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Indoor aerodynamics and Ventilation Design in Big Enclosed Spaces

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ABSTRACT

The technique of computational fluid mechanics is applied to simulate indoor air movement and convective heat transfer induced by thermal sources in big enclosures. This is achieved by solving a system of partial differential equations on conservation of momentum, enthaply and mass with turbulent effect described by the k-ε model. The equations are discretized using finite difference method and solved by the Semi-Implicit-Method for Pressure-Linked-Equations-Revised (SIMPLER) scheme. Examples taken to illustrate the capability of the technique are the air movement in an air-conditioned indoor gymnasium and an office. Illustration of the predicted results for critizing the ventilation efficiency and performance of the mechanical system in those big spaces are demonstrated.

KEYWORDS Air-conditioned space, computational fluid dynamics and heat transfer, field model, k-ε model

INTRODUCTION

The technique of field modelling (see e.g. Spalding 1980; Alamadari et al. 1986; Whittle 1987); is a popular design tool to predict the airflow and temperature induced by mechanical systems inside enclosures. A set of partial differential equations describing conservation of momentum, enthalpy, and sometimes chemical species is solved numerically from the present knowledge on computational fluid dynamics and heat transfer (Patankar 1981; Minkowycz et al. 1988). Results predicted can be used to assess the performance of Heating, Ventilation and Air-conditioning (HVAC) systems installed in buildings with different configurations. This method is especially suitable for predicting the airflow and temperature distribution in big air-conditioned spaces such as a gymnasium, an atrium, a factory, or a shopping mall. Optium design on the diffuser spacing, cooling load, and location of exhaust for contaminants can be achieved.

Works appeared on the literature include the vorticity/stream function approach for computing the air diffuser performance index (Nielsen 1974, 1975); sizing of heat using two-dimensional laminar flow (Chu et al. 1976); studies on nonbuoyant and weakly buoyant flow induced by mechanical heating systems for ventilation assessment (Hjertager and Magnussen 1977); the isothermal flow induced by ceiling mounted diffusers with the standard-dissipation model and a 'large eddy simulation' approach on a rectangular enclosure (Sakamoto and Matsuo 1980); the k-ε model approach with the CHAMPION code (Almadari et al. 1986, Reinartz and Renz 1984); the three-dimensional simulation and cooling load calculation based on the PHOENICS code (Chen et al. 1988; Chen and Van der Kooi 1988); the vector potential approach (Ozoe et al. 1980, Yamazaki et al. 1987) on a ventilated cubic enclosure; the works of Murakami (1988, 1989, 1990) on clean room design; works with moisture (Chow 1989); the atrium air-conditioning air flow (McLean 1990, Whittle 1990, Seymour 1992); and the works with low-Reynolds-number k-ε model on air supply parameters and boundary conditions by Chen et al. (1991a, 1991b). There are many others such as those papers presented at the Annual Meeting of the Architectural Institute of Japan (1991) and a review is presented by Moser (1991) concerning the IEA Annex 20 activities related to indoor aerodynamics. There, the associated commercial and research packages for fluid flow simulation are also listed.

This article illustrated how a field model is used as a design tool to study the air flow and temperature using a self-developed computer package (Chow 1989, Chow et al. 1991). Besides solving the set of equations for mean flow and enthalpy using the k-ε model (Spalding 1980), an additional equation is included to describe the moisture ratio for the air-conditioned space (Chow 1989). The simulation showed that satisfactory results can be predicted in simulating aerodynamics in an indoor air-conditioned gymnasium and an office. The model is found to be useful for building services engineers in designing HVAC system.

FIELD MODEL

A field model (Spalding 1980; Patankar 1981) is able to predict the turbulent convective air flow induced by the thermal sources within an enclosure. The average values of an air variable ϕ defined by the following expression can be computed relatively easily using a turbulent model:

$$\phi_t = \phi + \phi' \qquad ...(1)$$

where ϕ_t is the instaneous value and ϕ' is the fluctuation.

