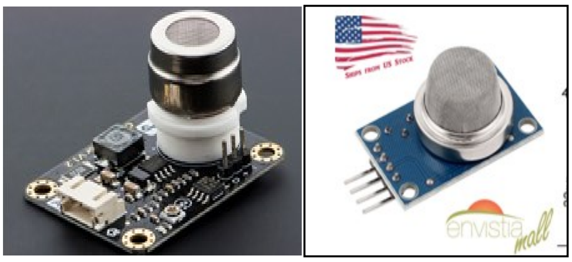


**Figure 9: MG811AND MQ135 gas sensor for $CO_2$ measurement**

**Figure 10: Carbon dioxide regulator flow rate and the steel vessel**

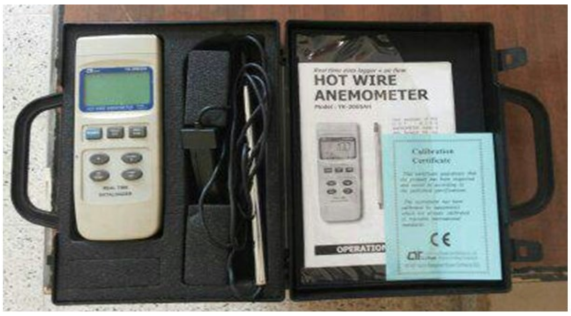


**Figure 11: Hotwire Anemometer air velocity measurement**

The wall temperature was measured by using an Infrared thermometer MODEL: GM700 from BENETECH Company, as shown in Figure 12. Increasing the measurement accuracy in practice is done by repeating the measurement successively to increasing confidence in values through statistical data. Quantities: average (or mean). Finally, the characteristics of a system device are shown in Table 12. A mechanical ventilation system was used as a full fresh air Single-Package air conditioner model 50EES—ZP. The packaged unit is shown in Figure 13.

**Figure 12: Infrared thermometer GM700**

**Figure 13: Packaged unit**

Under steady-state room conditions, the index of indoor air quality ($\varepsilon_c$) is defined as the ratio of the difference between the average concentrations in the interior ($C_s$) and the concentration exit air ($C_e$), and the difference of the room average concentration ($C_p$) and ($C_s$) [26].

$$\boldsymbol{\varepsilon_c} = \frac{C_e - C_s}{C_p - C_s} \tag{10}$$

The contaminant removal effectiveness takes values from zero to infinity. Table 12 shows ventilation cases where $\varepsilon_c$ values approach their lower and upper limits according to ASHRAE Standard 62-2001[27]. The air change effectiveness $\varepsilon_a$ of the space is calculated by relying on the calculation of the shortest time $\tau n$ of the switch that is extracted by knowing the size of the space and the average time for air exchange $2\tau P$.

$$\tau_n = \frac{V}{Q} \tag{11}$$

$$\varepsilon_a = \frac{\tau n}{2\tau P} \tag{12}$$

The air change effectiveness $\varepsilon_a$ takes values from 0 to 1. Table 13 shows ventilation cases where ε and $\varepsilon_a$ values approach their lower and upper limits according to ASHRAE Standard 62-2001[27].

**Table 12: Limits for contaminant removal effectiveness**

| | | | |
|---|---|---|---|
| Contaminate Removal Effectiveness | Upper limit | $\varepsilon_c \longrightarrow \infty$ | Contaminant source at the outlet Flow field does not have influence |
| | Perfect mixing | $\varepsilon_c \longrightarrow 1$ | Complete and instantaneous mixing Position of a contaminant does not have influence |
| | Lower limit | $\varepsilon_c \longrightarrow 0$ | Contaminant source is in the recirculation area, which is completely separated from the bypass area |

**Table 13: Details of limits for air exchange efficiency**

| | | | |
|---|---|---|---|
| Air exchange efficiency | Upper limit | $\varepsilon_a \longrightarrow 1$ | Ideal piston flow |
| | Perfect mixing | $\varepsilon_a \longrightarrow 0.5$ | Complete and instantaneous mixing |
| | Lower limit | $\varepsilon_a \longrightarrow 0$ | Bypass area and recirculation area are completely separated |

