Numerical study of the influence of internal heat source in naturally ventilated offices

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Abstract

The natural ventilation in a building performs a fundamental role in users' welfare and health, when used as a bioclimatic and architectural strategy, it can provide brighter projects. Increasing the indoor air speed ventilation has an important performance maintaining indoor air quality, contributing to thermal interactions between human body and environment, which can significantly reduce the building's energy cost. In this study, a numerical research of natural convection in single-sided and cross ventilation is developed using the finite volume method, taking account the internal heat sources influence. The numerical model is used to determine the effects of wind and thermal forces combined by analyzing the stream functions and temperature distribution within an office. The turbulence model is the two differential equations for the low Reynolds number. The building's internal heat sources influence is analyzed herein. The results show that the single-sided ventilation presents excellent conformity with the results obtained in literature. The influence of the internal heat source is considered and it has been observed that the best source position is near to the air exit for both single-sided ventilation and cross ventilation.

keywords: natural ventilation, single-sided ventilation, cross ventilation, thermal comfort, numerical solution.

1. Introduction

Natural ventilation can be used as an important strategy for local passive cooling and achieving a more comfortable indoor environment. The air renewal in a room, where appropriate, provides health and comfort to the occupants. In the summer, the natural ventilation's primary purpose is to increase the human heat dissipation by convection and evaporation to achieve a comfort feeling. In the winter, the natural ventilation's function is to keep the internal air quality, at a smaller air renovation rate than during the summer (Ashrae 62, 2004).

The external natural airflow to the inside is basically for two purposes: wind action and thermal buoyancy. The ventilation is linked to the temperature difference due to thermal buoyancy. The wind force ventilation provides the airflow from a positive pressure region through the opening, in the facade with front to the prevailing wind direction, and the air exhaustion, through the opening located on the opposite facade, which has a nega-

tive pressure. The airflow depends on the wind's speed and direction as well as the size of the openings (Heiselberg, Svidt and Nielsen, 2001; Tantasavasdi, Srebric, and Chen, 2001; Li and Delsante, 2003; Lipin and Hien, 2007; Visagavel and Srinivasan, 2009).

The airflow depends on the wind's speed and direction as well as the size of the openings. Marian (2013) analyzes the influence of different types of windows and furniture in natural ventilation in office environments. The maximum window-type air showed higher efficiency compared to vertical sliding window, providing a higher volume and a better distribution of air in the room.

The thermal efficiency of natural ventilation depends on the external air temperature and the internal airflow of the enclosed area. Airflow velocity is the most important factor in the evaluation of thermal sensation during ventilation, since it provides a hot or cold sensation, even when the air temperature remains con-

stant. 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 airflow can be altered according to the size and position of the strategically placed openings, and also by the shape and volume of the enclosed areas, for a specific climatic condition of the enclosed area (Papakonstantinou, Kiranoudis and Markatos, 2000; Allocca, Chen and Glicksman, 2003).

When there is no incidence of wind, the buoyancy-driven flow becomes the only element responsible for the air renewal in the building and represents a simple situation for natural ventilation. If there are wind incidences, this action should be in conjunction with the buoyancy-driven flow, so that these movements sum up to result in the most efficient natural ventilation. For this result to happen, it is fundamental that the configuration of the indoor airflow originates from the wind action

alone, and that the flux direction comes from the difference in the mentioned temperature sum. When there is no combining of these two phenomena, the opposite could result in some inconveniences, such as greater pressure due to the wind from the upper openings when compared to those originated in the stuck effect, impeding the escape of smoke and dust generated internally (Heiselberg, Svidt and Nielsen, 2001; Li and Delsante, 2003).

In this context, the study of the influence of power and the position of an internal heat source in an office, it helps to understand the direction of air currents and the degree of warming within this environment.

1.1 Cross ventilation

Among the usual engineering solutions, cross ventilation is very efficient and provided by openings in opposite walls (or different) with pressure differential caused by the wind action. The cross ventilation can provide a higher renovation rate than in the single-sided ventilation, uniformly ventilating the building (Allard and Ghiaus, 2005). So it is important that there is no obstruction between the windows and/or openings in the opposite facades, Fig. 1.

