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Full Length Research Paper

Combined forced and free convection heat transfer in a semiporous open cavity

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An analysis of heat transfer inside a semi porous two-dimensional rectangular open cavity was numerically examined. The open cavity consists of two vertical walls closed to the bottom by a uniform heat flux. One vertical wall is a porous wall and fluid inflows normal to it. The other wall transfers the same uniform heat flux to the cavity. The study shows how natural convection effects may improve the forced convection inside the open cavity. The main motivation for this research is its application for electronic equipment where the cooling devices used for the electronic equipment are frequently based on natural and forced convection and the equipment may reach dangerous limits of 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 Grashof numbers at the heated wall and in the bottom.

Key words: Semiporous open cavity, forced and free convection, heat transfer.

INTRODUCTION

The heat transfer in enclosures is studied for a variety of engineering applications. Results were presented in research surveys such as in Catton and Edwards (1976), Kakaç et al. (1987), and it became a main topic in convective heat transfer textbooks (Bejan, 1994). Usually the enclosures are closed and natural convection is the only 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 used (Chan and Tien, 1985; Hess and Henze, 1984; Penot, 1982). Ramesh and Merzkirch (2001) presented a study combining laminar natural convection and surface radiation from side-vented open cavities with opening top; Gunes and Liakopoulos (2003) studied, by a spectral element method, the three-dimensional free convection in a vertical channel with spatially periodic, flush-mounted heat sources; Korichi and Oufer (2005) presented a numerical study of convective heat transfer between a fluid and three obstacles placed on the lower and upper wall of a rectangular channel. Cheng and Lin (2005) present an optimization method of thermoelectric coolers using genetic algorithms and Vasiliev (2006) presented a short review on the micro and miniature heat pipes used as electronic component coolers. Delgado-Buscalioni et al.(2001) presented a numerical and theoretical investigation on the natural convection in an inclined 2 and 3 D side-heated enclosures for low Prandtl-number-fluid. Devices used to cool electronic equipment are frequently based on forced convection (Sparrow et al, 1985). Altemani and Chaves (1988) presented 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 continues that study and also investigates two vertical parallel plates opened at the top and closed at the bottom by a uniform heat flux, as indicated in Figure 1. One of the vertical plates is porous and there is a normal forced fluid flowing through it. The opposite vertical plate supplies the same uniform heat flux to the cavity. In addition to the forced convection, the analysis considers the influence of natural convection effects. The maximum temperature is obtained for the uniformly heated plate and to the bottom for the parameters governing the heat transfer: Reynolds (Re) and Grashof (Gr) numbers.

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Analysis

The conservation equations of mass, momentum and

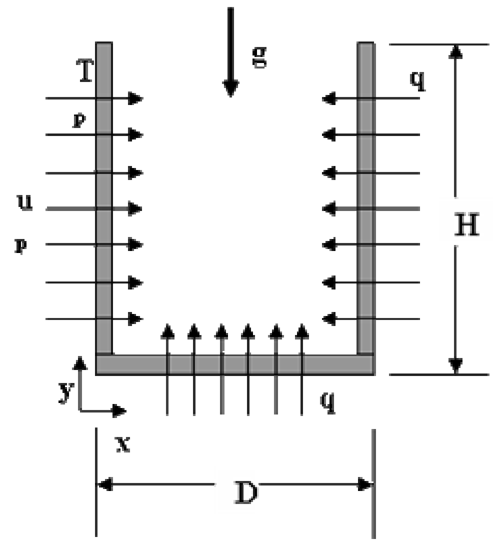

Figure 1. Coordinate system and thermal boundary conditions of the open cavity.

energy, as well as their boundary conditions, will be expressed for the system indicated in Figure 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, that is, density variations are accounted 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:

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

The specific mass is related to the temperature according to the Boussinesq approximation:

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

The pressure is now expressed in terms of a modified pressure defined as

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

With eqs. (2) and (3), the term (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 expression relates the buoyancy forces to the temperature difference ($T–T_p$). According to this formulation, the specific mass will be assumed constant and equal to ρ_p in all 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 difference and laminar flow conditions assumed.

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

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

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

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

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

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

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

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

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

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.