## 5. Results and Discussion

*I. First case*

The first case study is executed by calculating the data within the space through the operation of the ceiling air conditioning system at a flow rate of 1512 $m^3$/h and supply temperature $T_S$(20 $^0$C). The ventilation system flow rate is135$m^3$/h, and the air change rate is 7.8/h, as shown in Table 14, and Table 15. Figure 14 reveals the ppm contour at the section of (x-y) z [1.25 m]. This section shows the distribution of carbon dioxide concentration in the center of the room and near the occupants of the room where the majority of the concentration is 980 ppm except that the area near the exhalation of the nose of people has a very high concentration and low near the grill for ventilation where the rates are close to 1000 ppm as recommended of ASHRAE standards when ventilating is equal to 7.5 L/S. person. Figure 15, reveals the velocity contour at the section of (x-y) z [1m], it is noticed in this section, which is above the form of similarity, that the velocity rates are low and close to thermal comfort because of the impact of barriers and be close to zero near the walls in addition to distribution regularly throughout the room to provide a comfortable atmosphere for occupants.

- **Comparison of simulation and experimental results**

***a.*** **Concentration simulation.**

Figure 16**,** at plan (x,y) and (z= 1.25 m) in the diagram, represents the comparison between the experimental and theoretical values of the pollutant concentration. They are high in the room near the floor of the room, decrease with the rise, and are affected in the middle of the space due to their ventilation effect in addition to their proximity to the intake of the ceiling air conditioner. There are slight differences between these two results, and the errors ratios of them are do not exceed ٢%, we could be used as an estimating and calculations in the lecture room.

***b.*** **Velocity simulation**

Figure 17**,** at plan (x,y) and (z= 1 m) in the diagram, represents the comparison between the practical and theoretical values of air velocity in the space. The velocity varies between higher values near the center area due to the effect of the velocity of the ventilation currents and its proximity to the source of cold air and decreases towards the ceiling. The errors ratios of them are under 1٤% in the occupied zone, we could be used as an estimating and calculations in the lecture room.

**Table 14: Numerical and experimental $CO_2$ concentration in the test room**

| Numerical | | | | | Experimental | | | | | error percentages |
|---|---|---|---|---|---|---|---|---|---|---|
| Ventilation Methods | ACH 1/h | Average air $CO_2$ ppm | | | Ventilation Methods | ACH 1/h | Average air $CO_2$ ppm | | | |
| | | 0.2 | 1.2 | 1.8 | | | 0.2 | 1.2 | 1.8 | |
| Vent 135 m3 / h RC1512 m³/h | 7.8 | 976 | 957 | 962 | Vent135 m3 / h RC 1512 m³/h | 7.8 | 971 | 924 | 945 | 2% |

**Table 15: Numerical and experimental air velocity in the test room**

| Numerical | | | | | Experimental | | | | | error percentages |
|---|---|---|---|---|---|---|---|---|---|---|
| Ventilation Methods | ACH 1/h | Average air velocity m/s | | | Ventilation Methods | ACH 1/h | Average air velocity m/s | | | |
| | | 0.2 | 1.2 | 1.8 | | | 0.2 | 1.2 | 1.8 | |
| Vent135m3 / h RC1512 m³/h | 7.8 | 0.51 | 0.62 | 0.30 | Vent 135 m3 / h RC1512 m³/h | 7.8 | 0.47 | 0.49 | 0.29 | 14% |

**Table 16: Numerical and experimental $CO_2$ concentration in the test room**

| Numerical | | | | | Experimental | | | | | error percentages |
|---|---|---|---|---|---|---|---|---|---|---|
| Ventilation Methods | ACH 1/h | Average air $CO_2$ ppm | | | Ventilation Methods | ACH 1/h | Average air $CO_2$ ppm | | | |
| | | 0.2 | 1.2 | 1.8 | | | 0.2 | 1.2 | 1.8 | |
| Vent171m3 / h RC1512 m³/h | 9.9 | 850 | 836 | 831 | Vent171m3 / h RC1512 m³/h | 9.9 | 873 | 870 | 819 | 2% |

