Proper Supply Air Distribution Inside A Ship Cabin For A Best Ventilation Efficiency In Basrah City

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Abstract

This paper focuses on the thermal comfort inside a selected Iraqi ship's (GOUGE) cabin space whom working in the Umqaser port. Because of limited space and low ceiling height of these cabins, a low mixing ratio between the supplied conditioned air and the indoor air is expected. This lack of homogeneity of the air mixture causes many health problems for the crew and reduces their effectiveness. A computational fluid dynamics (CFD) has been used within the program (ANSYS 15) for the purpose of analysis and solving the equations of mass, momentum and energy and to compare the numerical results obtained from the program with the experimental results for distribution the temperature, velocity and relative humidity of the air conditioned inside Iraqi ship's cabin, where the results showed good matching.

A full scale test cabin has been designing and building as a simulator cabin with the same specifications of the real cabin to validate the measurements which carried out on this test cabin with the experimental and numerical results on the real cabin. In this study different supply diffusers are tested, such as square and circular ceiling diffusers, side wall register and slot diffuser. It was found that the ceiling square diffuser is the proper diffuser for the cabin. This study focuses on the air changing per hour, also on the air diffusion inside the cabin, taking into consideration the climatic basrah where the temperature and high humidity.

Key words: Air Diffusion performance Index (ADPI), Computational Fluid Dynamic (CFD). Air supply location, Space Diffusion Effective Factor (SDEF), Air Changing per Hour (ACH).

افضل توزيع للهواء داخل قمرة باخره عراقية بما يحقق اعلى كفاءة تهويه في الظروف المناخية لمدينة البصرة .

الخلاصه

تقود هذه الدراسه الى تحليل ثلاثي الابعاد للراحه الحراريه داخل قمرة الباخره العراقيه دهوك نوع (حفار) العامله في ميناء ام قصر جنوب العراق.نتيجة للارتفاعات القليله لغرف الباخره ذات الحيز المحدود فان الهواء المجهز الى داخل القمره لا يتم خلطه بشكل جيد مع هواء القمره وخروج قسم من الهواء غير المخلوط خارج القمره دون الاستفاده منه، وهذا النقص في التجانس للهواء المخلوط يسبب عدة مشاكل صحيه مما يؤثر على نشاط الطاقم وقلة كفائته .
اجريت المحاكاة باستخدام برنامج ديناميك السوائل الحسابيه (CFD) ضمن البرنامج (ANSYS R15) لغرض التحليل وحل معادلات الكتله والزخم والطاقه ومقارنة النتائج المستحصله من البرنامج مع النتائج العمليه لتوزيع درجات الحراره والسرعه والرطوبه النسبيه للهواء داخل قمرة الباخره،وتم تصميم وبناء نموذج تشبيهي كامل يحمل نفس مواصفات القمره البحريه الحقيقيه الغرض منه مطابقة النتائج العمليه والنظريه التي تمت قرائتها على هذا النموذج وتصديقها مع النتائج العمليه والنظريه للقمره البحريه الحقيقيه وقد حصل توافق جيد بين هذه النتائج. من خلال دراسة الانواع التاليه من موزعات الهواء

، سقفي مربع، سقفي دائري، موزع جانبي وموزع شقي جانبي و تبين من خلال نتائج الدراسه ان افضل موزع هواء يلائم قمرة الباخره هو الموزع المربع ومن موقع سقفي. وتتناول الدراسه معدل تغيير الهواء في الساعه (ACH), وعامل انتشار الهواء داخل قمرة الباخره في ضروف البصره المناخيه حيث درجة الحراره والرطوبه العاليه.

كلمات مرشده: عامل الاداء الحراري (ADPI) ديناميكية الموائع الحاسوبيه(CFD)، موقع تجهيز الهواء،انتشار الهواء في قمرة السفينه(SDEF)،معدل تغير الهواء في الساعه(ACH).

1. Introduction

The shipping industry and the transfer of goods and passengers between international ports needs especially interest to maintain the transfer of trade between nations. Indoor air quality on ships not received much attention in spite of exposure the crew for health hazardous symptoms, as a result of combined working and living in limited space, good indoor air quality is important for crew's health and their productivity [3]

There are differences between the environment conditions of the ships with the land where the ships can make passages from zone to another during the day under different environmental conditions, and the crew members or passengers spend on board long time during the trip that causes thermal stresses or unhealthy conditions [1]. Great variations in surrounding air parameters need special requirements for ships to design a marine air- conditioning for continues operation during voyage around the year.

Because of the low internal heights and limited spaces as cabins, kitchens, engine rooms and other spaces comparing with the building, the air supply dose not mixed well with the indoor air and exhaust out as small cold air volumes without benefit of it [11].