The set of equations describing conservation laws on ϕ_t can be transformed into a form in ϕ with the fluctuation ϕ' terms separated out, i.e.

$$\frac{\partial}{\partial t}(\rho\phi) + \text{div}\,[\rho\,\vec{V}\,\phi - \Gamma_\phi\,\text{grad}\,\phi] = S_\phi \qquad ...(2)$$

where ρ is the density of air, S_ϕ is the source term of ϕ and $\vec{V}$ is the air velocity vector which can be expressed in terms of its components u, v, and w in a three-dimensional Cartesian coordinate (x-y-z) system as :

$$\vec{V} = u\hat{x} + v\hat{y} + w\hat{z} \qquad ...(3)$$

The k-ε model is used in this study and now ϕ becomes the mean values of u, v, w, enthalpy h, moisture content (or humidity ratio) f and turbulence parameters k, ε; Γ_ϕ is the effective diffusivity for ϕ. The corresponding effective diffusivity and source terms are shown in Table 1. Note that the turbulent viscosity is expressed in terms of k and ε as :

$$\mu_t = \frac{C_D \rho k^2}{\varepsilon} \qquad ...(4)$$

The moist air is treated as a mixture of dry air and water vapor. Therefore, the temperature T is calculated from the enthalpy h and moisture content f (Wang 1986) :

$$h = C_p T + f h_t \qquad ...(5)$$

This model is only an approximate one that does not include a thermodynamic calculation and so C_p is taken to be 1870 kJ/(kg·K), h_t is 2501 kJ/kg.

The set of equations given by (2) are solved numerically using the control volume method and the power law scheme. The Semi-Implicit-Method for Pressure-Linked-Equations-Revised (SIMPLER) algorithm (e.g. Patankar 1981) is used to solve for u, v, w, h, f, k, and ε. Air is assumed to obey the ideal gas law with the density can be calculated from the temperature T :

$$P = \rho RT \qquad ...(6)$$

The under-relaxation technique (Patankar 1981) is used to ensure stability by limiting the changes occurring in the coefficient of the variable ϕ by an appropriate relaxation factor α. The solid boundary condition is described by the wall function (Launder and Spalding 1973). The derivative of variables normal to a free surface is taken as zero. The atmospheric pressure is specified outside the free boundaries. The initial patterns for velocity components u, v, and w inside the enclosure are guessed; pressure is assumed to be at one atmospheric pressure, the values k and ε are defined as 10^{-12}J/kg and 10^{-12}J/(kg·s) Iteration is performed until the results converge.

INDOOR AIR-CONDITIONED GYMNASIUM

Numerical experiments were performed on two cases. The first case is an indoor air-conditioned gymnasium shown in Figure 1. This air-conditioned gymnasium is a multi-purpose hall designed for indoor sports activities like badminton and basket ball games. The area is fully air-conditioned year round. The air conditioning system is basically an "all air" system with chilled water supply from the central air-conditioning plant. The chilled water flow to the cooling coil is controlled automatically in response to fluctuations in return air temperature and hence reducing the cooling load of the room. Two ceiling hung air handling units are installed at the corners of the gymnasium. Each is designed to handle 6300 l/s with the return air installed at the underside. The space is of length 33 m, width 19 m, height 8.22 m and is enclosed by four solid walls. Air is blowing out through linear air grille with speed 5 m/s, enthalpy 28.0 kJ/kg and moisture content 0.0048 kg/kg. Because linear air grille is used and the air space is very big, the whole enclosure is not considered. The part of interest is located at the centre and of width 3 m. It is divided into 10400 (i.e. 26 x 20 x 20) control volumes as shown in Figure 2. This is different from simulating airflow induced by smoke extraction fan (Chow and Leung 1989) where the whole atrium has to be considered.

