Simulation of Thermal Comfort in the Lecture Room Equipped with Direct Expansion Air Conditioner and Ventilation System

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Abstract

This research investigation in airflow characteristics and thermal comfort inside a lecture room contained load under cooling by a ceiling air conditioner of 4-way cassette with 45° discharger angle, and ventilating system. A modified version of a three-dimensional software FLUENT was used to simulate the complex flow inside the room. The computational program was validate by comparing the numerical results with experiential result of temperature and air velocity measurement. The comparison showed a good agreement. The results showed that the ceiling air conditioner with mixing ventilation system is more effective in achieving the airflow mixing, temperature distribution, and thermal comfort as recommended by interval environment condition of ANSI/ASHRAE Standard 55-2010.

Keywords: Temperature Distribution (TD), Computational Fluid Dynamics CFD, Thermal Comfortable (TQ), 4-way Cassette Air-conditioner and Performance Coefficient of Air Distribution

محاكاة الراحة الحرارية داخل غرفة محاضرة مجهزة بمكيف هواء ذات التمدد المباشر ونظام التهوية

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الخلاصـــة

هذا البحث يبحث في تدفق وخواص الهواء والراحة الحرارية (TC) داخل غرفة محاضرات تحتوي على حمل تبريد بواسطة مكيف هواء سقفي رباعي الاتجاهات بزاوية تصريف °45، مع منظومة تهوية. استخدمت نسخة معدلة من برنامج FLUENT لمحاكاة التدفق ثلاثي الابعاد المعقد داخل الغرفة. تحقق من صحة البرنامج الحسابية من خلال مقارنة النتائج العددية مع النتائج التجريبية لقياس درجات الحرارة وسرعة الهواء. اظهرت المقارنة توافقا جيدا. أظهرت النتائج أن مكيف الهواء السقفي ونظام التهوية الخلطي أكثر فاعلية في تحقيق تدفق الهواء وتوزيع درجة الحرارة والراحة الحرارية كما هو موصى به من قبل معيار الظروف البيئية الداخلية ANSI/ ASHRAE القياسية 55-2010.

الكلمات المفتاحية: توزيع درجات الحرارة وديناميك الموائع الحسابية والراحة الحرارية ومكيف هواء سقفي رباعي الاتجاهات ومعامل الاداء لتوزيع الهواء

البحث مستل من رسالة ماجستير للباحث الأول

Introduction

Good internal conditions depend on internal thermal comfort which is an indicator by measuring the effects of temperature and relative humidity in the interior space. a numerical investigation of thermal comfort in an office room with 4-way cassette air-conditioner Buildings account for about 40% of total energy consumption worldwide (Bamodu, *et al*., .2015) a comparison between an office with a ceiling and a wall cooling device and the extent of its impact on air distribution as roof equipment is commonly used in East Asia (Oluleke, *et al*., 2017) In this field to test ventilation using three methods mixing, displacement and stratification for the purpose of knowing the differences between them and provide thermal comfort (Wiley, *et al*., 2011). The low air change rates of a residential room from 1 to half and its effect on thermal comfort and the effectiveness of air change and ventilation of radioactive land in winter and summer illustrated by (Simone and Olesen 2013). Experimental and numerical analysis of the lecture room with a ceiling air conditioner and ventilation system and its impact on indoor air quality and thermal comfort (Noh, *et al*., 2005). the best operation conditions of 4-way cassette air-conditioner in a lecture room to satisfy both the TC and CO_2 limit, when the discharge angle, the discharge airflow and the ventilation rate is 30 degrees, 23 CMM, and 800 CMH respectively as mentioned by (Houri, *et al.*, 2009). Applying numerical results on variable ventilation systems during the year for the purpose of saving energy and comparing it with the field side which depends on the occupants' load suggested by (Lu, *et al*., 2010). Experiments, numerical analyses, modeling and practical results of a person sleeping in the room with the measurement of carbon dioxide rates by a sensor in the center of the room (Popio, *et al*., 2014). The numerical results of temperature and flow rates were analyzed and compared with the test chamber. It was found that the displacement ventilation and cooling system of roof panels are one of the most efficient types (Liu, *et al*., 2009) “The aim of this study is to uses simulate the thermal comfort inside a lecture room equipped with direct expansion air conditioner and ventilation system. The numerical work is based on non-isothermal, incompressible, three-dimensional, (K-ε) turbulence model, and solved using a computational fluid dynamic (CFD) approach, used laws of mass conservation, momentum with turbulence modeling, and energy conservation.