1.2 Single-sided ventilation

When the cross ventilation is not possible in some buildings due to the built project shape and position, single-sided ventilation is used, Fig. 1. The natural single-sided ventilation is characterized by openings in only one building facade (Papakonstantinou, Kiranoudis and Markatos, 2000; Allocca, Chen and Glicksman, 2003, Awbi, 2003).

2. Physical and mathematical model

The physical model is a typical office with dimensions 4.7 x 2.9 m and 2.8 m height. The entrance and exit air areas have 0.60 m x 0.60 m. A 700 W internal heat source simulates electronic equipment and people inside the office. The single-sided ventilation (inflow and outflow in the same plane) and cross ventilation (inflow and outflow on opposite planes), Fig. 1, are analyzed.

Figure 1
Physical and
mathematical model configuration.

Steady state, two-dimensional, turbulent and incompressible flow are considered. The air inflow temperature is known and all domain surfaces are insulated. Air velocity at exit is calculated using the expression (Etheridge and Sandberg, 1996):

$$Q_s = c_d A \sqrt{gh \frac{\Delta T}{T_e}} \qquad (1)$$

where Q_s is the volumetric flow (m³/s), h (m) is the height between the opening centers (single-sided ventilation) or the height opening (cross ventilation), A is the opening area (m²), g is equal 9.81m²/s, c_d is a non-dimensional coefficient for the load loss at opening and is equal to 0.6, ΔT is the temperature gradient between the outdoor and indoor environments, and T_e is the external temperature (K).

When the difference between the walls temperatures and the air temperature is small when compared to the air's absolute temperature in the domain entrance, Boussinesq approximation is used; that is, in terms of body force in the equations of Navier-Stokes, the specific mass is defined as a linear function of the temperature:

$$\rho = \rho_{ref} + \rho_{ref} \beta (T - T_{ref}) \qquad (2)$$

T is the air temperature, ρ the specific mass, ρ_{ref} the reference specific mass and β the volumetric expansion coefficient.

Mass, momentum and energy conservations, in time averages, are presented in their conservative and dimensional form as follows.

Mass conservation equation:

$$\frac{\partial (\rho \bar{u}_i)}{\partial x_i} \qquad (3)$$

where ρ is the specific mass and *u* the velocity vector.

Momentum conservation equation:

$$(4) \quad \frac{\partial(\rho \bar{u}_j \bar{u}_i)}{\partial x_j} = - \frac{\partial \bar{\rho}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_t)\frac{\partial \bar{u}_i}{\partial x_j}\right] + \frac{\partial}{\partial x_j}\left[(\mu + \mu_t)\frac{\partial \bar{u}_j}{\partial x_i}\right] + \rho_{ref}\, g\, \beta\, (\bar{T} - \bar{T}_{ref})\, \delta_{2i}$$

where μ is the dynamic viscosity, μ_t is the turbulence viscosity.
Energy conservation equation:

$$(5) \quad \frac{\partial(\rho \bar{u}_j T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\frac{\lambda}{C_p} + \frac{\mu_t}{\sigma_t}\right)\frac{\partial T}{\partial x_j}\right]$$

Where λ is the thermal conductivity, c_p the specific heat under constant pressure and σ_t the Prandtl turbulence number for the energy equation.

Turbulence viscosity, in contrast to dynamic viscosity, is not a fluid property but a function of the turbulence state in the flux. This can vary significantly from one flux point to the next, as well as flux to flux. The definition of Reynolds tension is not part of the turbulence model, but only one structure to build each model. The transport equation for turbulent kinetic energy, derived from Navier-Stokes equations, is:

$$(6) \quad \frac{\partial(\rho \bar{u}_j K)}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu_t + \frac{\mu_t}{\sigma_K}\right)\frac{\partial K}{\partial x_j}\right] + P_K + G_B - \rho\varepsilon$$

where k is the dissipation rate of the turbulent kinetic energy, P_k is the shear production of turbulent kinetic energy, and G_B is the buoyant production of turbulent kinetic energy, all of which are defined in the following way:

$$(7) \quad P_K = \mu_t\left(\frac{\partial \bar{u}_j}{\partial x_i} + \frac{\partial \bar{u}_i}{\partial x_j}\right)\frac{\partial \bar{u}_j}{\partial x_i} \qquad G_B = - \frac{\mu_t\, g\, \beta}{\sigma_T}\frac{\partial \bar{T}}{\partial x_2}$$

The turbulence model (Xu, Chen, and Nieuwstadt, 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.

