IJCRT.ORG ISSN : 2320-2882

INTERNATIONAL JOURNAL OF CREATIVE RESEARCH THOUGHTS (IJCRT)

An International Open Access, Peer-reviewed, Refereed Journal

Numerical and Experimental Study of The Air-Conditioning System of a Large Sports Hall

Ph.D. Student. Mohammed Alsheekh

Department of Mechanical Engineering, Engineering College, University of Basra.

Abstract

This research studies the internal environmental conditions in a large mechanically-conditioned sports hall experimentally and numerically. The Computational Fluid Dynamics (CFD) using the Engineering Software (ANSYS/FLUENT) conducts this analysis. The (Design Modeler) program designed the inner space of the hall, and the usual numerical findings of the program were compared and ratified with the experimental data gathered in a large sports hall at the Basra University Athletic Education College during the ten-day campaign. The metrics include the properties of the airflow at various internal space sites, as well as the surface temperature of the internal materials. In the model, various systems are applied to examine the ambient conditions occurring in the hall under various ventilation and professional conditions, which contribute to the various cooling trends. The numerical findings and the experimental tests include the amplitude of the airflow, temperature, relative humidity, and concentration of CO2 in the hall's door atmosphere when this hall is empty and when a sporting event is involved. There is a comparison of the numerical and experimental findings, and they indicate strong consensus.

Keywords: Large Enclosure, Athletic Hall, CFD model, ANSYS, IAQ.

1. Introduction

We should care about indoor air quality when we are indoors. HVAC systems adjust the indoor air by filtration, humidification, dilution, and cooling. Air-conditioning buildings have their external microbial emissions and not necessarily from the outside. Air-conditioning devices can be a cause of fungal contamination. Indoor air quality can be measured using CO_2. Indoor air quality is determined by CO_2 levels in open spaces (ASHRAE Standard 62, 2001). Ventilation is intended to remove indoor toxins. A decent fan would be expected to prevent indoor air pollution. [1]. There are three numerical approaches: finite difference, finite element, and spectral methods. This paper deals exclusively with the finite volume system, a special formulation of finite differences that is fundamental to the better-known CFD codes - CFX/ANSYS, FLUENT, PHOENICS, and STAR-CD. [2]. CFD codes are used for optimal indoor airflow analyses and dispersion of contaminants for health and safety purposes. Therefore, many studies do not adequately combine theoretical and experimental approaches to examining stadium air quality. Proper ventilation and fresh air play a key role in the control of indoor air quality and thermal comfort. Some tests are carried out with a computational fluid dynamics code, on a large mechanically ventilated athletic hall as (PHOENICS). [5]. The technology has improved knowledge of the thermal environments and has contributed to the development of more rational thermal testing methods. Despite the sophistication of the models, the techniques used to simulate air exchange have become more sophisticated, making it difficult to test the thermal environment. simulation programming that allows the measurement of both thermal comfort and stress indices to enhance the quality of life and, on the other hand, reduces the risk of thermal stress. [11]. The growing demand for air-conditioned buildings and the resultant demand for electrical energy has prompted research into passive cooling, such as the Euro pear Union-funded PASCOOL programmer. One of the tasks in this programmer is to develop comfort criteria appropriate to free-running non-air-conditioned buildings. [12]. Computational fluid dynamics is a useful tool for predicting the diffusion of airborne contaminants in a room. The CFD program called (VORTEX). Was used to provide a microscopic prediction of the diffusion of the tracer gas in the whole space as well as air velocity and temperature data. The CFD simulations were found to be particularly useful for understanding, in greater detail, the air diffusion process in the room. [13]

Therefore, this study investigates numerically and experimentally the environmental conditions prevailing in a large indoor (wrestling Hall) under different mechanical ventilation schemes and occupation conditions by ANSYS15 (CFD).

2. Experimental Procedure

A 10-day experimental campaign in the frame of a research project is accomplished in an indoor wrestling hall within the Sport Education College, University of Basra. This hall is surrounded by the close vicinity includes heavy-traffic roads at about 1 km and the sea at about 2 km to the southwest. The height of the indoor space is 8 m, the arena is 265 m^2, and the capacity of the hall is (35-50) people. The windows are normally closed and the heating–ventilating–air conditioning (HVAC) system operates according to the needs. Measurements were taken at different locations in the hall with and without the HVAC system is in operation. This information is shown in Figures (1and4).

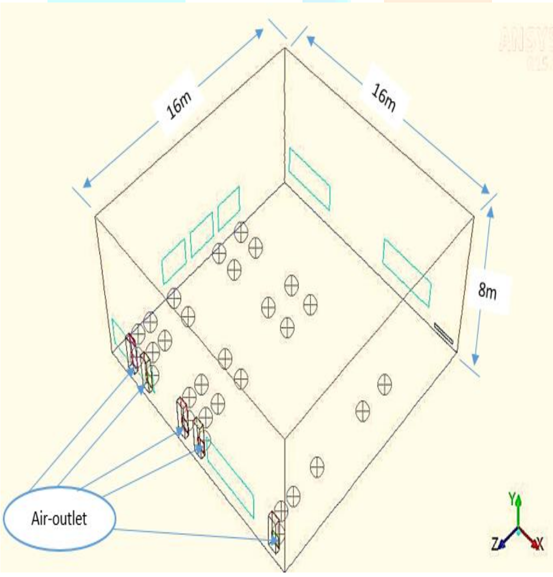

Fig.1 Plane view of some points for Basic Case1.

Fig.2infrared thermometer.

Fig.3 Anemometer + Psychrometer. [8]

2.1 Measuring Instruments

The main instrument used is (Data logging / Printing Anemometer + Psychrometer) [8]. This device simultaneously measures and displays air velocity, temperature, relative humidity, wet bulb temperature, and air Volume (CFM / CMM). Surface temperatures of indoor materials were measured with an infrared thermometer. These instruments are shown in Figures (2and3).

3. Theoretical Model, Initial and Boundary Conditions

The ANSYS 15 (CFD) code [7] solves the time-averaged conservation equations of mass, momentum, energy, and chemical species in steady three-dimensional flows: [2]

$$\frac{\partial(\rho\emptyset)}{\partial t} + div(\rho\emptyset v) = div(\Gamma grad\emptyset) + S_{\emptyset} \quad \text{----------- (1)}$$

Rate of increase		Net rate of flow		Rate of increase		Rate of increase
Of Ø of fluid	+	of Ø out of	=	of Ø due to	+	of Ø due to
Element		fluid element		diffusion		sources

Where; ρ, v, Γ, and $S_{\emptyset}$ are density, velocity vector, effective exchange coefficient ofØ, and source rate per Unit volume, respectively.

The discretization of the domain is followed by the reduction of the previous equations to their finite domain form using the hybrid formulation of the coefficients and the solution technique employs the SIMPLEST algorithm (an improved version of the well-known SIMPLE algorithm). The standard (k-ε) turbulence model is applied, while buoyancy effects are considered, to improve convergence, under relaxation which was used.

As seen in (Fig.1) the dimensions of the objects are real, the geometry is as detailed as possible according to the plans of the building and the blueprints of the mechanical ventilation system, always taking into account computational efficiency. The domain size is (16m x16m x 8m), and it includes 3 rows of spectators' seats and 5 inlet air fans of split unit devices. It should be noted that the model configurations were set so that the best balance is achieved among convergence, grid independence, and runtime saving, due to the high complexity of the domain geometry as can be shown in Figures (4and5).

Model configurations concerning boundary and initial conditions, as well as settings information of the studied cases, are given below: Fresh air comes in the hall via split unit fans air inlets of Figures (1and4), the dimensions of each inlet are 0.3m x 0.5m, with mean axial z-velocity of 5m/s and turbulence intensity of 5%, respectively, according to the experimental measurements in all cases. The boundary condition and the temperature locations can be shown in table (1).

Table 1: Information and input model data of the studied cases.