Material and Methods

Symbols, Greek, & abbreviations

G_K: Generation of turbulence kinetic energy due to the mean velocity gradients

T: Temperature

ρ : Air density

Φ: Independent variables

S_Φ: Source term of the general flow property

J: Diffusion flux (Kg/ m^2.s)

σε,σk: Turbulent prandtl number K and Ɛ

Γ: Diffusion coefficient

μ: Dynamic viscosity of fluid (Kg/m s)

ρ: Density of fluid (Kg/m^3)

A: area, unit m^2

ASHREA: American Society of Heating Refrigerating and Air Conditioning Engineer

CFD: Computational Fluid Dynamics

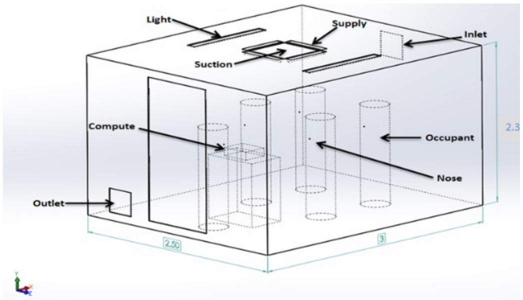
ELR: Experimental lecture Room

U, V, W: Velocity component in x, y, and z- directions m/s

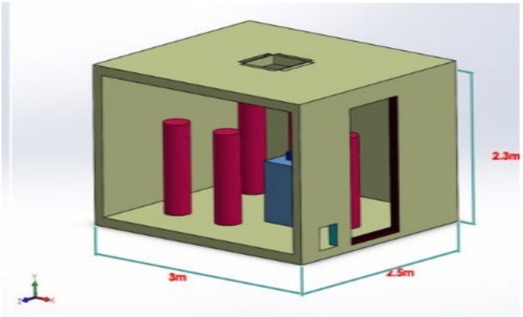
ACH: Air change per hour

TC: Thermal comfort

The governing equations are thereby time-averaged navier-stokes and continuity equations given as mass conservation by was Conducted using ANSIS FLUENT software. And experimentally by designing a full- scale lecture room dimension (L= 3m, W=2.5m, H=2.3m) The inlet cold air is supplied by with 4-way cassette ceiling type air-conditioner measuring the temperature and humidity at fix positions. Which connecting to control panel and from the control panel to the computer testing it under Iraqi weather conditions.



(a)



(b)

Figure (1) (a and b) Basic Dimensions of the Lecture Room Model Geometry

Numerical 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 presented in Figure (1). Which shows that the basic dimensions of the model were geometry. As (John, and Anderson 2009), for the computational model 12. The geometry is generated by using SolidWorks to build the lecture room components (Students, Lights, President, and Table). Two cases of lecture room were studied and modeled were given in (Table (1) a and b). The ANSYS software and SOLIDWORK will solve the governing integral equations for the conservation of mass and momentum, (Documentation, 2005). The occupants were modeled as the cylinder objects with cup; each human body was simulated by 0.3 x 1.2 m^2. All occupants and fluorescent lamps (2 x 20W) were located in the room. Computer (1*500W) was modeled with a metabolic rate of $60W/m^2$ (ASHRAE, 2004). Conservation Equations for Scalar property

$$\frac{\partial(\rho\Phi)}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j\Phi\right)=\frac{\partial j_{\Phi j}}{\partial x_j}+S_{\Phi} \qquad (1)$$

Governing Equations The three physical laws are

Continuity Equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad (2)$$

Momentum Equation of Newton's Second Law

In x-direction

$$\frac{\partial(\rho u)}{\partial t}+\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho vu)}{\partial y}+\frac{\partial(\rho wu)}{\partial z}=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right] \qquad (3)$$

In y-direction

$$\frac{\partial(\rho v)}{\partial t}+\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho wv)}{\partial z}=-\frac{\partial p}{\partial y}+\mu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right]+Sbj \qquad (4)$$

In z-direction

$$\frac{\partial(\rho w)}{\partial t}+\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}= -\frac{\partial p}{\partial z}+\mu\left[\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right] \quad (5)$$

