Numerical and Experimental study of the indoor environmental Conditions of a large Athletic hall (wrestling hall) using ansys 15

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Abstract

This study investigates numerically and experimentally the environmental conditions prevailing in a large mechanically ventilated athletic hall, with the aid of the computational fluid dynamics by ANSYS 15 program. The indoor space of the building was simulated in the ANSYS 15 program environment and the model results were validated against experimental data collected during a 7-day campaign in the hall. The measurements included airflow characteristics at different locations of the indoor space, as well as surface temperatures of the indoor materials. Having obtained good agreement between experimental and numerical results, different scenarios were applied in the model to investigate the environmental conditions prevailing in the hall under different ventilation and occupational conditions. These regard air-conditioning and cooling modes, as well as empty and full hall during an athletic event. The velocity, temperature and relative humidity were studied and results revealed dynamic behavior of the fields, significantly altering with the different considered cases.

Keyword: Large Enclosure, Athletic Hall, CFD model, IAQ.

الخلاصة

هذه الدراسة تتحرى بشكل عددي و تجريبي الشروط البيئية التي تسود في قاعة رياضية كبيرة و مكيفة ميكانيكيا ، وتتم هذه الدراسة بواسطة ديناميك السوائل الحسابية (CFD) ضمن برنامج (ANSYS 15) الهندسي ، الفضاء الداخلي للبناية تم بنائه بواسطة برنامج (ANSYS 15) و ان النتائج العددية النموذجية في البرنامج تم مطابقتها و تصديقها مع البيانات التجريبية التي جمعت اثناء حملة سبعة أيام في القاعة الرياضية ، تضمنت المقاييس خصائص التيار الهوائي في المواقع المختلفة من الفضاء الداخلي بالإضافة الى درجات الحرارة السطحية من المواد الداخلية وقد حصل التوافق الجيد بين النتائج التجريبية و العددية ، المخططات المختلفة طبقت في النموذج لتحري الشروط البيئية التي تسود في القاعة تحت التهوية المختلفة و الشروط المهنية ، التي تتعلق بأنماط التبريد المختلفة بالإضافة الى القاعة الفارغة والكاملة اثناء الحدث الرياضي ، تم دراسة سرعة التيار الهوائي و درجة الحرارة والرطوبة النسبية بشكل ملحوظ في الحالات المدروسة المختلفة .

(1)Introduction

The Computational fluid dynamics (CFD) codes have become an important tool in the research field of air quality, in both the indoor and outdoor environments.

They are currently applied for investigations of indoor airflow fields for building design and optimum ventilation purposes and for pollutants dispersion in working areas for health and safety reasons. However, few studies combine theoretical and experimental methods to investigate air quality in stadiums and athletic halls, where a large number of people are present during events and athletes train and compete. Proper ventilation and supply of fresh air play a significant role in the control of indoor air quality and thermal comfort given that metabolism is intense due to the overcrowding of people. [1]

The last decade has been characterized by a significant increase of worldwide scientific database in indoor environments. People recognize that indoor air quality may be more important than outdoor air quality because they spend over 70% of their time indoors. Applications of heating, ventilation and air conditioning system are known to modify the indoor air quality by means of filtration, humidification, dilution and cooling the outdoor air entering the occupied space. For instances, adequate filtration of outdoor air intake of air-conditioning system through a well-maintained filter can be effective in preventing outdoor microbial contamination associated with outdoor sources of environmental microbes in air-conditioning buildings .Air-conditioning system has also been shown to contribute to the rising fungal contamination in indoor air from various components in the system. [5]

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.

(2) Methodology
(2.1) Experimental Procedure

A 7-day experimental campaign in the frame of a research project is accomplished in an indoor wrestling hall within the athletic Education College, University of Basrah. This hall is surrounded with 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 area 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.

The instrument used is (Data logging / Printing Anemometer + Psychrometer) [8] this device simultaneously measures and displays air velocity, temperature, humidity, wet bulb temperature, and air Volume (CFM / CMM). Surface temperatures of indoor materials were measured with an infrared thermometer. This can be seen in Figures (6and7).

(2.2)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:

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

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

Where; ρ, v, Γ and $S_{\emptyset}$ are density, velocity vector, effective exchange coefficient of∅, and source rate per Unit volume, respectively, for a solved for variable ∅. [2]

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 turbulence model is applied, while buoyancy effects are considered. To improve convergence, under relaxation 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 independency, and runtime saving, due to the high complexity of the domain geometry as can be shown in Figures (2and3).

(2.3) 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 CFD program was validated by comparing the flow patterns, vertical profiles of temperature, velocity, relative humidity and turbulence intensity of the measured data and computed results for a Large Athletic Hall (Wrestling Hall). This can be seen in (Fig. 4).

