CONVECTIVE COOLING OF ELECTRONIC EQUIPMENT

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Abstract. *An analysis of heat transfer inside a semi porous two-dimensional rectangular open cavity was numerically examinated. The open cavity comprises two vertical walls closed to the bottom by an uniform heat fluxt. One vertical wall is a porous wall and an inflow of fluid occurs normal to it. The other wall transfers the same uniform heat flux to the cavity. It shows how natural convection effects may enhance the forced convection inside the open cavity. The main motivation for this research is its application for electronic equipment where the devices used for the electronic equipment cooling are frequently based on natural and forced convection where the equipment may reach a dangerous limit temperature reducing its efficiency. Governing equations are expressed in Cartesian coordinates and numerically handled by a finite volume method. Results of the maximum temperature are presented for both Reynolds and Grashoff numbers at the heated wall and in the bottom..*

Keywords*: Electronic cooling, finite volume, natural and forced convection*

1. INTRODUCTION

The heat transfer in enclosures has been studied for a variety of engineering applications. Results have been presented in research surveys such as in Bruchberg et al. (1976), Kakaç et al. (1987), and it has become a main topic in convective heat transfer textbooks (Bejan, 1984). Usually the enclosures are closed and natural convection is the single heat transfer mechanism. There are however several applications in passive solar heating, energy conservation in building and cooling of electronic equipment, where open cavities are employed (Chan and Tien, 1985, Hess and Henze, 1984 and Penot, 1982). Ramesh and Merzkirch (2001) present a study of steady, combined laminar natural convection and surface radiation from side-vented open cavities with top opening; Gunes and Liakopoulos (2003) study, by a spectral element method, the three-dimensional free convection in a vertical channel with spatially periodic, flush-mounted heat sources; Cheng and Lin (2005) present a optimization method of thermoelectric coolers using genetic algorithms and Vasiliev (2006) presents a short review on the micro and miniature heat pipes used as electronic component coolers.

Devices employed for the cooling of electronic equipment are frequently based on forced convection (Sparrow et al., 1985). Altemani and Chaves (1988) present a numerical study of heat transfer inside a semi porous two-dimensional rectangular open cavity for both local and average Nusselt numbers at the heated wall and for the isotherms and streamlines of the fluid flowing inside the open cavity. This paper presents a continued work where one makes a numerical analysis of the heat transfer inside a semi porous two-dimensional rectangular open cavity. It is made by two vertical parallel plates opened at the top and closed at the bottom by an uniform heat flux and open at the top, as indicated in Fig. 1. One of the vertical plates is porous and there is a normal forced fluid flow passing through. The opposite vertical plate supplies the same uniform heat flux to the cavity. In addition to the forced convection, the analysis considered the influence of natural convection effects. The maximum temperature are obtained for the uniformly heated plate and to the bottom for the parameters governing the heat transfer: Reynolds (Rep) and Grashoff (Gr) numbers.

2. ANALYSIS

The conservation equation of mass, momentum and energy, as well as their boundary conditions, will be expressed for the system indicated in Fig. 1. Due to the low velocities usually associated with permeable walls, the natural convection will be considered in the analysis. It is assumed that the flow is laminar and occurs under steady state conditions.

The natural convection will be treated via the Boussinesq approximation, i.e., density variations are accounted for only when they contribute to buoyancy forces. In this problem, the buoyancy term is obtained from the y momentum equation terms representing the pressure and body forces:

$$-\frac{\partial p}{\partial y}-\rho \cdot g \tag{1}$$

where g is the acceleration of gravity; ρ is the specific mass and $\frac{\partial p}{\partial y}$ is the pressure gradient in y direction.

The density is related to temperature according to the Boussinesq approximation:

$$\rho=\rho_p-\rho_p \cdot \beta \cdot (T-T_p) \tag{2}$$

where ρ is the density, β is the coefficient of thermal expansion, T is the temperature and T_p indicates the temperature of the fluid inlet at the porous wall.
In Fig. 1, x and y are the Cartesian coordinates, D is the width of the open cavity, q is the surface heat flux and H is the height of the open cavity. In (2) T_p indicates the inlet temperature of the fluid at the porous wall and ρ_p the corresponding density. The pressure is now expressed in terms of a modified pressure defined as

$$p^*=p+\rho_p \cdot g \cdot y \tag{3}$$

where p^* is the modified pressure. With eqs. (2) and (3), eq. (1) can be expressed by

$$-\frac{\partial p^*}{\partial y}+\rho \cdot g \cdot \beta \cdot (T-T_p) \tag{4}$$

The second term in this equation relates the buoyancy forces to temperature differences (T – Tp). According to this formulation, the density will be assumed constant and equal to ρ_p in all the equations, so that the subscript p may be deleted. It is also assumed that all other fluid properties are constant. Viscous dissipation and compression work are not considered in the analysis, according to the low velocities, moderate temperature differences and laminar flow conditions assumed.