Elsafty, A.F, el ,was studied the ventilation efficiency inside ship's cabin using the GAMBIT program for drawing and meshing the model and the fluent program to solve the governing equations, he found the proper location of supply air from a side wall of the ship's cabin working in a red sea. The objective of this work is to study the indoor air quality on Iraqi ship's cabin whom working in um qaser port where high temperature and humidity. For matching the numerical with the experimental results, the computational fluid dynamic (CFD) has been used within the program (ANSYS 15).

2. Tested cases

2.1 Real cabin

An Iraqi ship's cabin of (3.4mx2mx2.1m) for second engineer is shown in figure 1(a) was selected to investigate experimentally the air flow patterns, air temperature and relative humidity distribution. The content of the cabin includes one bed, a small table, a chair, a refrigerator, a small wardrobe and a mirror with small lamp source. The cabin ceiling is equipped with lamps and there are two small windows in the side marine wall. Conditioned air is supplied to the cabin space through circular duct 20 cm diameter with square diffuser fixed at ceiling, 1.4m from the front wall of the ship, see figure 1(a). The return is exhausted through a grille of (33.4cm wide x 15.5 cm height) dimensions, fixed in the lower part of the cabin door, 26cm from the floor.

2.2 Full scale model

The full scale cabin was manufactured in the workshops of the general Iraqi ports company, with the same specifications of the real cabin and equipped with same contents as in the real one. A supply air square diffuser is used in this cabin, which is fixed in the cabin ceiling at distance 1.5m from the front wall, as shown in figure 1(b).

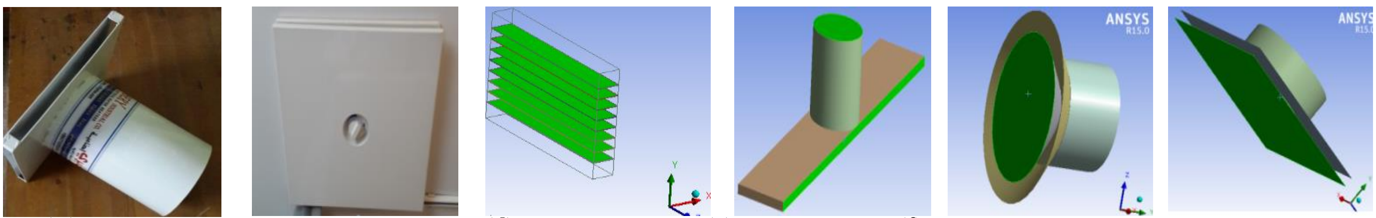
(a) (b)

Figure 1: Real cabin (a), full scale cabin (b)

2.3 Studied supply air diffusers

The following diffusers were studied in this study:

- Square diffuser which locally manufactured in the work shop.
- Real square diffuser of the ship's cabin.
- Wall register diffuser, slot diffuser, circular diffuser and square diffuser which were drawn by (design modeler) using (ANSYS R15) program as shown in figure 2.



(C) (d) (e) (f)

Figure 2: Types of diffusers, (a) Manufactured diffuser, (b) Real diffuser, (c) Register wall diffuser, (d) Slot diffuser (CFD), (d) Circular diffuser (CFD), (f) Square diffuser (CFD).

3. Field measurements

3.1 Temperature and relative humidity Measurement Tested devices

The reading were taken through the following devices; fluke971 temperature and humidity meter, of temperature range from-20˚Cto 60˚C, accuracy $\pm$0.5˚C on 0 to 45˚C, and relative humidity range from 5 to 95 % figure 3(a)[4]. An Infrared thermometer GM 300, of temperature range -50 to 380ºC, accuracy ± 1.5ºC, is also used as shown in figure 3 (b) [5].

Fig.3: Temperature and relative humidity measurement devices (a) and (b)

(a) (b)

Fig.4: velocity measurement devices (a) and (b)

3. 2 Velocity measurement devices

An EXTECH Data logging /printing Anemometer 451181, shown in figure 4(a), was used to measure the air velocity in same locations of domain points. Measuring Range is from 0.4 to 25 m/s, resolution 0.1m/s, accuracy $\pm$3. Fluke 922 was used to measure the supply air velocity inside the duct, of range 1m/s to 80 m/s, accuracy is 2.5 m/s. figure 4(b) [6].

4. Theoretical analysis

In this study, thirty points at different heights are selected to cover the whole cabin. These points are distributed in four levels (20cm, 100cm, 180cm, and 195cm) above the ground as shown in figure 5. The air supplied into the cabin for the tested cases by register grill diffuser, square diffuser, circular diffuser and slot diffuser which drawn by design modular of ANSYS 15 program, as shown in figure 2.