Basic case: It corresponds to a selected day from the experimental campaign. The hall is empty and the HVAC system operates in the air-conditioning mode,
Without cooling.

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

Event case: The hall is half-full with 20 spectators attending an athletic event and ventilation conditions are the same as in the basic case.

Model configurations concerning boundary and initial conditions, as well as settings information of the cases studied, are given below: Fresh air comes in the hall via split unit fans (air inlets of Figs. 1and5), the dimensions of inlet are 0.3m x 0.5m, with mean axial z-velocity of 4m/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).

(3) Design and performance of thermal comfort.

There are several factors that can account for the design and performance of thermal comfort, one of them:

Air Diffusion Performance Index (ADPI), can be calculated from the following general relationships: [9]

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

Where: *(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), must be (V_x<0.36 m/s).

$$ADPI = \frac{N_\theta}{N} * 100\% \text{--------------- (3)}$$

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

The performance of an air distribution system within a room/zone can be rated in terms of ADPI (the Air Diffusion Performance Index). 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 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 a space has to be assessed and verified on-site. This on-site measurement should be conducted in accordance with 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]

Fig. 1 Domain of the athletic hall

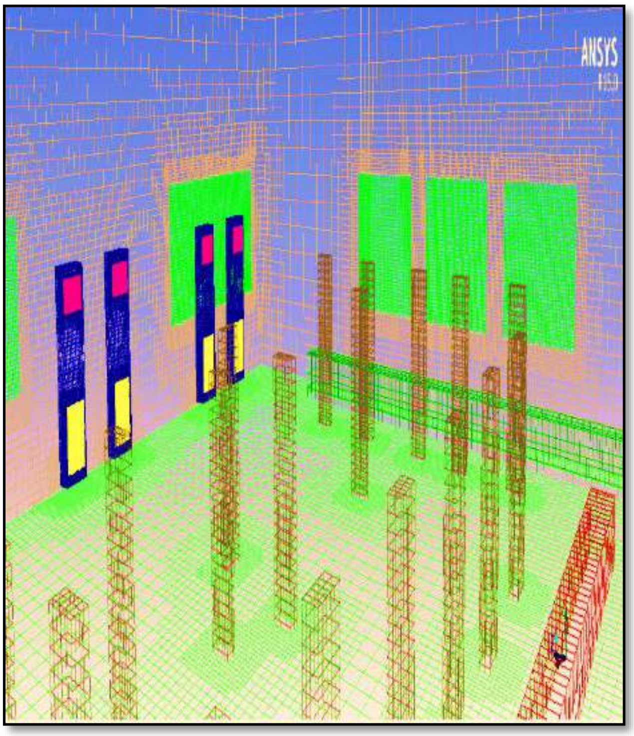


Fig. 3 Plane view of the geometrical grid for the Event Case.

Fig. 2 Plane view of the geometrical grid For the Basic Case.

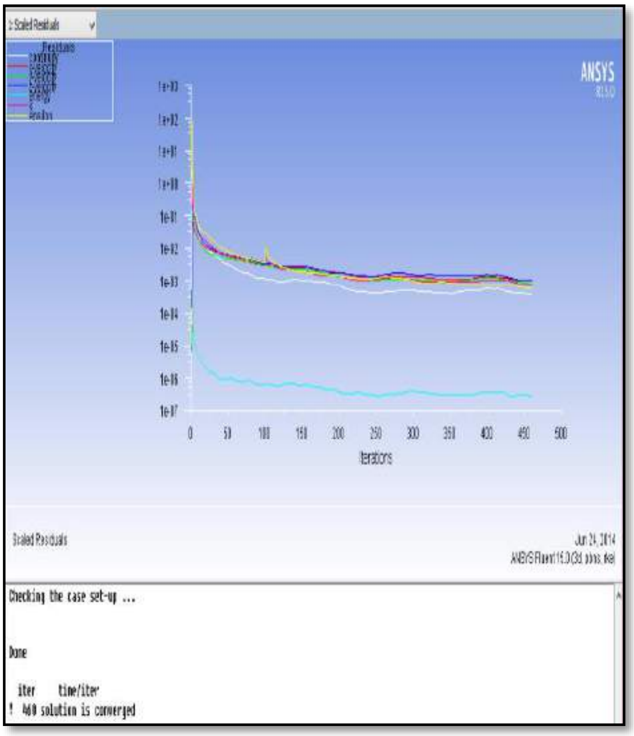


Fig.4 Validation of the Basic Case.