Data	Basic case1	Cooling case1	Event case1	Case2
Inlet air temp. (°C) of split unit 1	39.6	18.1	18	18
Inlet air temp. (°C) of split unit 2	39.8	18	18	18
Inlet air temp. (°C) of split unit 3	40	18.2	18.1	18
Inlet air temp. (°C) of split unit 4	39.9	18.3	18.1	18
Inlet air temp. (°C) of split unit 5	39.5	18	18	18
Ceiling surface temp. (°C)	43-44	42	41	40-41
Floor surface temp. (°C)	40-40.5	39	38.4	38-38.8
East wall surface temp. (°C)	41-41.5	41	40	39-41
North wall surface temp. (°C)	41-41.5	41	40	39-41
Windows temp.(°C)	45	43.5	42	42
Surface temp. of 1st seat level (°C)	40	38.4	37.3	37-37.6
Surface temp. of 2nd seat level (°C)	40.5	38	37.2	37-37.4
Surface temp. of 3rd seat level (°C)	40	38.1	37	37

4. Design and performance of thermal comfort.

Many indexes can be used to account for the design and performance of thermal comfort, among these indexes is the air diffusion performance index (ADPI). This index can be calculated as follows [9]: -

$$EDT(\theta) = (T_x - T_{av}) - 8(V_x - 0.15) \quad \text{----------------------- (2)}$$

Where: *EDT (θ)* Effective draft temperature (K), *(T_x)* Local air stream dry-bulb temperature (°C), *(T_{av})* Average (set-point) room dry-bulb temperature (°C), *(V_x)* Local airstream centerline velocity (m/s), which must be *(V_x*<0.36 m/s).

$$ADPI = \frac{N_\theta}{N} x100 \quad \text{--- (3)}$$

Where: *(N_θ)* Number of points measured in the occupied space that falls within ($-1.5 < \theta < +1$K), (N) Total number of the point measured in the occupied space

The performance of an air distribution system within a room/zone can be rated in terms of the Air Diffusion Performance Index (ADPI). Among the several evaluation methods used to design air distribution systems based on flow rate, sound data, isovels, and comfort criteria, the (ADPI) selection method is quite commonly used. The selection process takes advantage of ADPI's correlation with the ratio of the isothermal throw of the diffuser and the characteristic length of the system in the room. This paper clarifies what (ADPI) is and is not, what is involved in the selection process and how it fares against the industry standards and benchmarks in ventilation and thermal comfort. Several factors that may potentially cause any deviation to the predicted value of (ADPI) during the design stage are discussed. The post-installation (ADPI) that reflects the actual (ADPI) rating for space has to be assessed and verified on-site. This on-site measurement should be conducted per a set of guidelines given in the (ASHRAE) Standard 113. A detailed (CFD) analysis can also provide an alternative solution to verify the actual "as-built" ADPI rating. [9]

5. Validation of CFD Program

A commercial (CFD) program was used for the computations. By default, the code uses the finite-volume method and the upwind-difference-scheme for the convection term. The convergence criterion was set such that the respective sum of the absolute residuals must be less than 10^{-3}. [6]

The validation of (CFD) program is obtained by comparing the measured data and the numerical results of the flow patterns, vertical profiles of temperature, CO_2 concentration, velocity, and turbulence intensity inside the large athletic hall (wrestling hall). as shown in Figure (9).

6. Results and Discussion

The factors affecting the comfortable conditions inside the large athletic hall, i.e., velocity, temperature, relative humidity, and CO_2 concentration are discussed through the following studied cases.

6.1 Case 1 (Actual Case)

This case represents the (wrestling hall) of the college of physical education at the University of Basra. This hall is of (16m x16m x 8m height) dimensions, and it includes 3 rows of spectators' seats and 5 inlet air fans of (5-ton split unit each). An experimental study and numerical study using the engineering (ANSYS 15) program are achieved on this hall with different conditions of the indoor environment. It should be noted that the model configuration was set so that the best balance is achieved among convergence, grid independence, and runtime saving, due to the high complexity of the domain geometry, figures (5 and 6).

6.1.1 Basic case1

In this case, the hall to be studied was empty and the HVAC system operates in the air-conditioning mode without cooling, i.e., only the fan is operated. The experimental measurements are accomplished, comprising, air velocity temperature and relative humidity, inside the hall at a certain air velocity of the fan. The numerical values of the same air properties using the (ANSYS 15) program are also accomplished corresponding to the same inlet air velocity used in the previous experiment test. Two inlet air velocities of the HVAC fan, are studied. These velocities are (4 m /s) and (5 m/s) but in this study chosen only (5 m/s).

The comparison of experimental data and the numerical result seems a good agreement as given by table (2), which includes the experimental and numerical results for several points and the points at the arena, for the two studied inlet velocities, figure (1) shows the points in the (x-y) plane toward the (-ve z-axis) within the area of thermal. The comfort of human feeling.

It was observed that the temperature differences are between (1-2°C) in large halls as shown in figures (13 and 14). The result of the studied hall is compared with graphical figures (10 and 11) of velocity vector-only because the flow in the large space is isothermal and the variances are shown in the chart (1 and 2). The comparison with the published data reveals a good agreement, e.g., for the relative humidity, it can be shown that “The difference between experimental and numerical values is between 10% and 20%, considering several similar results reported in the field of indoor airflow measurements and simulations” [10].

6.1.2 Cooling Case1

This is a hypothetical case; the hall is assumed to be empty and the HVAC system operates in the cooling mode.

A typical summer day is considered in this case, with an outdoor temperature of 47°C and an initial indoor temperature of 41°C. The same experimental and numerical steps followed in the basic case are applied, and the same inlet air velocity of (5 m/s) is tested. The (ANSYS15) program, CFD, and Fluent 15 are applied for the selected points inside the hall at the levels (y=1.75m, y=0.5m) the obtained result is shown in table (3), and can be seen in figures (16-18).

6.1.3 Event Case1

The hall is half-full with 20 spectators attending an athletic event. The ventilation conditions are tested when the HVAC system operates in the air – conditioning mode with and without cooling. The (ANSYS15) program, CFD, and Fluent 15 are applied for the selected points in the numerical study at the levels (y=0.5m, y=0.9m, y=3m). Chosen velocity less than (0.36m/s) to obtain (ADPI) according to the requirement to calculate (EDT). The value was (ADPI=53).

The result of this case is shown in table (4). In this case, it is observed that when the hall is semi or fully loaded with people the thermal comfort factor (ADPI) is decreased, even if the inlet air velocity is increased the (ADPI) will not be enhanced. This indicates that there is a lock in the air-conditioning load of the tested hall, and the air distribution system applied is not favorable. See the charts (3 and 4) and figures (19 and 20) which are related to this case.

6.2 Case 2 (Modification Case)

A modified case is proposed for the college of physical education at Basrah University. The engineering (ANSYS15) program is used to distribute the conditioned air according to an engineering division of the hall into useful zones, as shown in figures (7 and 8). This case is studied by selecting points inside the hall as previously done in the other cases at the levels (Y = 0.5m: Y = 0.9m: Y = 3m). With the same inlet air velocity (5 m/s). With this new distribution, the (ADPI) is improved from the value (53%) to (77%). the results of this case can be seen in table (5), the charts (3 and 4), and figures (23 and 24).

6.3 Debating Remarks

From the above-studied cases, and observing the air velocity distribution, it can be shown that the velocity in the breathing zone at the horizontal line (z-plane) at (y=0.5m, y=0.9m, and y=1.75m) above the floor, is high, because this region near the air supply grill, and it is decreased away from this region toward the opposite side of the supply air inlet. It can be shown also that the air velocity is increased as the air change per hour is increased.

Now observing the temperature profile in the breathing zone at the horizontal line (z-plane) at (y=0.5m, y=0.9m, and y=1.75m) above the floor, it can be shown that the temperature values are low near the supply air inlet, and they have increased away from this region because of the heat released from the occupants. It can be shown also that the air temperature is decreased as the air change per hour is increased.

The relative humidity inside the hall shows the opposite trend of the air temperature, i.e. as the temperature is increased the relative humidity is decreased and vice versa. The results are shown in tables (2, 3,4and5), the chart (3 and 5), and figures (15, 18, 21, and 25).

Finally observing the CO_2 concentration distribution inside a large athletic hall, it is clear that CO_2 concentration becomes very high in the region of the occupied hall with athletes and spectators. This concentration is decreased in the supply air inlet region, and it is decreased also as the air change per hour is increased. It is also decreased with increasing the inlet air velocity because of better removal of CO_2 concentration. These results are listed in tables (4and5), chart (8) and can be shown in figures (22, 26, and 27).

7. Conclusions

From the results obtained in this study, the following conclusions can be made:

1- The engineering (ANSYS 15) program can be used to redesign the air conditioning system of a large athletic hall and to select a new air distribution system. So that the (ADPI) is improved, with saving time, cost and effort.

2- In this study, and by using the (ANSYS15) program it is found that the air conditioning system and the air distribution system of the wrestling hall of the faculty of physical Education of Basrah University have some problems. With a little improvement in these two systems, it is found that the (ADPI) is improved from (53%) to (77%).

3- It is recommended for this college, that, if the air conditioning split unit are redistributed according to an engineering manner, and maybe add some other units to the hall the human comfort will be enhanced more and more and this will increase the air diffusion performance factor to its maximum value.