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 and in the bottom. Expressed in dimensionless terms, the boundary conditions become:

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

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

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

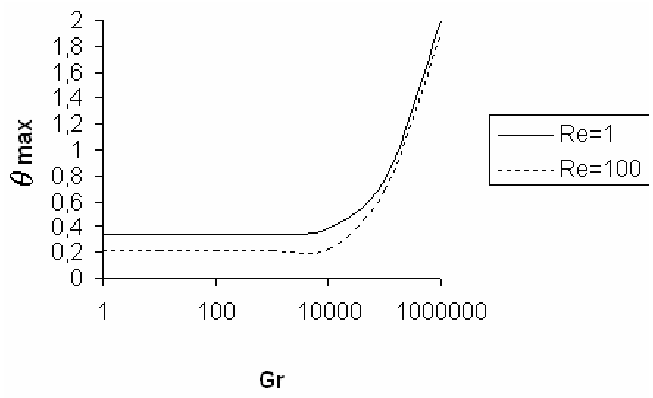

Figure 2. Maximum dimensionless temperature as a function of Grashof number for Re=1 and Re=100.

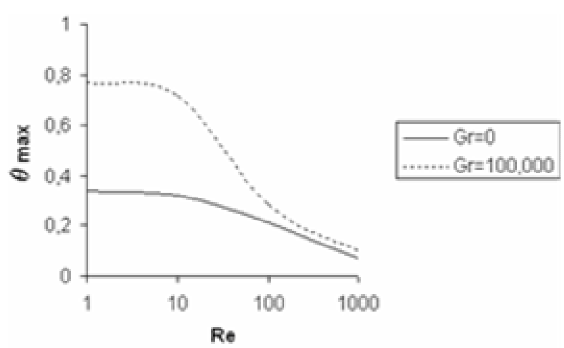

Figure 3 . Maximum dimensionless temperature as a function of Reynolds number for Gr=0 and Gr=1x10^5.

The dimensionless velocity component normal to the permeable wall ($u_p \frac{D}{\nu}$) is one parameter of this problem and it will designate the porous wall Reynolds number, Re_p. The outflow boundary of the open cavity, at Y equal to H/D 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 this paper, a single value, equal to 0.72, was assigned to the Prandtl number and H/D = 1.

METHODOLOGY

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

To solve the problem numerically, it integrates the Eqs. (1) to (3) related to u and v variables, already described in dimensionless cylindrical coordinates on a generic volume control. Integration is done following the volume control method formulation developed by Patankar (1980) where potential law schematic is taken, to calculate the flux term through the limits of each internal control volume.

Distinguished equations make a coupled system involving u, v and T variables. The numerical solution in this system is solved using simple schematic purposed by Patankar (1980). To solve this simultaneous mathematical equations that come from distinguish process, it uses line-to-line iterative method.

As initial step it is considered the following first approximation u = v = 0 (stagnated fluid) and T = 0 for all domain. In each process, there was a need for updating u, v and T values and u and v equations were solved three times for each iteration. To reach T values it was solved only once by iteration. Such process was widely useful in cases where Gr and Re are high, that causes stronger convetion streams. The acceptance standard of a solution as converged is based on the maximum mistake possible inside the whole calculations range. The obtained results convergence was accepted when relative changes in the dependent variables were below 1.0×10^{-5}.

RESULTS AND DISCUSSION

The maximum dimensionless temperature θ_{max} is shown on Figure 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 x 10^4. After this value, T_{max} increases abruptly, caused by influence of the buoyancy effects.

The maximum dimensionless temperature θ_{max} is shown in Figure 3 as function of the Reynolds number for the modified Grashof number equals to 0 and 100,000. It is noticed that the behavior of the curve is affected by Gr. For Gr equals 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} decreases 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 in Figures 4 and 5. Figure 4 shows the effects of the forced convection and Figure 5 shows a larger penetration into the temperature in the upper part of the cavity when Grashof increases imposed by natural convection effects.