3. Numerical model

A 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). Power Law interpolation scheme is used to evaluate the fluxes on the faces of the control volumes. Pressure-velocity coupling is assured by the algorithm SIMPLEC developed by Van Doormaan and Raithby (1984). The TDMA (Tri-diagonal Matrix Algorithm) algorithm resolves the system of resultant discretized equations. The block correction algorithm is implemented to convergence acceleration. Convergence is achieved when the normalized residual is less than 1×10^{-5}. The mesh presents a non-uniform distribution on nodes in both directions of the domain, with the control volumes being less new in the walls and greater in the center of the cavity, Fig. 2a. The number of nodes are 101x 61 points in x and y directions, respectively. The mesh test presents discrepancies smaller than 2% when the average exit temperature is evaluated.

The air change rate (ACH - air flow per indoor volume) as a function of the internal heat source, is presented in Fig. 2b for single-sided ventilation. As the heat load increases, velocity increases too. Results are compared with Eq. (1) and Allocca, Chen and Glicksman (2003) showing excellent agreement between the numerical and semi-analytical models.
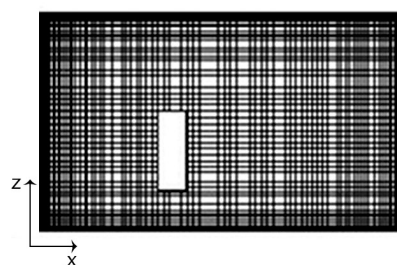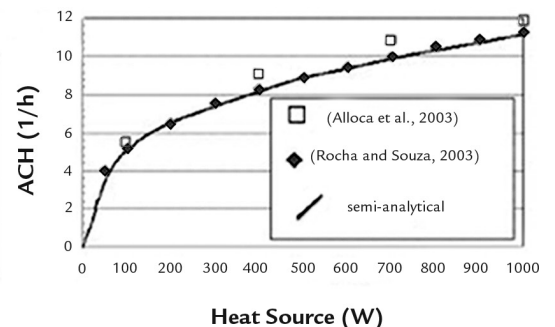

Figure 2
a) Computational mesh;
b) numerical model validation

4. Results

4.1 Internal heat source position

Figures 3 and 4 presents isotherm profiles when an internal heat source (700 W) had its position changed from left to right side in the office, respectively. All cases revel around the source, a bubble with high temperature.
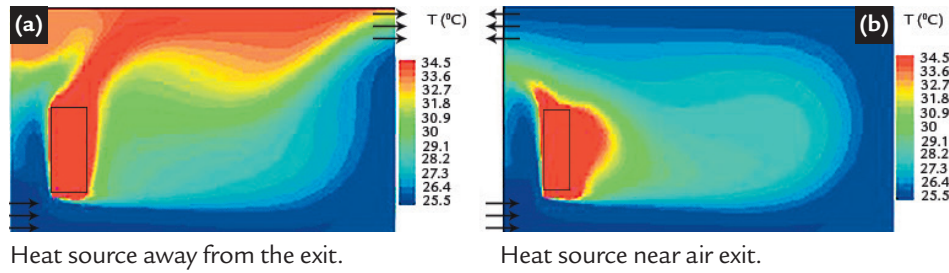

(a) Heat source away from the exit.

(b) Heat source near air exit.

Figure 3
Heat transfer on left side of the office.

Figure 3 shows the results that when the internal heat source is on left side, a lower recirculation is formed in the middle plane of the office and a large region of discomfort can be seen, with temperatures ranging from 27 to 31 °C. Figure 3a shows results when a cross-ventilation configuration is used. On top of the office appears one region with high temperatures caused by the mass flow passing near the heat source.

Temperatures vary from 29 to 35 °C on top of the office. Figure 4b shows the results for a single-sided ventilation configuration. No high temperatures where encountered on top of the office.

