# SINGLE-SIDED NATURAL VENTILATION IN AN ENVIRONMENT INFLUENCED BY AN INTERNAL HEAT LOAD

**Henor Artur de Souza, henor@em.ufop.br[1]**
**Luiz Joaquim Cardoso Rocha, ljoaquim@em.ufop.br[2]**

[1]Universidade Federal de Ouro Preto, Escola de Minas/ Programa de Pós-Graduação em Engenharia Civil - Área de Construção Metálica, Ouro Preto, MG, CEP. 35.400-000
[2]Universidade Federal de Ouro Preto, Escola de Minas/ Departamento de Engenharia de Controle e Automação e de Técnicas Fundamentais, Ouro Preto, MG, CEP. 35.400-000

***Abstract.*** *Natural indoor ventilation performs fundamental functions for human health and well-being by maintaining contaminant concentration at a safe level and contributing to body-heat transfer to the environment. When used as an architectural and bio-climatic strategy, projects can be more spacious and better illuminated, which in turn, can significantly reduce the enery cost for the building. Using the finite volume method, presented herein is a numerical study of single-sided natural ventilation in constructed environments. A numerical model is used for determining the combined effects of wind and thermal forces, which in turn, determine the air flow and temperature distribution in the designated area. For the turbulence model, two differential equations for a low Reynolds number LNR k-$\varepsilon$ were used. Then, the influence of the area's internal heat sources on the efficiency of the single-sided natural ventilation was analyzed. The results show excellent agreement between numerical and semi-analytical model.*

***Keywords****: natural ventilation, thermal comfort, numerical solution.*

## 1. INTRODUCTION

The ventilation of internal spaces is fundamental for providing a thermally comfortable and healthy environment, without creating significant cross currents of air that could proportion local discomfort and noise (ASHRAE 62, 2004). Ventilation, in general, represents any mechanism where air in a closed area is substituted by external air that enters in the area by means of intentional or non-intentional openings. When appropriately applied, natural ventilation is an efficient mechanism, since it can provide a cost-efficient means for cooling enclosed areas.

The thermal efficiency of natural ventilation depends on the external air temperature and the internal air flow of the enclosed area. Air flow velocity is the most important factor in the evaluation of the thermal sensation during ventilation, since it provides a hot or cold sensation, even when the air temperature remains constant. On the other hand, the air temperature is influenced by many factors so that it becomes almost impossible to control with some form of mechanized conditioning. However, the internal air flux can be altered in function of the size and position of strategically placed openings, and also by the shape and volume of the enclosed areas, for a given climatic condition of the enclosed area (Papakonstantinou *et al*, 2000; Scigliano & Hollo, 2001; Allocca *et al*., 2002).

When natural ventilation can be a sufficient strategy for obtaining internal area comfort, project resources should be utilized. Among these resources are: the shape and direction of the building, the project forecast for open-spaced vertical ventilation, and the utilization of elements to direct the flux of air indoors. The building ventilation plan should make maximum use of local dominant winds. Besides the above-mentioned resources, some building project modifications should be evaluated, checking primarily the influence of the positioning of the openings in the walls, and later, the effect of the relative position of these buildings on the lot, and the comfort and health of its occupants. Also, there should be taken into account, the presence of obstacles and some variable factors, such as direction and frequency of the winds and the difference of the internal and external air temperature. (Emmerich *et al*, 2001).

When there is no incidence of wind, the chimney effect become the sole element responsible for the air renewal in the building and represents a simple situation for natural ventilation. If there are incidences of wind, this action should be in conjunction with the chimney effect, so that these movements sum together to result in the most efficient natural ventilation. For this to occur, it is fundamental that the configuration of the air flux, indoors, originates from the wind action, alone, and the direction of the flux from the difference in temperature sum up. When there is no combining of these two phenomena, the opposite of the same could result in some inconveniences, such as greater pressure due to the wind from the upper openings in relation to those originating in the chimney effect, impeding the escape of smoke and dust generated internally. (Heiselberg *et al*., 2001; Li *et a*l., 2003).