**Table 17: Numerical and experimental air velocity in the test room**

| Numerical | | | | | Experimental | | | | | error percentages |
|---|---|---|---|---|---|---|---|---|---|---|
| Ventilation Methods | ACH 1/h | Average air velocity m/s | | | Ventilation Methods | ACH 1/h | Average air velocity m/s | | | |
| | | 0.2 | 1.2 | 1.8 | | | 0.2 | 1.2 | 1.8 | |
| Vent171 m3 / h RC1512 m³/h | 9.9 | 0.3 | 0.23 | 0.23 | Vent171 m3 / h RC1512 m³/h | 9.9 | 0.29 | 0.2 | 0.2 | 10% |

*II. Second case*

The second case study is performed by calculating the data within the space through the operation of the ceiling air conditioning system at a flow rate of1512 $m^3$/h and supply temperature $T_S$ (19 $^0$C). The ventilation system for fresh air is in the case of operation. The ventilation system flow rate is 171$m^3$/h, and the air change rate is 9.9 /h, as shown in Table 16, and Table 17.

***a.*** **Carbon dioxide distribution**

Figure 18, shows the ppm contour at the section of (x-y) z [1.25 m], this section reveals the distribution of carbon dioxide concentration in the center of the room and near the occupants of the room where the majority of the concentration is 780 ppm, except the area near the exhalation of the nose of occupants has a very high concentration and low near the grill for ventilation where the rates are close to 780 ppm as recommended by ASHRAE standards when ventilating is equal to 9.5 L/S. person.

***b.*** **Velocity distribution.**

Figure 19, shows the velocity contour at the section of (x-y) z [1m], it is noted in this section, the velocity rates are low near to thermal comfort because of the impact of barriers and are close to zero near the walls in addition to distribution regularly throughout the room to provide a comfortable atmosphere for occupants

***c.*** **Comparison of Simulation and Experimental**

i. $CO_2$ concentration simulation

Figure 20, at plan (x,y) and (z= 1.25 m) in the diagram, represents the comparison between the process and theoretical values of the pollutant concentration. They are high in the room near the floor of the room and decrease with the rise and are affected in the middle of the space due to their ventilation effect in addition to their proximity to the intake of the ceiling air conditioner. There are slight differences between these two results, and the errors ratios of them are do not exceed ٢%, we could be used as an estimating and calculations in the lecture room.

ii. Velocity simulation

Figure 21, at plan (x,y) and (z= 1 m) in the diagram, represents the comparison between the practical and theoretical values of air velocity in the space. The velocity varies between higher values near the center area due to the effect of the velocity of the ventilation currents and its proximity to the source of cold air and decreases towards the ceiling. There are slight differences between these two results, and the ratio of error them are under 1٠%, we could be used as a comparison of simulation and experimental results.

**Table 18: Details of a concentration and location of the chamber**

| Ventilation rate | $C_e$ | $C_s$ | $C_p$ | Level/meter |
|---|---|---|---|---|
| 7.5L/S.P | 978ppm | 380ppm | 976 ppm | 0.2 |
| 7.5L/S.P | 978ppm | 380ppm | 955 ppm | 1.2 |
| 7.5L/S.P | 978ppm | 380ppm | 944 ppm | 1.8 |
| 9.5L/S.P | 912ppm | 380ppm | 867 ppm | 0.2 |
| 9.5L/S.P | 912ppm | 380ppm | 824 ppm | 1.2 |
| 9.5L/S.P | 912ppm | 380ppm | 804 ppm | 1.8 |
| Average 7.5L/s.p | 978ppm | 380ppm | 958 ppm | 1.15 |
| Average 9.9L/s.p | 912ppm | 380ppm | 831 ppm | 1.15 |

**Table 19: Details of air exchange efficiency for chamber**

| Ventilation rate | $V$ | $Q$ | $\tau n$ | $\tau P$ | $\varepsilon_a$ |
|---|---|---|---|---|---|
| 7.5L/S.P | 17.25m$^3$ | 135m$^3$/h | 7.6 minute | 5.1 minute | 0.45 |
| 9.5L/S.P | 17.25 m$^3$ | 171m$^3$/h | 6.05 minute | 6.5 minute | 0.46 |