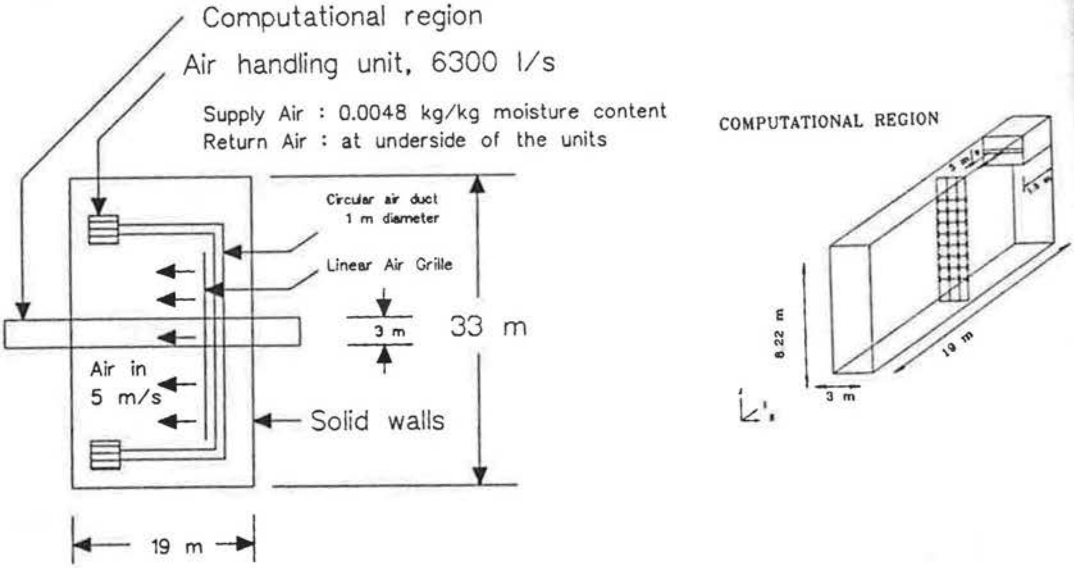

Figure 1: Indoor air-conditioned gymnasium

Figure 2: Indoor air-conditioned gymnasium (control volumes)

The gymnasium is kept at 18°C and a moisture content of 0.0054 kg/kg in winter. Field measurement on the horizontal air speed, temperature and relative humidity are performed at positions as in Figure 2. This corresponds to the computational domains at I = 15, K = 3, 5, 7, 9. The predicted results on the velocity vector are presented in Figure 3. The horizontal air speeds, air temperature and relative humidity at the monitoring points are shown in Figures 4a to d. The predicted results agree reasonably with the measured data. The models seems to be useful for building services engineers in designing HVAC systems.

