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ASSESSMENT OF MATHEMATICAL MODELLING OF HEAT TRANSFER AND AIR FLOW PATTERNS INSIDE AN AIR-CONDITIONED MOVIE THEATRE

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SUMMARY

The present paper is devoted to investigate the proper design of air supply system in a large air-conditioned amphitheatre. The paper deals with numerical investigations of the influence of ventilation and air conditioning supply and extract diffusers/grills on air flow pattern and the temperature in a large movie theatre. This work focuses on the analyses of air flow patterns and thermal behavior in movie theatres where large number of audience that are seated. The effectiveness of an air flow system is commonly determined by the successful removal of sensible and latent loads from occupants, equipment etc in a large theater. Fresh air supply to the theatre is expected to replace warm and void air and to reduce thermal load in the zone in question. This is targeted to attain desired temperature regulation for human thermal comfort conditions. The present mathematical model is inherent in a packaged Computational Fluid Dynamics (CFD) commercially available FLUENT Code that solves the partial differential equations governing the transport of mass, three momentum, energy and species in a fully turbulent three dimensional domain under steady state conditions. Numerical computations were carried out with more than 600000 orthogonal three dimensional control volumes. Grid nodes were densely located in the vicinity of the heat sources and accounted for sensible and latent heat load from individual humans. The primary objective of the present work is to assess the airflow characteristics, Heat transfer and carbon dioxide concentrations in large air-conditioned configurations with large number of occupants.

The paper indicated good agreement with reported experimental data in full scale geometries. The paper ends with a brief discussion and conclusions statements.

KEYWORDS

Indoor air quality, CO_2 concentration and thermal comfort

INTRODUCTION

This paper attempts to enhance the air conditioning system effectiveness by proposing supply air distributions and optimizing the air supply boundary conditions. This would require comparisons of air and flow properties with prevailing thermal comfort standards values, such as ASHRAE handbook [1] and ANSI/ASHRAE standard 62–2004, [2] .Finally this would enable assessing how far is the proposed design from the required comfort conditions in the occupancy region.

ASHRAE handbook [1] represents the general design criteria for fully air-conditioned commercial and public buildings; “for theatres and auditoriums within the occupied zones, air conditions should be maintained as; relative humidity is held below 55%, temperatures are held within the 68 to 72 °F range, and the air movement below 25 fpm to maintain the concentration of contamination at an acceptable level”. While ANSI/ASHRAE standard 62–2004, [2] , limited the indoor CO_2 concentration up to 1000 ppm in order to satisfy a healthy environment.

CFD MODEL

The present numerical investigation was based on solving the governing equations that described airflow inside the theatre by a CFD program FLUENT 6.2, [3] . This numerical approach solves the partial differential equations governing the transport of mass, three momentum, energy and species in a fully turbulent three dimensional domain under steady state conditions. The standard $k - \varepsilon$ model for turbulence closure was used as appropriate to this type of configurations, Khalil [4] . An orthogonal three-dimensional grid was superimposed on the solution domain and comprised more than 600000 numerical cells. Details of the present numerical technique, governing equations, boundary and inlet conditions can be found in the work of Abdel-Samee [5] , where reference should be made regarding grid independency tests, convergence and numerical stability. The different governing partial differential equations are typically expressed in a general form as:

$$\frac{\partial}{\partial x}\rho U\,\Phi+\frac{\partial}{\partial y}\rho V\,\Phi+\frac{\partial}{\partial z}\rho W\,\Phi=\frac{\partial}{\partial x}\left(\Gamma_{\Phi,eff}\,\frac{\partial\,\Phi}{\partial x}\right)+$$
$$\frac{\partial}{\partial y}\left(\Gamma_{\Phi,eff}\,\frac{\partial\,\Phi}{\partial y}\right)+\frac{\partial}{\partial z}\left(\Gamma_{\Phi,eff}\,\frac{\partial\,\Phi}{\partial z}\right)+S_{\Phi}\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Where ρ is the air density and Φ is the dependent variable, S_Φ = Source term of Φ, and U, V, W are the velocity vectors, and, $\Gamma_{\Phi,\mathrm{eff}}$ is the effective diffusion coefficient. The effective diffusion coefficients and source terms for the various differential equations are listed in the Table 1.