Single-sided natural ventilation is characterized by a single opening or more than one, situated in the same plane of the building (wall), in a zone of equal or slightly different pressure, making it difficult and serving as an obstacle for air circulation, Fig. (1).

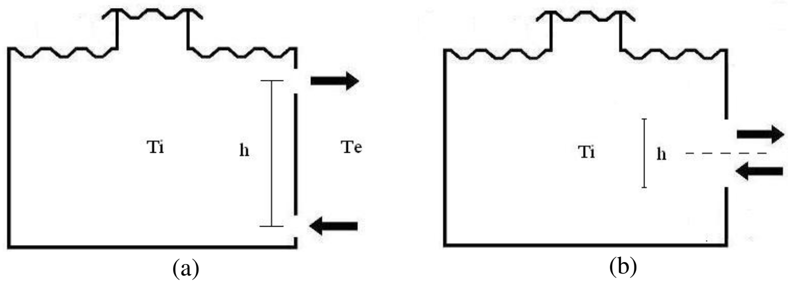



**Figure 1. Single-sided natural ventilation with air flux being generated through (a) a lower and upper openings, (b) single opening.**

The most commonly used methods for evaluating the distribution of indoor air flux are: the experimental method and the numerical methods. The numerical method places itself as a most widely-used tool in function of its flexibility in the variation of the shape of the object and the alteration of the parameters that influence the flux of air, when adequate mathematical models and appropriate computational programs are available.

In this work, the influence of the indoor heat sources on the efficiency of the single-sided natural ventilation was analyzed, taking into consideration lower and upper openings configuration.

## 2. MATHEMATICAL FORMULATION

### 2.1. Studied model

A typical office environment with dimensions of 4.7 x 2.9 m and ceiling 2.8 m off the floor, with two openings placed on the same wall were considered. The areas of the entrance and exhaust for the air are equally 0.60 m x 0.60 m. A source of internal energy, varying from 50 to 1000 Watts, is utilized to simulate the existence of electronic equipment indoors.

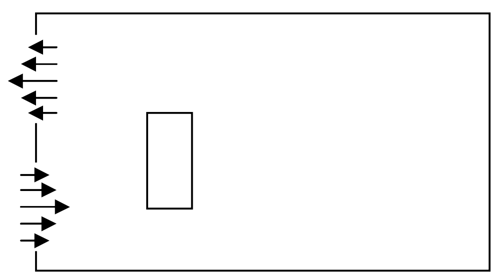


**Figure 2. Configuration of the studied model.**

In resolution of the governing equations, the following hypotheses were considered: bi-dimensional flux, turbulence, noncompressibility and a steady-state regime.

As boundary conditions are admitted that the air temperature of the entrance is known and all the surfaces of the domain are insulated. The air velocity at the exit is calculated using the expression (Etheridge & Sandberg, 1996),

$$Q_s = c_d A \sqrt{gh \frac{\Delta T}{T_e}} \tag{1}$$

where $Q_s$ is the volumetric flow ($m^3/s$), h (m) is the height between the opening centers (Fig. 1), $c_d$ is the non-dimensional coefficient for the load loss at the opening, considered equal to 0.6, $\Delta T$ is the temperature gradient between the outdoor and indoor environments, and $T_e$ is the external temperature. The flow in is calculated to satisfy the continuity equation,

$$\sum u_{2,j} = \sum u_{3,j} + \Delta u \; ; \qquad \Delta u = \frac{flow_{out} - flow_{in}}{\sum \rho_{2,j} A_{2,j}} \tag{2}$$

and,

$$\text{flow}_{out} = Q_s.\rho\ ; \qquad \text{flow}_{in} = \sum \rho_{3,j}.u_{3,j}.A_{3,j} \tag{3}$$

### 2.2. Mathematical and turbulence model

The equations for mass conservation, quantity of linear movement and energy, as well as time averages are present in their conservative and dimensional form as follows.