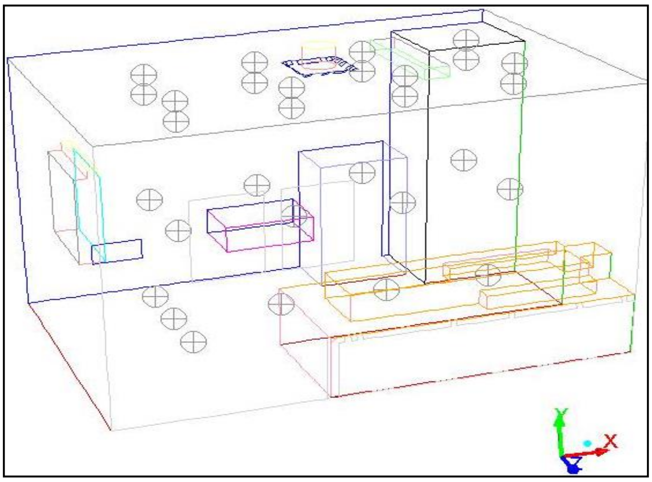


Figure 5: Measuring domain points (0.2m, 1m, 1.8m, 1.95m)

4.1 CFD Model

In this study (ANSYS R15) program is used to solve the governing equations [7, 8], these equations are:

Continuity equation:

$$\frac{\partial U_i}{\partial x_i} = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial xi}(\rho U_i U_j) - \frac{\partial}{\partial xi}\left\{\mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right\} + \frac{\partial p}{\partial x_j} + (\rho - \rho_0) = 0 \tag{2}$$

Eenegy equation:

$$\frac{\partial}{\partial x_i}(\rho U_i T) - \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{p_{rt}}\frac{\partial T}{\partial x_i}\right) \tag{3}$$

The transportation equations for k and ε are given by

$$\frac{\partial}{\partial x_i}(\rho U_i k) - \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{p_{rk}}\frac{\partial k}{\partial x_i}\right) - G - B + \rho\varepsilon = 0 \tag{4}$$

$$\frac{\partial}{x_i}(\rho U_i \varepsilon) - \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{p_{r\varepsilon}}\frac{\partial \varepsilon}{\partial x_i}\right) - \frac{\varepsilon}{k}(C_{1\varepsilon}G_K + C_{3\varepsilon}B) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{K} = 0 \tag{5}$$

The turbulent viscosity is given by:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \qquad \left(\frac{m^2}{s}\right) \tag{6}$$

The Reynolds stresses:

$$\overline{-u_i}u_j = v_t\left\{\frac{\partial u_i}{\partial u_j} + \frac{\partial U_j}{\partial U_i}\right\} - \frac{2}{3\delta_{ij}k} \qquad \left(\frac{m^2}{s^2}\right) \tag{7}$$

The production of turbulent kinetic energy due to mean velocity gradients (J/m^3), and due to buoyancy can be given as:

$$G = \mu_t \frac{\partial U_i}{\partial x_j}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) \tag{8}$$

$$B = -\beta g_i \frac{\mu}{\sigma T}\frac{\partial T}{\partial x_i} \tag{9}$$

The volumetric expansion coefficient of air is

$$\beta = -\left(\frac{\partial p}{\partial T}\right) \tag{10}$$

Where ρ is density of air (kg/m^3),U_iU_J are mean velocities along coordinate axes (m/s) , p is local total pressure (pa), g is gravitational acceleration (m/s^2), ρ_0 is reference density of air (kg/m^3), δ_{ij} is kronecker delta

The standard values;

$C_{1\varepsilon}$=1.44, $C_{2\varepsilon}$ =1.92, $C_{3\varepsilon}$ =0.8, C_μ =0.09,σ_k = 1.0 and σ_k = 1.3 [9].

4.2 Mesh quality

The quality of the mesh is a significant role in the accuracy and stability of the numerical computation. Checking the quality of the grid depending on the cell type in the mesh (Tetrahedral, hexahedral, polyhedral, etc.)

ANSYS R15 was used and the geometry was drawn by design modeler (DM) in the same program then imported to fluent for meshing, set up, run and (CFD POST) to show the results. Several different types of meshes were applied to obtain the appropriate mesh to produces the desirable accuracy for the problem to be solved, a suitable method used for meshing was (cut cell) [10]. The structure grids type were successful for meshing the geometry as shown in figure 6, the number of cells for all cases for converged solution were between (200,000-380,000) cells.