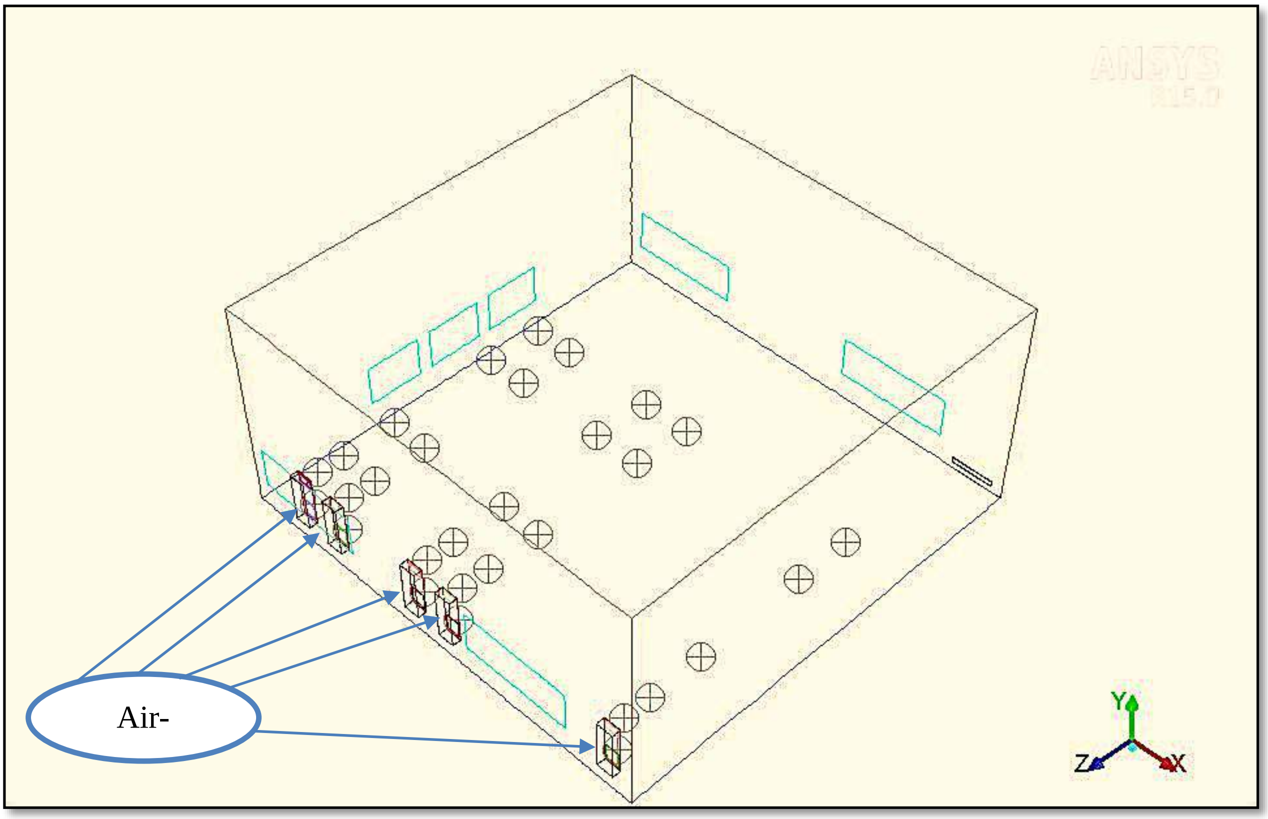


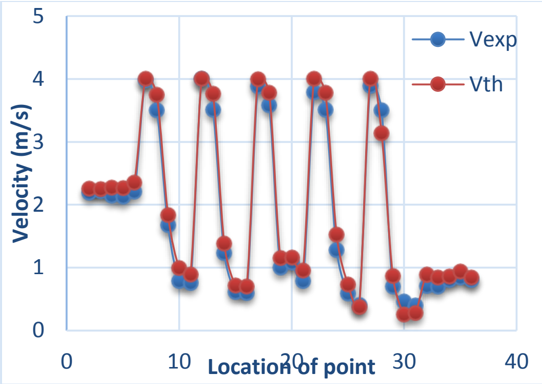
Fig. 5 Plane view of some points for Basic Case