Convergence and Stability

The simultaneous and non-linear characteristics of the finite difference equations necessitate that special measures are employed to procure numerical stability (convergence); these include under relaxation of the solution of the momentum and turbulence equations by under-relaxation factors which relate the old and the new values of Φ as follows,

$$\Phi=\gamma\,\Phi_{\mathrm{new}}+(1-\gamma)\Phi_{\mathrm{old}} \qquad (2)$$

where γ is the under-relaxation factor. It was varied between 0.2 and 0.3 for the three velocity components as the number of iteration increases. For the turbulence quantities, γ was taken between 0.2 and 0.4 and for other variables between 0.2 and 0.6. The required iterations for convergence are based on the nature of the problem and the numerical conditions (grid nodes, under-relaxation factor, initial guess, etc.). So the computational time required to obtain the results is based on many factors. The computational number of iterative steps is selected according space cell (spatial difference) to yield converged solutions, Kameel et al [6] . The validity of the present computational technique was assessed previously in the open literature, for example, Khalil [4,7]; where reference should be made for more detailed readings. A complete validation between the used numerical CFD program (FLUENT) with another experimental results was made by Cheong et al [8]. This validation was based on geometry of air-conditioned lecture theatre; good agreements with experiments were shown.

MODELING OF HUMAN PRESENSE

Due to the difference in human skin surface temperature and indoor environment temperature, there is heat exchange between human body and the indoor environment. In air-conditioned spaces, a steady-state thermal equilibrium is usually maintained between the human body and the indoor environment. So, all metabolic heat production is released. The human body and face is modeled according to the description of figure 1 and treated as a wall at a constant temperature of its skin temperature. The skin temperature is a function of metabolic rate, Abdel Samee [5] ; then the skin temperature is 34 °C according to 60 W/m^2 of metabolic rate ,ASHRAE, [1] generated from human body seated at rest. The human face was assumed to dissipate water vapor mass fraction of 0.042 kg_w / $kg_{d.a}$ at 37 °C (being body core temperature). This was used as moisture gain from the audience respiration to the theatre airflow, and zero diffusive sweat mass flux from skin surface is assumed.

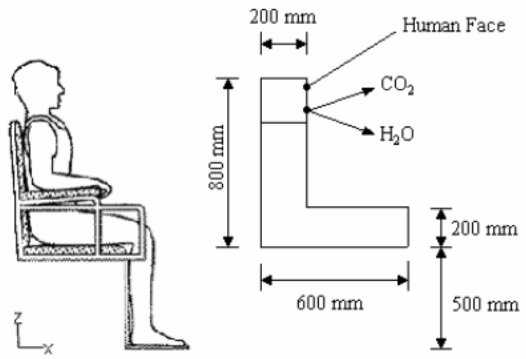


Figure 1: The human body modeling.

ANSI/ASHRAE Standard 62-2004, [2] specified the rate of CO_2 generation from human byproduct respiration with the physical person activity. For the present work, CO_2 concentration in the human respiration equal 0.03125 L_{CO2} /L_a.

GEOMETRICAL CONFIGURQATIONS

The present study was performed on theatre dimensions of 30.0 m long, 20.0 m wide, with varied ceiling height of 10.0 m at front to 8.0 m at the rear. Modeling of the effect of audience presence inside this theatre was developed with full audience loads, which are indicated on the floor and balcony regions as shown in figure 2. The present work describes the air flow characteristics for a given design that incorporated 47 ceiling mounted air supply diffusers (29 of which are mounted at the main ceiling plus 18 diffusers mounted at balcony floor level and 18 extract ports on both side walls. Table 2 summarizes the boundary conditions and related air openings areas for this designed case of the theatre. The present case was designed with 16.23 air changes per hour, air enters theatre volume through air supply ports with CO_2 concentration of 0.00038 (380 ppm), and the surface temperature of all internal walls assumed to be at 298 K.