Figure 4 presents results when the internal heat source is on right side of the office. In single-sided ventilation, Fig. 4a, when the heat source is away from the air exit, a large recirculation is formed in almost all of the office,

making the temperature remain below 29 °C near the air entrance and rise to 31 °C near on internal heat source. Around the heat source, the temperature reaches 35 °C. However, when the heat source is near the air exit, Fig. 4b, it is observed in practically all the office that the temperature remains uniform and equal to 25.5 °C. Only in the heat source region is there observed higher temperature values.
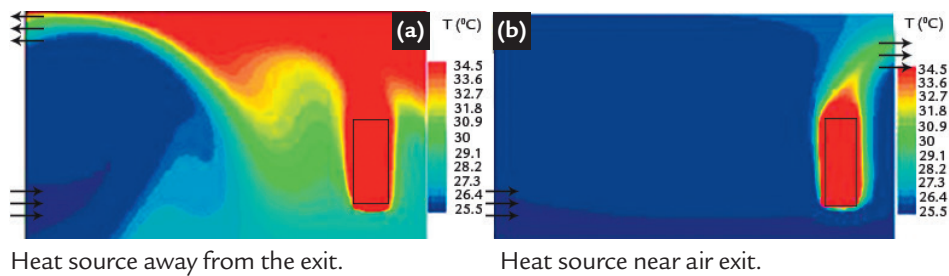

(a) Heat source away from the exit.

(b) Heat source near air exit.

Figure 4
Heat transfer on right side of the office.

4.2 Internal heat source power

Figure 5 shows air renovation (rate per hour) varying with the heat source for single-sided ventilation and cross ventilation. It is observed, in cross ventilation case, that internal air renovation rate is higher than in single-sided ventilation. This fact is evidenced by the internal vortexes elevation (Fig. 7). The internal heat source increase causes a higher variation in air renovation rate in the single-sided ventilation.
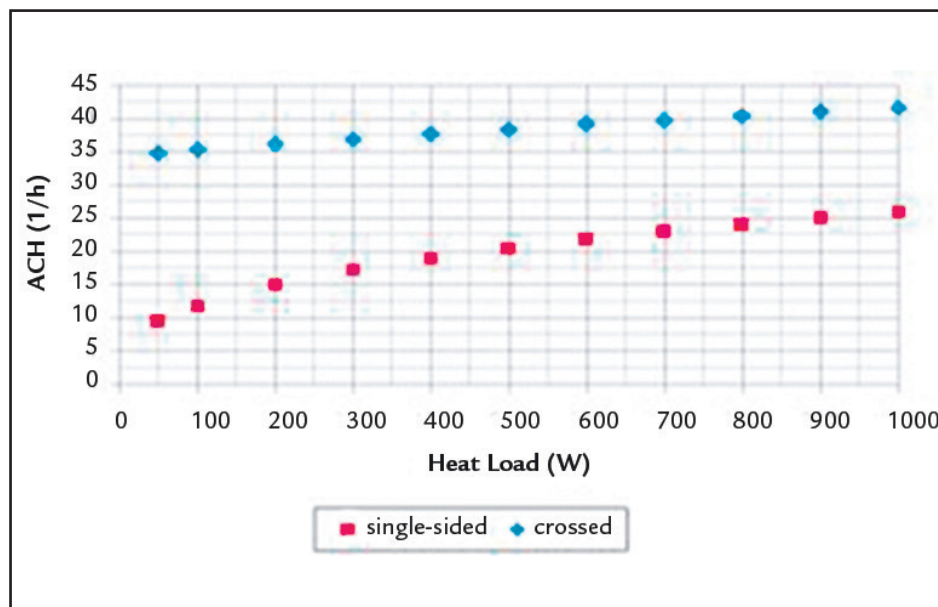

Figure 5
Air change rate versus heat source: single--sided ventilation and cross ventilation.