Table 2: Measured and simulated air velocities, temperatures, and relative humidities at measured points and CFD points of the domain for the basic case.

point	locations			V_{exp} m/s	V_{Nu} m/s	Error %	T_{exp} k	T_{Nu} k	Error %	Rh_{exp} %	RH_{Nu} %	Error %
	X(m)	Y(m)	Z(m)									
points average of outlet 1				2.55	2.835	10.0	314	313.925	-0.02	21.3	21.021	-1.3
points average of outlet 2				2.49	2.825	11.9	313.5	313.901	0.13	21.1	21.045	-0.3
points average of outlet 3				2.51	2.849	12.1	313.9	313.857	-0.01	20.9	21.105	1.0
points average of outlet 4				2.68	2.864	6.5	314	313.943	-0.02	20.7	21.017	1.5
points average of outlet 5				2.61	2.962	11.9	313.8	313.813	0.00	20.5	21.161	3.1
1	1.75	1.75	15	4.89	5.004	2.2	313	312.602	-0.13	20.3	22.583	10.1
2	1.75	1.75	14	4.50	4.649	3.2	312.4	312.705	0.10	20.1	22.460	10.5
3	1.75	1.75	12	1.98	2.194	9.7	312.8	313.313	0.16	19.9	21.743	8.5
4	1.75	1.75	8	1.23	1.257	1.9	313	313.523	0.17	22	21.501	-2.3
5	1.75	1.75	6	0.94	1.068	11.9	313	313.545	0.17	21.2	21.476	1.3
6	3.25	1.75	15	5.00	5.003	0.0	312.4	312.802	0.13	20.4	22.343	8.7
7	3.25	1.75	14	4.10	4.670	12.3	312.4	312.883	0.15	21.5	22.247	3.4
8	3.25	1.75	12	1.48	1.764	16.2	313	313.483	0.15	21.3	21.546	1.1
9	3.25	1.75	8	1.11	1.259	11.5	313.5	313.582	0.03	21.1	21.434	1.6
10	3.25	1.75	6	0.89	1.105	19.2	314	313.577	-0.13	20.9	21.439	2.5
11	6.75	1.75	15	4.70	5.002	6.0	313	313.002	0.00	20.7	22.106	6.4
12	6.75	1.75	14	4.07	4.708	13.6	313	313.062	0.02	20.5	22.036	7.0
13	6.75	1.75	12	1.12	1.457	22.8	313.7	313.665	-0.01	20.3	21.339	4.9
14	6.75	1.75	8	1.08	1.434	24.8	313.8	313.659	-0.05	20.1	21.347	5.8
15	6.75	1.75	6	0.89	1.192	25.2	314	313.638	-0.12	19.9	21.371	6.9
16	8.25	1.75	15	4.81	5.005	3.9	313	312.902	-0.03	21.8	22.225	1.9
17	8.25	1.75	14	4.51	4.693	3.9	313	312.978	-0.01	22	22.134	0.6
18	8.25	1.75	12	1.66	1.908	13.2	313.4	313.583	0.06	22.2	21.432	-3.6
19	8.25	1.75	8	0.68	0.877	22.6	314	313.629	-0.12	22.4	21.380	-4.8
20	8.25	1.75	6	0.41	0.468	12.8	314	313.599	-0.13	22.6	21.414	-5.5
21	14.75	1.75	15	4.98	5.006	0.5	312.7	312.503	-0.06	22.8	22.704	-0.4
22	14.75	1.75	14	3.68	3.870	5.0	312.5	312.788	0.09	23	22.360	-2.9
23	14.75	1.75	12	0.95	0.994	4.9	313	313.413	0.13	21.1	21.628	2.4
24	14.75	1.75	8	0.40	0.332	-21.2	313.4	313.6	0.06	21.7	21.413	-1.3
25	14.75	1.75	6	0.41	0.346	-18.8	314	313.692	-0.10	22.3	21.309	-4.6
26	1.75	0.5	15	0.50	0.505	1.6	314	313.855	-0.05	22.9	21.126	-8.4
27	3.25	0.5	15	0.40	0.420	4.6	314.2	313.834	-0.12	21	21.149	0.7
28	6.75	0.5	15	0.41	0.435	5.1	314.3	313.847	-0.14	20.9	21.134	1.1
29	8.25	0.5	15	0.47	0.558	15.5	314	313.885	-0.04	21	21.092	0.4
30	14.75	0.5	15	0.41	0.455	9.3	314	313.847	-0.05	21	21.134	0.6

Table 3: Measured and simulated air velocities, temperatures, and relative humidities at measured points and CFD points of the domain for the cooling case.

Point No.	Locations			V_{exp} m/s	V_{Nu} m/s	Error %	T_{exp} k	T_{Nu} k	Error %	Rh_{exp} %	RH_{Nu} %	Error %
	X(m)	Y(m)	Z(m)									
points average of outlet 1				2.43	2.844	14.7	303.7	305.014	0.4	17.9	17.222	3.9
points average of outlet 2				2.71	2.779	2.4	304.5	303.511	-0.3	19.3	18.745	3.0
points average of outlet 3				2.10	2.818	25.3	303.5	304.096	0.2	18.6	18.116	2.7
points average of outlet 4				2.10	2.804	25.2	303.9	304.352	0.1	19.7	17.858	10.3
points average of outlet 5				2.28	2.972	23.2	304.1	304.759	0.2	19.6	17.486	12.1
1	1.75	1.75	15	5.03	4.902	-2.7	295.1	291.588	-1.2	30.8	38.281	-19.5
2	1.75	1.75	14	4.00	3.835	-4.4	297.2	294.28	-1.0	34.6	32.392	6.8
3	1.75	1.75	12	2.02	2.470	18.4	298.3	297.412	-0.3	28.6	26.784	6.8
4	1.75	1.75	8	1.17	1.469	20.2	300.2	299.397	-0.3	23.8	23.800	0.0
5	1.75	1.75	6	1.72	1.344	-27.9	300.2	299.576	-0.2	23.9	23.549	1.5
6	3.25	1.75	15	4.69	4.898	4.2	294.1	291.474	-0.9	39.4	38.557	2.2
7	3.25	1.75	14	4.07	4.104	0.7	293.4	293.379	0.0	36.8	34.242	7.5
8	3.25	1.75	12	1.15	1.708	32.8	295.1	298.696	1.2	26.1	24.808	5.2
9	3.25	1.75	8	0.63	0.671	6.3	303.4	301.217	-0.7	21.1	21.389	-1.4
10	3.25	1.75	6	0.53	0.651	18.2	303.7	301.188	-0.8	21.2	21.427	-1.1
11	6.75	1.75	15	4.71	4.898	3.8	293.6	291.672	-0.7	33.2	38.080	-12.8
12	6.75	1.75	14	3.23	3.745	13.7	297.2	294.362	-1.0	34.4	32.229	6.7
13	6.75	1.75	12	1.45	1.806	19.9	300.1	298.781	-0.4	25.6	24.684	3.7
14	6.75	1.75	8	0.73	0.867	16.0	303.5	301.526	-0.7	21.1	21.008	0.4
15	6.75	1.75	6	0.49	0.738	33.3	303.7	301.892	-0.6	21.2	20.567	3.1
16	8.25	1.75	15	4.86	4.903	0.8	293.3	291.775	-0.5	38.6	37.835	2.0
17	8.25	1.75	14	3.61	3.802	5.0	297.8	294.412	-1.2	34.1	32.129	6.1
18	8.25	1.75	12	1.77	1.670	-5.8	300.1	299.321	-0.3	25.9	23.906	8.3
19	8.25	1.75	8	0.72	0.375	-93.0	301.4	302.666	0.4	19.3	19.668	-1.9
20	8.25	1.75	6	0.44	0.357	-22.2	304.3	302.733	-0.5	19.1	19.591	-2.5
21	14.75	1.75	15	5.01	4.909	-2.0	292.2	291.484	-0.2	32.6	38.533	-15.4
22	14.75	1.75	14	3.51	3.759	6.5	294	294.384	0.1	34.4	32.184	6.9
23	14.75	1.75	12	0.94	1.489	37.0	300	299.525	-0.2	24.1	23.621	2.0
24	14.75	1.75	8	0.40	0.276	-45.1	303.6	303.066	-0.2	19.5	19.220	1.5
25	14.75	1.75	6	0.44	0.286	-52.2	303.7	302.898	-0.3	19.4	19.406	0.0
26	1.75	0.5	15	0.50	0.438	-14.9	304	304.14	0.0	18.9	18.074	4.6
27	3.25	0.5	15	0.41	0.547	25.2	304.1	302.798	-0.4	19.4	19.519	-0.6
28	6.75	0.5	15	0.42	0.400	-4.2	303.8	303.22	-0.2	19.2	19.050	0.8
29	8.25	0.5	15	0.54	0.797	32.4	303.6	303.796	0.1	18.4	18.432	-0.2
30	14.75	0.5	15	0.47	0.529	11.5	302.4	304.011	0.5	17.2	18.208	-5.5

Table 4: Simulated air velocities, temperatures, and relative humidities for CFD points of the Event Case domain.