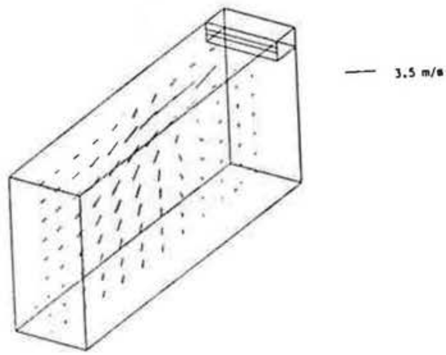

Figure 3: Predicted velocity vectors

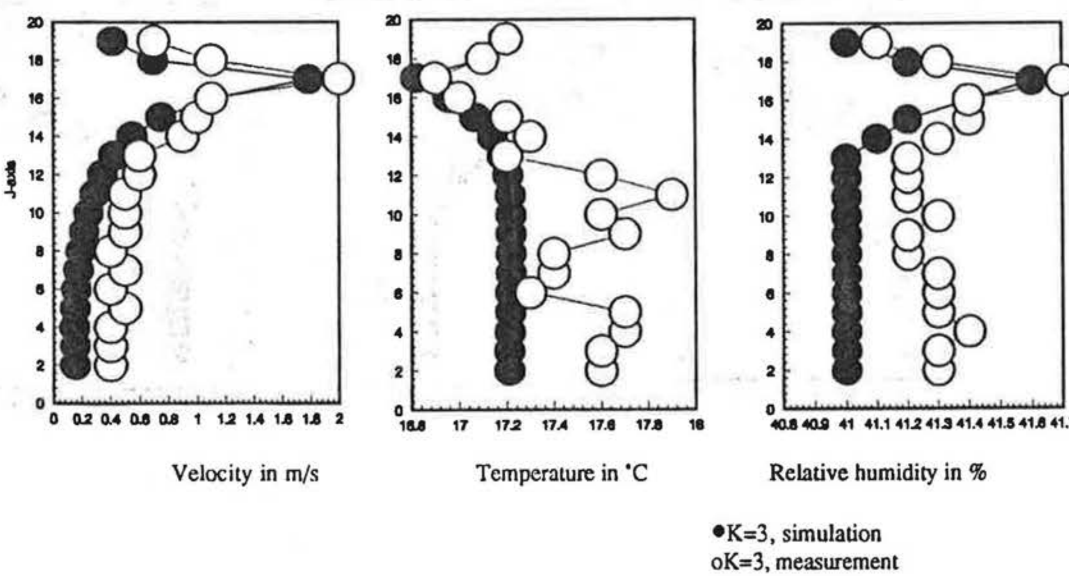

Figure 4a: Predicted results for the gymnasium

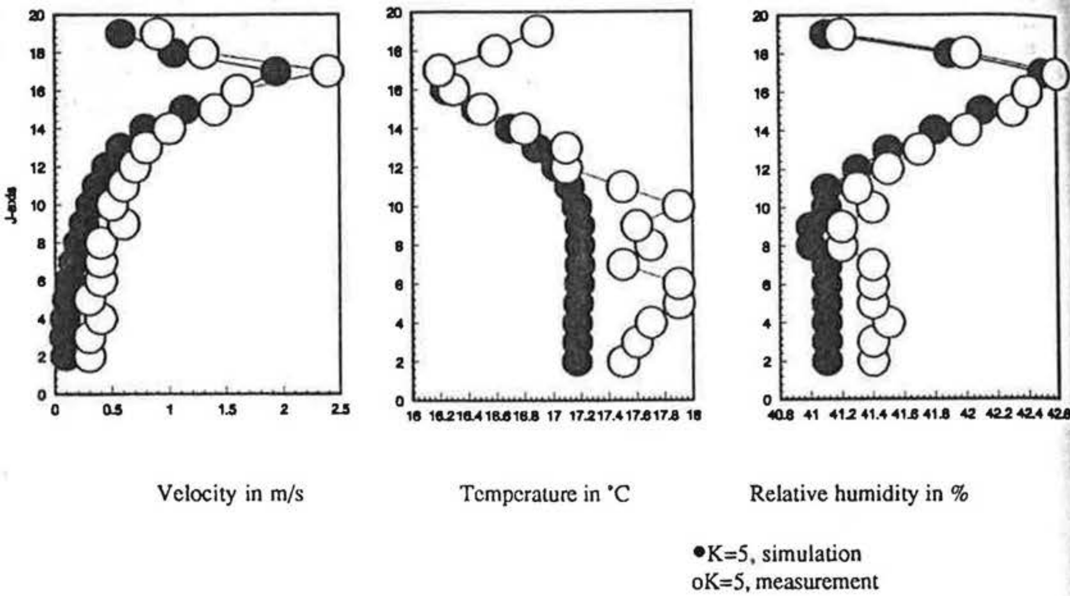

Figure 4b: Predicted results for the gymnasium

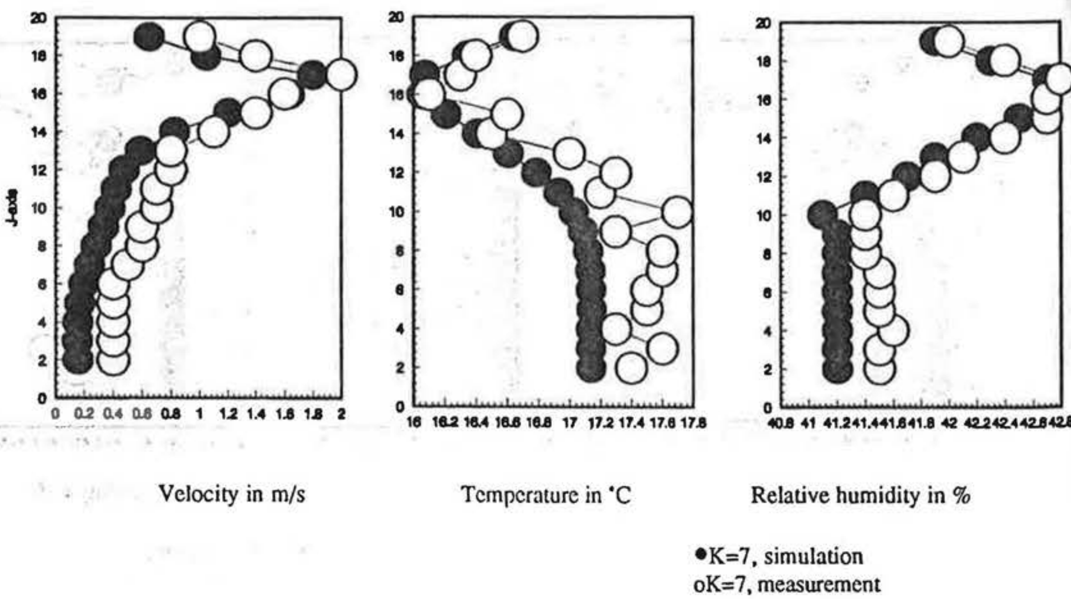

Figure 4c: Predicted results for the gymnasium

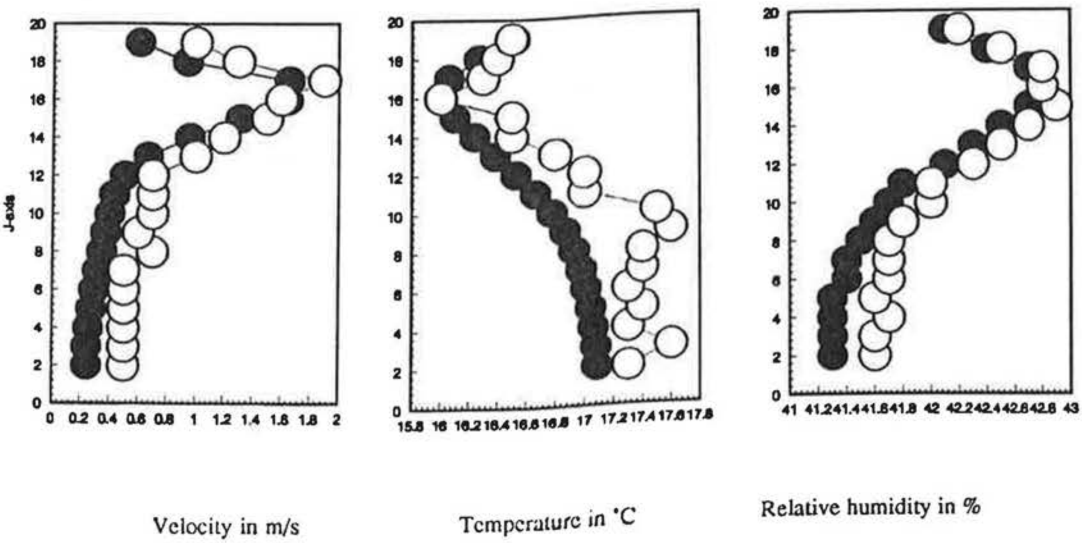

Figure 4d: Predicted results for the gymnasium

One of the key tasks in the commissioning exercise of the air distribution system is to ensure that the air stream at any point immediately above the badminton field will not adversely affect the motion of the badminton ball in the game. An air velocity of not more than 0.25 m/s is recommended. The predictions from the field model would help to determine the appropriate air velocity and flow direction that will mantain a satisfactory air circulation and an acceptable environment for a badminton game. The operation conditions of this HVAC system satisfied the criterion as indicated by the field modelling results.

AIR-CONDITIONED OFFICE

The second case is an air-conditioned office of length 4.5 m, width 2.4 m and height 3 m as shown in Figure 6. The air conditioning for the small office room is provided by a constant air volume, variable air temperature system. A constant air volume (CAV) box for the office supplies a fixed amount of cool air to offset the cooling load of the room. The office is divided into 8700 (i.e. 30 x 10 x 29) control volumes as in Figure 6. Air of temperature 16°C and moisture content 0.0091 kg/kg is supplied at the air grille. Horizontal air speed, air temperature and relative humidity are measured at points corresponding to the positions at I = 15, K = 10, 14, 18, 22 which are shown also in Figure 6. Results on the velocity and vector is presented in Figure 7 and those air variables are plotted from Figures 8a to d. Very good agreement between the predicted and experimental results are achieved.

