A NUMERICAL STUDY OF INFLUENCE OF THE ASPECT RATIO IN COMBINED FORCED AND FREE CONVECTION HEAT TRANSFER IN A SEMIPOROUS OPEN CAVITY

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Abstract. *A geometric 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 adiabatic horizontal wall. One vertical wall is a porous and an inflow of fluid occurs normal to it. The other wall transfers an 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 the work is its application for electronic equipment where the devices used for the electronic equipment cooling are frequently based on natural and forced convection. Governing equations are expressed in cartesian coordinates in the stream function formulation and numerically handled by a finite volume method. Results of the aspect ratio are presented for both local and average Nusselt numbers at the heated wall.*

Keywords*: Forced and free convection, Semiporous open cavity, Combined heat transfer*

1. Introduction

The heat transfer in enclosures has been studied for a variety of engineering applications. Results have been presented in research surveys (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, Penot, 1982).

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 geometry on 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 horizontal adiabatic surface 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 an uniform heat flux to the cavity. In addition to the forced convection, the analysis considered the influence of natural convection effects. Local and average Nusselt numbers are obtained for the uniformly heated plate for several values of aspect ratio for the parameters governing the heat transfer: Re_p and Gr.

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., specific mass 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 specific mass 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 specific mass, β is the coefficient of thermal expansion, T is the temperature and T_p indicates the inlet temperature of the fluid at the porous wall.

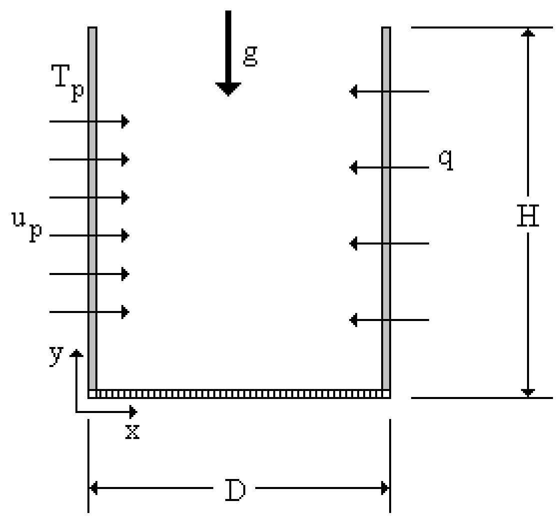

Figure 1. Coordinate system and thermal boundary conditions of the open cavity

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 equation (2) T_p indicates the inlet temperature of the fluid at the porous wall and ρ_p is the corresponding specific mass. 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 equations (2) and (3), the equation (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 ($T - T_p$). From this formulation, the specific mass 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 and U and V are the dimensionless velocities.

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

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

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

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

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

In equations (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 Grashof number, defined by

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

and the second is the Prandtl number of the fluid. In (10) Gr is the modified Grashof number. 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 (U_p) 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. Expressed in dimensionless terms, the boundary conditions become:

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

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

$$Y = 0 \ ; \ U = 0 \ ; \ V = 0 \ , \ \frac{\partial \theta}{\partial Y} = 0 \tag{11c}$$

where Re_p 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, Re_p. 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, Re_p and Gr. For a fixed particular fluid, there are still three parameters governing the heat transfer: H/D, Re_p and Gr. In the present work, a single value, equal to 0.72, was assigned to the Prandtl number.

The differential equations (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}. A carefully examination of the number of grid points will be presented with the results.

3. Results and discussion

From the velocity field solutions, a steam function defined as

$$\psi = \int_0^Y U \cdot dY \tag{12}$$

where ψ is the stream function and it was evaluated along lines X = constant, with $\psi = 0$ at X = Y = O.