Point No.	location			V_{Nu} m/s	T_{Nu} k	RH_{Nu} %	CO_2 Kmol/m^3	EDT
	X(m)	Y(m)	Z(m)					
1	0.5	0.5	2	0.089656	304.909	17.34213	0.000179	3.087579
2	0.5	0.5	13	0.094675	304.1983	18.05705	0.000182	1.52993
3	4	0.5	3	0.181778	305.7208	16.56523	0.000181	2.355539
4	4	0.5	13	0.265107	302.7708	19.596	0.000185	-1.26108
5	11	0.5	12	0.305292	302.8033	19.56509	0.000208	-1.55006
6	10	0.5	12	0.297353	302.741	19.63384	0.000202	-1.54886
7	12	0.5	14	0.266446	302.9426	19.40763	0.000203	-1.09997
8	13	0.5	13	0.229425	302.79	19.57939	0.000205	-0.95636
9	14	0.5	1	0.304621	303.7619	18.51944	0.000211	-0.58606
10	14	0.5	14	0.195231	303.0063	19.33685	0.000203	-0.46659
11	0.5	0.9	1	0.267156	304.1579	18.09824	0.000179	0.109619
12	0.5	0.9	13	0.068662	302.8181	19.54171	0.000181	0.35778
13	2	0.9	2	0.251388	302.8158	19.54447	0.000182	-1.10633
14	4	0.9	13	0.224906	302.1224	20.34169	0.00018	-1.58781
15	10	0.9	9	0.276736	302.5956	19.8009	0.000209	-1.52928
16	10	0.9	12	0.215363	302.5822	19.81407	0.000201	-1.05166
17	12	0.9	14	0.246176	302.7581	19.61475	0.000203	-1.12231
18	13	0.9	13	0.209616	302.6156	19.7773	0.000206	-0.97229
19	14	0.9	1	0.330056	303.2569	19.0624	0.00021	-1.29458
20	14	0.9	14	0.172172	302.776	19.59464	0.000203	-0.51241
21	2	3	11	0.31187	301.1882	21.47488	0.000178	-3.21779
22	0.5	3	13	0.231973	302.1484	20.31214	0.000184	-1.61838
23	4	3	8	0.306338	301.5724	21.00202	0.000185	-2.78931
24	4	3	13	0.154633	302.1992	20.25328	0.000187	-0.94884
25	10	3	9	0.20424	302.1953	20.26194	0.000202	-1.34961
26	10	3	12	0.174403	302.3608	20.06841	0.0002	-0.94538
27	12	3	14	0.330681	302.5616	19.83786	0.000202	-1.9949
28	13	3	13	0.315739	302.4944	19.91566	0.000205	-1.94253
29	14	3	3	0.314467	302.1726	20.28889	0.000203	-2.25418
30	14	3	12	0.259413	302.3684	20.06179	0.000208	-1.61792

Table 5: Simulated air velocities, temperatures, and relative humidities for CFD points of the Case2 domain.

Point No.	Location			V_{Nu} m/s	T_{Nu} k	RH_{Nu} %	CO_2 Kmol/m^3	EDT
	X(m)	Y(m)	Z(m)					
1	2	0.5	2	0.215593	301.4313	21.14371	6.94E-05	-0.37567
2	0.5	0.5	13	0.081719	302.9296	19.39101	7.86E-05	2.193681
3	4	0.5	3	0.354139	303.7336	18.51827	8.16E-05	0.818272
4	4	0.5	13	0.183768	301.4867	21.07687	7.40E-05	-0.06562
5	10	0.5	6	0.193672	300.9079	21.79894	7.12E-05	-0.72365
6	10	0.5	12	0.256899	300.8479	21.87635	7.42E-05	-1.28949
7	12	0.5	14	0.09928	301.7008	20.81682	7.56E-05	0.824371
8	13	0.5	13	0.164032	301.3125	21.29214	7.59E-05	-0.08199
9	14	0.5	1	0.289727	301.3676	21.22408	7.64E-05	-1.03241
10	14	0.5	14	0.094157	301.7595	20.74609	7.57E-05	0.924034
11	2	0.9	1	0.104214	300.8909	21.82053	7.07E-05	-0.02501
12	0.5	0.9	13	0.149673	301.9541	20.51432	7.92E-05	0.674488
13	7	0.9	2	0.053266	302.2166	20.20492	7.87E-05	1.708223
14	4	0.9	13	0.191076	301.3534	21.2408	7.33E-05	-0.25745
15	10	0.9	4	0.112302	300.7556	21.99724	8.31E-05	-0.22507
16	10	0.9	12	0.309243	300.543	22.26973	7.36E-05	-2.01314
17	12	0.9	14	0.118082	301.4865	21.07757	7.56E-05	0.459651
18	13	0.9	13	0.175417	301.0968	21.56137	7.61E-05	-0.38874
19	14	0.9	1	0.217386	301.1018	21.55542	7.72E-05	-0.71948
20	14	0.9	14	0.075809	301.4197	21.15953	7.53E-05	0.73104
21	2	3	2	0.298754	300.5546	22.25522	7.57E-05	-1.9176
22	0.5	3	13	0.297363	301.2813	21.33147	7.86E-05	-1.17977
23	4	3	8	0.245227	300.9678	21.72362	7.41E-05	-1.07622
24	4	3	13	0.27316	301.5492	21.00045	7.38E-05	-0.71832
25	10	3	9	0.240382	300.5518	22.25857	7.47E-05	-1.45349
26	10	3	12	0.275086	300.5285	22.28937	7.62E-05	-1.75439
27	12	3	14	0.280706	301.1784	21.45915	7.63E-05	-1.14947
28	13	3	13	0.277045	300.78	21.9642	7.73E-05	-1.51859
29	14	3	3	0.218096	300.3967	22.46277	7.83E-05	-1.43027
30	14	3	12	0.282199	300.381	22.48368	7.90E-05	-1.95881

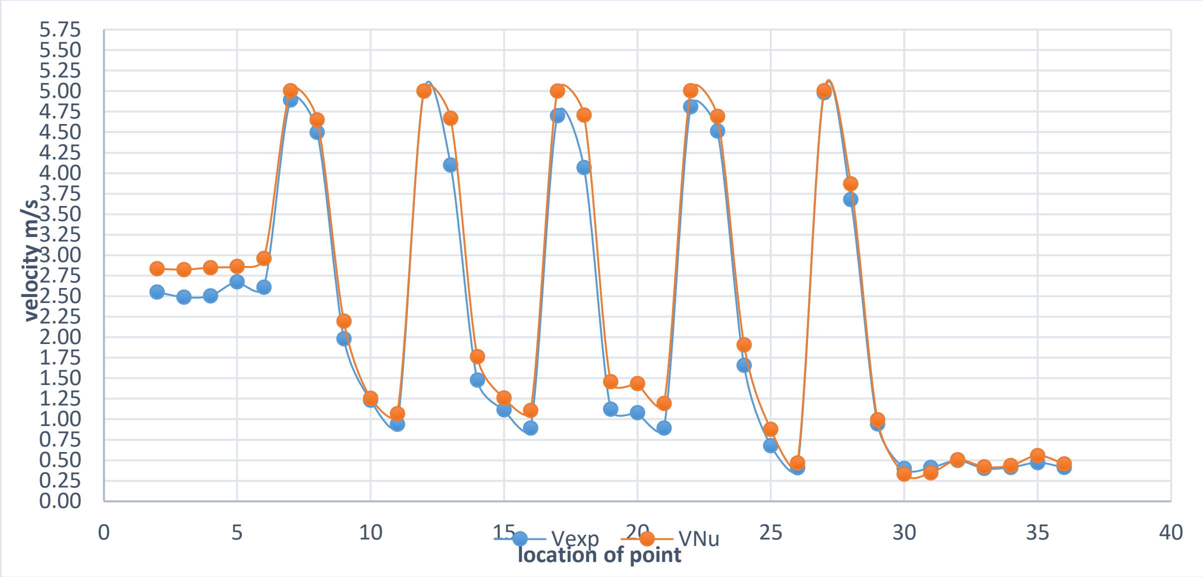

Chart .1 comparison between V_{exp} and V_{Nu}.

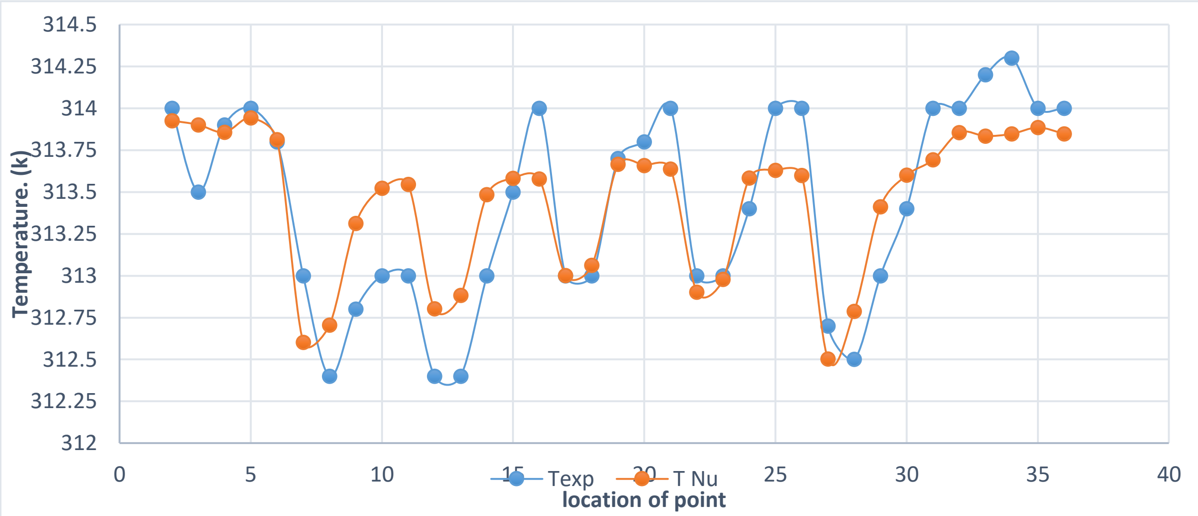

Chart .2 comparison between T_{exp} and T_{Nu}.