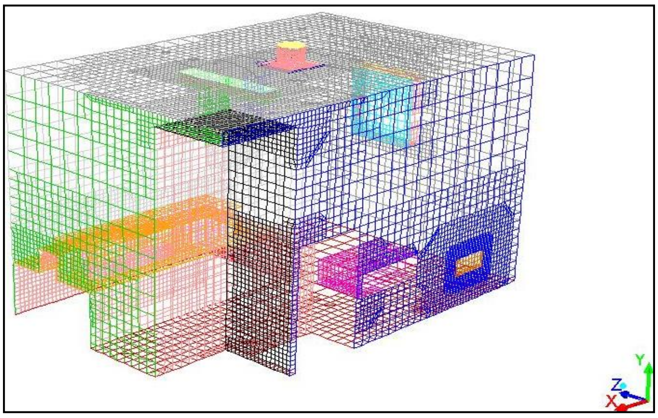


Figure 6: Model after meshing generation (ANSYS15)

5. Boundary conditions

The boundary conditions included the inlet and outlet of the air temperature, velocity and relative humidity, the velocities at the walls are considered to be zero because no-slip conditions are assumed, the outlet pressure is assumed to be ambient pressure, the temperature of ceiling, floor, walls, furniture and the sources of lighting and sailors are also taken in consideration as a boundary conditions.

6. Air changing per hour (ACH)

The demand of renewing the air of the space is based on space construction, heat load, and space environment characteristics such as shade, direction of exposure, convection, the radiant and the concentration of the contaminants. The approach of (ACH) [14] is defined as:

CMM = (Air velocity x Ventilation area) (11)

CMM = Supply air flow rate (m^3/min)

Air changing per hour, ACH = (CMM)/Room volume (12)

7. Space diffusion effective factor

Due to small volume and low height of the ship's cabin, the supply air dose not mixed well with indoor air because the time is insufficient for mixing air process inside the cabin so this process leads to have some unmixed small cold air volumes outlet exhausted to the ambient. The space diffusion effective factor (SDEF) used to distinguish between the effective mixing air and ineffective mixing air, is defined as:

$$\text{SDEF} = \frac{T_{ex} - T_s}{T_r - T_s} \tag{13}$$

T_{ex} = exhaust air temperature ˚C

T_s = supply air temperature ˚C

T_r = room air temperature ˚C

SDEF ≥ 1 effective temperature, SDEF ≤ 1 not effective temperature [11]

8. Degree of thermal comfort

The effective draft temperature θ (the difference in temperature between any point in the occupant zone and the control condition) can be used to indicate the degree of thermal comfort in an air- conditioned space [12].

$$\theta = (T_x - T_c) - 8\,(V_x - 0.15) \tag{14}$$

Where

Tx- is the space temperature at a specific location (ºC)

Vx- is the space air velocity at a specific location (m/s)

Tc- is the mean air temperature of the space or set point (ºC).

0.15 m/s is the desirable mean air velocity of the space.

During a cooling mode in commercial and public buildings if the space conditions are (t = 18ºC–26.5ºC, Vr = 0.5 m/s, and RH = 30–70%) [1], and the prefer velocity (vr) is from 0.15 m/s to 0.25m/s [6].

The air diffusion performance index is given by approach:

$$ADPI = \frac{N_\theta}{N}\ X\ 100 \tag{15}$$

Where

N_θ = The number of measuring points in occupied zone in which (-1.7ºC<θ< 1.1ºC).

N = the total number of points measured in occupied zone [11, 13]

9. Experimental and numerical results

Tables 1, 2 and 3 contain the experimental measurements which carried out on the real and full scale cabins and validated with the numerical results which obtained by the CFD program. The vertical and horizontal planes were also defined to verify the velocity, temperature and relative humidity contours as shown in figures 7, 8 and 9. The results for temperature and velocity which obtained experimentally and numerically are shown in figures 10 and 11.

10. Results and discussion

In this study by using the computational fluid dynamic program (CFD), the numerical results of the program have been ratificated with experimental results. Two inlet velocities (2m/s and 2.9m/s) are considered for 9 cases as shown in table 4, the following cases are:

1- In cases (1 and 2) the air is supplied through a wall register diffuser.
2- In case (3 and 4) the air is supplied through a wall register diffuser located over the door.
3- In cases (5 and 6) the air is supplied through a square diffuser located at ceiling.
4- In cases (7 and 8) the air is supplied through a circular diffuser at ceiling.
5- In case (9) the air is supplied by a slot diffuser located on the side wall of the cabin.

Among these nine cases, case (5) is credited case due to the best results for the values of air diffusion performance index (ADPI), the air changing per hour(ACH) and the space diffusion factor (SDEF) , at the inlet air velocity (2m/s), tables (1and 3).

In this case have been survey 22 points inside the cabin at the levels (y= 0.2m, y = 1m and y = 1.8m) table (1). It was observed that the values of temperature, velocity and relative humidity at the level (0.2m), have good distribution, while at the level (1m), there were some unwanted draft temperatures (θ), such as points (S10 and S12), due to the effective of refrigerator flux, and low velocity because of its collide with the locker, while at level (1.8m) the distribution is good, and on the level (1.95m) there are unwanted draft temperature as points S23, S25 and S29, see table (2). Because of the limited volume there is a problem related with the space diffusion effective factor (SDEF) which indicates the uncontrolled small cold air volumes exhausted out the cabin without the benefit of it due to insufficient time for mixing with the indoor air, as shown in points (S2, S3, S6, S8, S10, S11 and S20), and it is concluded that the (SDEF) is proportional with the size conditioned.