From the solution of the temperature field, the local heat transfer coefficient at the heated wall and a corresponding Nusselt number were expressed as

$$Nu(Y) = h(Y) \cdot \frac{D}{k} \quad , \quad h(Y) = \frac{q}{T_w(Y) - T_p} \tag{13}$$

where T_W indicates the local temperature of the heated wall. With the definition of the dimensionless temperature (equation 5c), the Nusselt number becomes

$$Nu(Y) = \frac{1}{\theta_w(Y)} \tag{14}$$

were Nu is the local Nusselt number and θ_w is the dimensionless heated wall temperature. An average Nusselt number for the heated wall was obtained from

$$\overline{Nu} = \overline{h} \cdot \frac{D}{k} \quad , \quad \overline{h} = \frac{q}{\overline{T}_w - T_p} \tag{15}$$

where $\overline{h}$ is the convective heat transfer coefficient and $\overline{T}_w$ indicates the average heated wall temperature. Expressed in dimensionless variables, the average Nusselt number becomes.

$$\overline{Nu}(Y) = \frac{1}{\overline{\theta}_w} \tag{16}$$

where $\overline{Nu}$ is the average Nusselt number and $\overline{\theta}_w$ is the dimensionless average heated wall temperature. $\overline{\theta}_w$ is evaluated by integrating the dimensionless temperature distribution along the heated wall:

$$\overline{\theta}_w = \frac{D}{H} \int_0^{H/D} \theta_w(Y) \cdot dY \tag{17}$$

3.1 Nusselt number distributions

In Fig. 2, the Nusselt number distributions for the aspect ratios of 2, 4 and 8, obtained for a pair of values of Re_p and Gr, are compared. As the aspect ratio increases, the minimum temperature of the heated wall moves closer to the bottom of the cavity. This temperature characterizes the start up of a natural convection boundary layer along the heated wall. Thus, these profiles indicate a stronger effect of natural convection as the aspect ratio of the cavity increases. It is also noticed that the Nusselt number distributions in the region controlled by natural convection match each other. In this region the heated wall temperature distributions seem to be independent of the aspect ratio of the cavity.

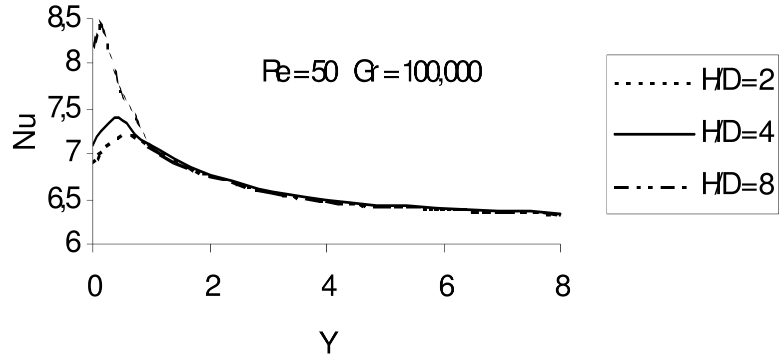

Figure 2. Influence of the aspect ratio for Re_p = 50 and Gr = 100,000.00

3.2 Average Nusselt number

Initially, the suitability of the grid size employed, considering the results obtained is presented in Fig. 3, related to the average Nusselt number defined in equation (16). In both, the absence or the presence of natural convection effects, a grid of 30x30 was adequate for the aspect ratio of 2 used in most of our results. Increasing the number of grid points from 900 to 1600, the average Nusselt number would change about 0.2%.