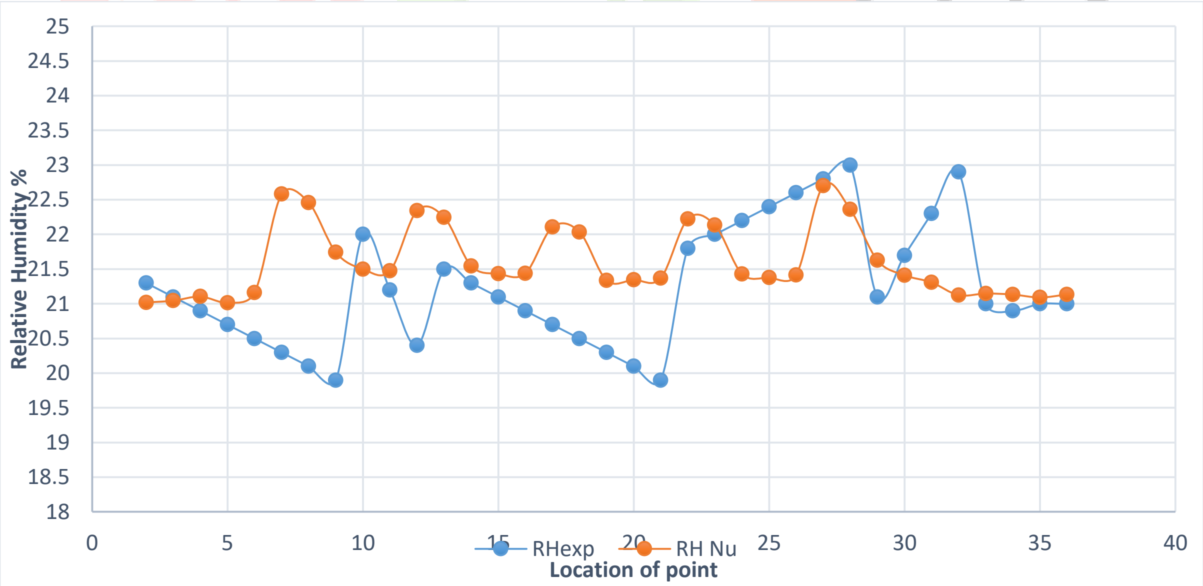

Chart .3 comparison between RH_{exp} and RH_{Nu}

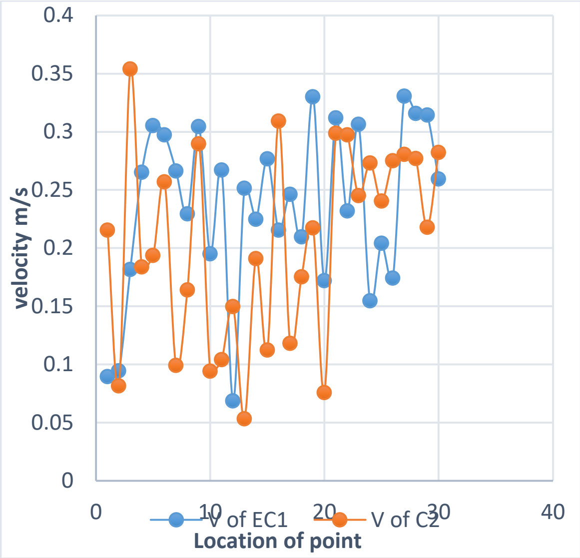

Chart .4 comparison between V_{exp} and V_{Nu}.

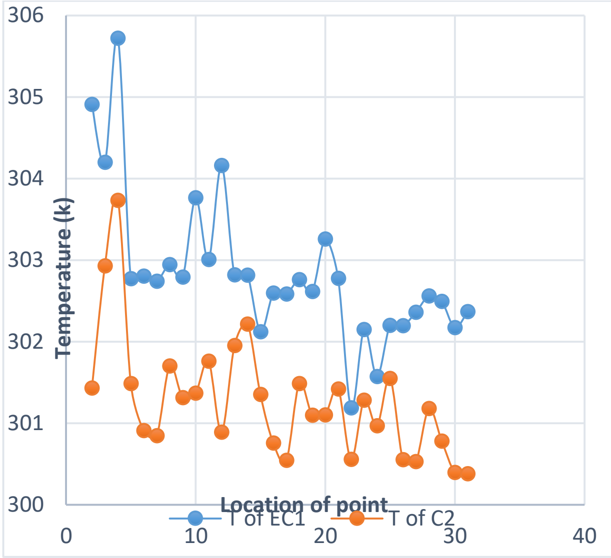

Chart .7 comparison between T_{exp} and T_{Nu}

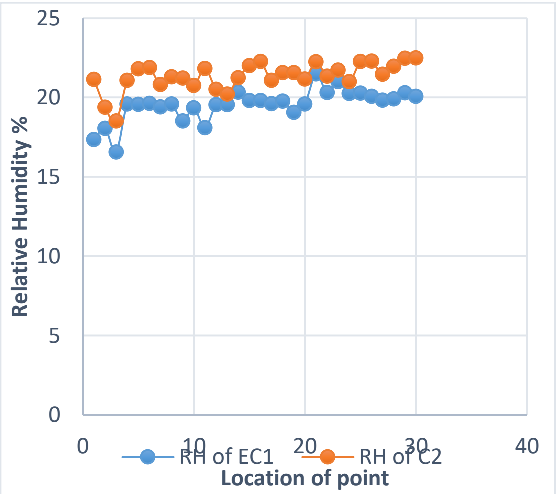

Chart .5 comparison between RH_{exp} and RH_{Nu}.

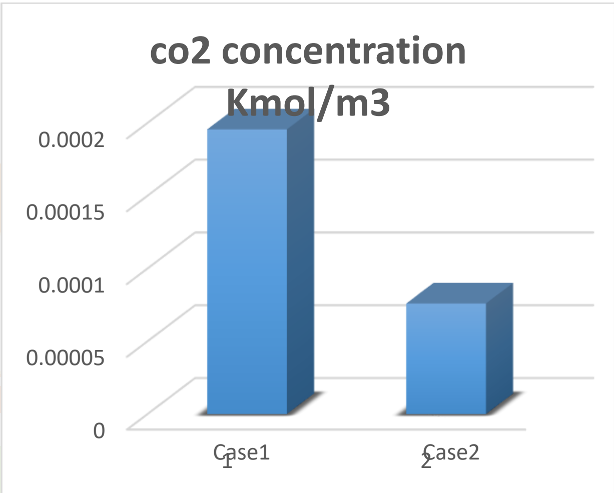

Chart .8 comparison for CO_2 between C1 and C2.

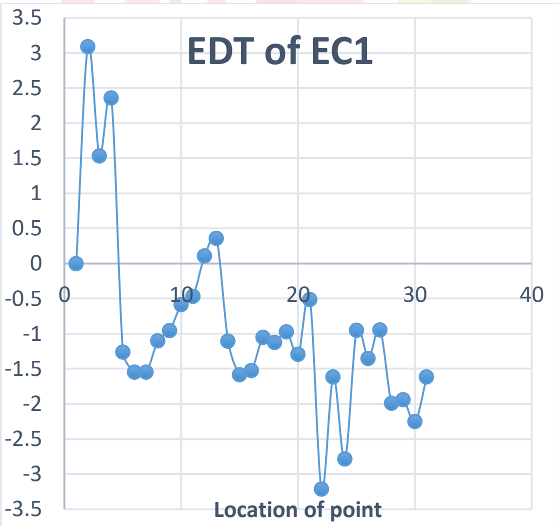

Chart .6 (EDT) for Event Case1.

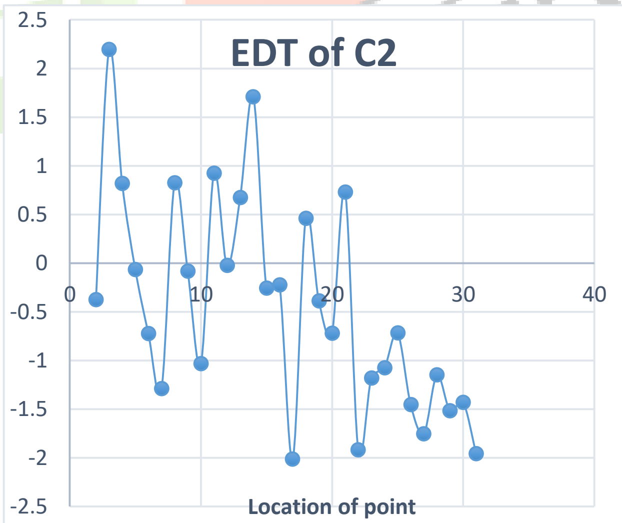

Chart .9 (EDT) for Case2.

Fig.4 Domain of the athletic hall.