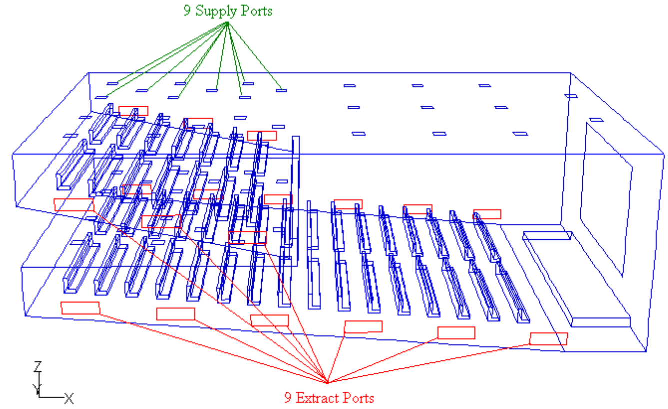


Figure 2: The theatre and seated audience modeling.

Air supply diffusers arrangements at both main ceiling surface and below the balcony surface (sub-ceiling surface) are shown in figure 3.

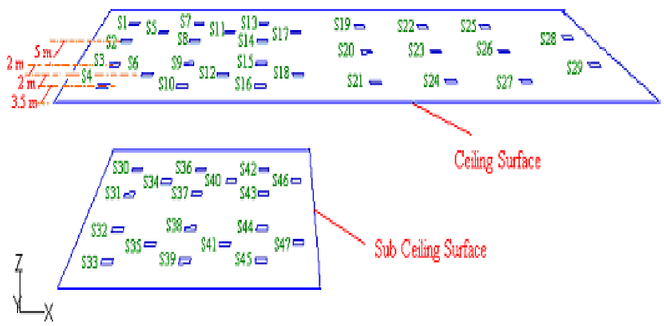
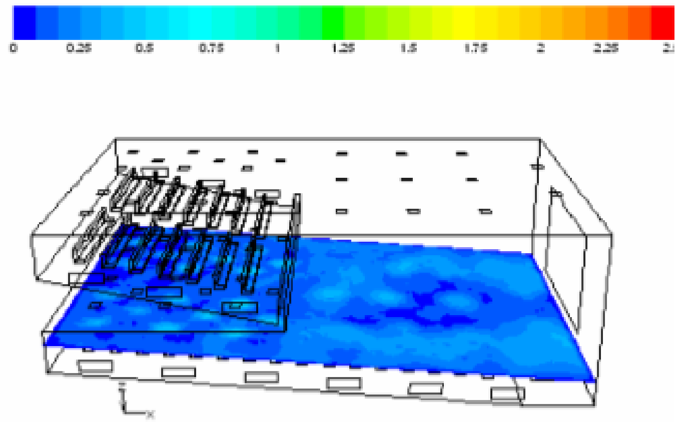


Figure 3: Air supply ports arrangement.

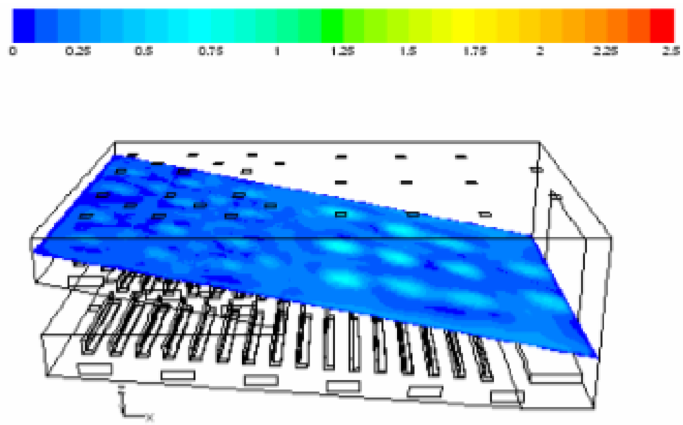
All supply velocities normal to supply grills with 5 % turbulence intensity according to previous recommendations of Khalil [4] .While turbulence length scale of 0.6 m was assumed in accordance with FLUENT [3] , as being the smallest dimension for the supply grill opening.

RESULTS AND DISCUSSION

Figures 4 to 7 presented the complete flow field predictions for air velocity, air temperature, and relative humidity and carbon dioxide concentrations inside the theatre. These predictions were developed at planes parallel to the theatre floor and balcony levels at 1.7 m from the finished floor, in conformity with ASHRAE requirements [1] .