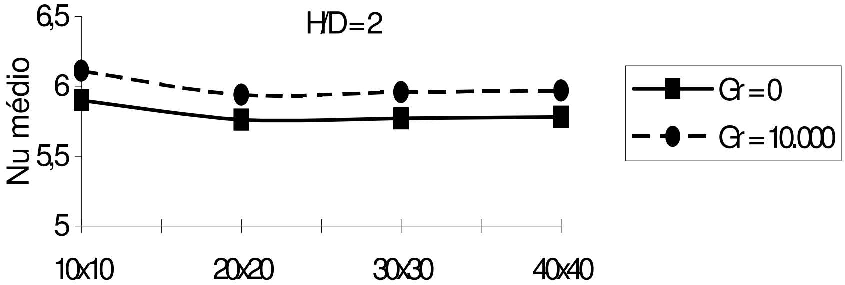

Figure 3. Suitability of the grid fineness employed for Re_p = 50

The average Nusselt numbers are shown in Fig. 4 as function of the aspect ratio H/D equal to 0.5, 1, 2, 4, 8 and 16 obtained for values of Re_p equal to 1, 5, 10, 15 and 20 and with the modified Grashoff number equal to zero. Considering the range of Reynolds number analyzed the average Nusselt number as shown, seem to be independent of the aspect ratio of the cavity H/D for H/D>4.

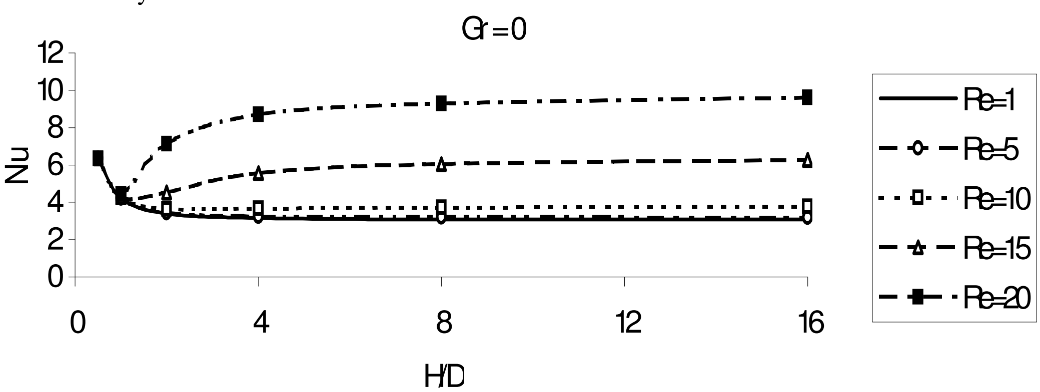

Figure 4. Influence of the aspect ratio for Gr = 0

Another view is presented in Fig.5, where Nu is shown as a function of H/D and parameterized to Gr, for Re_p = 1.0. This Nusselt seems to decrease slightly with Re_p. Nusselt attains a limit value practically independent of H/D and the

Grashof number, for the largest value of the aspect ratios greater than 2.

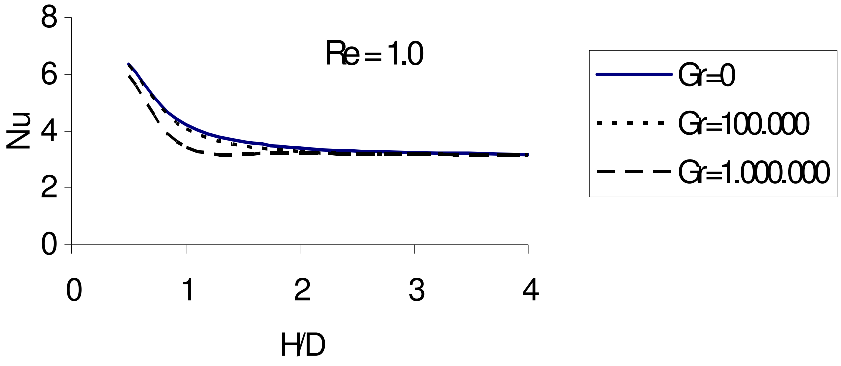

Figure 5. Influence of the aspect ratio for $Re_p = 1.0$

3. Conclusions

The results obtained show that the forced convection inside the semiporous opened cavity studied may be greatly enhanced by natural convection effects. When Gr is small enough, just forced convection controls the heat transfer. In this case, the upper portion of the heated plate becomes the most convenient region for cooling purposes. When Gr increases, natural convection effects may become dominant and then the lower portion of the heated plate constitutes the coldest region. When the aspect ratio of the open cavity increases there seems to be an increase of the role played by natural convection effects.

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