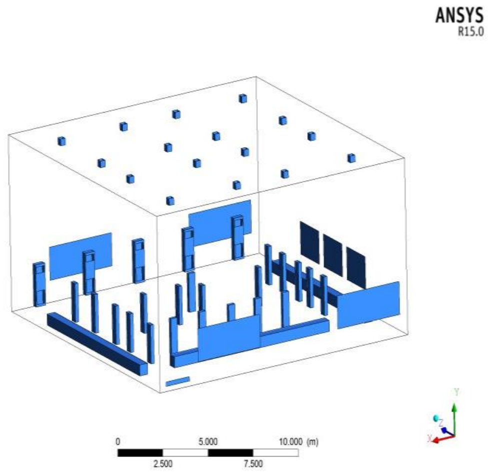

Fig.7 Geometrical domain of Case2.

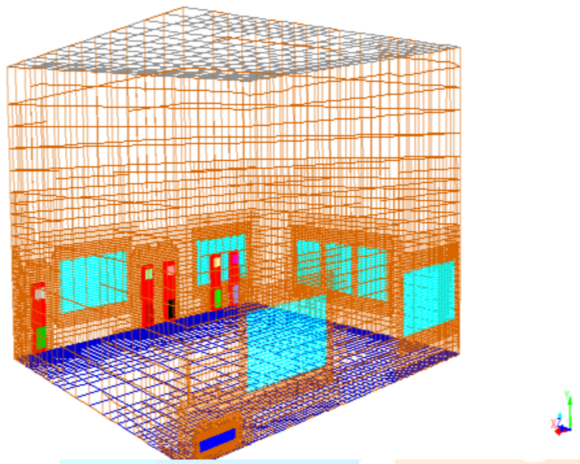

Fig.5 Plane of the geometrical grid For the BC1.

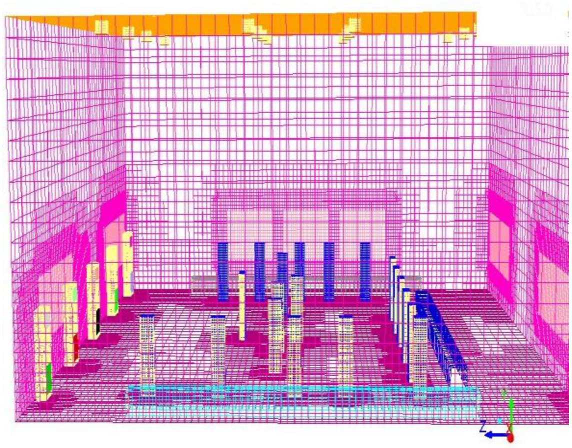

Fig.8 Mesh domain of Case2.

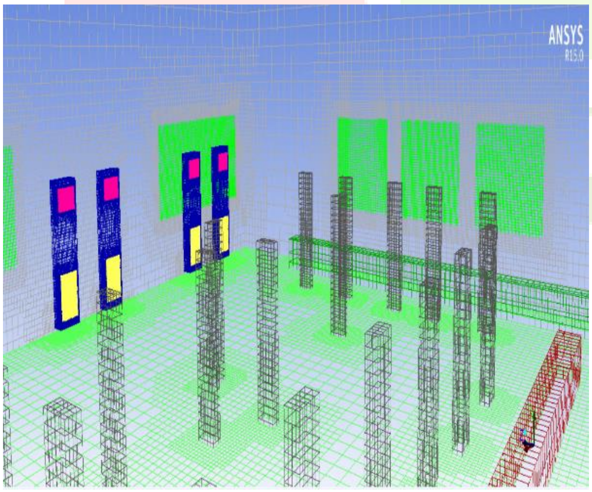

Fig.6 Plane of the geometrical grid for the EC1.

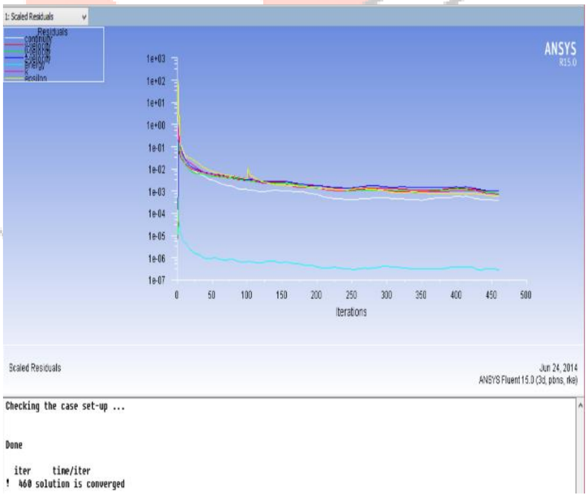

Fig.9 Validation of the Basic Case1.

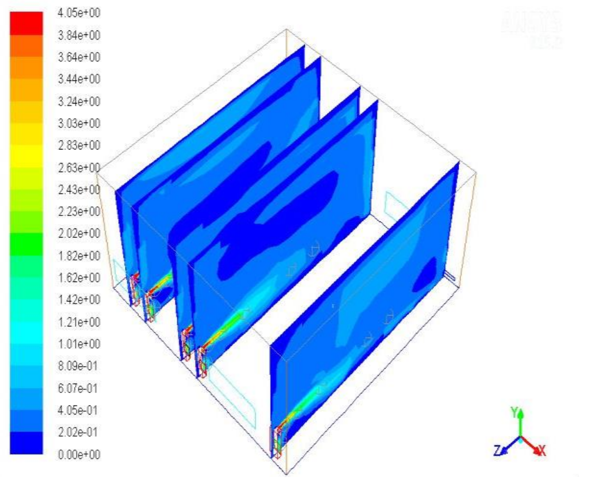

Fig.10: Contours of velocity magnitude for Basic Case1 at x-plane.

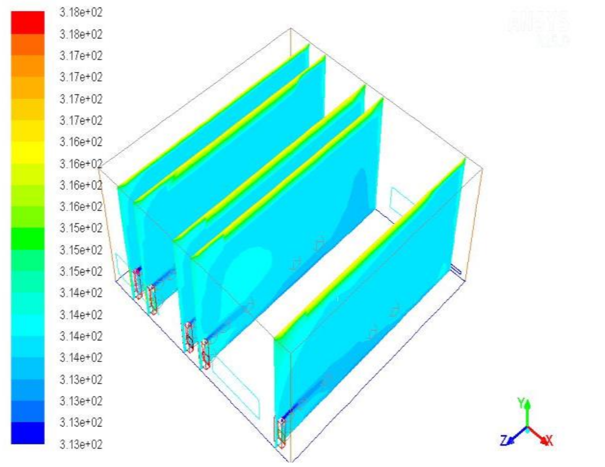

Fig.13: Contours of static Temperature for Basic Case1 at x-plane.

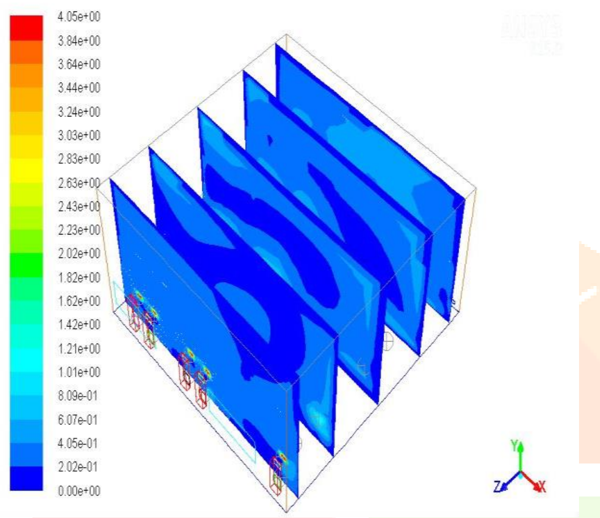

Fig.11: Contours of velocity magnitude for Basic Case1 at z-plane.

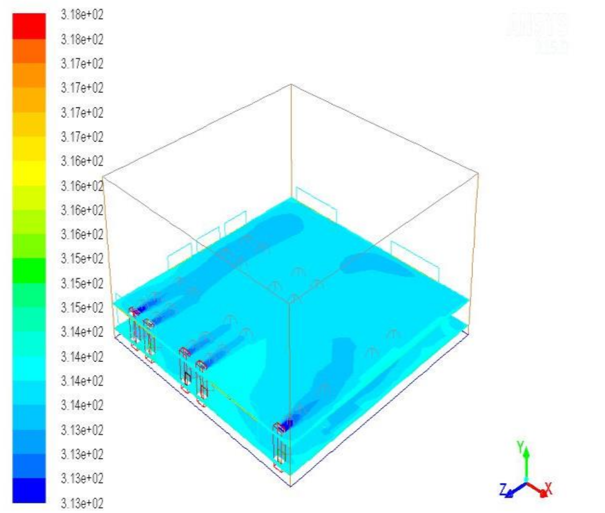

Fig.14: Contours of static Temperature for Basic Case1 at y-plane.