Mass Conservation,

$$\frac{\partial(\rho \bar{u}_i)}{\partial x_i} = 0 \tag{4}$$

Momentum,

$$\frac{\partial(\rho \bar{u}_j \bar{u}_i)}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu + \mu_t \frac{\partial \bar{u}_i}{\partial x_j}\right) + \frac{\partial}{\partial x_j}\left(\mu_t \frac{\partial \bar{u}_j}{\partial x_i}\right) + \rho_{ref}\, g\beta\left(\bar{T} - \bar{T}_{ref}\right)\delta_{2i} \tag{5}$$

where $\mu$ is the dynamic viscosity, $\mu_t$ is the turbulence viscosity, which in contrast to the dynamic viscosity, is not a property of the fluid, but, yes, a function of the state of turbulence in the flux. This can vary significantly from one point of flux to the next, as well as flux to flux. The definition of the Reynolds tension is not part of the turbulence model, but only in the structure to build each model. The main problem is in the artifact to be able to determine the distribution of the turbulence viscosity.

Energy Conservation.

$$\frac{\partial(\rho \bar{u}_j \bar{T})}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\frac{\kappa}{c_p} + \frac{\mu_t}{\sigma_t}\frac{\partial \bar{T}}{\partial x_j}\right) \tag{6}$$

where $\kappa$ is the thermal conductivity and $c_p$ is the specific heat under constant pressure, where $\mu_t$ is the turbulence viscosity and $\sigma_t$ the Prandtl turbulence number for the energy equation.

The turbulence model selected (Xu *et al*, 1998) is a two-differential equation for a low Reynolds number, LRN k-ε, originally developed by Jones and Launder (1972) and modified by Lam and Bremhorst (1981) and by Davidson (1990). This model reduced to the original Jones and Launder (1972) model, when the flux is occurring far from the wall. The transport equations for turbulent kinetic energy and for the dissipation rate of this energy are derived from Navier-Stokes equations.

Conservation of Turbulent Kinetic Energy

$$\frac{\partial(\rho \bar{u}_j k)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + G_B - \rho\varepsilon \tag{7}$$

where $\varepsilon$ is the dissipation rate of the turbulent kinetic energy. $P_k$ is the production term of k, and $G_B$ is the term associated to the thrust, all of which are defined in the following manner:

$$P_k = \mu_t\left(\frac{\partial \bar{u}_j}{\partial x_i} + \frac{\partial \bar{u}_i}{\partial x_j}\right)\frac{\partial \bar{u}_j}{\partial x_i} \qquad \text{and} \qquad G_B = -\frac{\mu_t g\beta}{\sigma_T}\frac{\partial \bar{T}}{\partial x_2} \tag{8}$$

Conservation of the Dissipation Rate of Turbulent Kinetic Energy.

$$\frac{\partial(\rho \bar{u}_j \varepsilon)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] - \frac{\varepsilon}{k}\left(f_1 c_{1\varepsilon} P_k + c_{1\varepsilon} G_B - f_2 c_{2\varepsilon}\rho\varepsilon\right) \tag{9}$$

with

$$f_1 = 1 + \left( \frac{0,14}{f_\mu} \right)^3 \qquad \text{and} \qquad f_2 = \left[1 - 0,27 \exp\left(-\mathrm{Re}_t^2\right)\right]\left[1 - \exp\left(-\mathrm{Re}_n\right)\right] \tag{10}$$

and the Reynolds numbers $\mathrm{Re}_t$ and $\mathrm{Re}_n$ are given by:

$$\mathrm{Re}_t = \frac{\rho k^2}{\mu \varepsilon} \qquad \mathrm{Re}_n = \frac{\rho \sqrt{k}\, n}{\mu} \tag{11}$$

where $n$ is the normal distance to the closest wall. $f_\mu$ is a function of the turbulent Reynolds number and is defined in the following form:

$$f_\mu = \exp\left[ - \frac{3,4}{\left(1 + \mathrm{Re}_t / 50\right)^2} \right] \tag{12}$$

