GEOMETRY EFFECTS ON INTERNAL FLOW PATTERNS

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ABSTRACT

The aim of the work presented in this paper is to look for the changes in internal flows due to the combined action of thermal effects and atmospheric wind (natural ventilation) caused by external geometry. Two different configurations are presented as an illustration of the model application.

KEYWORDS

Natural Ventilation; Internal Flows; Numerical Turbulence Models.

GEOMETRY EFFECTS ON INTERNAL FLOW PATTERNS

In the present work a numerical simulation of velocity and temperature fields promoted by the combined action of thermal effects inside a parallelepipedic space in communication with the outside through small openings is presented. Two different geometric configurations are analized.

The rational approach of the problem combines a 3-D numerical simulation of the thermal and dynamic governing equations by means of a k-ϵ two-equation turbulence model with experimental data on wind pressure and pressure drop coefficients through the wall openings.

This method has shown to be effective to simulate this kind of internal flows avoiding experimental problems in particular those concerning the modelation of the combined action of thermal effects and natural wind.

NUMERICAL MODEL

Model Equations

Using the k-ϵ to equation turbulence model the governing equations of the flow can be written as follows:

$$\frac{\partial U_i}{\partial x_i} = 0 \qquad (1)$$

$$\frac{\partial}{\partial x_i}(\rho\ U_i\ U_j) - \frac{\partial}{\partial x_i}\left\{ \mu_t \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right)\right\} + \frac{\partial P^+}{\partial x_j} + (\rho - \rho_0)\ g_j = 0 \qquad (2)$$

$$\frac{\partial}{\partial x_i}(\rho\ U_i\ T) - \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{Pr_t} \frac{\partial T}{\partial x_i} \right) - S_T = 0 \qquad (3)$$

$$\frac{\partial}{\partial x_i}(\rho\ U_i\ k) - \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{Pr_k} \frac{\partial k}{\partial x_i} \right) - G - B + \rho\epsilon = 0 \qquad (4)$$

$$\frac{\partial}{\partial x_i}(\rho\ U_i\ \epsilon) - \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{Pr_\epsilon} \frac{\partial \epsilon}{\partial x_i} \right) - C_1 \frac{\epsilon}{k}\ (G + B)\ (1 + C_3 R_f) - C_2 \rho \frac{\epsilon^2}{k} = 0 \qquad (5)$$

$$\mu_t = \rho\ C_\mu \frac{k^2}{\epsilon} \qquad (6)$$

where:

$$G = \mu_t \frac{\partial U_i}{\partial x_j}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \qquad (7)$$

$$B = -\ \beta\ g_i \frac{\mu_t}{Pr_t} \frac{\partial T}{\partial x_i} \qquad (8)$$

$$\beta = -\ \frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right) \qquad (9)$$

$$P^+ = P + \rho \frac{2}{3} k \qquad (10)$$

with:

C_μ	C_1	C_2	Pr_k	Pr_ϵ	Pr_t
0.09	1.44	1.92	1.0	1.3	0.7

In the form presented the momentum equations were subtracted by the static pressure equations, $\partial P_0/\partial x_i + \rho_0\ g_j = 0$ (11), to show the nature of the buoyant term $(\rho - \rho_0)g_j$, (12).

The values of the numerical constants in the transport equations for k and ϵ and in the equation for evaluate μ_t are the same as those in the original paper by Launder and Spalding (1972). In the transport equation for ϵ, R_f is a flux Richardson number defined as $R_f = 0.5\ B_l/(B+G)$ (13), (Rodi, 1984) where B_l is the buoyancy production of only the lateral energy components, and C_3 is a numerical constant equal to 0.8, (Hassain, 1980). In the present application it was assumed that vertical shear layer is dominant and R_f was set equal to zero.

Numerical Method

In the discretization of the equations a staggered grid system was adopted (scalar quantities in the center of the main cells and velocity components at the center of the cell surfaces to which they are normal) and an hybrid scheme (Spalding, 1981), was employed for the convective difusive terms. The finite-difference equations obtained were solved using the well known SIMPLE algorithm, (Patankar and Spalding, 1972) and the primitive variable computed in a semi-implicit line-by-line procedure over the staggered finite difference grid. Readers interested in the detail of this code can refer to Dias Delgado (1989).

Boundary Conditions

Openings. The pressure outside each of the openings was taken as a boundary condition assuming:

$$P_{out} = C_p \; 1/2 \; \rho \; U_0^2 \tag{14}$$

where C_p is the pressure coefficient outside the opening and U_0 the reference wind velocity. The velocity component normal to the opening, U, was linked with the internal pressure by the pressure equation:

$$\delta P = \zeta \; 1/2 \; \rho \; U \, |U|; \tag{15}$$

where ζ is the experimental value of the pressure drop coefficient of the opening.