Conservation of Energy

$$\frac{\partial}{\partial_x}(u\mathrm{T})+\rho\frac{\partial}{\partial_y}(wT)=\frac{\partial}{\partial_x}(\Gamma_{eff.h}\frac{\partial T}{\partial x})+\frac{\partial}{\partial_y}(\Gamma_{eff.h}\frac{\partial T}{\partial y})+\frac{\partial}{\partial z}(\Gamma_{eff.h}\frac{\partial T}{\partial z})+S_T \quad (6)$$

Turbulence Modeling

$$\frac{\partial}{\partial x}(\rho uK)+\frac{\partial}{\partial y}(\rho vK)+\frac{\partial}{\partial z}(\rho wK)=\frac{\partial}{\partial x}(\frac{\mu t}{\sigma k}\frac{\partial k}{\partial x})+\frac{\partial}{\partial y}(\frac{\mu t}{\sigma k}\frac{\partial k}{\partial y})+\frac{\partial}{\partial z}(\frac{\mu t}{\sigma k}\frac{\partial k}{\partial z})+G_k-\rho\varepsilon \quad (7)$$

Model constants: the constants of standard k-Ɛ model were given in Table (2).

Convergence Criteria

Residuals of scaling for Convergence Criterion of continuity, momentum, Energy were presented in Table (3).

Boundary Conditions

Boundary conditions specify the flow and thermal variables on the boundaries of the physical model were presented in Table (4).

Mesh Resolution

To achieve the results of high-resolution solution must increase the number of cells and this requires a computer with a very high processor or dual processor for the Windows system and the cells may reach 4 million cells and 8 Gigabyte RAM. For the present cases, a number of meshes of (4.3 Million Cells) were taken as shown in Table (5).

Experimental Work

Thermal comfort inside the space was based on several measurements, including temperature, air velocity and relative humidity, which must comply with the international standard of distribute and rates of variation. the parameters that are measured in this research were.

1. Temperature and relative humidity which were measured by advance sensors.
2. Air velocity which was measured by Modified device.

System Description

A full-scale chamber was established for the purpose of conducting practical experiments and taking the results by providing standard conditions lecture room. The Experimental lecture Room (ELR) was designed and constructed in Mechanical Engineering Department at University of Technology. The tested lecture room of 2.5W*3L*2.3H m^3. Dimensions have a single insulated door in one wall. The (ELR) was constructed of polyurethane foam sandwich panel (PU Sandwich Panel) which was used for air-conditioned rooms and controlled-environment rooms with either positive or negative temperatures. Velocity and CO_2 concentration can reach stability taken over 3hours period the instrument readings of CO_2 concentration for these strategies, within did reach constant readings. The volume due to supply air (7.8 ACH), will neglect the infiltration of air and consider the exchange of air as nothing and take into account the effect of the load force is due to the natural convection currents due to the buoyancy force was formed after calculating the (non- parameter Richardson number (Koufi, *et al*., 2017), the laboratory chamber characteristics showed in the Figure (2).

Table (1) (a) First Case Study Parameter

Ventilation Flow Rate m^3/h	Air Cooled Flow Rate m^3/h	ACH	Ts ℃	T_{vent} ℃	Velocity (s) m/s	Velocity (vent) m/s
135	1512	7.8	20	37	4.2	0.4

Table (1) (b) Second Case Study Parameter

Ventilation Flow Rate m^3/h	Air Cooled Flow Rate m^3/h	ACH	Ts ℃	T_{vent} ℃	Velocity (s) m/s	Velocity (vent) m/s
171	1512	9.9	19	37	4.2	0.52

Table (2) Model Constants of the Standard k-ε Model ANSYS Workbench, (2005)

σk	σε	C1ε	C2ε	Cμ
1.0	1.3	1.44	1.92	0.09

Table (3) Scaling Residual

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

Table (4) Boundary Conditions of Lecture Room Model

Part	Type	Boundary Conditions
Wall	Wall	T_o = 54 °C-Thermal Convection (h) = 8.2 ($W/m^2.K$)
Door	Wall	T_o= 54 °C-Thermal Convection (h) = 8.2 ($W/m^2.K$)
Ceiling Air Condition (4way)	Velocity Inlet-out Flow	Velocity Inlet = 4.2 m/s Out Flow/ Gage Pressure = 0 pa T = -20 °C
Outlet Sections	Flow Out	Gage Pressure = 0 pa
Input Section(ventilation)	Velocity Inlet	-Velocity = (0.4-0.52) m/s -T=37 °C - Φ=41
Human Body	Wall	-T=34 °C -Heat Flux(q) = 60 ($W/m^2.K$)
External environment conditions	Wall	-T=45 °C - Φ=41