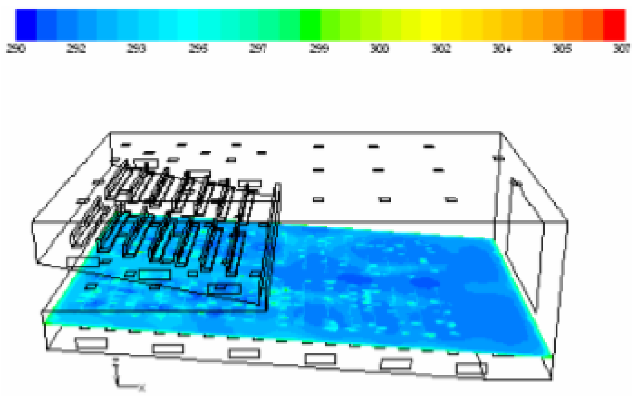


(A) At a plane at 1.7 m above Theatre floor level

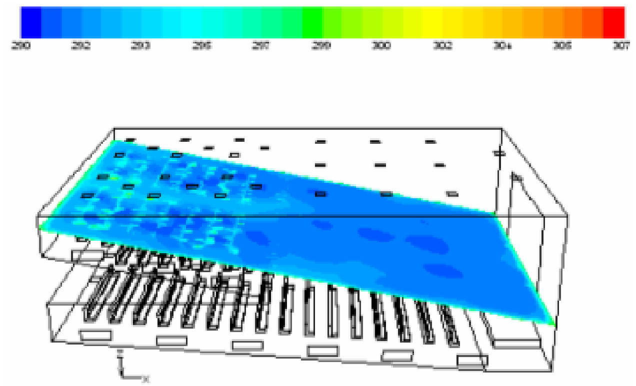


(B) At a plane at 1.7 m above Balcony level.

Figure 4: Velocity magnitude contours (m/s)

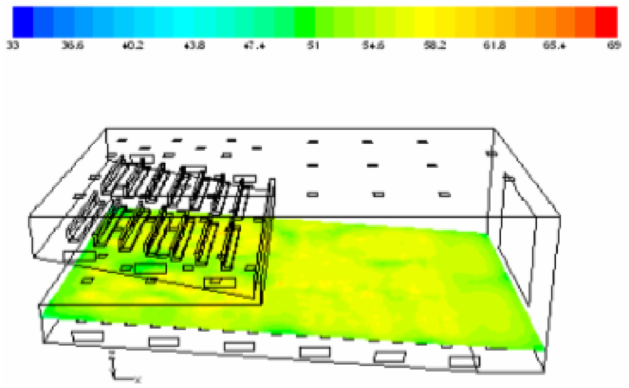


(A) At a plane at 1.7 m above theatre floor level

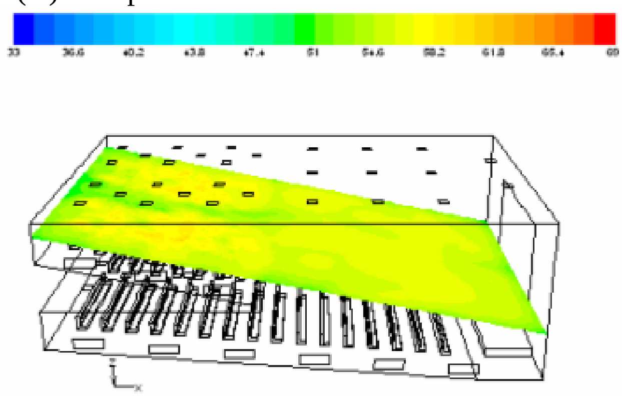


(B) At a plane at 1.7 m above balcony level.

Figure 5: Predicted air temperatures contours (K)



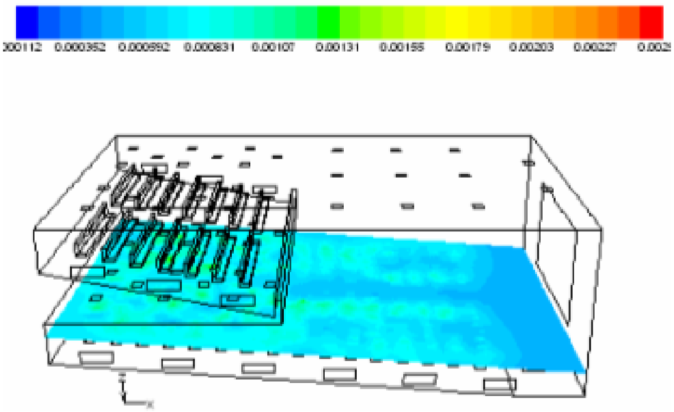
(A) At a plane at 1.7 m above theatre floor level