Figures (7, 8 and 9) show the contours of distribution of the air temperature, velocity and relative humidity respectively through three horizontal planes parallel to the front side and three vertical planes parallel to the ceiling, .

Figures (10 and 11) show the validation of temperature and velocity between numerical and experimental results, which reveal with an acceptable agreement. The velocities at points S26, S28 and S30 in table 2 cannot be measured experimentally because of the limited range of the velocity device.

Table 1: Experimental and numerical results for case 5 at y = 0.2m**,** 1m and 1.8m

Sensors	Location(m)			Temperature°C		Velocity m/s	RH% EXP	RH% CFD	θ˚C	SDEF
	x	y	z	EXP	CFD	CFD				
S1	0.7	0.2	0.6	18.4	18.4	0.23	46.6	38	-0.98	1.07
S2	0.7	0.2	1.4	18.5	18.8	0.13	46.2	38	0.36	0.96
S3	0.7	0.2	1.0	18.5	18.8	0.12	46.2	38	0.44	0.96
S4	1.4	0.2	1.0	18.3	18.6	0.16	46.2	38	-0.08	1.00
S5	2.1	0.2	1.0	18.4	18.5	0.14	46.4	37	-0.02	1.00
S6	2.8	0.2	1.0	18.3	18.8	0.05	45.3	37	1.00	0.96
S7	0.7	1.0	0.6	18.1	18.5	0.12	45.0	38	0.14	1.02
S8	1.4	1.0	0.6	18.7	18.9	0.05	45.5	36	1.10	0.93
S9	2.1	1.0	0.6	18.7	18.7	0.11	45.7	37	0.42	1.00
S10	2.8	1.0	0.6	18.7	18.9	0.04	45.8	36	1.18	0.93
S11	0.7	1.0	1.25	18.8	18.8	0.04	45.8	37	1.08	0.96
S12	1.4	1.0	1.25	18.8	20.0	0.14	44.6	38	1.48	1.00
S13	2.1	1.0	1.25	19.6	18.7	0.16	44.2	36	0.02	1.00
S14	2.8	1.0	1.25	18.2	18.7	0.04	44.2	37	0.98	1.00
S15	0.7	1.8	0.60	19.0	18.6	0.08	43.5	37	0.56	1.00
S16	1.4	1.8	0.60	18.9	18.5	0.13	44.4	37	0.06	1.02
S17	2.1	1.8	0.60	18.8	18.4	0.13	44.3	38	-0.04	1.07
S18	2.8	1.8	0.60	18.9	18.1	0.11	44.2	37	-0.18	1.14
S19	0.7	1.8	1.25	18.6	18.8	0.16	42.5	38	-0.60	0.96
S20	1.4	1.8	1.25	18.6	19.2	0.17	42.0	38	0.44	0.80
S21	2.1	1.8	1.25	19.2	18.7	0.06	44.3	38	0.82	1.00
S22	2.8	1.8	1.25	18.8	18.2	0.15	43.3	38	-0.40	1.10
average				18.	18.7	0.11	44.0	37.5	0.35	1.00

Table 1: Experimental and numerical results for case 5 at y = 0.2m**,** 1m and 1.8m

Table 2: Validation of temperature and velocity for case 5 at y = 1.95m

Sensor	location			Temperature˚ C		Velocity m/s		RH%	˚Cθ	SDEF
	x	y	z	EXP	CFD	EXP	CFD			
S23	0.7	1.95	0.60	19.2	18.6	0.55	0.5	40.5	-2.80	1.17
S24	1.4	1.95	0.60	19.0	18.5	0.4	0.35	40.2	-1.70	1.13
S25	2.1	1.95	0.60	18.6	18.4	0.5	0.45	37.6	-2.20	1.70
S26	2.8	1.95	0.60	18.5.	18.1	-	0.11	37.5	-0.26	1.04
S27	0.7	1.95	1.25	18.7	18.9	0.7	0.85	34.4	0.12	0.93
S28	1.4	1.95	1.25	18.7	19.4	-	0.17	37.2	0.64	0.88
S29	2.1	1.95	1.25	18.3	18.7	0.55	0.60	40.0	-3.50	1.04
S30	2.8	1.95	1.25	17.8	18.5	-	0.23	40.0	0.74	1.02
average				18.6	18.6		0.40	38.0	-1.12	1.10