For low Reynolds numbers, Davidson (1990) modified the function $f_\mu$ in the definition for turbulent viscosity $\mu_t$.

$$\mu_t = f_\mu c_\mu \overline{\rho} k^2 / \varepsilon \tag{13}$$

$c_\mu$ is an empirical constant. The constants and the empirical functions utilized here were taken from the work of Davidson (1990), so that: $\sigma_t = 0.9;\ \sigma_k = 1.0; \sigma_\varepsilon = 1.3; c_\mu = 0.09; c_{1\varepsilon} = 1.44; and\ c_{2\varepsilon} = 1.92$.

The boundary condition for the turbulent kinetic energy is that $k = 0$ for all the walls. For the dissipation rate of this energy, the flux at the walls is zero. The non-dimensional form of these equations shows that the flux is governed by the following parameters: the aspect rate, the Grashof number and the Prandtl number, respectively,

$$\mathrm{RA} = \frac{H}{L}; \qquad \mathrm{Gr} = \frac{g\beta \Delta T H^3}{\nu^2}; \qquad \mathrm{Pr} = \frac{\nu}{\alpha} \tag{14}$$

An analysis of the conservation equations shows that these can be rearranged and put in a simplified generic form.

$$\frac{\partial\left(\rho u_j \phi\right)}{\partial x_j} = \frac{\partial}{\partial x_j}\left( \Gamma \frac{\partial \phi}{\partial x_j} \right) + S \tag{15}$$

where $\phi$ is the dependent variable, $\Gamma$ is the diffusion coefficient, and S is the source term.

### 2.3. Numerical Model

The selected numerical model for the resolution of the equations for mass conservation, linear movement quantity, energy, turbulent kinetic energy and its rate of dissipation, is the method of finite differences with formulation for control volumes, developed by Patankar (1980). This method utilizes the Power Law interpolation scheme to evaluate the fluxes on the faces of the control volumes. The pressure-velocity coupling is assured by the algorithm SIMPLEC developed by Van Doormaan and Raithby (1984). The system of resultant discretized equations is resolved by algorithm TDMA (Tri-diagonal Matrix Algorithm). To implement convergence acceleration, an algorithm for correction by blocks was used and this transferred the information from the boundaries to the interior of the domain more quickly. As such, the dependent variable reached its correct level more rapidly. The equations are considered converged when the normalized residual is less than $1x10^{-5}$.

The mesh utilized presented a non uniform distribution of the nodes in both directions of the domain, with the control volumes being less new the walls and greater in the center of the cavity, Figure 3. The number of nodes are 101x 61 points in directions x and y, respectively. The mesh test presented discrepancies smaller than 2% when the average exit temperature is evaluated.

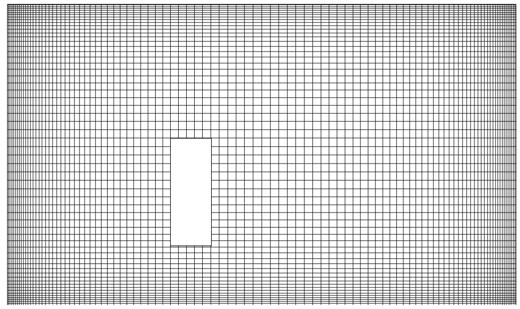


**Figure 3. Computational mesh.**

## 3. RESULTS AND DISCUSSIONS

The evolution of the entrance air velocity as function of the internal heat source is presented in Figure (4). As the heat load increases the velocity increases too. In Figure (5) the rate of air renovation per hour as function of the internal heat source is presented too. The value obtained in this work is compared with that obtained by semi-analytical model and the results encountered in Alloca et al (2003). The results show excellent agreement between numerical and semi-analytical model.