One of the main goals of this project was to verify the computational models using the experimental data were obtained from the test room, which is located at the University of Technology Iraq**.** The overall difference between the numerical results and the experimental once is about 7.5 percent. A reasonable good agreement was achieved for temperature between numerical results and experimental results. Possible reasons for the differences between the numerical and experimental data are:

a. Size of the computational domain.
b. Non-uniformity for the inlet mass flow rate for the inlet (grill, nose, supply). Our model assumed that the flow rate in each one was equal.
c. Experimental blind regions. The experimental data were recorded for data points. This leaves large amounts of the experimental space were no data where recorded. The computational grid was significantly finer than the experimental one.

iii. Ventilation effectiveness to remove contaminant $\varepsilon_c$

From Table 12, the efficiency of the removal of pollutants shows the presence of three levels; zero, one, and infinity which depends on the number of pollutants source, location, space area, and regularity. Where the number one indicates that the mixing occurs in a very good condition. Which is a useful measure for designers in the choice of ventilation strategy that suitable for the location shown in Table 18, and Figure 22. the diagram shows the values of the effectiveness of the removal of pollutants relative to the source locations of the chamber that the case of the flow of 7.5 liters/second per person is close to the one, especially in the area near the nose and mouth of students, and this ratio increases when the increase of ventilation flow like 9.9 liters/second per person due to increased air switch parameters in the chamber. Figure 23, illustrates the overall effect of the removal of pollutants for the two ventilation cases by relying on the concentration rates for all areas of the space where the ratios are close to one, which represents a good mixing quality of the space.

iv. Air exchange effectiveness $\varepsilon_a$

one can see in Table 19, that the value of the air change efficiency is close to the half where the mixing is ideal, but the reason for the rise is the small space, where the size of the space has a significant impact on the mixing to allow more efficient circulation of hot air and smaller the volume, and more ventilation that is closer to the displacement system consumes more energy due to increased loads.

## 6. Conclusions

The main control of indoor air quality and maintaining the health of lecturers and students reduce the concentration of carbon dioxide under 1000ppm. The results were practically analyzed in the test room and compared with the analytical results obtained from the application of computational fluids dynamics, and the results were satisfactory in terms of their convergence. The important points in the Experimental and Numerical Work are:

1. $CO_2$ level, an index of indoor air quality can be detected by the ratio of air space volume occupants and ventilation strategy.
2. The air velocity distribution is low variation near the occupied zone due to its indirect from the current source and is within the recommended thermal comfort ranges.
3. The Ventilation and discharge airflow one can be suggested to satisfy both the indoor air quality and thermal comfort according to the [ASHRAE Standard 62, ANSI / ASHRAE Standard 55-2010], with rates of 7.5 L / s. person and 9.5 L / s. person respectively achieving rates close to 1000 ppm. Percentage compared ventilation effectiveness 8% from 9.5 L / s. person up to 7.5 L / s. person.
4. The ceiling device comprises the specifications that achieve thermal comfort and internal air quality, including the angle of air drainage where it can be controlled and the more it achieves more thermal comfort but because of the small room space, we chose a 45-degree angle**.** A good mixing leads to a reduction in expenditures by quickly achieving the required conditions.
5. The value of the air change rate, pollutant removal efficiency were extracted in the research. These are very important criteria, especially for designers to know the quality of the equipment and the ventilation method that satisfies the purpose of the place before installation.
6. Effectiveness values of air change and removal of contaminants were within the values of close mixing, this means achieving a high internal air quality by increasing the air life of the space and not leaving dead areas away from ventilation and currents that prevent the deposition of the genes and make the sides of the space in semi-equal concentrations.