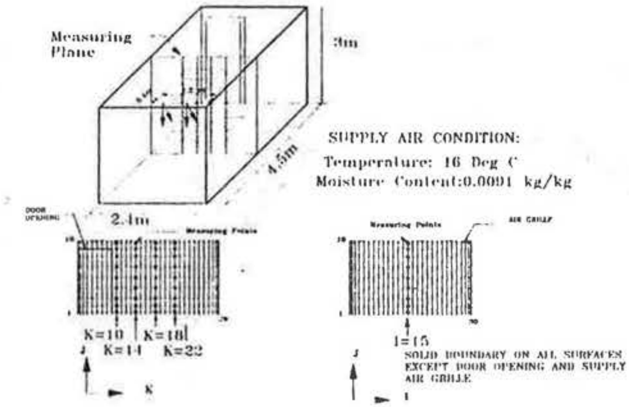

Figure 6: Air-conditioned office building

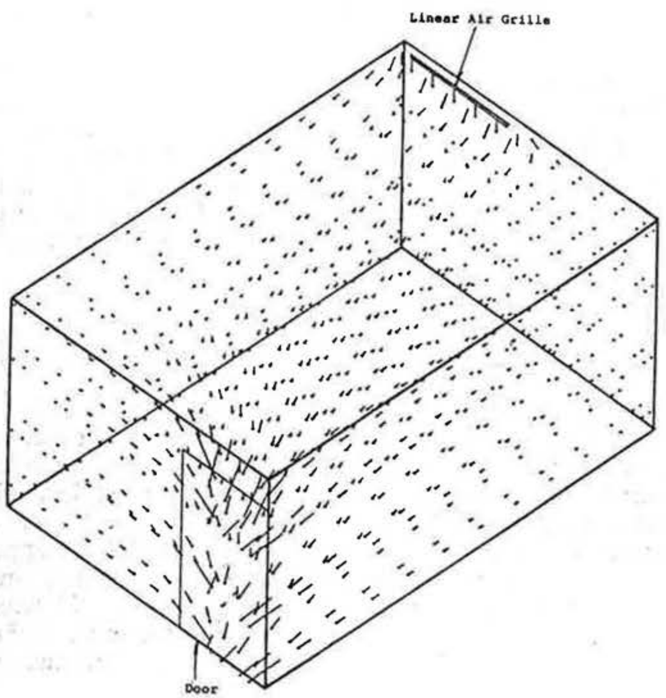

Figure 7: Predicted velocity vectors

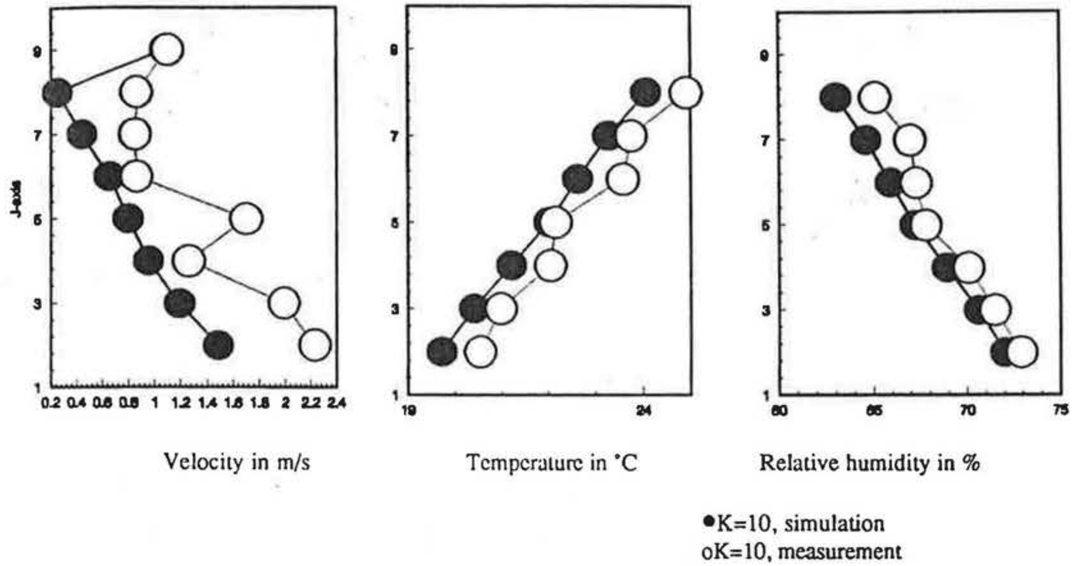

Figure 8a: Predicted results for the office building

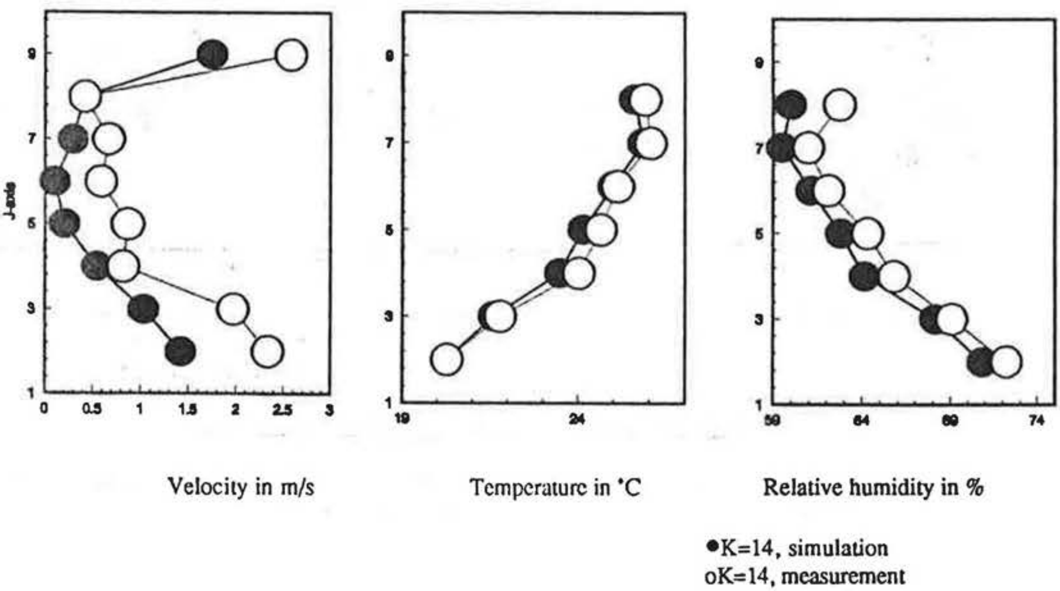

Figure 8b: Predicted results for the office building

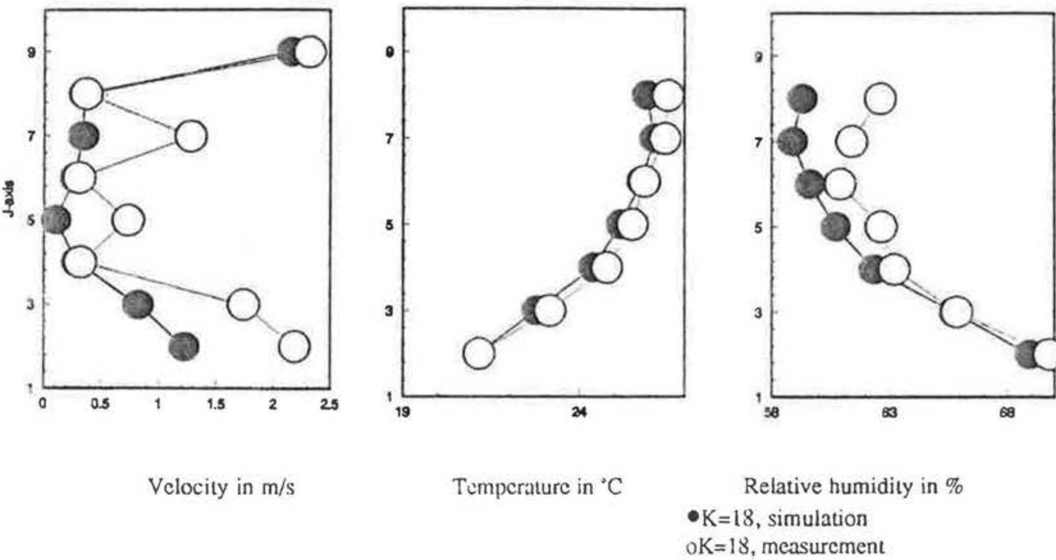