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

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

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

Point No.	Locations			V_{exp} m/s	V_{th} m/s	T_{exp} k	T_{Nu} k	RH_{exp} %	RH_{Nu} %
	X(m)	Y(m)	Z(m)						
points average of outlet 1				2.195	2.2624276	313.8	313.90848	20.6	21.063747
points average of outlet 2				2.221	2.2513804	314	313.90964	21	21.062483
points average of outlet 3				2.151	2.2722647	313.5	313.86801	21	21.110161
points average of outlet 4				2.131	2.2648635	313.9	314.00629	21	20.957156
points average of outlet 5				2.211	2.3529201	314	313.84778	21	21.132594
1	1.75	1.75	15	3.981	4.0036712	313.2	312.60104	21.6	22.584687
2	1.75	1.75	14	3.506	3.7488418	313	312.68161	21.8	22.487541
3	1.75	1.75	12	1.677	1.8409307	313.5	313.26901	21.1	21.793309
4	1.75	1.75	8	0.788	0.9938601	312.9	313.50818	21.7	21.51766
5	1.75	1.75	6	0.759	0.8949071	312.9	313.52466	20.8	21.498837
6	3.25	1.75	15	4.004	4.0030689	314	312.80087	21.6	22.344561
7	3.25	1.75	14	3.513	3.7641978	313	312.86511	21.9	22.268002
8	3.25	1.75	12	1.234	1.3831714	312.9	313.46289	21.2	21.569553
9	3.25	1.75	8	0.611	0.7159041	313.4	313.6492	20.9	21.356991
10	3.25	1.75	6	0.598	0.7013043	313.7	313.63037	21.7	21.378379
11	6.75	1.75	15	3.881	4.003036	313.5	313.00079	21.8	22.107236
12	6.75	1.75	14	3.588	3.7865133	313.6	313.04742	21	22.052343
13	6.75	1.75	12	1.001	1.1597604	313.4	313.63239	21	21.376073
14	6.75	1.75	8	1.087	1.160279	313.5	313.65195	21	21.353878
15	6.75	1.75	6	0.789	0.955303	313.7	313.65424	21.2	21.35129
16	8.25	1.75	15	3.793	4.0056858	313	312.90091	21.8	22.225451
17	8.25	1.75	14	3.511	3.7837541	313.1	312.96408	21.4	22.150633
18	8.25	1.75	12	1.281	1.5217694	313.5	313.60651	21	21.405497
19	8.25	1.75	8	0.586	0.7325229	313.9	313.68277	20.6	21.318963
20	8.25	1.75	6	0.409	0.3737494	314	313.63708	21	21.370752
21	14.75	1.75	15	3.901	4.0043802	313	312.50131	21.9	22.705635
22	14.75	1.75	14	3.501	3.1413331	313	312.77518	21.6	22.375293
23	14.75	1.75	12	0.701	0.8661697	313.2	313.39767	21.5	21.64454
24	14.75	1.75	8	0.461	0.2566139	313.6	313.61172	21	21.399586
25	14.75	1.75	6	0.401	0.2766746	313.7	313.69684	20.7	21.303037
26	1.75	0.5	15	0.712	0.8897354	314	313.84055	20.5	21.141085
27	3.25	0.5	15	0.698	0.8444201	314.1	313.83344	21	21.149071
28	6.75	0.5	15	0.801	0.8614534	313.8	313.84171	20.7	21.139804
29	8.25	0.5	15	0.842	0.9377113	313.6	313.97055	21	20.995869
30	14.75	0.5	15	0.803	0.8408837	314	313.83591	20.1	21.146329



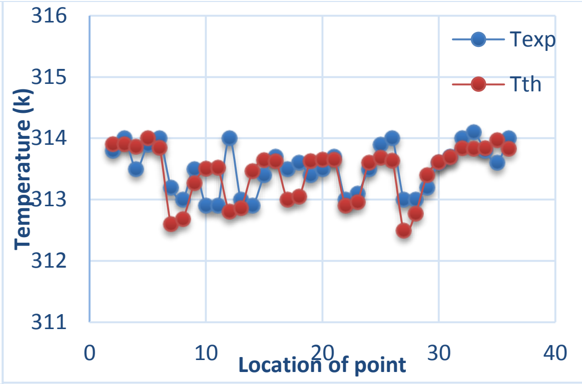


Chart .1 compare between V_{exp} & V_{Nu}. **Chart .2 compare between T_{exp} & T_{Nu}.**

Table 3: simulated air velocities and temperatures and relative humidity at CFD points of the domain for the Event Case.