In order to obtain the conservation equations in dimensionless form, the following variables were defined:

$$X=\frac{x}{D}\ ,\quad Y=\frac{y}{D} \tag{5a}$$

$$U=u \cdot \frac{D}{\nu}\ ,\quad V=v \cdot \frac{D}{\nu} \tag{5b}$$

$$P=\frac{p^*}{\left(\frac{\rho \cdot \nu^2}{H^2}\right)}\ ,\quad \theta=\frac{T-T_p}{\left(\frac{q \cdot D}{k}\right)} \tag{5c}$$

where D is the width of the open cavity, P is the dimensionless pressure, θ is the dimensionless temperature, θ_w is the dimensionless heated wall temperature and U and V are the dimensionless velocities.

The equations expressing conservation of mass, x and y momentum and energy then become

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{6}$$

$$U \cdot \frac{\partial U}{\partial X}+V \cdot \frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\nabla^2 U \tag{7}$$

$$U \cdot \frac{\partial V}{\partial X}+V \cdot \frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\nabla^2 V+Gr \cdot \theta \tag{8}$$

$$U \cdot \frac{\partial \theta}{\partial X} + V \cdot \frac{\partial \theta}{\partial Y} = \frac{\nabla^2 \theta}{Pr} \quad (9)$$

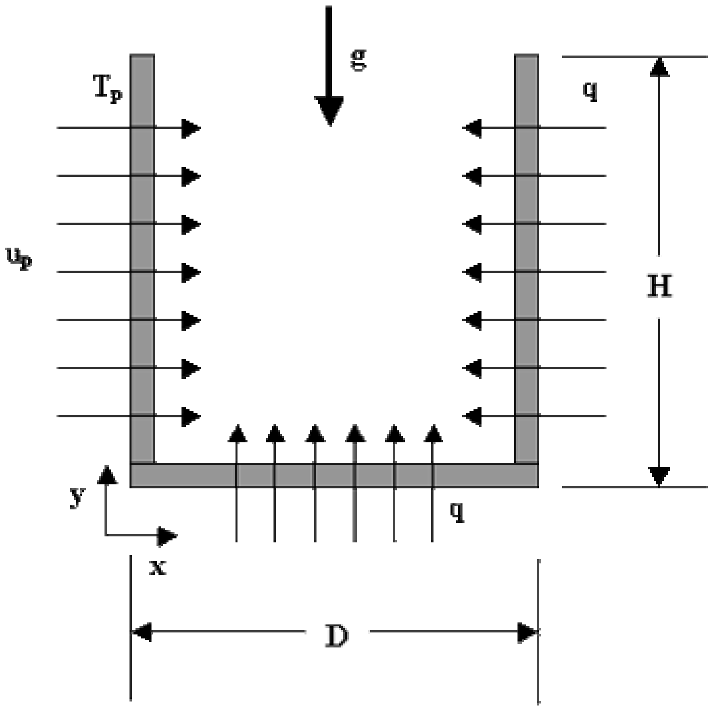

Figure 1. Coordinate system and thermal boundary conditions of the open cavity

In eqs. (7) to (9) ∇^2 is the Laplace operator in cartesian coordinates. These equations are coupled and present two independent parameters, Gr and Pr. The first is the modified Grashoff number, defined by

$$Gr = \frac{g \cdot \beta \cdot q \cdot D^4}{k \cdot \nu^2} \quad (10)$$

and the second is the Prandtl number of the fluid. In eq. (10) ν is the kinematic viscosity.