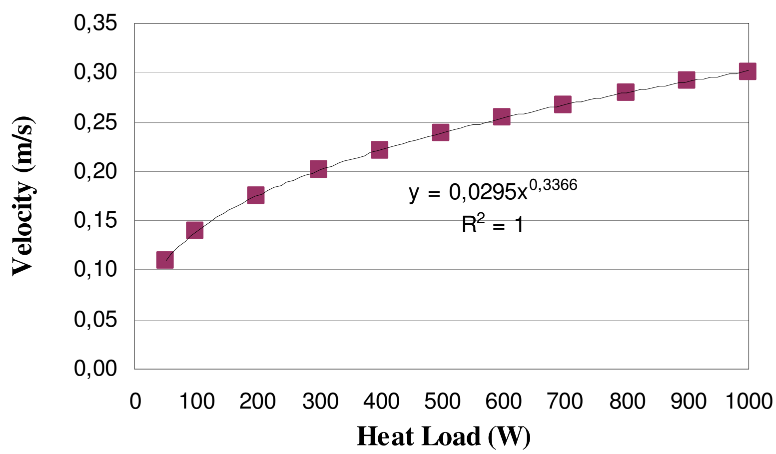



**Figura 4. Velocity over a range of internal heat loads.**

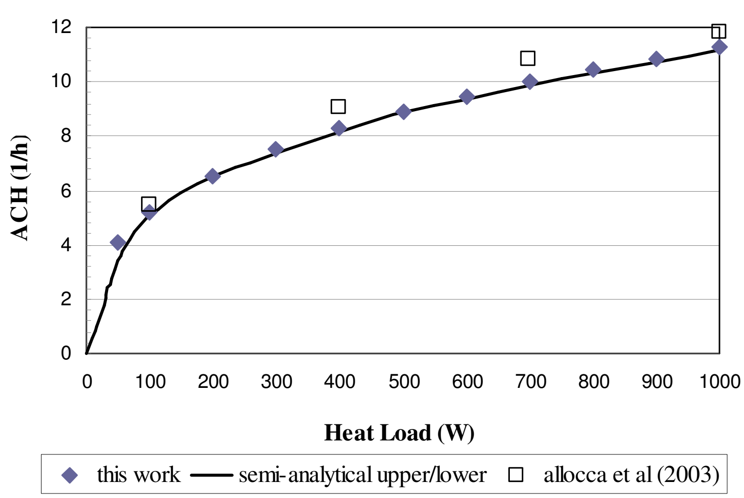



**Figura 5. Air change rate over a range of internal heat loads.**

In Figure 6, the isotherms distribution and vector velocity are presented with internal heat source variations. As isotherms are constructed simply by uniting the points of the same temperature and are spaced at 0.5 $^{o}C$ intervals between themselves.