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

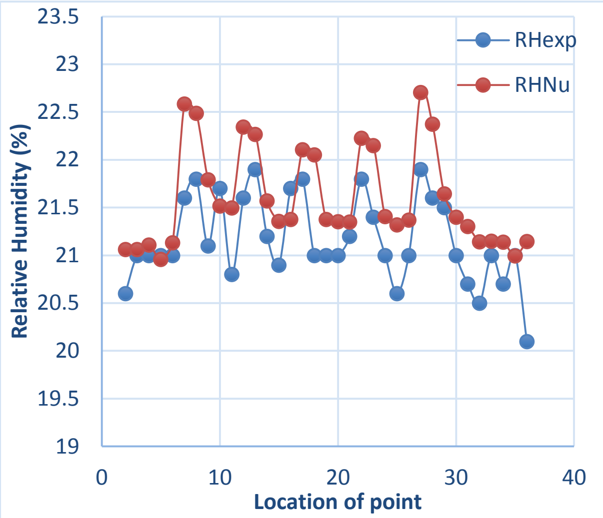


Chart .3 compare between RH_{exp} & RH_{Nu}.

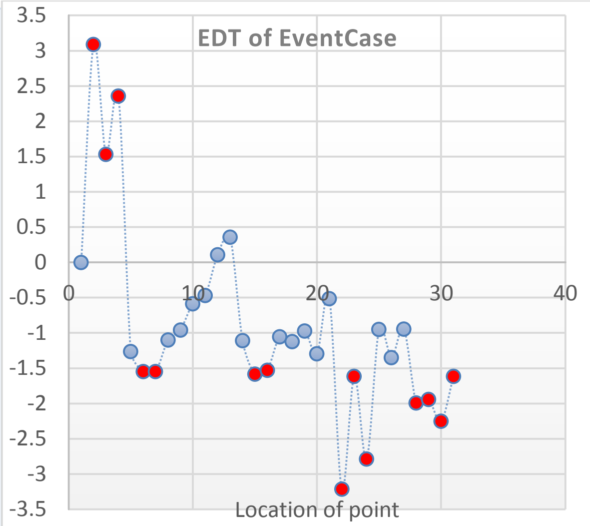


Chart .4 (EDT) for Event Case.

(4) Results and Discussion
(4.1) Basic Case

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 supply are (4 m /s) and (5 m/s), but in this case chosen only (4 m/s).The comparison experimental data and numerical result, seems a good agreement as given by the table (2), which includes the experimental and numerical results for several points and for the points at the arena, for the two studied inlet velocities, Comfort of human feeling. It was observed that the temperature differences are between (1-2°C) in large halls as shown in figures (11 and 12). The result of the studied hall are compared with graphical figures (8 and 9) of velocity contour 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.

(4.2) Cooling Case

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 velocity in the hall is standard velocities are (4m/s and 5m/s) for the type split unit to better indoor air mixing. has been taking the experimental results and the theoretical to this case , as in the basic case, for the purpose of validation of result , and then took the points of the program (ANSYS15, CFD, Fluent15) on levels (y=0.5,y=0.9,y=3) velocity of less than (0.36 m/s) for the purpose of calculating (ADPI)as in the event case and the value when velocity supply (5 m/s) is (ADPI=53), which indicates the presence of errors in the design of the cooling system of the sports hall, which needs to raise the velocity of the air to (7.5 m/s standard velocity for large hall according to ASHRAE standards [4]) or redistributed air conditioning split unit according to an engineering manner for get thermal comfort . This can be seen in fig.14, 15 and16.

(4.3) Event Case

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) for the purpose of obtaining (ADPI) according to the requirement to calculate (EDT). The value was (ADPI=53%).The result of this case is shown in table (3). 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 enhanced . This indicates that there is lock in the air-conditioning load of the tested hall, and the air distribution system applied is not favourable. See the chart (4) and figures (17, 18 and 19) which are related to this case.

(5) 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. By a little improvement in these two systems, it is found that the (ADPI) is improved from (53%) to more (80%).

3- It is recommended for this college ,that ,if the air conditioning split unit are redistributed according to an engineering manner ,and may be 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 .

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Nomenclature:

(CFD) Computational Fluid Dynamics.

(HVAC) Heating Ventilating Air Conditioning.

(IAQ) Indoor Air Quality.

(ρ) Density. (Kg/m^3)

(V) Velocity vector. (m/s)

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

Fig.6 infrared thermometer

Specifications	Range	Resolution	Accuracy
ft/s	60 to 4930ft/s	1ft/s	±3% rdg
m/s	0.4 to 25m/s	0.1m/s	±3% rdg
Temperature	-4 to 144°F/-20 to 60°C	0.1°F/°C	±1°F/°C
Humidity	0 to 100%RH	0.1%	±3%
CFM/CMM	0 to 99999	0.1/1	±3%
Wet Bulb	-7.6 to 158°F/-22 to 70°C	0.1°	
Dimensions	8.1 x 2.75 x 2" (208 x 70 x 53mm)		
Weight	9oz (260g)		