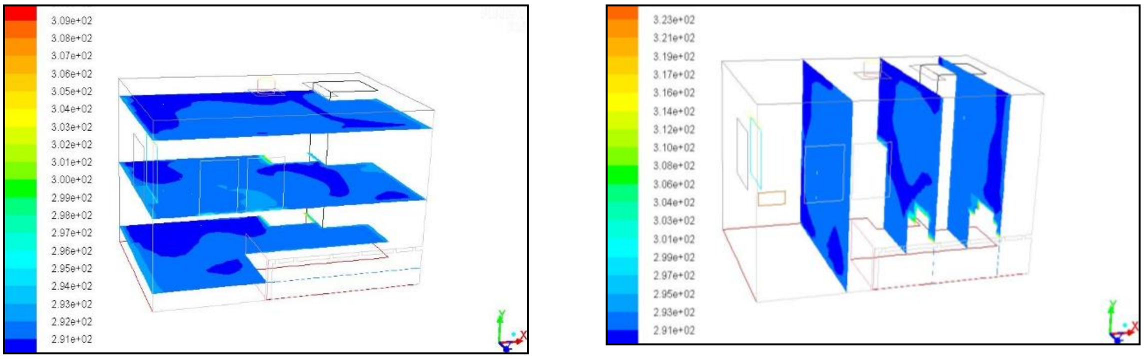
Table 3: Experimental and numerical results for full scale cabin

sensors	Location (m)			Temperature °C	Temperature °C	Velocity m/s	RH%	RH%	˚Cθ
	x	y	z	CFD	EXP	CFD	CFD	EXP	
S1	0.7	0.2	0.6	18.7	18.4	0.25	45	34	-1.1
S2	0.7	0.2	1.4	18.9	18.5	0.14	45	34	-0.02
S3	0.7	0.2	1.0	18.9	18.7	0.27	45	34	-1.06
S4	1.4	0.2	1.0	18.9	18.8	0.26	45	34	-0.98
S5	2.1	0.2	1.0	20.1	18.7	0.10	42	34	1.50
S6	2.8	0.2	1.0	19.8	19.1	0.14	43	34	0.88
S7	0.7	1.0	0.6	19.3	19.2	0.07	43	34	0.94
S8	1.4	1.0	0.6	18.8	18.3	0.11	45	35	0.12
S9	2.1	1.0	0.6	19.9	18.7	0.10	45	34	0.50
S10	2.8	1.0	0.6	19.2	18.6	0.24	44	34	-0.52
S11	0.7	1.0	1.25	19.8	18.7	0.10	42	34	1.20
S12	1.4	1.0	1.25	18.8	19.0	0.09	45	34	0.46
S13	2.1	1.0	1.25	19.7	18.3	0.20	43	34	0.30
S14	2.8	1.0	1.25	19.2	18.3	0.13	44	34	0.22
S15	1.4	0.2	1.0	19.3	19.0	0.08	44	35	0.86
S16	1.4	1.8	0.60	20.1	19.3	0.02	42	34	2.14
S17	2.1	1.8	0.60	18.7	18.9	0.04	47	34	0.58
S18	2.8	1.8	0.60	19.0	18.8	0.11	45	33	0.32
S19	0.7	1.8	1.25	19.3	18.6	0.14	44	34	0.38
S20	1.4	1.8	1.25	18.8	18.5	0.15	45	34	-0.20
S21	2.1	1.8	1.25	18.9	18.2	0.05	45	32	0.70
S22	2.8	1.8	1.25	18.4	19.2	0.17	44	32	-0.76

Table 3: Experimental and numerical results for full scale cabin

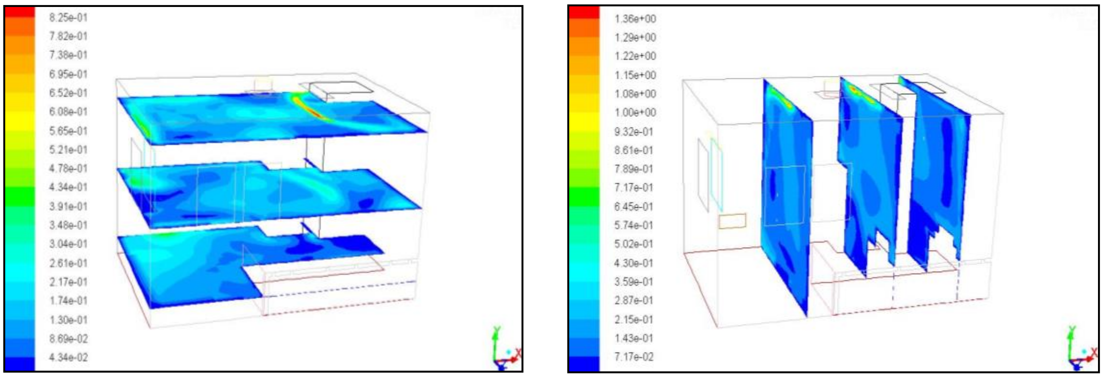
Table 4: ADPI, ACH, RH% and SDEF for studied 9 cases

