NUMERICAL SIMULATION OF INDOOR AERODYNAMICS IN BIG ENCLOSED SPACES

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ABSTRACT

The technique of field modelling is applied to predict the 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 describing the conservation of momentum, enthaply and mass. The k-E model is used to describe the turbulent effect. The equations are discretized using finite difference method and solved by the Semi-Implict-Method-for Pressure-Linked-Equations-Revised (SIMPLER) scheme. Examples taken to illustrate the capability of the model are the smoke movement pattern in an atrium fire, part of an air-conditioned atrium, the ventilation of a big cargo terminal and a fire compartment for validating the field model. The predicted results are particularly useful for critizing the performance of the mechanical system in those big spaces.

(1) INTRODUCTION

The technique of applying higher-level mathematical model (see e.g. Whittle 1987, 1988; Alamadori et al. 1986), or known as field modelling technique (Spalding 1980), has now become a 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 (Barker and Lam 1986; Jedrzejewska-Scibak and Lipinski 1988) of HVAC systems installed in buildings with different configurations as well as smoke movement pattern (Markatos et al 1982, Bos et al 1984, Chow 1989, Rhodes 1989) in an enclosed fire and hence optimum design can be achieved. This method is much better than extracting information from physical models (Jackman 1973) as the experimental studies will be too expensive. Also, the results achieved for a certain configuration might not apply to others. Further, predicting the thermally induced airflow and temperature distribution for air-conditioned spaces such as an atrium, factory, or shopping mall, are especially useful in assessing diffuser spacing, cooling load, and dispersion of contaminants.

Ostrach (1972); Turner (1973); Jaluria (1980); Bejan (1984); and Arpaci and Larsen (1987) review the extensive studies on natural convection inside an enclosure carried out during the past two decades. Others 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); studies on the two-dimensional flow induced by a vertical isothermal surface (Plumb and Kennedy 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); using the k-ε model with the CHAMPION code (Reinartz and Renz 1984, Almadari et al. 1986, Markatos et al 1982, Bos et al 1984); the three-dimensional simulation using k-ε based on the PHOENICS code and for cooling load calculation of ACCURACY (Chen et al. 1988, Chen et al 1988) on the smoke movement prediction JASMINE (Cox and Kumar 1987); the vector potential approach (Ozoe et al. 1980, Yamazaki et al. 1987) on a ventilated cubic enclosure; the more recent works of Murakami (1987, 1988, 1990) on clean room design; works with moisture (Chow 1989) and on atrium air-conditioning air flow (IMechE 1990).

Works of this kind becomes more and more popular and this article illustrated how it can be used as a design tool. The temperature and aerodynamics are predicted by a self-developed computer package (Chow 1989, Chow and Leung 1989) for solving the set of conservation equations for mean flow and enthalpy using the k-ε model (Spalding 1980).

Section (2) of this paper describes the field model. Section (3) describes the numerical experiments that have been performed on an atrium fire and Section (4) describes on the simulation in part of an air-conditioned atrium, Section (5) on the study in a ventilated cargo terminal, and Section (6) on validating the field model by a full-scale physical model (Markatos et al 1982). The model is useful for building services engineers for designing mechanical system. The paper ends in Section (7) with a conclusion.

(2) FIELD MODEL

A field model (Spalding 1980; Patankar 1981; Chow and Leung 1988) is able to predict the convective field of flow and temperature induced by the thermal sources within an enclosure. The resultant air flow is turbulent because the inertial force due to density differences is much greater than the viscous one. An analytical solution for the air flow and temperature cannot be achieved. Even present computer systems have difficulty in predicting the instantaneous momentum, density, pressure, enthalpy, and moisture. But the average of these variables is useful for engineering and can be computed using turbulent model. Any average value is related to the instaneous value ϕ_t by :

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

where 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 :

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

Different turbulent models are proposed to close this set of equations. The k-ε model is used in this model. In this way, 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 :