Fig. 7 Anemometer + Psychomotor

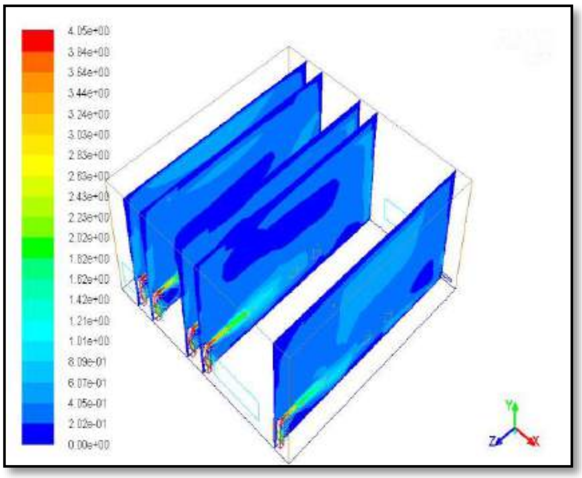


Fig.8: contours of velocity magnitude for Basic Case at x-plane.

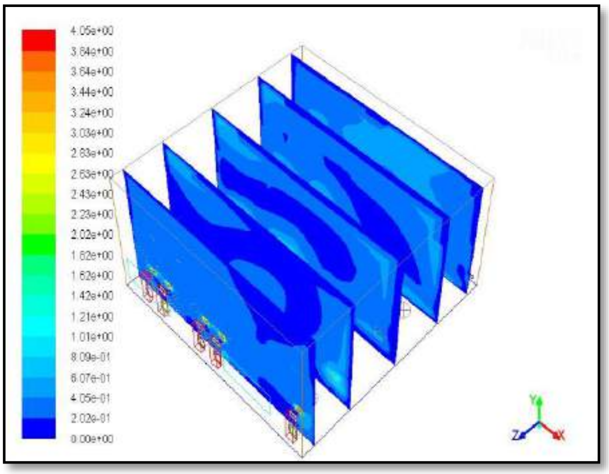


Fig.9: contours of velocity magnitude for Basic Case at z-plane.

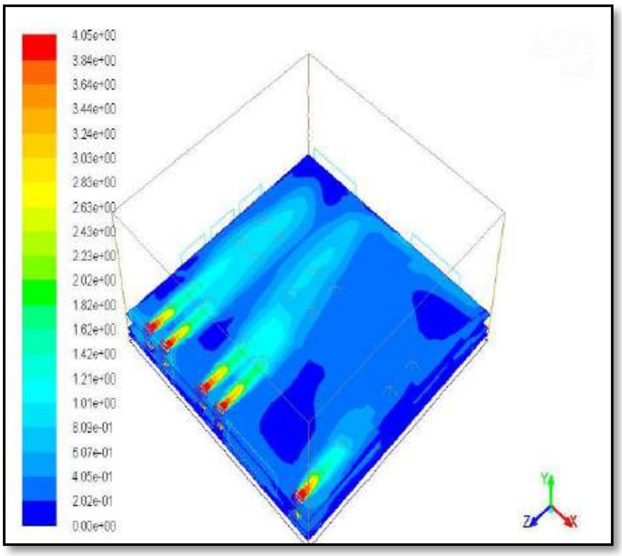


Fig.10: contours of velocity magnitude for Basic Case at y-plane.

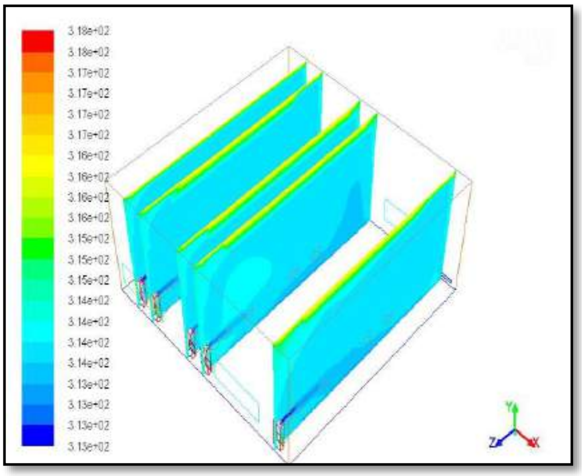
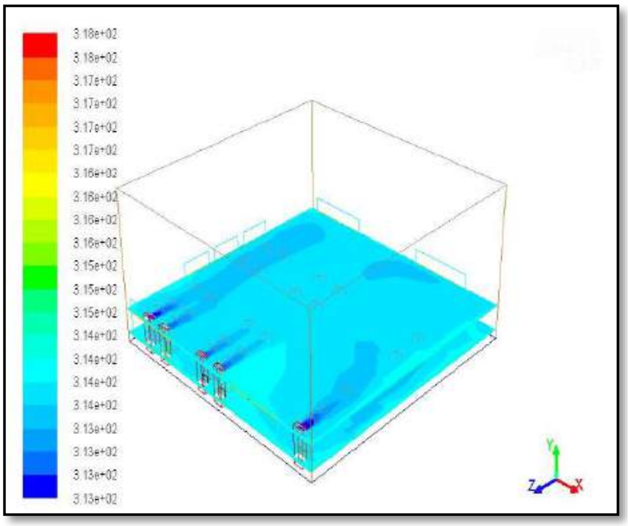


Fig.11: contours of static Temperature for



Basic Case at x-plane. Fig.12: contours of static Temperature for Basic Case at y-plane.