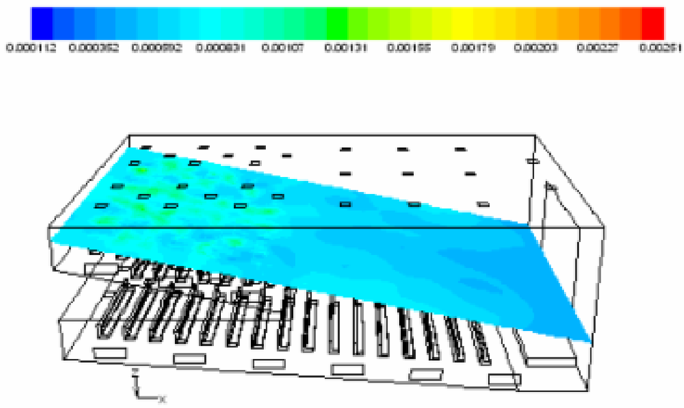
(B) At a plane at 1.7 m above balcony level.

Figure 6: Predicted relative humidity contours, Rh%

As displayed in the previous figures showing air flow characteristics; it was observed that there are uniform flow velocity magnitudes along all occupied zones (balcony and floor zones) except for some limited small stagnant zones. Also the same behavior was identified for the air dry bulb temperature and relative humidity patterns.



(A) At a plane at 1.7 m above theatre floor level



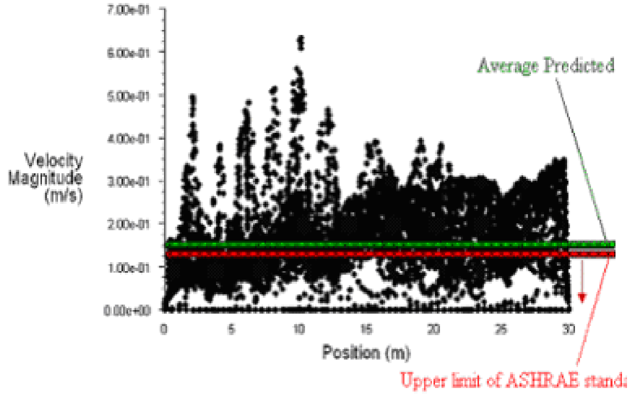
(B) at a plane at 1.7 m above balcony level.

Figure 7: Predicted CO_2 concentration contours

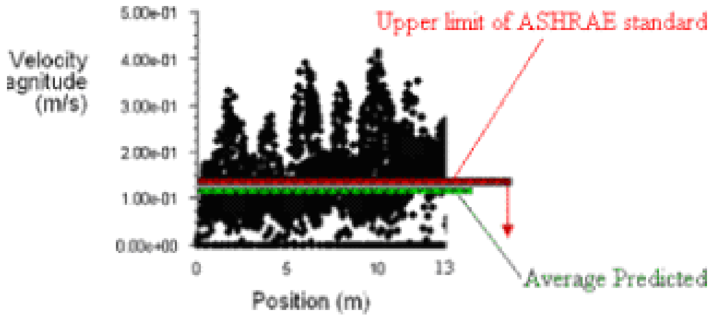
As for the CO_2 concentrations; the predicted CO_2 concentrations along all occupied zones were found to be well below the threshold limit given in ASHRAE recommendations, ASHRAE, [2] .

Occupied zones air properties vs. recommended values

For any air-conditioned space design, a comparison between the predicted air flow properties within the occupied zones and the corresponding ASHRAE comfort values should be performed to assess the attainment of comfort or not. In the next few figures, comparisons between predicted air flow velocities, relative humidity and CO_2 values at different locations within the occupied zones (here at a tilted plane at 1.7 m from the floor and balcony levels) and ASHRAE standard comfort conditions ASHRAE [1,2] are listed. These comparisons are shown in figures 8 to 11 (each of the dots indicated on the figures 8 to 11 represents a predicted value in the planes at 1.7 m from theatre floor level or at the balcony level).