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

For the enthalpy equation, (i.e. $\phi = h$) the source term S_h will have a fire source described by the thermal power $\dot{Q}_f$:

$$S_h = \dot{Q}_f \quad ...(5)$$

The set of equations given by (2) are solved numerically using the control volume method. The enclosure is first divided into a number of control volumes as shown in Figure 1. A variable ϕ at the node P is expressed in terms of the values at its neighbor, i.e. at points X+, X-, E, W, N, and S through the coefficients a_{X+}, a_{X-}, a_E, a_W, a_N, and a_S after integrating over the control volume :

$$a_P\phi_P = a_{X+}\phi_{X+} + a_{X-}\phi_{X-} + a_E\phi_E + a_W\phi_W + a_N\phi_N + a_S\phi_S + b \quad ...(6)$$

where

$$a_P = a_{X+} + a_{X-} + a_E + a_W + a_N + a_S + \rho_P^{\circ}\frac{\Delta\tau}{\Delta t} - S_P\,\Delta\tau \quad ...(7)$$

$$b = S_c\,\Delta\tau + \rho_P^{\circ}\,\phi_P^{\circ}\frac{\Delta\tau}{\Delta t} \quad ...(8)$$

with ϕ_P° and ρ_P° are the values at P in an earlier time interval Δt. The source term S_f is linearized over the control volume Δτ as :

$$\int S_f\, d\tau = S_P\,\phi_P\,\Delta\tau + S_C\,\Delta\tau \quad ...(9)$$

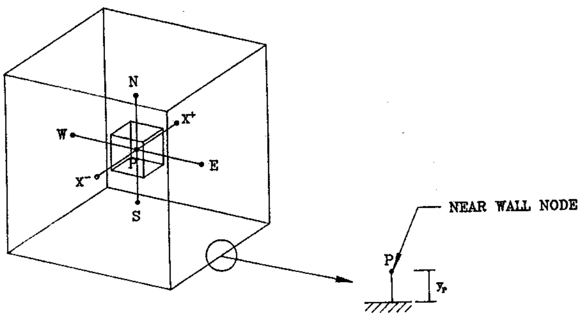

FIGURE 1 : CONTROL VOLUME

The coefficient a_{X+} is expressed by the Power Law scheme as :

$$a_{X+} = D_{X+}\,A(|P_{X+}|) + [-F_{X+}, 0] \quad ...(10)$$

with

$$A(|X|) = [0, (1 - 0.1|X|)^5] \quad ...(11)$$

[a, b] : greater value of a and b.

$$F_{X+} = (\rho u)_{X+}\,A_{X+} \quad ...(12)$$

$$P_{X+} = F_{X+} / D_{X+} \quad ...(13)$$

$$D_{X+} = A_{X+}\,\Gamma_{X+} / \delta_{X+} \quad ...(14)$$

where Γ_{X+} is the diffusion coefficient evaluated at the point X+. Similar expressions can be derived for F_{X-}, P_{X-}, D_{X-}, etc. Numerical schemes, such as the SIMPLER (Semi-Implicit Method for Pressure Linked Equations) or the PISO (Pressure-Implicit-Splitting-Operator) are used to solve for u, v, w, h, k, and ε. The details are described elsewhere (e.g. Patankar 1980 and Issa et al 1986) and will not be repeated here. The air is assumed to obey the ideal gas law; so density can be calculated from the temperature T :

$$P = \rho RT \quad ...(15)$$

Further, an under-relaxation technique (Patankar 1981) is used to ensure stability by limiting the changes occurring in the coefficient a_ϕ of the variable ϕ with an appropriate relaxation factor α. The value of ϕ computed at a certain iteration (denoted by ϕ_P^{new}, is related to its value at a previous one ϕ_P^{old} by :

$$\phi_P^{new} = \alpha\phi_P^{new} + (1 - \alpha)\,\phi_P^{old} \quad ...(16)$$

Equation (5) together with (16) are solved using a tridiagonal matrix (TDM) techique. The solid boundary condition is imposed by the wall function (Launder and Spalding 1973). Here, the shear stress τ_w is a constant value for a near-wall node and the velocity parallel to the wall is fitted by a logarithm function, i.e. :

$$\tau_w = \frac{K\rho_P\,C_D^{1/2}\,k_P^{1/2}\,u_P}{in(Ey^+)} \quad ...(17)$$

with

$$y^+ = \frac{C_D^{1/4}\,\rho_P\,k_P^{1/2}\,y_P}{\mu_l} \quad ...(18)$$

P is a near-wall node at a distance y_P away from the wall. No-slip condition is used to compute near-wall velocity components. For the enthalpy equation, an adiabatic wall condition is applied at the near-wall grid node. Near-wall turbulence parameters k and ε are computed under the assumptions :

$$\int_o^{y_P} \varepsilon dy = C_D^{3/4}\,k_P^{3/2}\frac{1}{K}\ln(Ey^+) \quad ...(19)$$

$$\varepsilon_P = \frac{C_D^{3/4}\,k_P^{3/2}}{Ky_P} \quad ...(20)$$

The derivative of variables normal to a free surface is taken as zero as suggested by Harlow and Welch (1965) and atmospheric pressure is specified outside the free boundaries. 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 $10^{-12}Jkg^{-1}$ and $10^{-12}Jkg^{-1}s^{-1}$. Iteration is performed until the results converge.