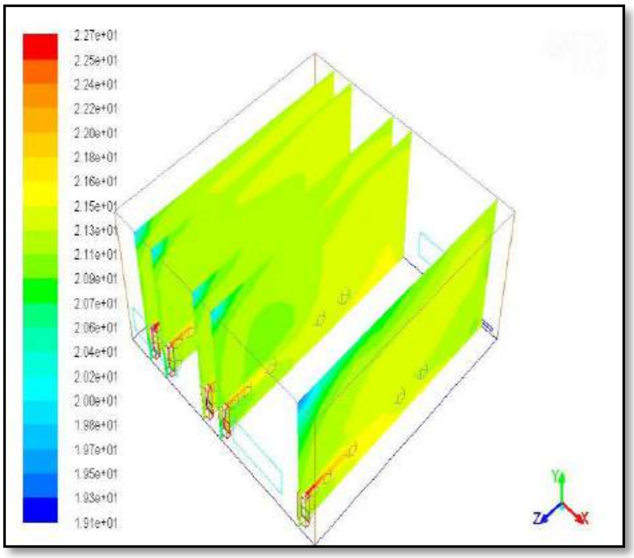


Fig.13: contours of Relative Humidity for Basic Case.

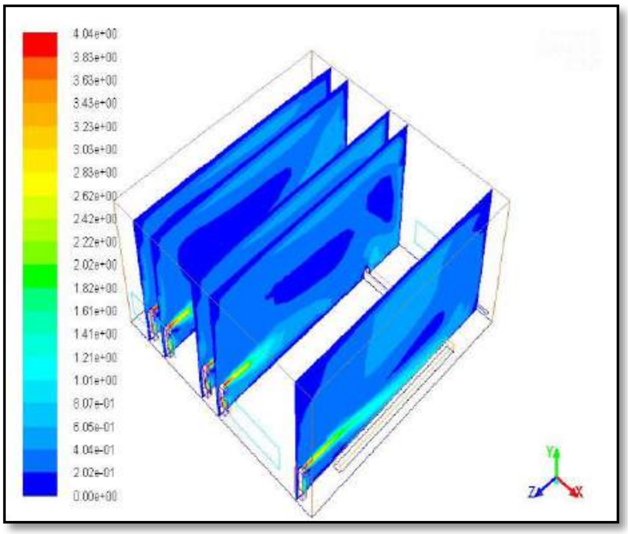


Fig.14: contours of velocity magnitude for Cooling Case.

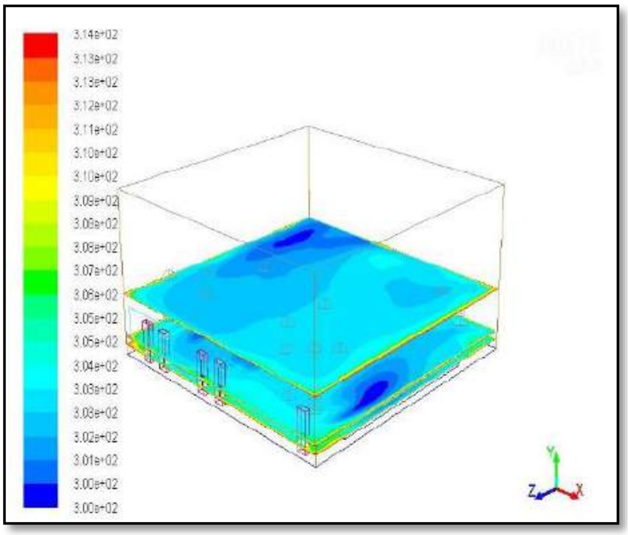


Fig.15: contours of static Temperature for Cooling Case.