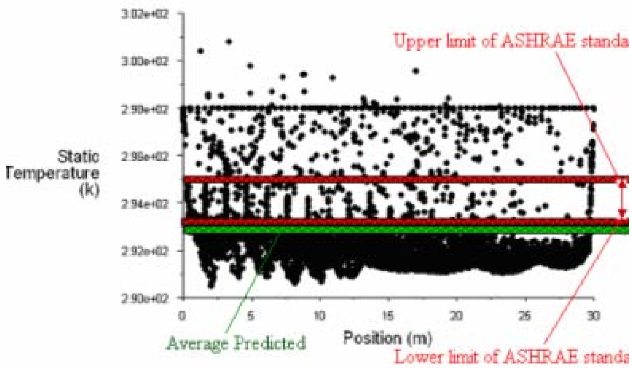


(A) Theatre floor level

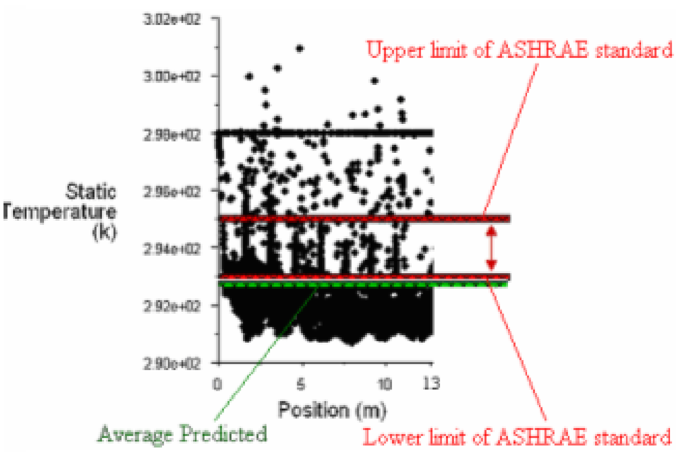


(B) Balcony level

Figure 8: Comparisons between the predicted average velocity and ASHRAE handbook [1] magnitudes in occupied zones.

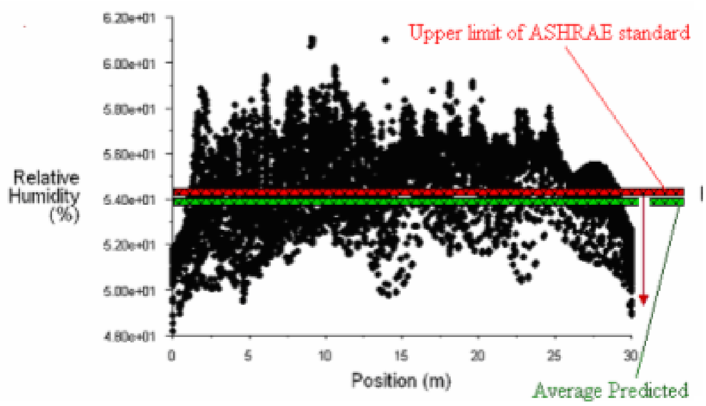


(A) Theatre floor level

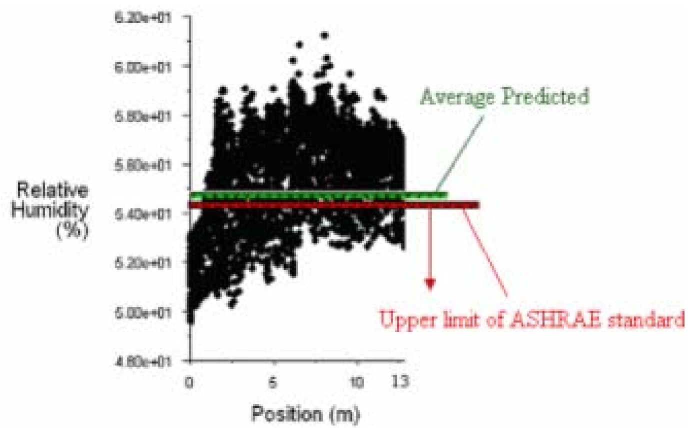


(B) Balcony level.

Figure 9: Comparisons between the predicted average temperature and ASHRAE Handbook [1] magnitudes in occupied zones.