(3) ATRIUM FIRE

The atrium (e.g. Saxon 1986, IMechE 1990), sometimes known as "the third-generation design" in languages of fire design (e.g. Morgan 1986), has become a staple design feature in most types new buildings in Hong Kong. Representative examples include the Hong Kong Bank, Bank of China and the Hong Kong Convention and Exhibition Centre. Actually, this is an old architectural concept having a history of over two thousand years. But in those modern buildings, it means an opening through two or more floor levels, that is closed at the top and is not defined as a mall. It can also be visualized as a centroidal, interior, daylit space in a buildings. Although the atrium design is appreciated by designers and building users for its aesthetic, functional and economical. However, regardless of other technical problems, special attention most be paid to the risk of fire.

Knowledge on the fire environment within the atira is very important for designing appropriate safety system vital element of architectural design. Previous fire research results (e.g. Markatos et al 1982) show smoke is the major problem to be solved. A large quantity of it will be evolved from thermal decomposition of a wide range of building and furniture materials such as timber, synthetic polymeric materials etc. Even a very low concentration of smoke will bring sereve problems. When there is a fire, smoke will be accumulated in the open space. It will be spreaded rapidly between floors through the unrestricted atrium in both the horizontal and vertical direction. Therefore, even a small fire in an atrium building will generate a very large quantity of smoke due to air entrainment. The dense and toxic smoke creates great difficulties for fire fighting.

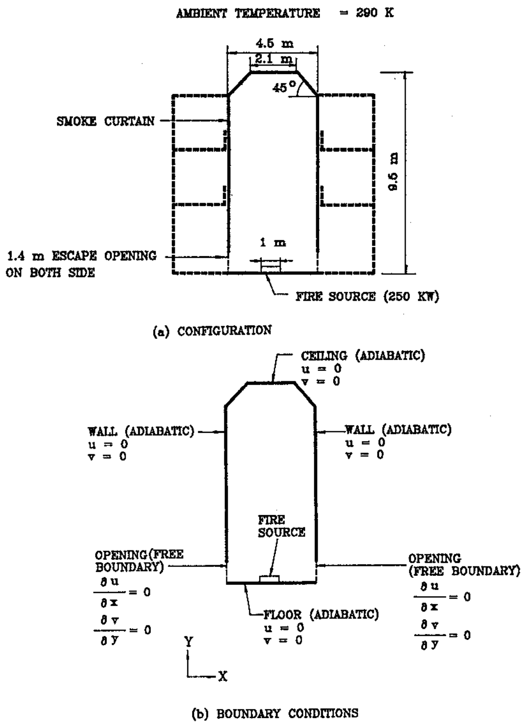

FIGURE 2 : ATRIUM EXAMPLE

In order to understand smoke movement in a common atrium building, the field modelling technique is used to simulate the fire environments within such structures. As described above, smoke spreading mechanism in an atrium is important and heat transfer from the fire source does not bring much problems at the growth stage. The combustion processes only occur at the plume. Therefore, it becomes a convective problem induced by a thermal source. To simplify the picture and save the computing time, a two-dimensional simulation is performed. The simulation is good enough for justifying the physical picture in a linear atrium (Saxon 1986). An atrium space shown in Fig. 2 with a height of 9.5m and width 4.5m, a fire of size 1m, thermal power 0.25 MW is used to demonstrate how a field model works. Results on air flow field and temperature predicted are shown in Fig. 3. Obviously, air is flowing upward and a hot layer is found at the top indicating smoke will be accumulated. To justify the result predicted, a 1/10 scaled model (Chow & Ng 1991) showing the flow pattern with aluminium powder is considered. Water is used as the fluid medium accounting for the scaling effect. An upward current is found and this is consistent with the predicted results as shown in Fig. 4.