At the three solid boundaries of the open cavity, the velocity components are null, except the velocity of injection of the fluid (Up) at the porous wall. The thermal boundary conditions comprise a uniform (reference) temperature at the porous wall and a specified heat flux at the heated vertical wall and in the bottom. Expressed in dimensionless terms, the boundary conditions become:

$$X = 0 \ ; \ U_p = u_p \frac{D}{\nu} = Re_p \ ; \ V = 0 \ , \ \theta = 0 \quad (11a)$$

$$X = 1 \ ; \ U = 0 \ ; \ V = 0 \ , \ \frac{\partial \theta}{\partial X} = 1 \quad (11b)$$

$$Y = 0 \ ; \ U = 0 \ ; \ V = 0 \ , \ \frac{\partial \theta}{\partial Y} = 1 \quad (11c)$$

where Rep is the porous wall Reynolds number.

The dimensionless velocity component normal to the permeable wall ($u_p \frac{D}{\nu}$) is one parameter of this problem and it will be designated the porous wall Reynolds number, Rep. The outflow boundary of the open cavity, at Y equal to H/D (where H is the height of the open cavity), is just a virtual boundary defining the calculation domain. In order to obtain a solution, two conditions must be satisfied at this boundary. First, there must be no backflow of fluid and second, there must be no diffusion from outside into the calculation domain. The first condition was verified checking the velocity profiles of each result obtained and discarding those results when a backflow was observed. The second was satisfied imposing artificially negligible partial derivatives of θ and U in the vertical direction at the outflow boundary. The velocity component V was corrected at the outflow boundary in order to satisfy the conservation of mass in the whole domain.

The problem presents four independent parameters: H/D, Pr, Rep and Gr. For a fixed particular fluid, there are still three parameters governing the heat transfer: H/D, Rep and Gr. In the present work, a single value, equal to 0.72, was assigned to the Prandtl number and H/D = 1.

3. METHODOLOGY

The differential eqs. (6) to (9) together with their boundary conditions (11), determine a coupled system involving the four variables U, V, P and θ. The equations were discretized using the control volume formulation described in (Patankar, 1980) and the solution was obtained employing the SIMPLE scheme. The convergence of the results was accepted when the relative change of the dependent variables was under 10^{-3}.

4. RESULTS AND DISCUSSION

The maximum temperature T_{max} is shown on Fig. 2 as function of the modified Grashof number for Reynolds number equal to 1 and 100. Considering the range of the modified Grashof number analyzed the maximum temperature is shown and as Gr is the ratio of buoyancy forces to viscous forces it can be seen that the influence of forced convection is dominant for Gr until 1×10^4. After this value, T_{max} is increasingly abrupt caused by influence of the buoyancy effects.

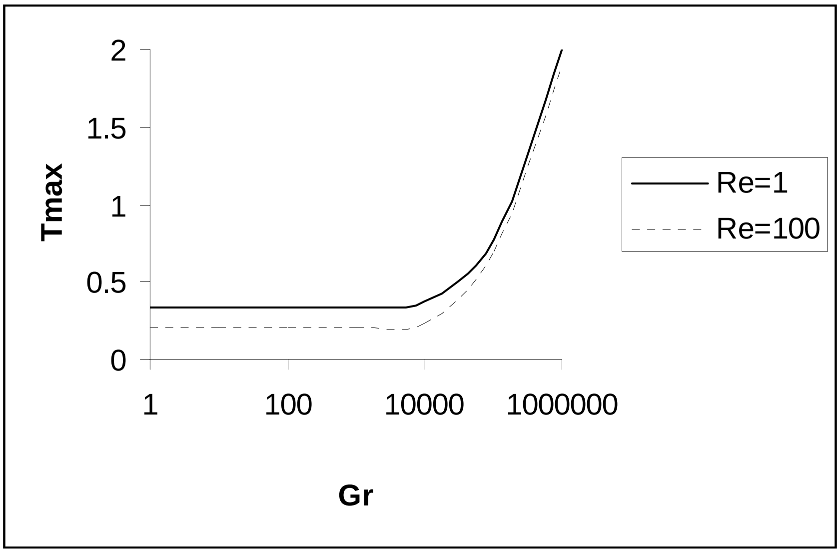

Figure 2. Maximum temperature as a function of Grashoff number for Re=1 e Re=100.

The maximum temperature T_{max} is shown on Fig. 3 as function of the Reynolds number for the modified Grashof number equal to 0 and 100,000. It is noticed that the behavior of the curve is affected by Gr. For Gr equal 100,000 buoyancy forces are bigger than viscous forces and it increases the cooler effect because the convective forces increase. Considering the range of the Reynolds number analyzed the maximum temperature as shown, T_{max} decrease with the Reynolds number. The influence of forced convection is dominant after Reynolds equal to 100.