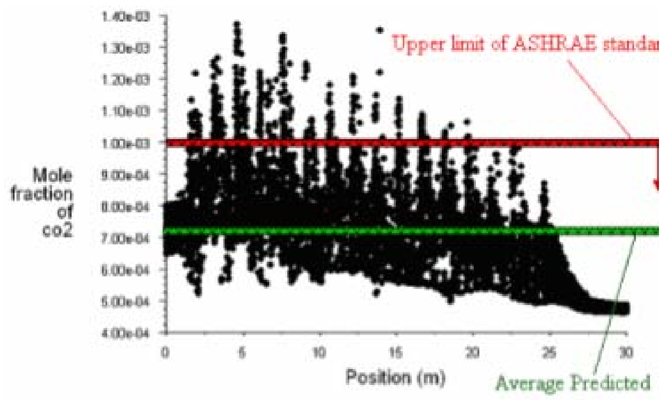


(A) Theatre floor level

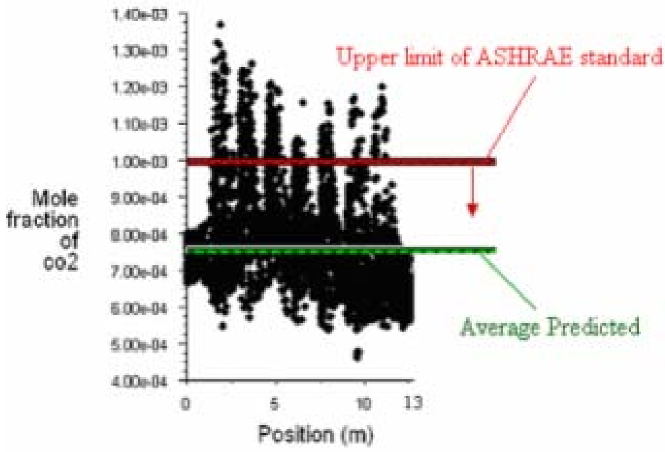


(B) Balcony level

Figure 10: Comparisons between the predicted average relative humidity and ASHRAE handbook [1] magnitudes in occupied zones.



(A) Theatre floor level



(B) Balcony level.

Figure 11: Comparisons between the predicted average CO_2 concentration and ASHRAE standard [2] Magnitudes in occupied zones.

CONCLUSIONS

The reported comparisons between the present predictions of air velocities, temperatures, relative humidity and carbon dioxide concentrations and those of ASHRAE Comfort and CO2 concentrations within the occupied zones; indicated qualitative agreements. There is a general recognizable qualitative agreement between recommended ASHRAE levels and those predicted in the present work. The design configurations shown in figures 2 and 3 yielded better air supply and return air utilization that yielded predicted CO_2 concentration within the occupied zones that are below the maximum allowable limit (1000 ppm). Naturally increasing the number of air return grilles would lead to more uniform air flow distribution and would minimize the stagnant air zones. The results shown in the present work indicated a good agreement between the averaged predicted air velocity, average temperature, average relative humidity and average CO_2 concentration with the ASHRAE recommended comfort conditions within the occupied zones in the theater.

NOMENCLATURE

C_1, C_2 and C_μ Turbulence Model Constants
H Enthalpy, kJ/kg
k Kinetic energy of turbulence, m^2/s^2
Rh Relative Humidity kg/kg
S_Φ Source term of Φ in governing equation
U, V, W velocities in x, y and z directions, m/s
V Velocity vector, m/s
ρ Air density, kg/m^3
$\Gamma_{\Phi, eff}$ Effective diffusion coefficient.
ε Dissipation rate
μ Viscosity, kg.m/s,
Φ General Dependent Variable.
d.a Dry air
W water vapor