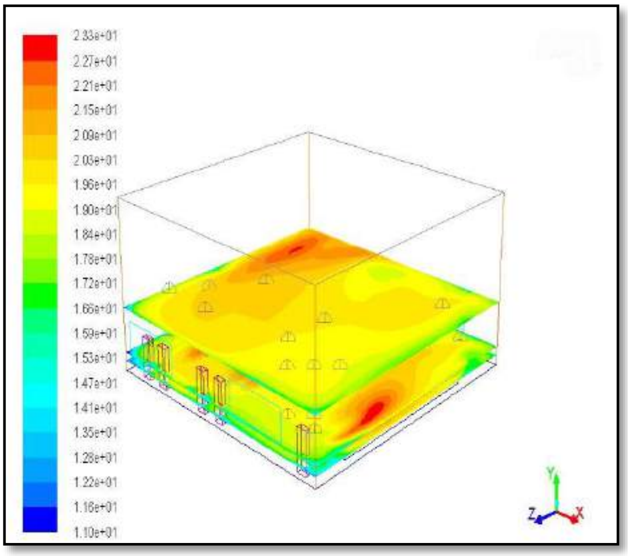


Fig.16: contours of Relative Humidity for Cooling Case.

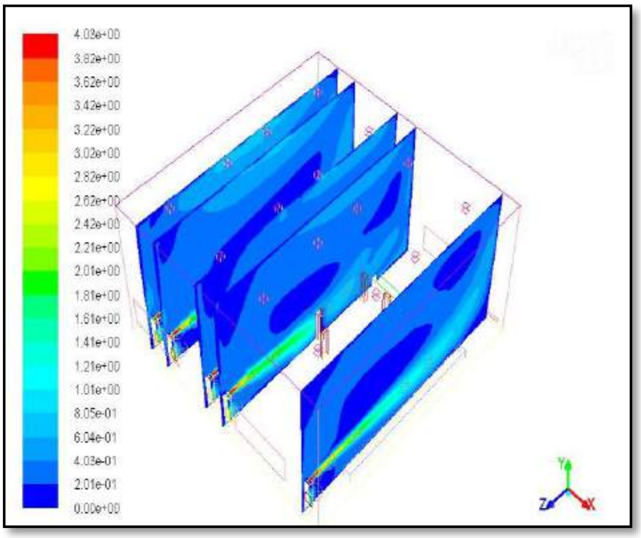


Fig.17: contours of velocity magnitude for Event Case.

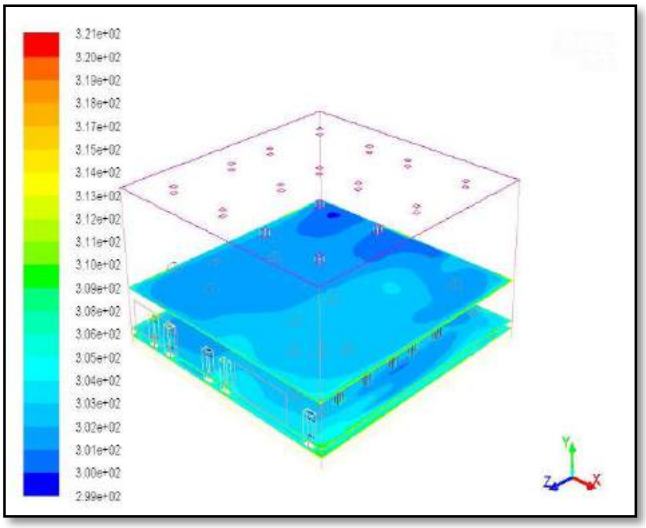


Fig.18: contours of static Temperature for Event Case.

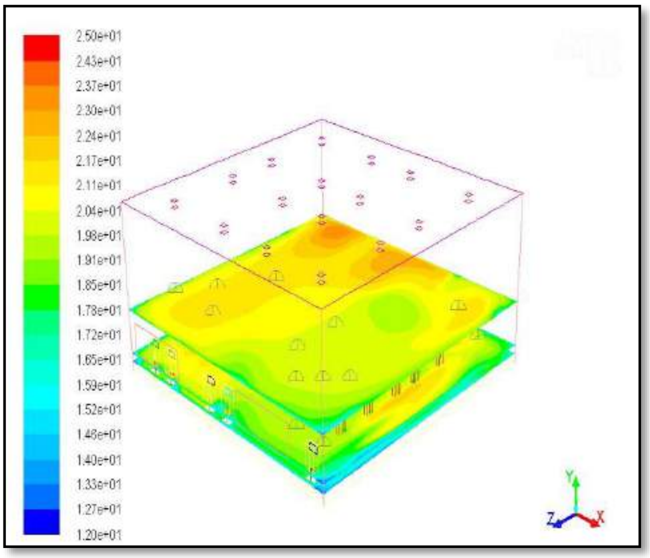


Fig.19: contours of Relative Humidity for Event Case.