The effects of natural convection in the maximum temperature are shown on Fig. 4 and Fig. 5. Figure 4 shows the effects of the forced convection and Fig. 5 shows a larger penetration into the temperature in the upper part of the cavity when Grashof increases imposed by natural convection effects.

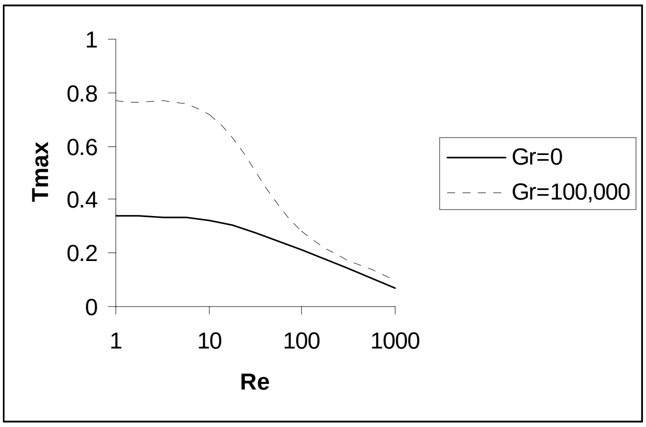

Figure 3. Maximum temperature as a function of the Reynold number for GR=0 e Gr=$1x10^5$.

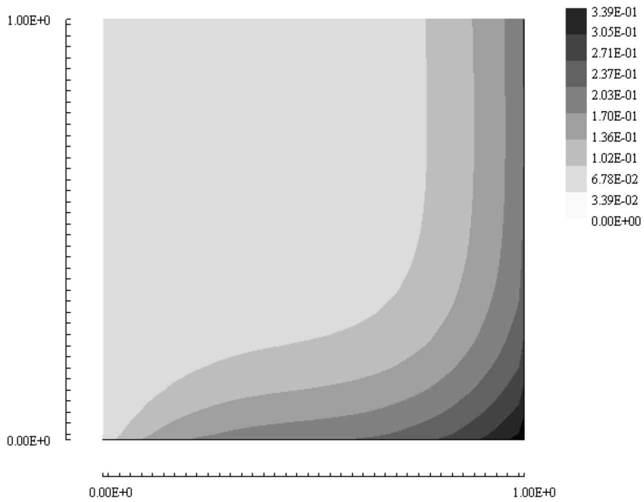

Figure 4. Temperature field for Re=10 and Gr=0.

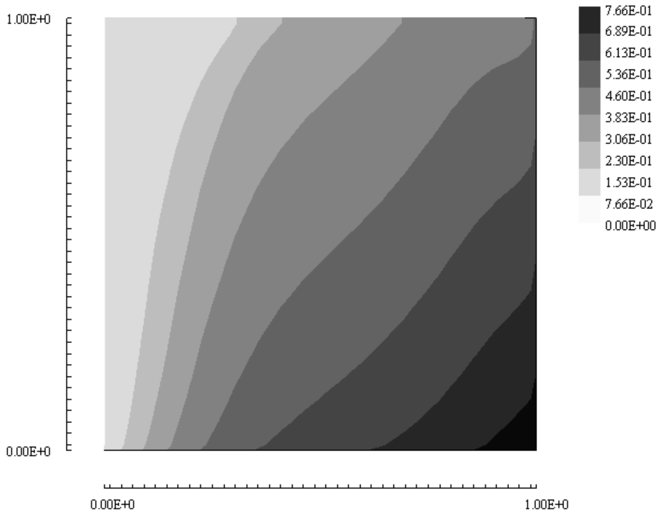

Figure 5. Temperature field for Re=10 and Gr=$1x10^5$.

5. CONCLUSIONS

This study can be applied in many industrial applications such as solar heating, energy conservation in buildings, refrigeration of electronic equipment and other systems where heat transfer occurs by forced or free convection. So, for cooling purposes, the results obtained show that the forced convection inside the semiporous open cavity studied may be greatly enhanced by natural convection effects. When Gr is small enough, just forced convection controls the heat transfer. When Gr increases, natural convection effects may become dominant and then the electronic equipment may reach a dangerous limit temperature. This study allowed identifying the biggest temperature regions when the system is submitted to combined and free convection, making possible to apply control actions, avoiding thermal damages to the devices that work with this cooling process. It concludes that the buoyancy term is fundamental when dealing with forced convection cooling.

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