Table (5) Different Element Size with Average of Air Parameters

Type of Mesh	Number of Elements	Air velocity m/s	Air temperature ℃
Mesh 1	3952551	0.33	26.1
Mesh 2	4642898	0.34	26.6
Percentage Difference		1.3%	1.8%

(a)

(b)

Figure (2) (a,b) Full-scale Lecture Room Geometry

Occupant's Model

The occupants were modeled as the cylinder objects with cup, each human body was simulated by (0.3D x 1.2 H) m^2 Human sitting and a thermal load of 100 watts in a stable state All occupants are modeled with a metabolic rate of 60W/m^2 (Turner, *et al.*2011).

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 as shown in Figure (3). Where the convection source of students and lectures is light and carry the computer. At the same time, temperatures. Humidity and velocity the air after one to three hours to obtain the status of the study through the sensors installed at the specified points and readings are repeated after five minutes to ensure the stability of the criteria after the signal is transferred from the sensors to the information collector. Then received by the computer for the purpose of Presentation and data presentation and save, and calculating the change in temperature relative humidity and velocity, and then comparing the results (ANSI / ASHRAE Standard 55-2010, 2011). The sensors were calibrated in center organization for standardizations quality control.

Figure (3) Inverter in Ceiling Cassette

Measurement Equipment

All Points were controlled by tow Arduino microcontroller as showed in Figure (4) and Table (5). All measured data including temperature, humidity and air velocity were transferred from sensors to a computer for displaying, and analysis. The sensors are shown in Figures 5, 6 and 7. The functions and models of sensors were recorded in Table (5) (Mahyuddin and Awbi 2010).

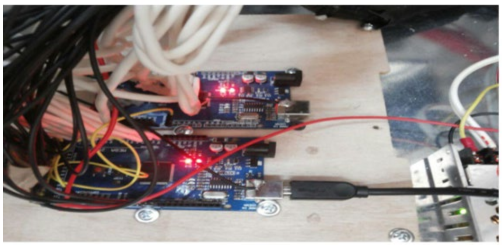


Figure (4) Component Panel

Figure (5) DHT22 (T-H) Sensor

Figure (6) Hot Wire Anemometer

Figure (7) Infrared Thermometer GM700

Increasing Precision by Multiple Measurements.

Increasing the measurement accuracy in practice was done by repeating the measurement successively for the purpose of increasing confidence in values through Quantities average (or mean) each 5minute measurement. All these equipment's were calibrated before used to ensure reliability and accuracy. The calibrate equipment's give to correct measures of physical parameters of environment. Finally, the characteristics of each system device are shown in Table (6).

Air Ventilation System

The air conditioning system used in this research for mechanical ventilation was a full fresh 50EES—ZP Single-Package air conditioner. Packaged unit shown in Figure (8). It was a domestic and commercial refrigeration unit designed on the basis of an air outlet and an air inlet returning to the unit.

Figure (8) Packaged Unit50EES—ZP

The Specific Location of Multiple Type Sensors

The positions of all the sensors were shown in Figure (9).

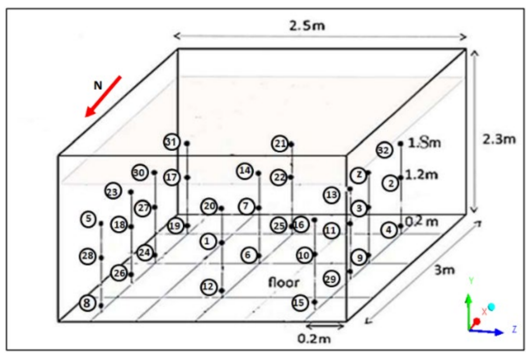


Figure (9) Specific Location of Sensors

Table (6) Characteristics of Measurement System

Sensor -model	Function	Range	Resolution	Accuracy
DHT22	Temperature	-40 ~80Celsius	0.1OC	<+-0.5Celsius
DHT22	Humidity	0-100%RH	0.1%RH	+-2%RH
Yk-2005AH	Air velocity	0.2 to 20 m/s	0.01 m/s	±0.025 m/s
Yk-2005AH	Temperature	-18 to 93°C	0.1°C	±0.3°C
GM700	Temperature	-50~700 Celsius	0.1 °C	±1.5 Celsius