Walls. The generalized log law, described in detail by Launder and Spalding (1974) was employed as boundary condition. The shear stress at the control volume adjacent to the wall τ_ω follows the relation:

$$\frac{U}{\tau_\mu} \; \rho \; C_\mu^{1/4} \;\; K^{1/2} = \frac{1}{k} \; ln \; (\; E \frac{y \; \rho \; C_\mu^{1/4} \;\; K^{1/2}}{\mu}); \tag{16}$$

where E is an integration constant which depends on the roughness of the wall, K is the Von-Karman constant and y is the distance to the wall. The value of ϵ in the control volume adjacent to the wall is assigned using the relation:

$$\epsilon = \frac{C_\mu^{3/4} \;\; k^{3/2}}{K \;\; y}; \tag{17}$$

The ambient temperature outside the walls T_0 was also introduced as a boundary condition, expressing the source term of the temperature equation per unity of area of the cell surface adjacent to the wall as:

$$S_T = - \frac{\mathcal{U}}{C_p} \; (T - T_0); \tag{18}$$

where $\mathcal{U}$ is the heat transfer coefficient of the wall.

Heat Sources. The heat power dissipated by the heating plate was introduced in a similar way equating:

$$S_T = \frac{\dot{q}_w}{C_p}; \tag{19}$$

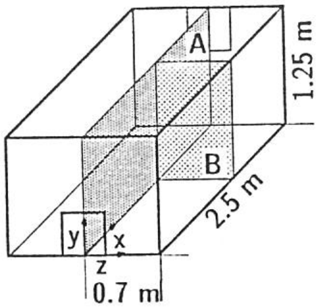
where $\dot{q}_w$ is the heat power dissipated per unity of area.

MODEL APPLICATION

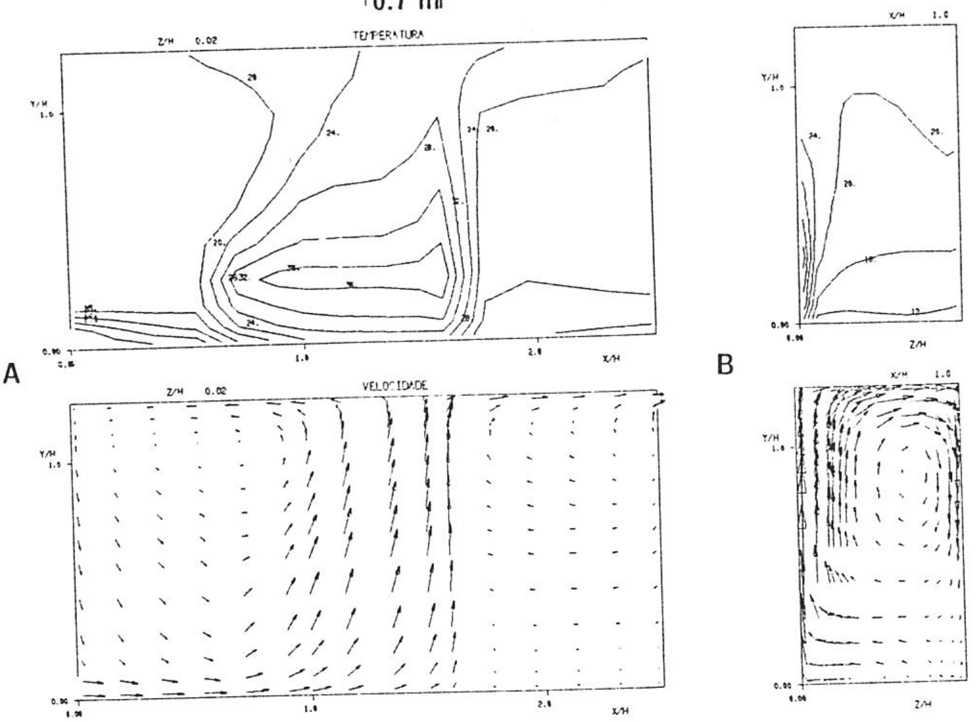
Two cases were simulated, one with alligned openings (case1) and the other with non-alligned openings (case 2). In both geometries the external dimensions are 2.5x1.25x1.4 m^3, and a vertical heating plate, as the thermal source, was considered with an uniform power of q_w = 2000 W/m^2 and an area of 0.5x0.255 m^2. The reference wind velocity was taken as U_0 = 0.3 m/s. Simmetry conditions on x-y plane were assumed in numerical simulations.

In both cases simulated the thermal effects are dominant. The temperature and velocity patterns shows the influence of the different position of the openings in the two cases. In case 2 the mean temperature is higher then in case 1, and this fact can be explained by a big recirculation bubble located near the simmetry wall, where the maximum temperatures are located. In both cases it is clear the effect of the buoyancy forces.