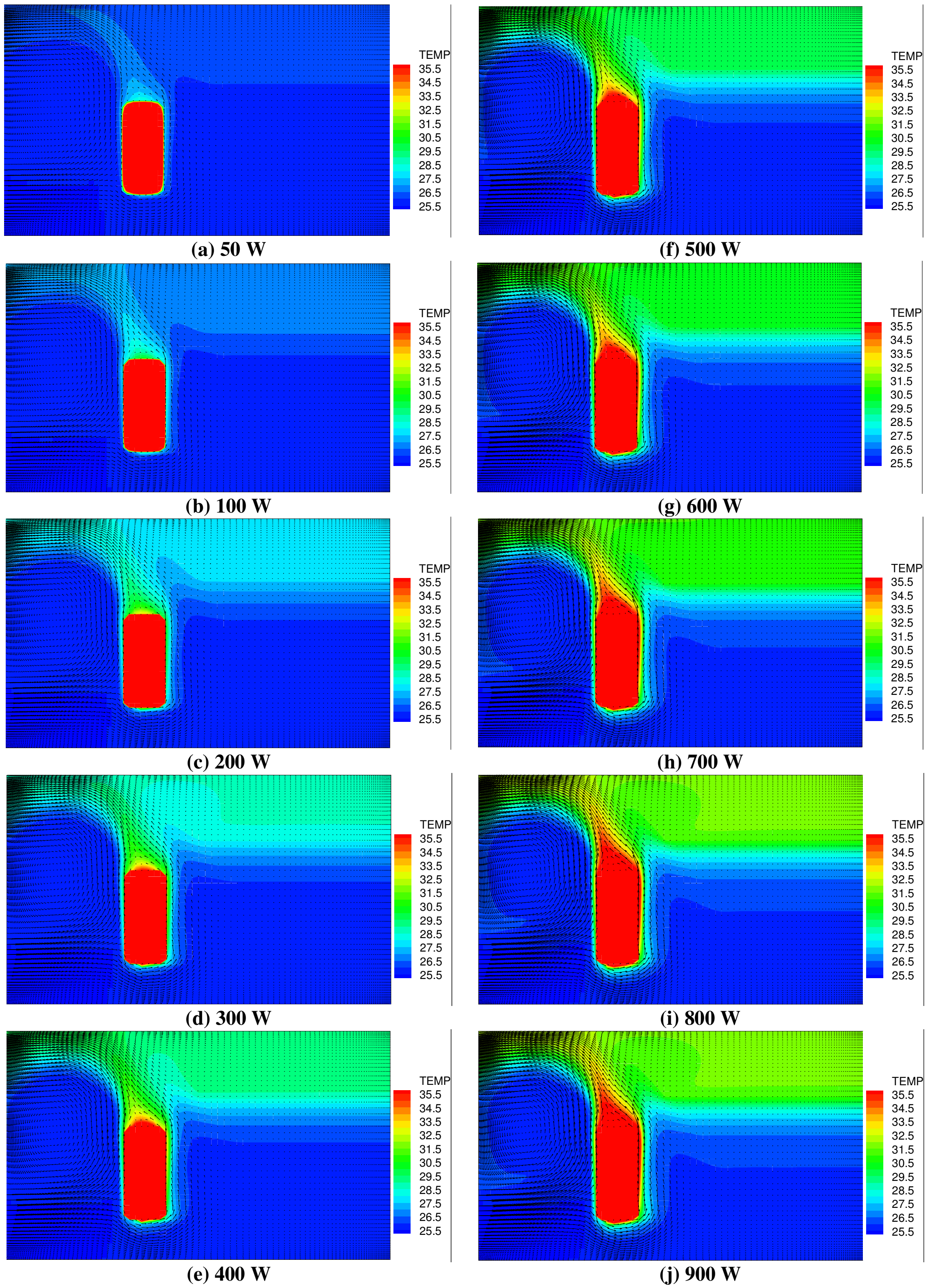