Table (7) Numerical and Experimental Air Temperature in the Test Room

Numerical						Experimental					
Ventilation Methods	AHC	Location	Average Air Temperature 0.2	1.2	1.8	Ventilation Methods	AHC	Location	Average Air Temperature 0.2	1.2	1.8
Vent 135 m^3/h RC 1512 m^3/h	7.8	First Contour	24.3	24.6	25	Vent 135 m^3/h RC 1512 m^3/h	7.8	First Contour	25.3	24.2	25.7
		Second Contour	24.2	25.0	28			Second Contour	25.1	23.3	25.3
		Third Contour	24.5	24.6	25.1			Third Contour	25.4	24.2	25.7
Percentage Error			0.4%								

Air Changes Per Hour

Air change rate was a measure of the air volume added to or removed from a space in one hour, divided by the volume of the space. ASHRAE, 2004.

$$ACH = \frac{Q}{V} \tag{8}$$

Results and Discussion

The case study was a practical and numerical comparison of a classroom model that accommodates five people equipped with a four-way roof air conditioner, the provision of thermal comfort and compatibility of the results. The first case study was carried out by calculating the data within the space through the operation of the roof air conditioning system at a flow rate of 1512 m^3/h, Ts (20°C), and angle (45°). The ventilation flow rate was 135 m^3/h and the air change rate was 7.8 per hours. As shown in the Tables (7 and 8). The second case study was carried out by calculating the data within the space through the operation of the roof air conditioning system at a flow rate of 1512 m^3 / h, Ts (19°C), and angle (45°). The ventilation flow rate is 171m^3/h and the air change rate is 9.9 per hours. As shown in the Tables (9 and10).

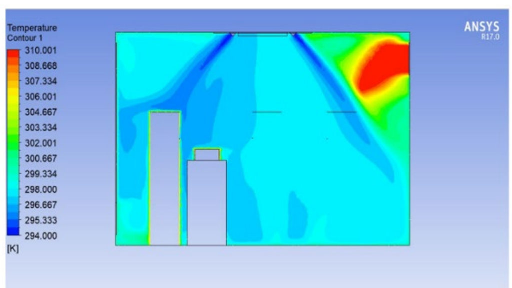


Figure (10) Temperature Contour

Figure (10) showed the temperature contour at the section of (x-y) z 1.25m in the section that represents the distribution of temperatures in the middle part of the space, the temperature decreases near the processing air currents up to 20 °C, while it increases near people due to thermal loads resulting from metabolic processes to 34 °C as well as the load resulting from the computer which reaches 37 °C. With a temperature of 24 to 25 degrees °C.

Distribution at (x and y) plan of z 1.25m, Figure (11) shows the contour velocity. At the section of (x-y) z (1m) note that in the velocity section near the center of the room is velocity ranges within the recommended limits with high velocity in the area near the supply.

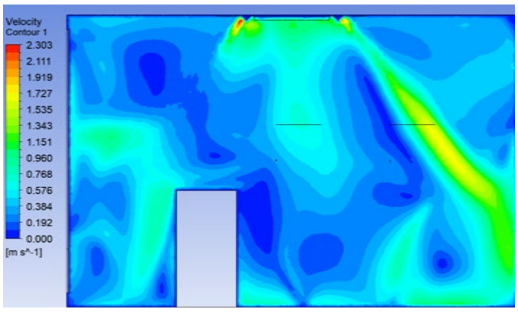


Figure (11) Velocity Contour Distribution at (x,y) Plan of z1m

Figure (12) illustrates the temperature ensures good distribution of temperatures and overcomes the thermal loads generated by occupants in addition to personal computer and lights where the prevailing temperature is 25 to 26 degrees and is within the range Previously Recommended thermal comfort.