Figures 6 and 7 present isotherms and vector velocities when the heat source is changed from 50 to 900 Watts in single-sided and in cross ventilation, respectively. Results show (Fig. 6) that the temperature increases when internal heat source increases, revealing a bigger stratification in the temperature profile, due to buoyancy, on right side of heat source. As indicated, on the left side of heat source, the vortexes created due to the internal heat source influence that increases the airflow when the heat source increases. However, in this case (single sided ventilation) and depending on the office height, the office air temperature is maintained within the comfort range (Ashrae 55, 2013). A suitable condition for the air quality cannot be obtained depending on the occupation profile, represented by the elevation of the heat source level. In crossed ventilation, (Fig. 7), the vortex formation, as also an influence of the heat source, makes the airflow more uniform in the office. In this case, the level of temperature near the source becomes higher as the internal source increases, which could lead to a discomfort situation in all points of the office.
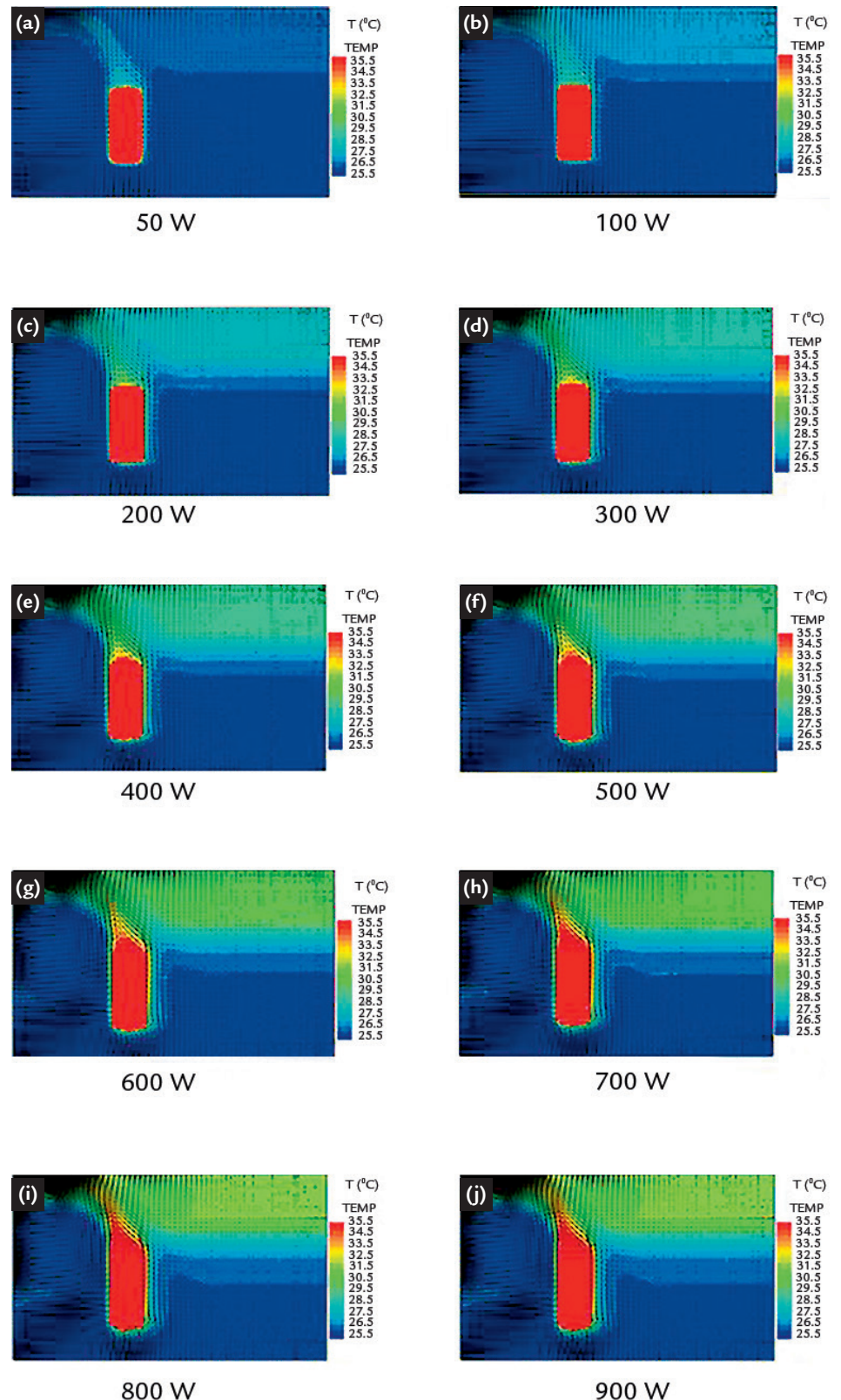