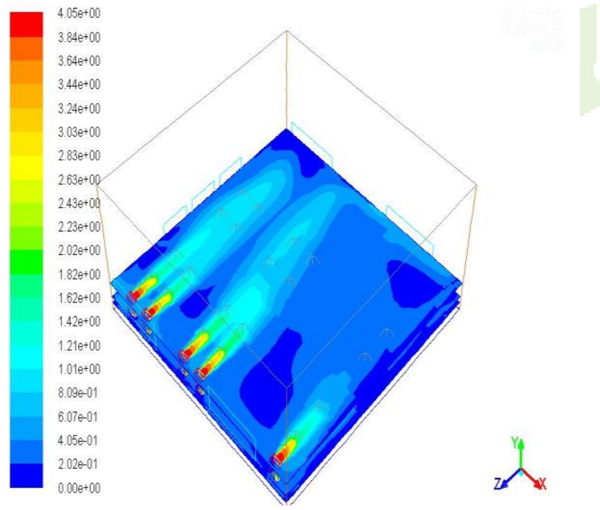

Fig.12: Contours of velocity magnitude for Basic Case1 at y-plane.

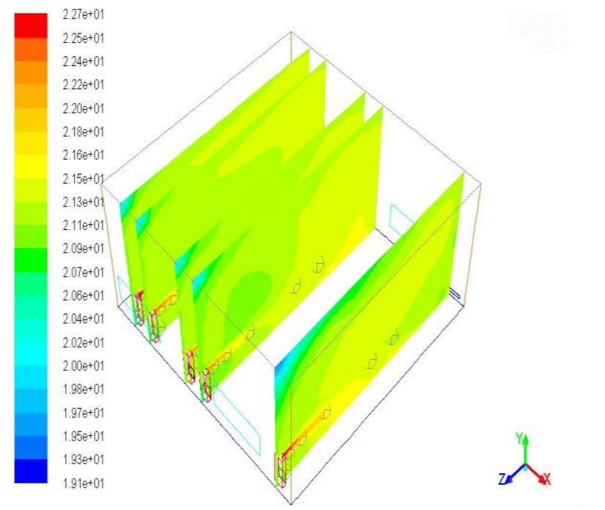

Fig.15: Contours of Relative Humidity for Basic Case1.

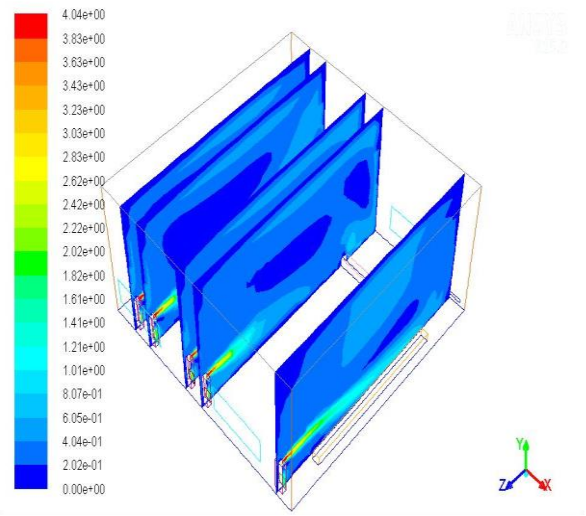

Fig.16: Contours of velocity magnitude for Cooling Case1.

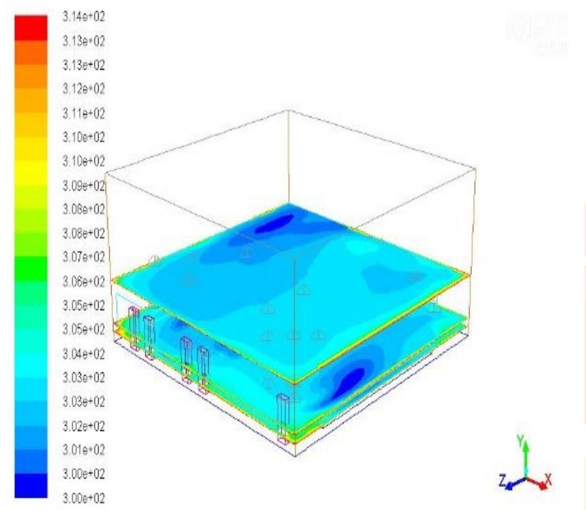

Fig.17: Contours of static Temperature for Cooling Case1.

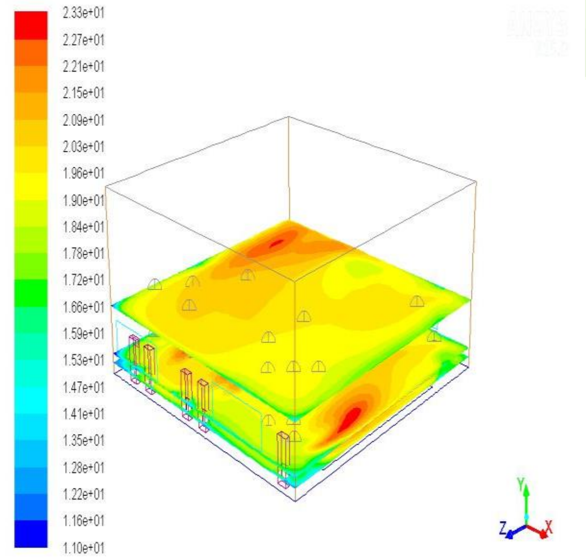

Fig.18: Contours of Relative Humidity for Cooling Case1.

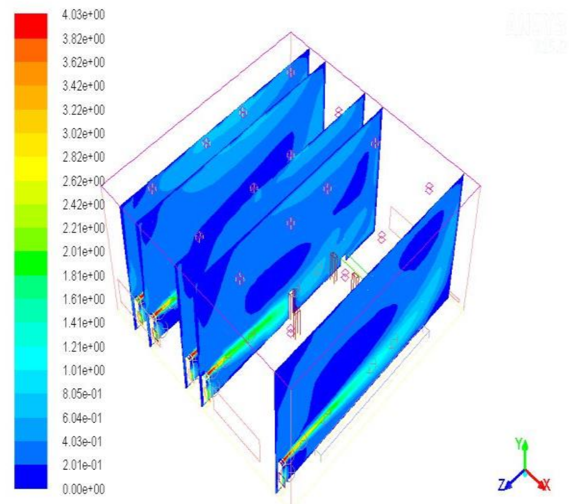

Fig.19: Contours of velocity magnitude for Event Case1.

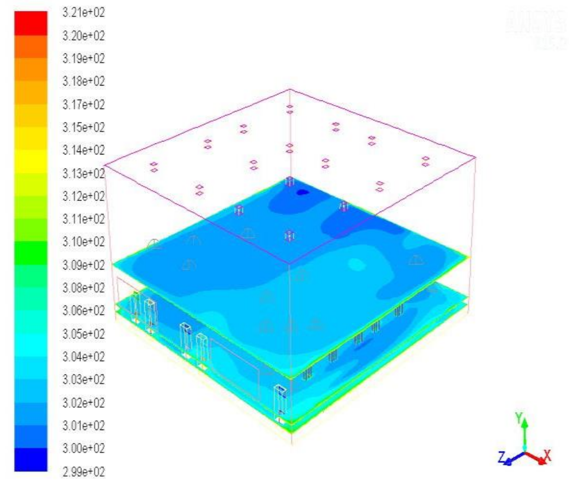

Fig.20: Contours of static Temperature for Event Case1.

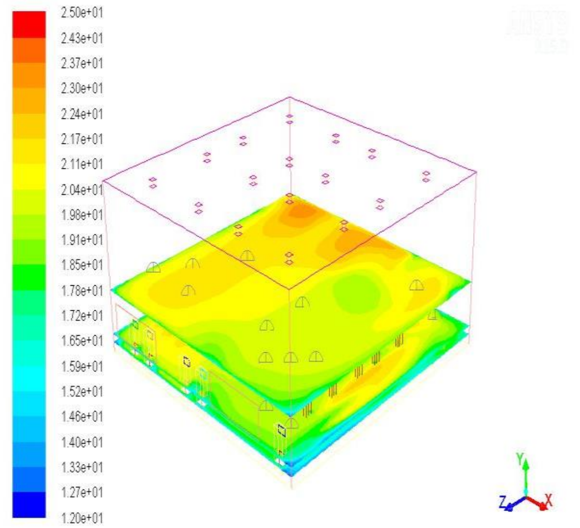

Fig.21: Contours of Relative Humidity for Event Case1.

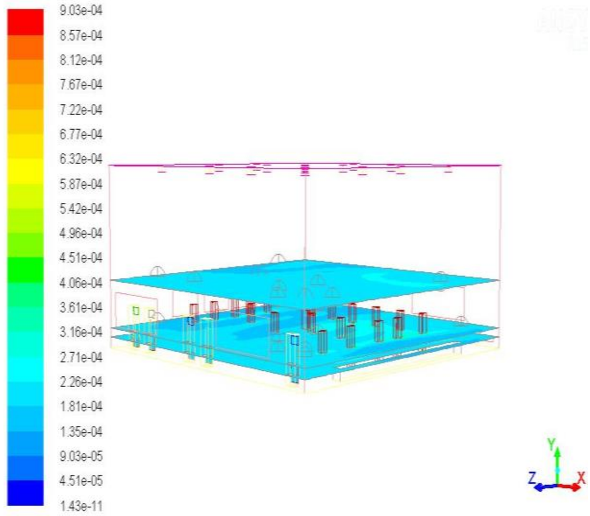

Fig.22: Contours of Co2 concentration for Event Case1.