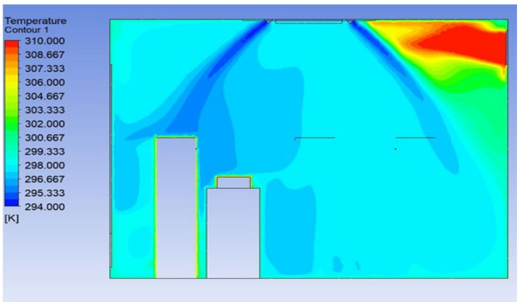


Figure (12) Temperature Contour Distribution at (x and y) Plan of z 1.25 m

Figure (13) similarity, 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 environment for occupants. Ceiling due to processing and suction.

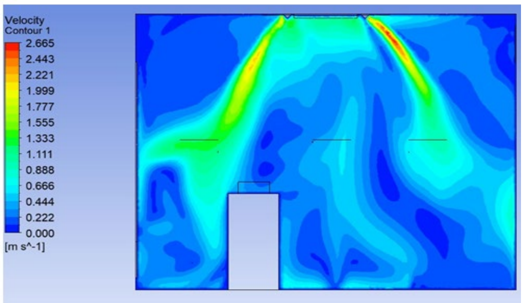


Figure (13) Velocity Contour Distribution at (x,y) Plan of z 1m

Comparison of Simulation and Experimental

Figure (14) at plan (x and y) and (z = 1.25 m) The behavior is similar to an increase in temperatures near the ceiling due to increased loads and the rise of hot air upward, and the temperature ranges are within the limits of thermal comfort.

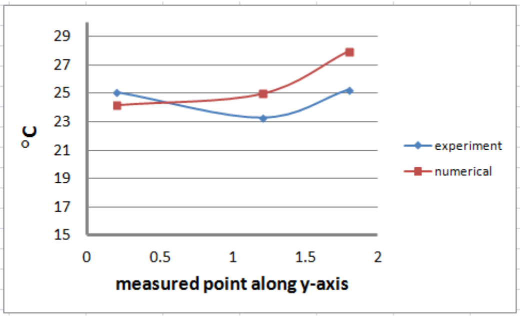


Figure (14) Comparison between Numerical and Experimental Results of Temperature Distance at Plan (x and y) and z= 1.25 m

Figure (15) at plan (x,y) and (z = 1 m) 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.

Table (8) Numerical and Experimental air Velocity in the Test Room

Numerical						Experimental					
Ventilation Methods	AHC	Location	Average Air Temperature			Ventilation Methods	AHC	Location	Average Air Temperature		
			0.2	1.2	1.8				0.2	1.2	1.8
Vent 135 m^3/h RC 1512 m^3/h	7.8	First Contour	0.36	0.42	0.48	Vent 135 m^3/h RC 1512 m^3/h	7.8	First Contour	0.34	0.41	0.42
		Second Contour	0.40	0.20	0.20			Second Contour	0.40	0.15	0.17
		Third Contour	0.53	0.60	0.32			Third Contour	0.48	0.46	0.31
Percentage Error			11%								

Table (9) Numerical and Experimental Air Temperature in the Test Room

Numerical						Experimental					
Ventilation Methods	AHC	Location	Average Air Temperature			Ventilation Methods	AHC	Location	Average Air Temperature		
			0.2	1.2	1.8				0.2	1.2	1.8
Vent 171 m^3/h RC 1512 m^3/h	9.9	First Contour	24.5	24.8	25.2	Vent 135 m^3/h RC 1512 m^3/h	9.9	First Contour	25.8	26.1	26.6
		Second Contour	24.5	25	25.8			Second Contour	25.3	25.5	26.4
		Third Contour	24.4	24.8	25.2			Third Contour	25.7	26	26.6
Percentage Error			4%								

Table (10) Numerical and Experimental Air Velocity in the Test Room

Numerical						Experimental					
Ventilation Methods	AHC	Location	Average Air Temperature			Ventilation Methods	AHC	Location	Average Air Temperature		
			0.2	1.2	1.8				0.2	1.2	1.8
Vent 171 m^3/h RC 1512 m^3/h	9.9	First Contour	0.37	0.47	0.31	Vent 135 m^3/h RC 1512 m^3/h	9.9	First Contour	0.32	0.40	0.25
		Second Contour	0.30	0.23	0.23			Second Contour	0.29	0.20	0.20
		Third Contour	0.34	0.39	0.33			Third Contour	0.30	0.36	0.32
Percentage Error			12%								