CASE 1

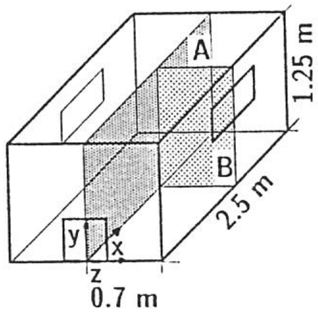


A
B
1.25 m
2.5 m
0.7 m
y
x
z



Z/H 0.02
TEMPERATURA
VELOCIDADE
X/H
Y/H
Z/H
A
B

CASE 2



A
B
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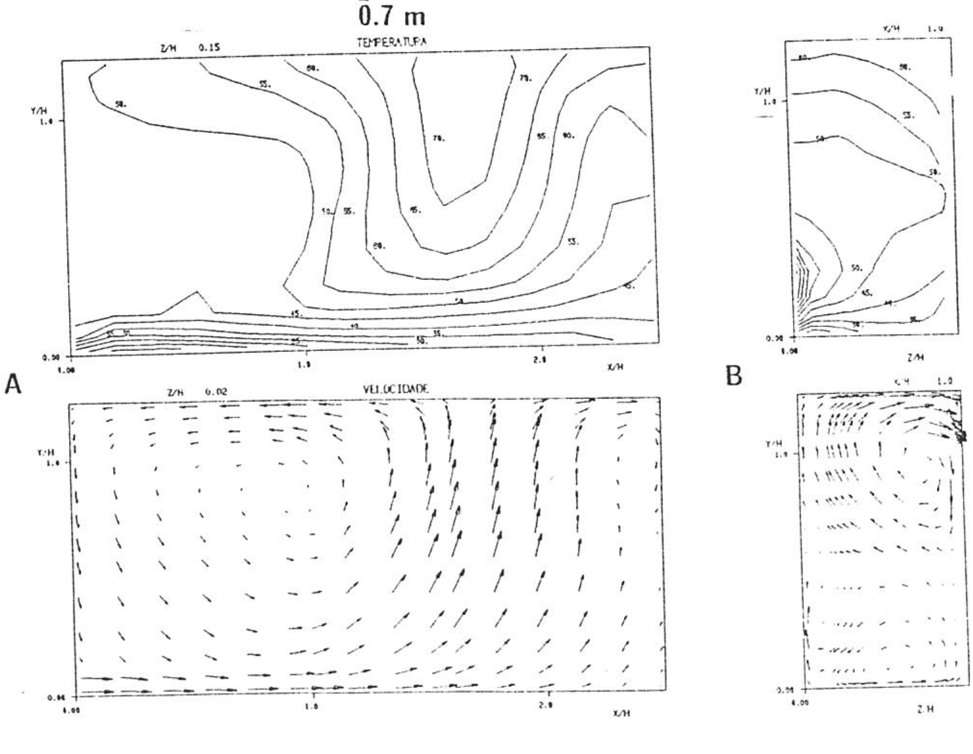
y
x
z



Z/H 0.15
TEMPERATURA
Z/H 0.02
VELOCIDADE
X/H
Y/H
Z/H
A
B

CONCLUSIONS

Results indicate this model can be used to accurately simulate the internal flow pattern inside parallelepipedic volumes. The model equation can include other different positions of the heating plate and the external openings.

In both configurations simulated, thermal effects are dominant. Different temperature and velocity patterns were obtained, which were governed by buoyancy effects.

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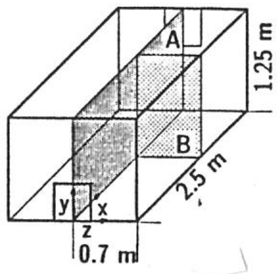
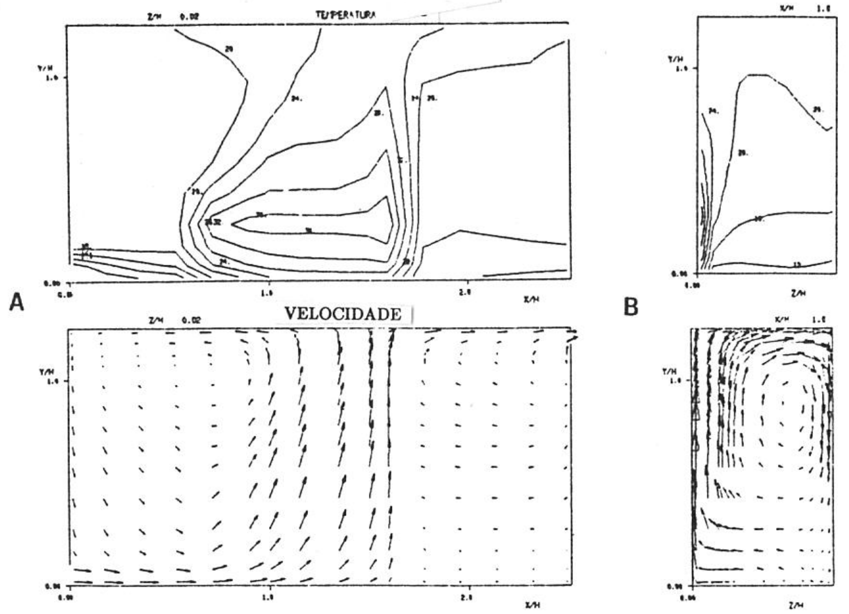
VENTILAÇÃO NATURAL

CASO 1





CASO 2