Sensor	Location			Register grill/side wall		Register grill/over door		Square diffuser/ceiling		Circular diffuser/ceiling		Slot diffuser/ side wall
				Case1	Case2	Case3	Case4	Case5	Case6	Case7	Case8	Case9
				2.9m/s	2m/s	2m/s	2.9m/s	2m/s	2.9m/s	2m/s	2.9m/s	1m./s
	x	Y	z	θ˚C	θ˚C	˚Cθ	˚Cθ	˚Cθ	˚Cθ	˚Cθ	˚Cθ	˚Cθ
S1	0.7	0.2	0.6	0.87	-1.13	0.79	0.08	-0.98	-0.68	-0.32	0.14	1.7
S2	0.7	0.2	1.4	2.28	1.19	-1,03	-1.89	0.36	0.08	-0.8	0.34	3.7
S3	0.7	0.2	1.0	-1.53	-0.09	-0.19	-0.83	0.44	0.48	0.74	0.94	2.0
S4	1.4	0.2	1.0	-2.65	-1.35	-0.19	-0.83	-0.08	0.48	-0.34	0.68	1.54
S5	2.1	0.2	1.0	3.01	-1.95	-0.03	-0.41	-0.02	0.32	0.86	0.92	-0.4
S6	2.8	0.2	1.0	4.23	-2.97	0.01	0.01	1.00	-0.08	0.78	0.56	S-0.88
S7	0.7	1.0	0.6	-2.81	-1.87	-0.03	-0.09	0.14	-0.20	0.04	0.24	-0.64
S8	1.4	1.0	0.6	-1.59	0.15	-0.35	-0.99	1.10	0.53	0.26	1.70	0.5
S9	2.1	1.0	0.6	-1.59	0.25	-0.31	-0.67	0.42	0.06	0.28	1.24	-0.86
S10	2.8	1.0	0.6	2.55	0.43	-4,03	0.19	1.18	-0.16	-0.3	0.64	-0.66
S11	0.7	1.0	1.25	2.57	0.89	-0.49	-0.65	1.08	0.12	0.64	1.50	-0.8
S12	1.4	1.0	1.25	4.07	-0.41	0.75	0.53	1.48	0.38	0.32	1.24	1.24
S13	2.1	1.0	1.25	0.39	-1.75	-0.19	-0.25	0.02	0.28	0.20	1.38	2.76
S14	2.8	1.0	1.25	0.11	-1.52	-4.95	3.73	0.98	0.04	0.78	1.54	2.16
S15	1.4	0.2	1.0	-1.15	-0.79	-0.11	0.29	0.56	0.32	1.22	3.44	0.76
S16	1.4	1.8	0.60	-2.07	0.53	0.93	0.55	0.06	2.26	1.5	1.82	0.06
S17	2.1	1.8	0.60	1.37	0.77	0.89	0.67	-0.04	1.11	0.32	1.50	0.5
S18	2.8	1.8	0.60	1.13	0.558	-0.15	-0.51	-0.18	0.26	0.24	0.52	3.16
S19	0.7	1.8	1.25	1.81	0.814	-0.13	-0.39	-0.60	0.26	-0.20	0.62	0.10
S20	1.4	1.8	1.25	1.73	0.886	-0.49	-0.75	0.44	0.20	0.02	0.06	-0.16
S21	2.1	1.8	1.25	-3.69	-2.71	0.51	0.51	0.82	-0.16	0.88	-0.18	-0.10
S22	2.8	1.8	1.25	1.45	0.79	2.89	-4.69	-0.40	-0.10	-1.81	0.62	0.30
ADPI%				27	68	86	86	90.9	90.9	86	67	63
ACH				23	16	12	18	16	24	16	16	16
RH%				50	44	42	37	37	37	38	30	40
SDEF%								68	40	63	54	59



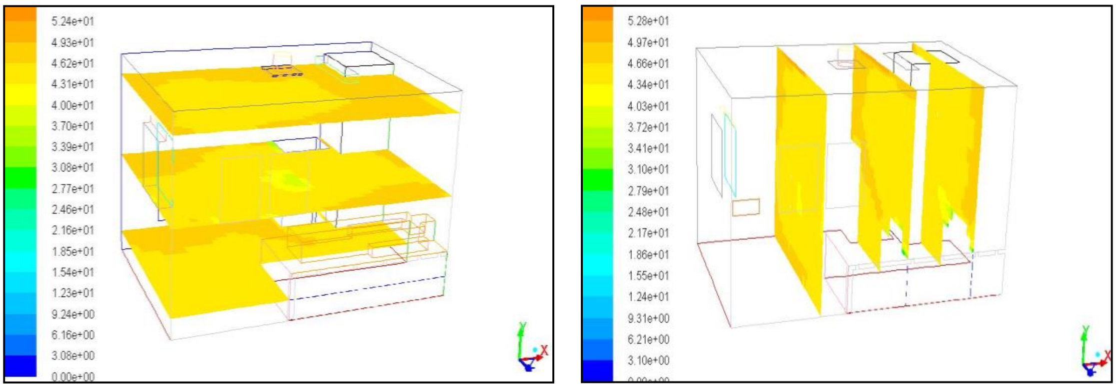
(a)Temperature vertical planes (b) Temperature horizontal planes