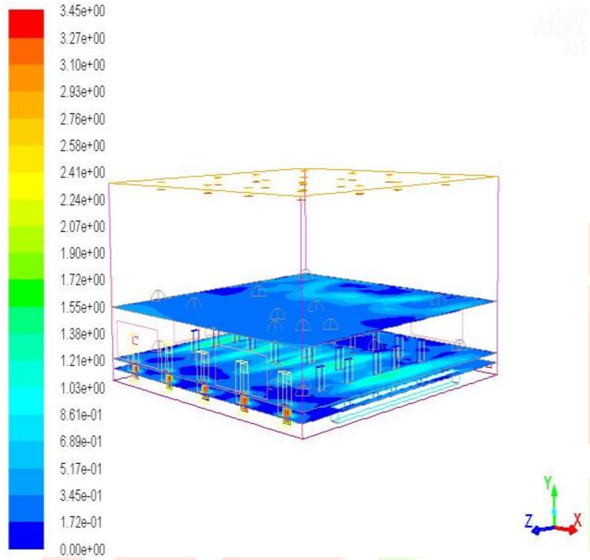

Fig.23: Contours of velocity magnitude for Case 2.

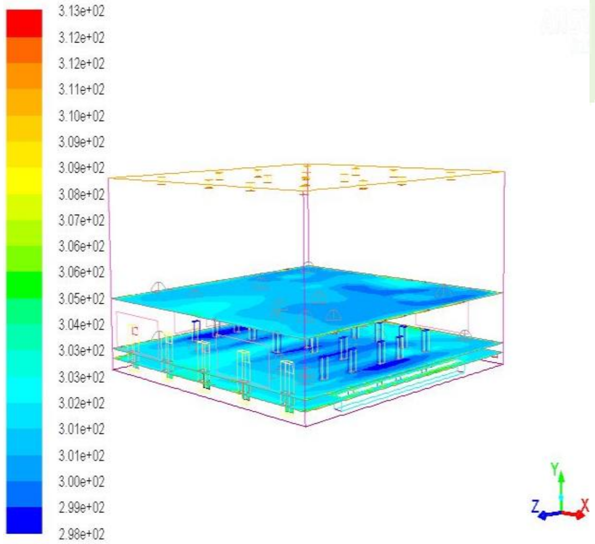

Fig.24: Contours of static Temperature for Case 2.

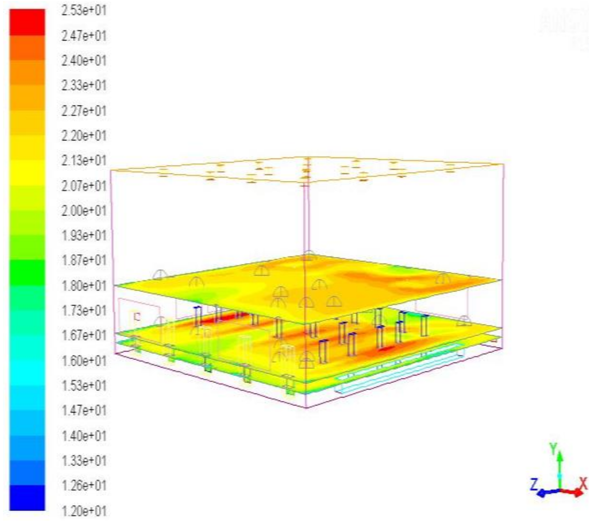

Fig.25: Contours of Relative Humidity for Case 2.

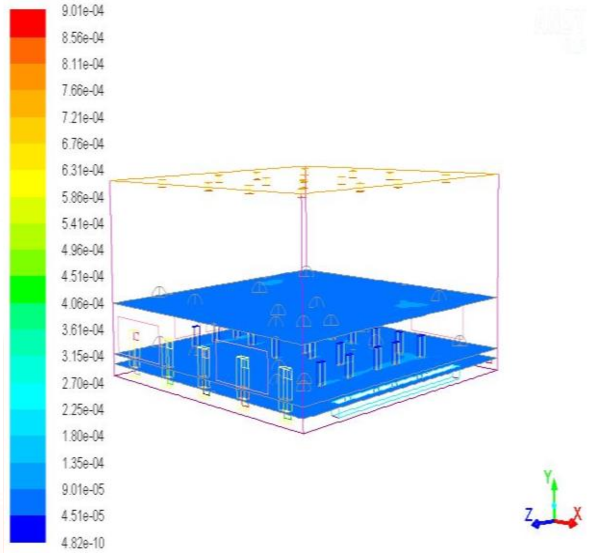

Fig.26: Contours of Co2 concentration for Case 2.

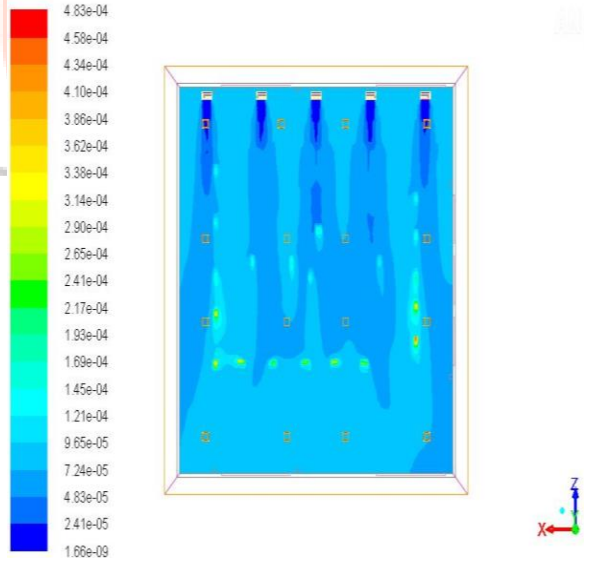

Fig.27: Contours of Co2 concentration for Case 2 at x-z plane.

Nomenclature:

(CFD) **C**omputational **F**luid **D**ynamics.

(HVAC) **H**eating **V**entilating **A**ir **C**onditioning.

(IAQ) **I**ndoor **A**ir **Q**uality.

(ρ) Density.

(v) Velocity vector.

(Γ) Effective exchange coefficient of Ø.

(Sø) source rate per Unit volume.

(CFM) ft^3/min

(CMM) m^3/min

(ADPI) **A**ir **D**iffusion **P**erformance **I**ndex.

(EDT) **E**ffective **D**raft **T**emperature.

(C1) Case1; **(BC1)** Basic Case1; **(CC1)** Cooling Case1; **(EC1)** Event Case1; **(C2)** Case2.

(exp) experimental; **(Nu)** numerical.

References

[1] K. Ponsoni and M. S. G. Raddi, (2010) "**Indoor Air Quality Related to Occupancy at an Air-conditioned Public Building**", Araraquara – SP Brazil.

[2] Versteeg H.K., Malalasekera W., "**An Introduction to Computational Fluid Dynamics**", Second Edition, England, (2007).

[3] Awabi H.B., "**Ventilation of Buildings**", London, New York, Taylor, Francis, (2005).

[4] ANSI/ASHRAE Standard 62.1 "**Ventilation for acceptable air quality**" Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, Canada, U.S., (2004).

[5] Stathopoulou O.I., Assimakopoulos V.D., "**Numerical Study of the Indoor Environmental Conditions of a Large Athletic Hall Using the CFD Code PHOENICS**", Athens, Greece, (2007).

[6] Chen Q., Glicksman L., "**Performance Evaluation and Development of Design Guidelines for Displacement Ventilation**", Massachusetts Avenue, Cambridge, (1999).

[7] ANSYS, Inc. (2013), Users Guide, 15 version.

[8] User's Guide of EXTECH, INSTRUMENTS, **Data logging / Printing Anemometer + Psychrometer,** Model 451181.

[9] Rusly E., AIRAH M., Gagliardini S., "**The truth about the Air Diffusion Performance Index (ADPI)**", TROX Australia, (2014).

[10] Bartak M., Cermak M., Clarke J.A., Denev J., Drkal F., Lain M., Macdonald I., Majer M., Stankov P., "**Experimental and numerical study of local mean age of air**", Proceedings of the 7th International Building Performance Simulation Association Conference, (2001).

[11] Alfano F.R.d., Palella B.I., Riccio G., "**A FRIENDLY TOOL FOR THE ASSESSMENT OF THERMAL ENVIRONMENTS**", Fisciano, Napoli, Italy, (2005).

[12] Baker N., Standeven M., "**Thermal comfort for free-running buildings**", Cambridge, UK, (1996).

[13] Xinga H., Hattonb A., Awbic H.B., "**A study of the air quality in the breathing zone in a room with displacement ventilation**", Oxford, UK, (2001).