Figure 8c: Predicted results for the office building

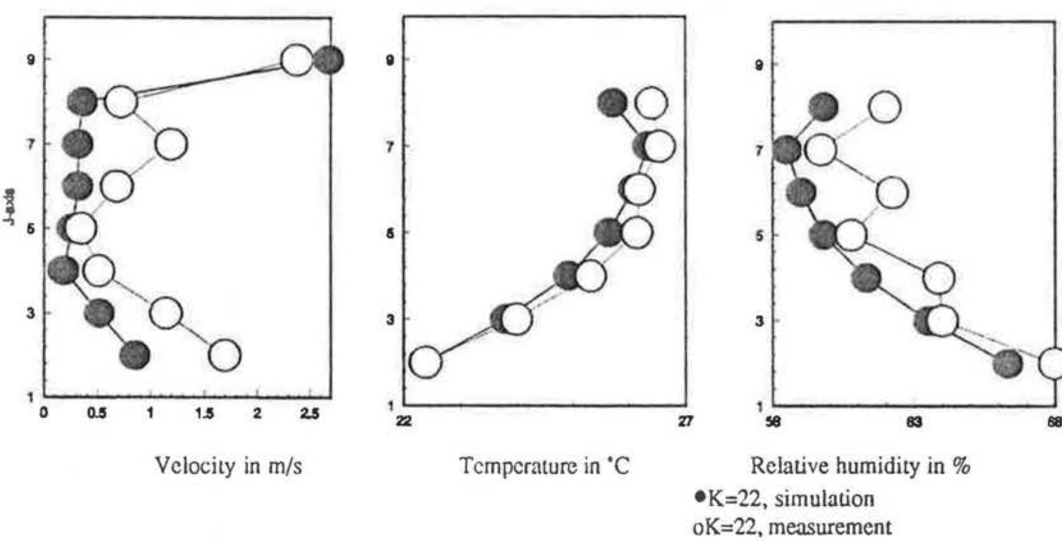

Figure 8d: Predicted results for the office building

The main purpose of the simulation exercises is to provide some ground work for the following subsequent analyses at that commercial building:

- to determine the flow field when there is a sensible heat and latent heat source in the room which is generated by an occupant or solar radiation.