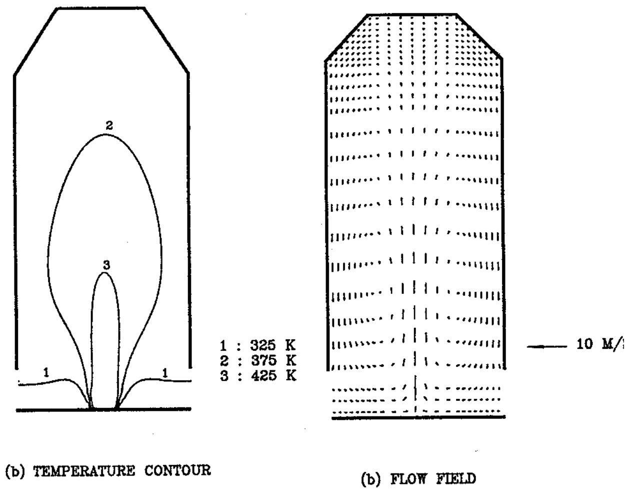

FIGURE 3 : PREDICTED RESULTS

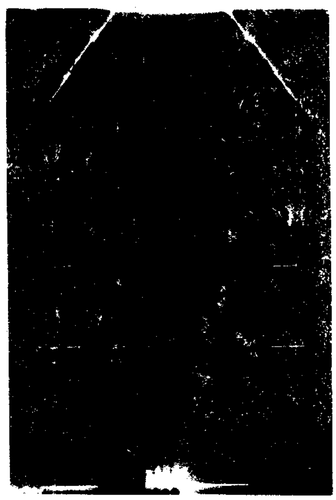

FIGURE 4 : SCALED MODEL FLOW PATTERN

(4) AIR-CONDITIONED ATRIUM

An air-conditioned atrium shown in Figure 5 is considered. Since only the air-conditioning system induced flow is simulated, it is unnecessary to consider the whole enclosure. The lower part is of interest and therefore simulation is different from the case for airflow induced by fire as in above. Obviously, the whole atrium has to be considered there. However, the atrium in this example is kept initially at 303 K and a moisture ratio of 0.0248. The enclosure is divided into 31 x 30 x 9 = 17670 nodes. Cool air enters the space at 1 m/s, air temperature of 293 K and a moisture ratio of 0.0134. The air moisture ratio f is also computed (Chow 1989) and coupled with the temperature. The transient temperature variation as indicated by the 295K contour surface is shown in Fig. 6 together with the air velocity vectors diagram at 121 sec. With this predicted result, the 'transient' cooling in the atrium space can be visualized. Obviously, effect of varying the spacing of the air diffusers, air outlet velocity and etc. can all be visualized by this sort of results.