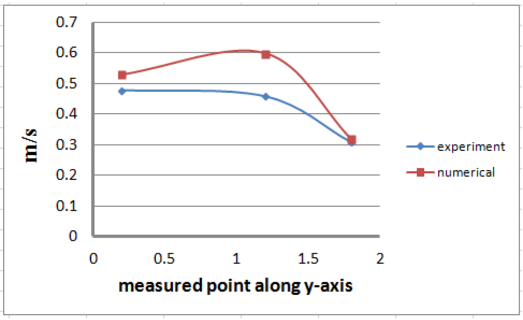


Figure (15) Comparison between Numerical and Experimental Results of Air Velocity Distance at Plan (x,y) and z= 1m

Figure (16) at plan (x,y) and (z= 1.25 m) The behavior is similar to an increase in temperatures near the ceiling due to increased loads and the rise of hot air upward, and the temperature ranges are within the limits of thermal comfort.

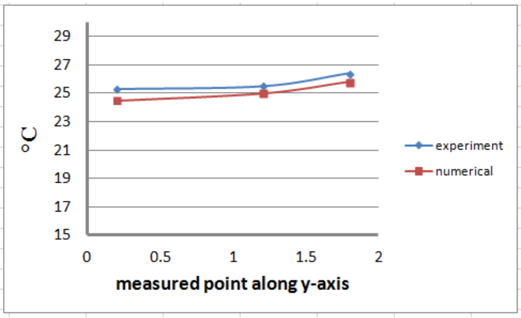


Figure (16) Comparison between Numerical and Experimental Results of Temperature Distance at Plan (x,y) and z= 1.25 m

Figure (17) at plan (x,y) and (z= 1 m). 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.

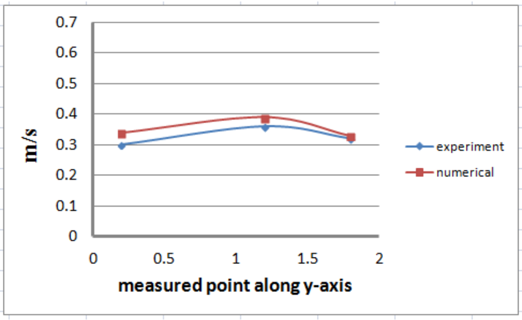


Figure (17) Comparison between Numerical and Experimental Results of Air Velocity Distance at Plan (x,y) and z = 1m

Numerical and Experimental Percentage Error

The comparison between numerical and experimental results for temperature distribution with three cases studied for Ts = (19-20) ℃ and flow rate equal to 1512 m^3/h and ventilation flow rate (135-171) m^3/h Tvent = (37) ℃ respectively. A reasonable good agreement was achieved for temperature, and velocity profile between numerical results and experimental results. The simulated data generally followed the experimental trends. The overall distribution between the numerical results and the experimental once is about (first case 4.3%), (second case 7.5%).

Conclusions

The four-way ceiling air conditioner with mechanical ventilation system is a very sophisticated system widely used in Japanese, Korean and Chinese universities. Main control to maintaining the health of lecturers and students by reducing the variances of temperature and air velocity the results were practically analyzed in the test room and compared with numerical results obtained from the application of dynamic computational fluids and the results were satisfactory in terms of their convergence. The important points of this study in the experimental and numerical work were

1. The four-way ceiling-type air conditioner turned out to be more suitable to be used in the Lecture room.

2. Ceiling type air conditioner gives little variation in velocity and temperature inside space.

3. Ventilation achieves an improvement

in indoor air quality according to the recommended Standards, with rates of 7.5 L/ s. Person.

4. Air temperatures are close to the contrast due to the mixing quality of the ceiling device, while the highest degree is near the Ceiling due to thermal loads, high hot air and an average temperature within the thermal comfort ranges.

5. The air velocity distribution is regular with the low velocity near the ground due to its distance from the current source, and is within the recommended thermal comfort ranges.

6. The results were practically analyzed in the test room and compared with the numerical results obtained from the application of computational fluids dynamic, and the results were satisfactory in terms of their convergence.

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