- to determine the diffusion of the indoor contaminants generated by the fabric or the occupant in the room.

Obviously the field modelling technique can be applied for this purpose.

CONCLUSION

The following conclusions can be drawn from the present analysis :

1. The three-dimensional distribution of airflow pattern, temperature, and moisture content in an air-conditioned space may be predicted using a field model. Here, turbulent effect is described by the k-ε model and the equations for the primitive variables are solved by the SIMPLER scheme (Patankar 1981). This is very useful for large spaces such as a gymnasium or an atrium where temperature and moisture gradients are found.

2. From the predicted results on the air flow, temperature and moisture content, it is possible to verify whether the air-conditioning design for the enclosures can provide a satisfactory environment. Examples on an air-conditioned gymnasium and a small office have been taken to illustrate the simulation. This can be applied to HVAC design for atrium, clean room etc.

3. On-site measurements would help in validating the predicted results. Measurements of the moisture distribution in larger buildings such as this air-conditioned gymnasium is particularly desirable in tropical areas as dehumidification is important.

Table 1: Variable ϕ

Variable ϕ	Effective diffusivity for ϕ Γ_ϕ	Source of ϕ : S_ϕ
1	0	0
u	μ_{eff}	$-\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}(\mu_t \frac{\partial u}{\partial x}) + \frac{\partial}{\partial y}(\mu_t \frac{\partial v}{\partial x}) + \frac{\partial}{\partial z}(\mu_t \frac{\partial w}{\partial x})$
v	μ_{eff}	$-\frac{\partial p}{\partial y} + \frac{\partial}{\partial x}(\mu_t \frac{\partial u}{\partial y}) + \frac{\partial}{\partial y}(\mu_t \frac{\partial v}{\partial y}) + \frac{\partial}{\partial z}(\mu_t \frac{\partial w}{\partial y}) + g(\rho - \rho_o)$
w	μ_{eff}	$-\frac{\partial p}{\partial z} + \frac{\partial}{\partial x}(\mu_t \frac{\partial u}{\partial z}) + \frac{\partial}{\partial y}(\mu_t \frac{\partial v}{\partial z}) + \frac{\partial}{\partial z}(\mu_t \frac{\partial w}{\partial z})$
f	$\frac{\mu_t}{\sigma_f} + \mu_l$	0
h	$\frac{\mu_t}{\sigma_h} + \mu_l$	0
k	$\frac{\mu_t}{\sigma_k} + \mu_l$	$G_k - \rho\varepsilon + G_B$
ε	$\frac{\mu_t}{\sigma_\varepsilon} + \mu_l$	$C_1 \frac{\varepsilon}{k}(G_k + G_B) - C_2 \rho \frac{\varepsilon^2}{k}$

with the following parameters :

$\mu_{eff} = \mu_t + \mu_l$; $\mu_l = 1.82 \times 10^{-5}$ kg/(m·s)

$G_B = \mu_t g \frac{1}{\rho}\frac{\partial \rho}{\partial y}$; $G_k = \mu_t \frac{\partial u_i}{\partial x_j}(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i})$ i,j = 1, 2, 3

$C_1 = 1.44$; $C_2 = 1.92$; $C_D = 0.09$; $\sigma_k = 1.0$; $\sigma_f = 1.0$; $\sigma_h = 1.0$; $\sigma_\varepsilon = 1.3$

NOMENCLATURE

C_1, C_2, C_D	=	empirical constants in the turbulence model
Cp	=	specific heat of the gas mixture at constant pressure
G_k, G_B	=	generation terms for the turbulent kinetic energy equation
S_ϕ	=	source term in the differential equation for ϕ
T	=	absolute temperature
f	=	moisture content (humidity ratio) of air
g	=	acceleration due to gravity
h	=	stagnation enthalpy of the moistened air
k	=	turbulent kinetic energy
p	=	static pressure
t	=	time
u, v, w	=	velocity components in the (Cartesian) co-ordinate directions x, y, and z, respectively, with y along the vertical direction
Γ_ϕ	=	effective exchange coefficient of the property
ε	=	turbulent energy dissipation rate
ϕ	=	general fluid property
σ_ϕ	=	turbulent Prandtl number of the property ϕ
α	=	relaxation factor

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