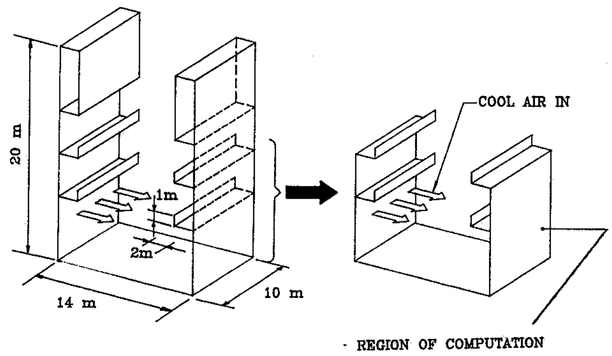

FIGURE 5 : AIR–CONDITIONED ATRIUM BUILDING

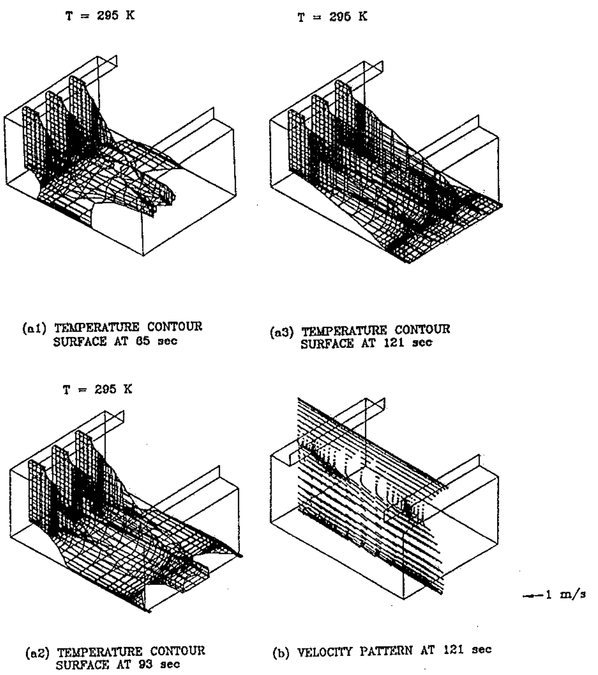

FIGURE 6 : PREDICTED RESULTS

(5) VENTILATED CARGO TERMINAL

A large enclosed cargo terminal of length 224m and width 128m shown in Fig. 7 is considered. The space is used for loading and unloading goods from the containers. It is installed with ventilation system composed of five bigger fans and one hundred and two smaller fans for providing sufficient ventilation. An existing ventilating rate of 158.8 m^3/s is available using this mechanical extraction system. At peak season, there will be hundreds of persons and a large number of vehicle tractors inside. In order to ensure sufficient air flow is achieved for avoiding stuffy air, a simulation is performed to find out the air movement at various locations under normal working situation. Again, a two-dimensional simulation is used for reducing the computing time and memory. The predicted result on the air flow pattern is shown in Fig. 8. Design operating points can be made by varying the fan speed to get good air movement pattern.

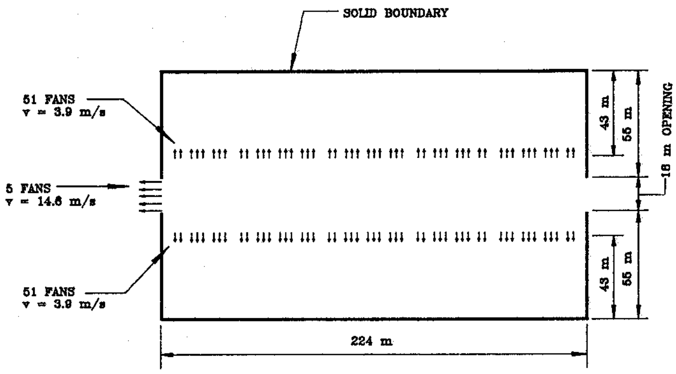

FIGURE 7 : CONFIGURATION OF CARGO TERMINAL

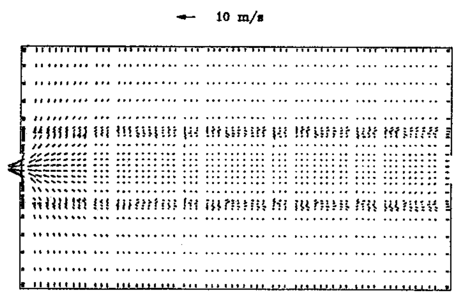

FIGURE 8 : PREDICTED FLOW FIELD FOR TUNNEL

(6) VALIDATION

The fire model (Chow & Leung 1989) can been validated by comparing with the experimental data measured at the Fire Research Station, United Kingdom (Markatos et al 1982). The configuration of the enclosure is shown in Fig. 9 with the fire size and location described. This is a good physical model for validating fire field model. The predicted temperature and velocity profile at 5.76m away from the rear wall are compared with the experimental data as shown in Fig. 10. Very good agreement between the two is achieved.