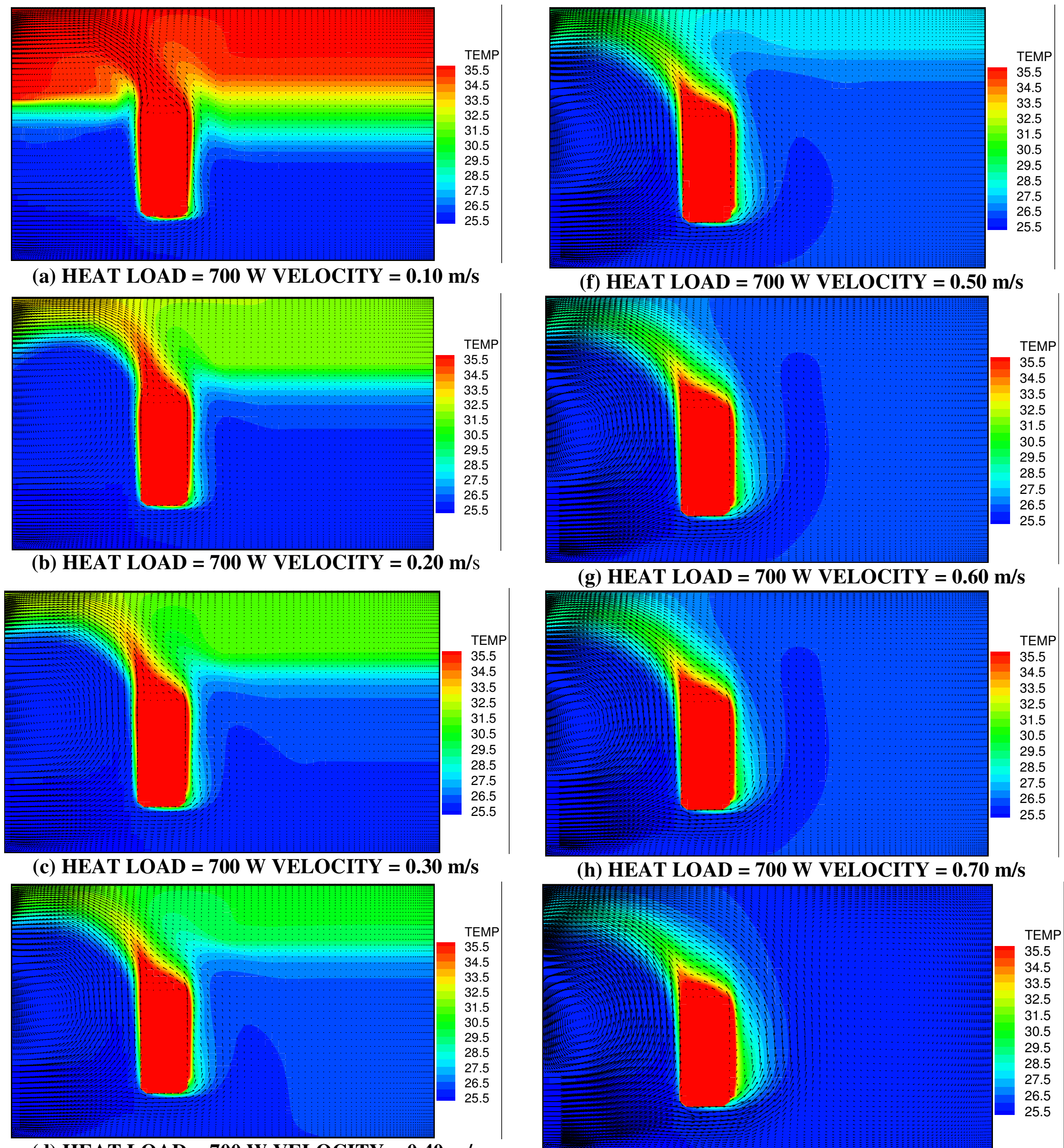