Figure 7: Temperature contours in vertical and horizontal planes for ceiling



(a) Velocity vertical planes (b) Velocity horizontal planes

Figure 8: Velocity contours in vertical and horizontal planes for ceiling



(a) Relative humidity vertical planes (b) Relative humidity horizontal plane

Figure 9: Relative humidity contours in vertical and horizontal planes

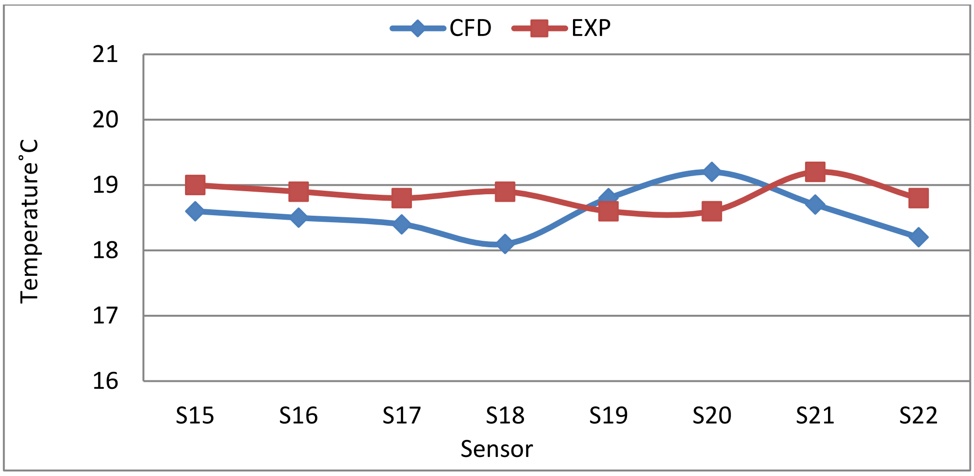


Figure 10: Temperatures validation at y = 1.8 m

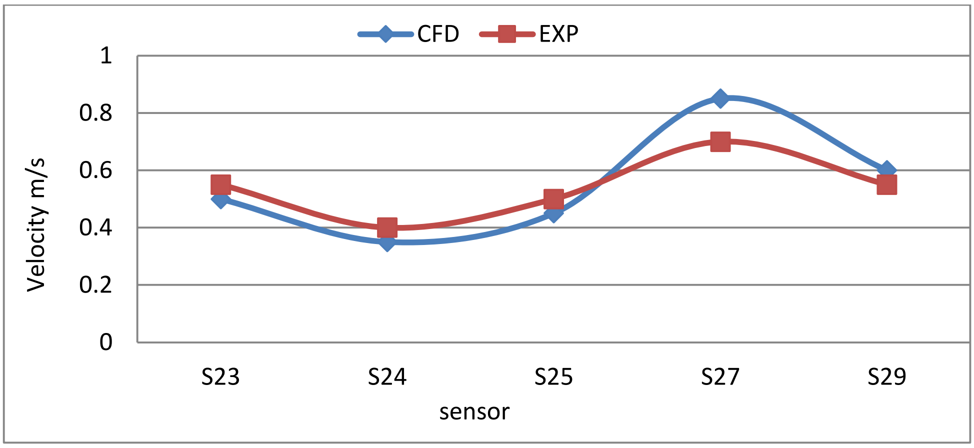


Figure 11: Velocity validation at y= 1.95m

11. Conclusions

From this study, the following points are concluded:

1- The air diffusion performance index (ADPI) inside the cabin was found to be (90.9%), the SDEF = 68, and ACH = 16 in case 5, when the square ceiling diffuser is used at the location (1.5m) from front side of the cabin while the ADPI = 86% when the square diffuser located at 1.4m from front side of the cabin, this leads that the case 5 is accredit case for distribution the temperature, velocity and relative humidity and the best case for space diffusion factor(SDEF).

2- The ADPI was found to be 86% in the full scale model cabin and the ADPI= 86% in case (7) where a circular ceiling diffuser is used for supplied air to the cabin

3- The preferable air velocity which gives the higher ADPI and SDEF values is 2m/s.

4- It was observed that the SDEF become a problem must be take more interest to treat the air diffusion inside the narrow space in future.

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