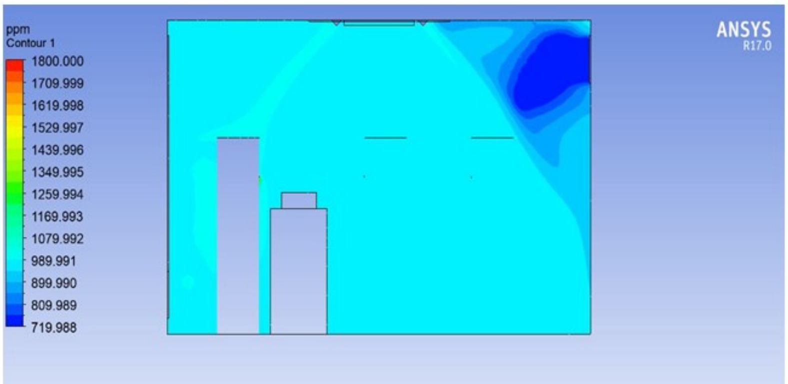


**Figure ١4: $CO_2$ contours distributions at the plane of (x-y) z [1.25m]**

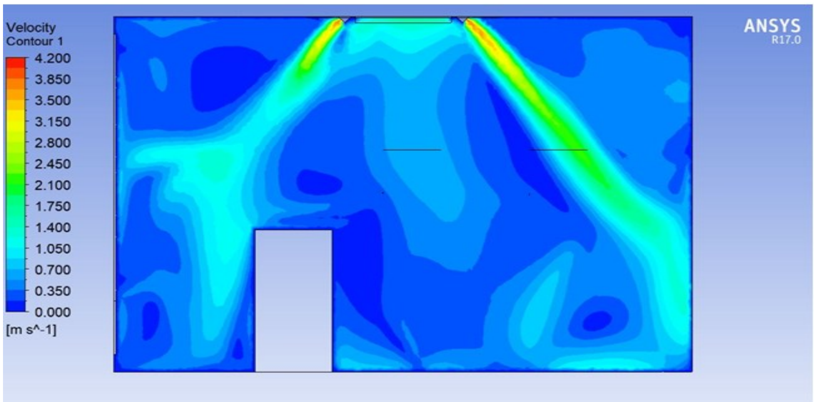


**Figure 15: Velocity contours distributions at the plane of (x-y) z [1m]**

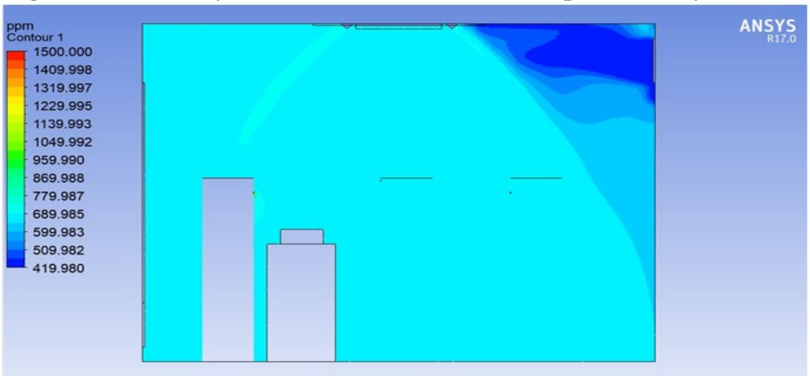


**Figure 16: ppm contours distributions at the plane of (x-y) z [1.25m]**

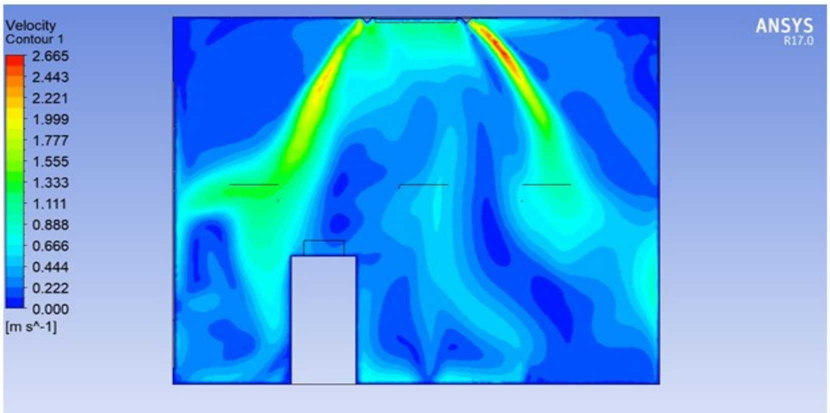


**Figure 17: Velocity contours distributions at the plane of (x-y) z [1m]**

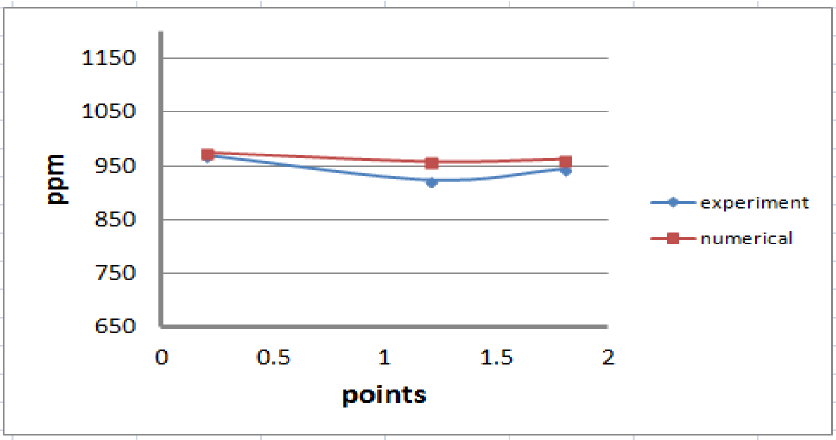


**Figure 18: A comparison of $CO_2$ concentration distance at plan (x,y) and (z= 1.25 m) in the lecture room**

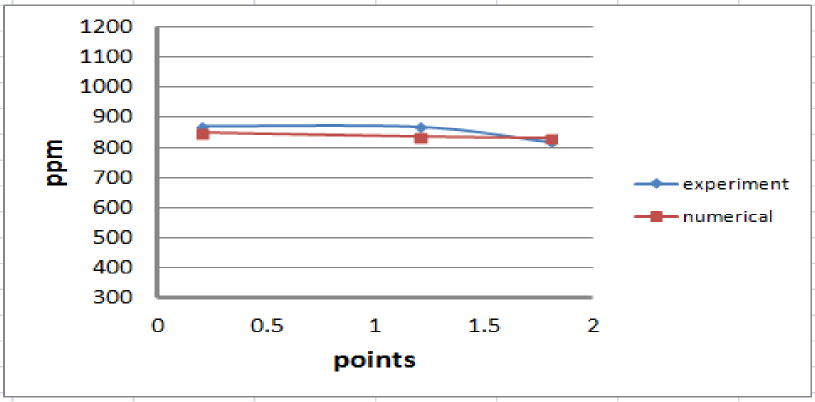


**Figure 19: A comparison of air velocity distance at plan (x,y) and (z= 1m) in the lecture**

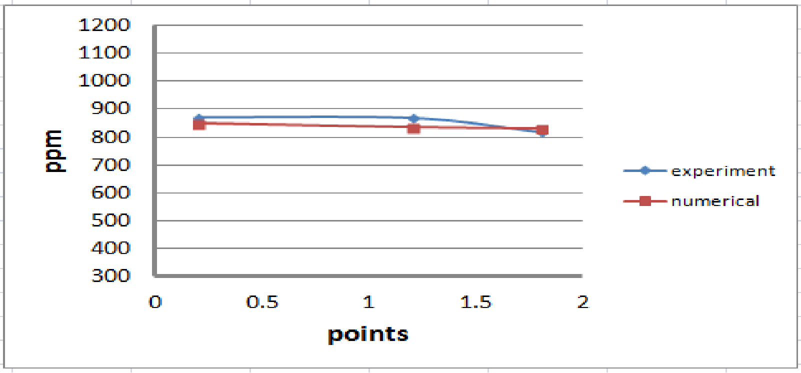


**Figure 20: A comparison of $CO_2$ concentration distance at plan (x,y) and (z= 1.25 m) in the lecture room**

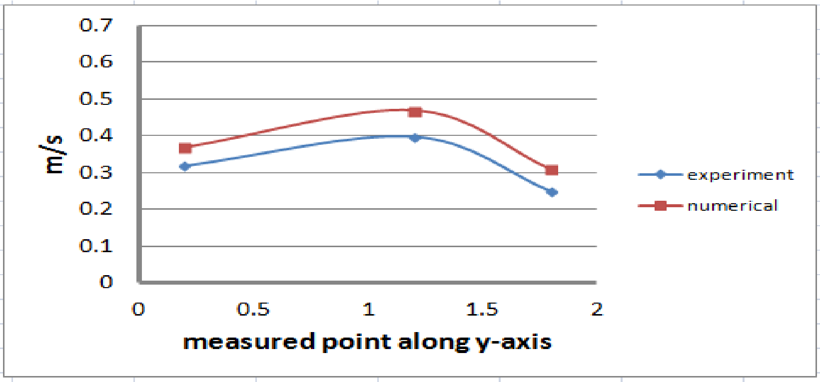