Figure 6
Temperature profile and vector velocity: single-sided ventilation.
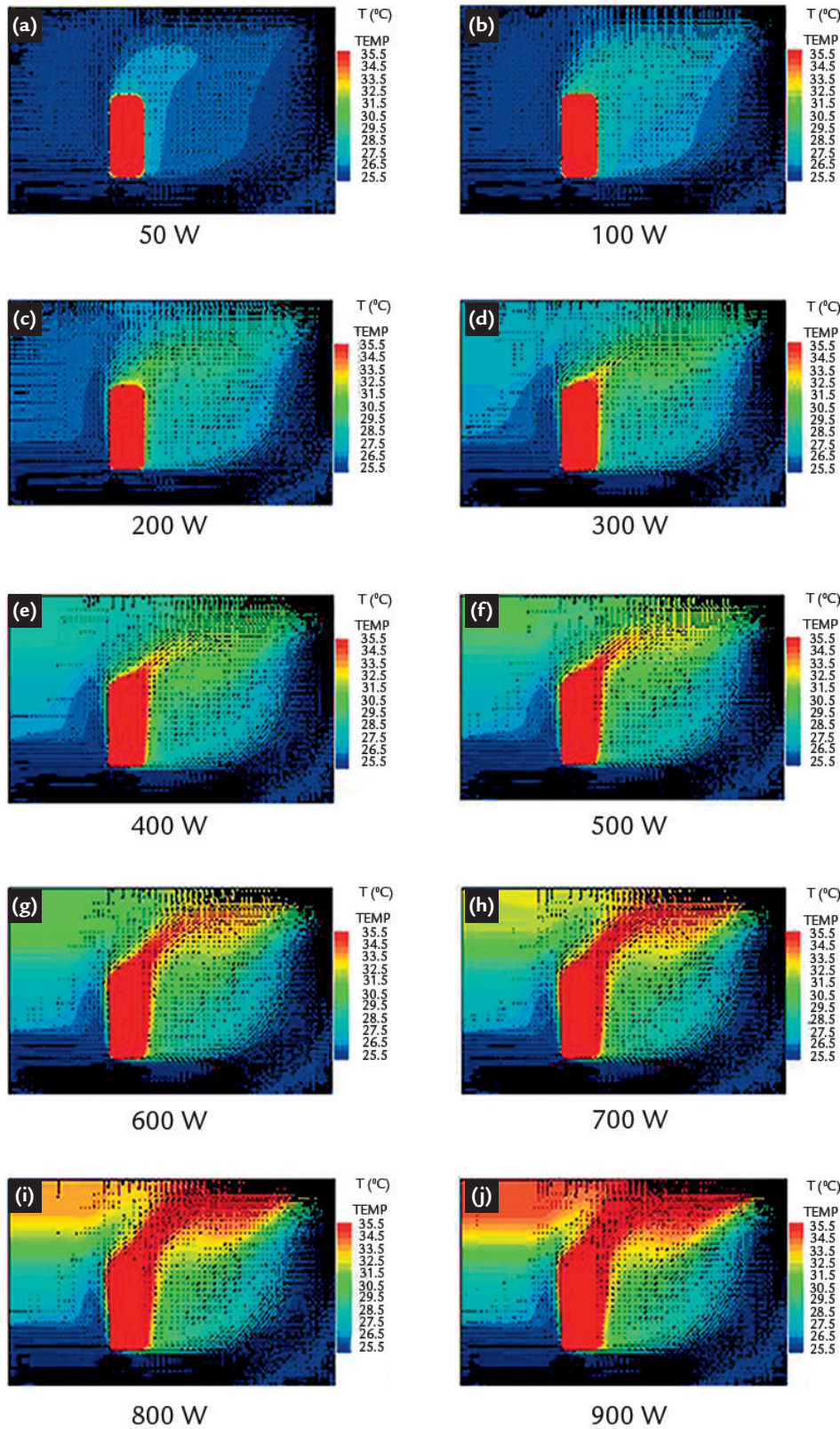
Figure 7
Temperature profile and
vector velocity: cross ventilation.

5. Conclusions

The results show that the temperature distribution within the room is influenced by the location of the internal heat source. The environment can present appropriate conditions for human comfort whenever the internal source is located next to the outlet, for a speed range between 0.25 m/s to 0.80 m/s.

According to the results, we conclude that the best configuration is one in which the heat source is near the exit of air. You can also see that crossed ventilation is more effective than unilateral ventilation, even when the heat source is far from the air exit.

In order to obtain an internal thermal environment that meets human needs of comfort, different sizes, shapes and positions of vents should be studied and modeled under different wind flow conditions (direction and speed) taking into account the profile of room occupancy. However this is a nontrivial solution of the problem and

in this context the numerical solution   is to play an important role, taking the   example the study presented here.

6. Acknowledgments

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