ACKNOWLEDGEMENTS

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REFERENCES

[1] ASHRAE Handbook. 2007, HVAC Applications.
[2] *ASHRAE Standard 62-2004*, Ventilation for Acceptable Indoor Air Quality, USA.
[3] FLUENT 6.2 Documentation, © Fluent Inc. 2005.
[4] E.E.Khalil, 2007. Numerical Simulation of Flow Regimes and Heat Transfer Interactions in Complex Geometries. Proceedings, 5th IECEC, St.Louis, AIAA-2007-4726, June 2007.
[5] W.Abdel-Samee,2007, Numerical Investigations of Flow Patterns and Thermal Comfort in Air-Conditioned Cinema Theatres. *MSc. Thesis*, Cairo University.
[6] R.A.Kameel. and E.E.Khalil ,2007. Computer Aided Design of Indoor Air Quality and Air Flow in Surgical Operating Theatres. Proceedings Roomvent 2007, Paper 1217, June 2007.
[7] E.E.Khalil, 2006 Energy Performance of Commercial Buildings in Egypt: A New Direction, Proceedings, Energy2030, Abu Dhabi, 2006.
[8] K.W.D.,Cheong,E. Djunaedy, Y.L., K.W.Chua. , Tham, S.C., Sekhar, N.H., Wong. and M.B .,Ullah. 2003. Thermal comfort study of an air-conditioned lecture theatre in the tropics. Building and Environment 38, pp. 63–73.

Table 1. Terms of Partial Differential Equations (PDE) equation 1.

	Φ	$\Gamma_{\Phi,eff}$	S_Φ
Continuity	1	0	0
X-momentum	U	μ_{eff}	$-\partial P/\partial x + \rho g + S_U$
Y-momentum	V	μ_{eff}	$-\partial P/\partial y + \rho g + S_V$
Z-momentum	W	μ_{eff}	$-\partial P/\partial z + \rho g(1+\beta\Delta t) + S_W$
H-equation	H	μ_{eff}/σ_H	S_H
RH-Equation	RH	μ_{eff}/σ_{RH}	S_{RH}
τ-age equation	τ	μ_{eff}/σ_τ	ρ
k-equation	k	μ_{eff}/σ_k	$G - \rho\,\varepsilon$
ε-equation	ε	$\mu_{eff}/\sigma_\varepsilon$	$C_1\,\varepsilon\,G/k - C_2\,\rho\,\varepsilon^2/k$

$\mu_{eff} = \mu_{lam} + \mu_t$ $\qquad$ $\mu_t = \rho\, C_\mu k^2 / \varepsilon$

$G = \mu_t\,[2\{(\partial U/\partial x)^2 + (\partial V/\partial y)^2 + (\partial W/\partial z)^2\} + (\partial U/\partial y + \partial V/\partial x)^2 + (\partial V/\partial z + \partial W/\partial y)^2 + (\partial U/\partial z + \partial W/\partial x)^2]$

$S_U = \partial/\partial x(\mu_{eff}\,\partial\Phi/\partial x) + \partial/\partial y(\mu_{eff}\,\partial\Phi/\partial x) + \partial/\partial z(\mu_{eff}\,\partial\Phi/\partial x)$

$S_V = \partial/\partial x(\mu_{eff}\,\partial\Phi/\partial y) + \partial/\partial y(\mu_{eff}\,\partial\Phi/\partial y) + \partial/\partial z(\mu_{eff}\,\partial\Phi/\partial y)$

$S_W = \partial/\partial x(\mu_{eff}\,\partial\Phi/\partial z) + \partial/\partial y(\mu_{eff}\,\partial\Phi/\partial z) + \partial/\partial z(\mu_{eff}\,\partial\Phi/\partial z)$

Turbulence model constants $C_1 = 1.44$, $C_2 = 1.92$, $C_\mu = 0.09$

$\sigma_H = 0.9$, $\sigma_{RH} = 0.9$, $\sigma_\tau = 0.9$, $\sigma_k = 0.9$, $\sigma_\varepsilon = 1.225$

Table 2. Theatre boundary conditions.

Inlet Air Conditions,			Supply descriptions		Extract descriptions	
V_S, m/s	T_S, K	RH_S, %	Modules (Ports)	Area, m^2	Modules (Ports)	Area, m^2
0.75 m/s for S1 to S6	289	60	6	0.6 × 0.6	18	1.8 × 0.6
1.00 m/s for S7 to S12	289	60	6	0.6 × 0.6		
1.25 m/s for S13 to S18	289	60	6	0.6 × 0.6		
2.50 m/s for S19 to S29	289	60	11	0.6 × 0.6		
0.75 m/s for S30 to S35	289	60	6	0.6 × 0.6		
1.00 m/s for S36 to S41	289	60	6	0.6 × 0.6		
1.25 m/s for S42 to S49	289	60	6	0.6 × 0.6		