**Figure 21: A comparison of air velocity distance at the plane (x,y), and (z= 1m) in the lecture**

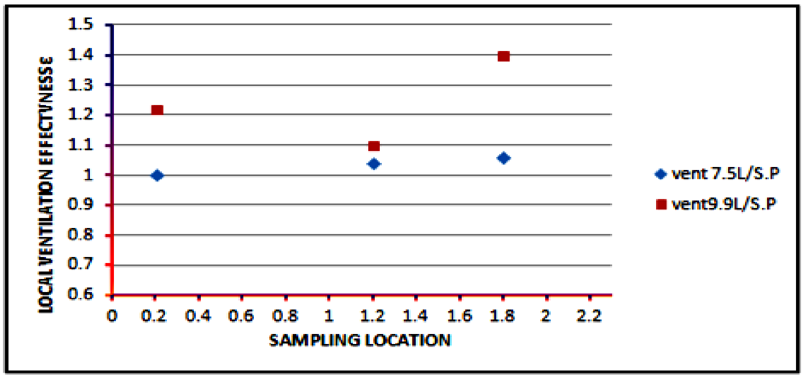


**Figure 22: Comprehensive ventilation effectiveness base on the average Concentration**

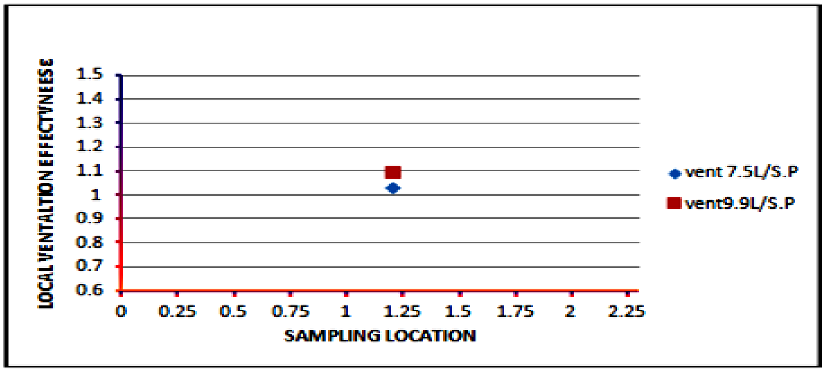


**Figure 23: Values of the effectiveness of The Removal of Pollutants location**

## Abbreviations

ASHRAE: American Society of Heating Refrigerating and Air Conditioning Engineer
CFD: Computational Fluid Dynamics
IAQ: Indoor air quality
ACH: Air change per hour

Subscripts
I: Inter
O: Outer
S: Supply
Eff: Effective condition

*II .N omenclature*

A: area, unit m2
Cμ,C2ε,Cεσε σk: Turbulence Model Constants
A: Area, (m2)
V: Volume(m3)
Q: Flow rate(m3/h)
C.p: Specific heat, (J/kg. k)
T: Temperature, K
J: Diffusion flux (kg/ m^2.s)
g: Gravitational acceleration (9.81 m/s^2)
Sm: Source term in momentum equation
μeff: Effective viscosity
P: Pressure, (Pa)
k: Kinetic energy, (J/Kg)

*III. Symbols, Greek*

σε, σk: Turbulent Prandtl number K and Ɛ
Γ: Diffusion coefficient
μ: Dynamic viscosity of the fluid (kg/m)
ρ: Density of fluid (kg/m3 )
εa: Air exchange effectiveness
τn:Time constant(min)
τp; room-averaged mean air age(min)
h: Coefficient of heat transfer, W/m^2.k
U, V, W: Velocity component in x, y, and z- directions m/s
G_b: Turbulence kinetic energy owing to the buoyancy
Gr : Grashof number
Re : Reynolds number
C: CO2 concentration unit (ppm)
Cs: CO2 concentration interspace (ppm)
Ce: CO2 concentration outer space (ppm)

Cp: average CO2 concentration in space (ppm)
vo : outdoor airflow rate per person. (L/minute)
N ; CO2 generation rate per person. (L/minute)
Q: is flow rate around 8.5 (L/minute)
S : cross-section area( mm2)
D : nostril diameter in this study 12 (mm).

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