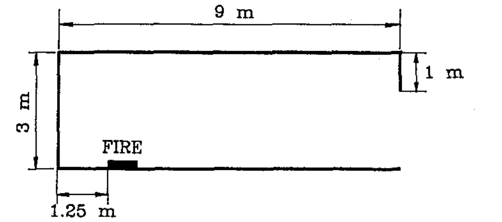

FIGURE 9 : GEOMETRICAL CONFIGURATION FOR F.R.S. CASE

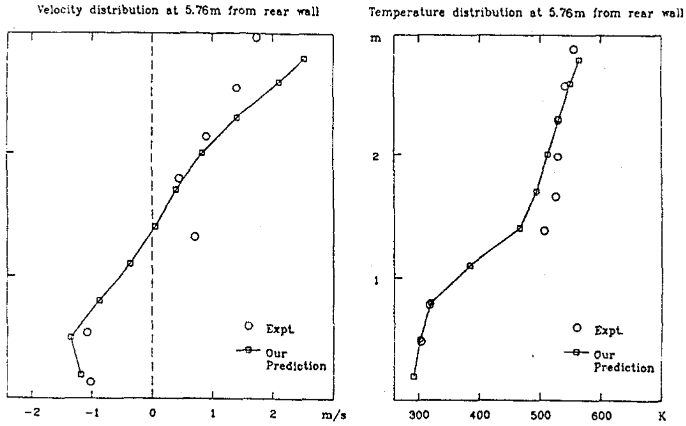

FIGURE 10 : EXPERIMENETAL DATA OF TWO DIMENSIONAL F.R.S. CASE

(7) CONCLUSIONS

The following conclusions can be drawn from the present analysis :

1. The distribution of airflow pattern and temperature induced by thermal sources in an enclosure can be predicted using the field modelling technique (Spalding 1980). Here, turbulent effect is described by the k-ε model and the equations for the primitive variables are solved by the SIMPLER scheme (Patankar 1980). This is very useful for studying smoke movement, air-conditioning ventilation etc. large enclosures such as an atrium.

2. An atria fire, an air-conditioned atrium and a ventilated cargo terminal are taken as examples for applying field model to predict airflow and heat transfer in large enclosures. From the predicted results, it is possible to assess the mechanical designs for the enclosures concerned.

3. The programs run on a minicomputer with a CPU time of up to 16 hr for each job. This is quite expensive but is tolerable when the precise indoor enviornmental conditions in large enclosures have to be known.

4. Further investigational works are to be carried out for improving the current situation. For simulating fire in the 'burning' region, combustion has to be considered in order to have a good estimate on the rate of heat released from the objects. Otherwise the model is only good for predicting smoke movement and the effectiveness of smoke control systems.

5. For simulating air-conditioned space, a detailed study on the coupled heat-mass transfer must be made to clarify the efficiency of moisture extraction. The thermodynamic equations are to be solved and moisture effects of building materials, especially, porous ones such as concrete, are to be studied.

6. Experiments in full-scale model should be carried out to provide relevant input data for 'tuning' the model. In-situ measurements on smoke movement, moisture extraction and etc. would also help in validating the predicted results. Measurements of the moisture distribution in larger buildings such as an air-conditioned atrium is particularly desirable in tropical areas as dehumidification is important. This is in progress and will be reported in separate papers.

TABLE 1

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}	$-\frac{\partial p}{\partial z} + \frac{\partial}{\partial x}(\mu_t \frac{\partial u}{\partial z}) + \frac{\partial}{\partial y}(\mu_t \frac{\partial v}{\partial z}) + \frac{\partial}{\partial z}(\mu_t \frac{\partial w}{\partial z})$
h	$(\frac{\mu_l}{\sigma_l} + \frac{\mu_t}{\sigma_t})$	0 (if no fire source) $\dot{Q}_f$ (in fire source)
k	$\frac{\mu_{eff}}{\sigma_k}$	$G_k - \rho\varepsilon + G_B$
ε	$\frac{\mu_{eff}}{\sigma_\varepsilon}$	$C_1 \frac{\varepsilon}{k}(G_k + G_B)(1 + C_3 R_f) - C_2 \rho \frac{\varepsilon^2}{k}$

with the following parameters :

$\mu_{eff} = \mu_t + \mu_l$

$\mu_t = \frac{C_D \rho k^2}{\varepsilon}$; $\mu_l = 1.82 \times 10^{-5}\ \mathrm{kgm^{-1}s^{-1}}$

$G_B = \mu_t g \frac{1}{\rho}\frac{\partial \rho}{\partial y}$

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

(u_i, u_j, u_k are now in tensor form)

$C_1 = 1.44$; $C_2 = 1.92$; $C_D = 0.09$; $\sigma_k = 1.0$; $\sigma_f = 1.0$

$\sigma_\varepsilon = 1.3$; R_f is the flux Richardson number

C_3 is given by : $G_B = - G_k (1 - C_3)$

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
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
K	=	karman constant
y_p	=	distance of near wall grid node from the wall

GREEK SYMBOLS

Γ_ϕ	=	effective exchange coefficient of the property
ε	=	turbulent energy dissipation rate
ϕ	=	general fluid property
μ	=	absolute viscosity of the gas mixture
σ_ϕ	=	turbulent Prandtl number of the property
α	=	relaxation factor

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