**Figure 6. Isotherms and vector velocity**

The temperature distribution exhibited typical stratification characteristics for this form of ventilation. It can be observed that the internal temperature increases as the internal heat source increases, producing, as such, a greater stratification of the internal temperature. There also occurs an increase in the air flux near the internal heat source, proportioning a greater air renovation in the environment. In Figure (7) presents the contour of indoor temperature and velocity vectors as function of velocity (ASHRAE 55, 2004).




**(a) HEAT LOAD = 700 W VELOCITY = 0.10 m/s**

**(b) HEAT LOAD = 700 W VELOCITY = 0.20 m/s**

**(c) HEAT LOAD = 700 W VELOCITY = 0.30 m/s**

**(d) HEAT LOAD = 700 W VELOCITY = 0.40 m/s**

**(f) HEAT LOAD = 700 W VELOCITY = 0.50 m/s**

**(g) HEAT LOAD = 700 W VELOCITY = 0.60 m/s**

**(h) HEAT LOAD = 700 W VELOCITY = 0.70 m/s**

**(i) HEAT LOAD = 700 W VELOCITY = 0.80 m/s**

**Figure 7. Contour of indoor temperature and velocity vectors**

As indicated swirling motions are created inside the room and the velocity airflow field resulted increased in buoyancy-driven flow patterns with the increased internal heat load.

The numerical method here presented has proven to be reliable, valuable tool in anlyzing buoyancy-driven single-sided ventilation over a range of conditions.

## 4. ACKNOWLEDGEMENTS

The authors give recognition and thanks to FAPEMIG and FUNDAÇÃO GORCEIX for their financial support.

## 5. REFERENCES

Alloca, C.; Chen, Q.; Glicksman, L. R.,2002, "Design analysis of single-sided natural ventilation", Energy and Buildings, London, Vol. 35, pp. 785-795.

ASHRAE- American Society of Heating, Refrigeration and Air Conditioning Engineers, 2004. "Ventilation for acceptable indoor air quality". ANSI/ASHRAE Standard 62.1: 2004, Inc., Atlanta, GA.

ASHRAE -American Society of Heating, Refrigeration and Air Conditioning Engineers, 2004. "Thermal Environment Conditions for Human occupancy". ANSI/ASHRAE Standard 55:2004. New York, USA, 2004.

Davidson, L. "Calculation of the Turbulent Buoyancy-Driven Flow in a Rectangular Cavity Using an Efficient Solver and Two Different Low Reynolds Number k-ε Turbulence Models". Numerical Heat Transfer, Part A, vol. 18, pp. 129-147, 1990.

Emmerich, S. J.; Dols, W. S.; Axley, J. W., 2001, "Natural Ventilation Review and Plan for Design and Analysis Tools", Colorado. Disponível em: < http://fire.nist.gov/bfrlpubs/build01/PDF/b01073.pdf>. Acesso em: 2005.

Etheridge, D.; Sandberg, M., 1996. "Building Ventilation: Theory and Measurement". Wiley, Chichester, UK.

Heiselberg, P., Svidt, K., Nielsen, P. V., 2001, "Characteristics of airflow from open windows", Building and Environment, Vol. 36, pp. 859-869.

J. P. Van Doormaan,G. D Raithby. Enhancements of the SIMPLE Method for Prediction Incompressible Fluid Flow. Numerical Heat Transfer, Vol.7, pp.147-163, 1984.

Jones, W.P., Lauder, B.E.. "The Prediction of Laminarization with Two-Equation Model of Turbulence". Int. J. Heat Mass Transfer, vol. 15, pp. 301-314, 1972.

Lam, C.K.G, Bremhorst, K.A. "A Modified Form of the k-ε Model for Predicting Wall Turbulence". J. Fluid Eng., Vol.103, pp.456-460, 1981

Li, Y.; Delsante, A., 2003, "Natural ventilation induced by combined wind and thermal forces", Building and Environment, London, Vol. 36, pp. 59-71.

Papakonstantinou, K. A.; Kiranoudis, C. T.; Markatos, N. C., 2000, "Numerical simulation of air flow field in single-sided ventilated buildings", Energy and Buildings, Vol. 33, pp. 41-48.

Patankar, S.V. "Numerical Heat Transfer and Fluid Flow". Hemisphere, New York, 1980.

Scigliano, S., Hollo, V., 2001, "IVN – Índice de ventilação natural", Editora Pini, São Paulo, 279 p.

Xu, W., Chen, Q., Nieuwstadt, F. T. M., 1998. "A new turbulence model for near-wall natural convection". International Journal of Heat and Mass Transfer, vol. 41, pp. 3161 – 3176.

## 6. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.