Conclusions

This study can be applied in many industrial applications

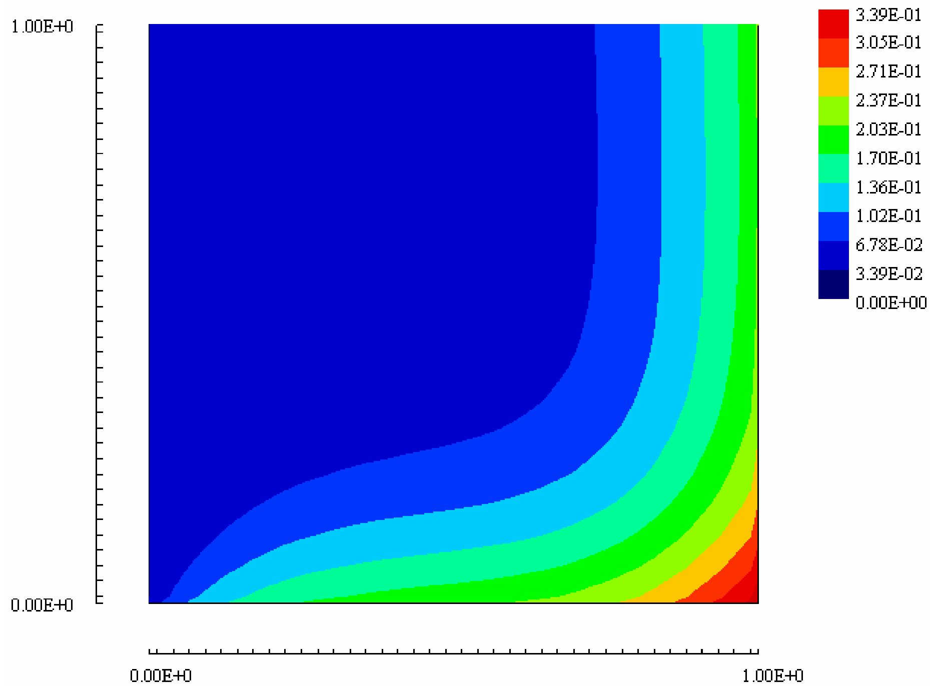

Figure 4. Dimensionless temperature field for Re = 10 and Gr = 0.

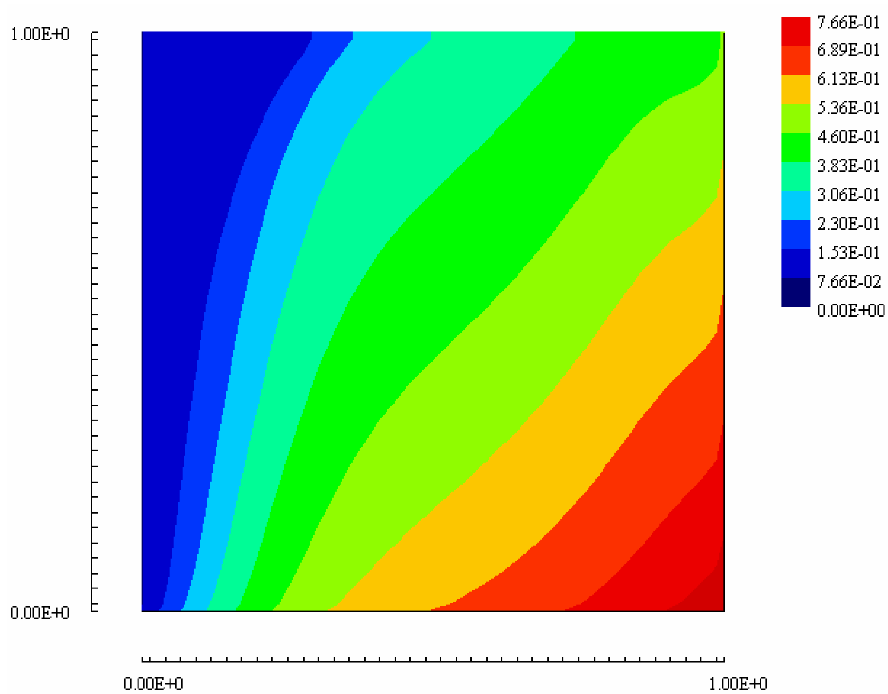

Figure 5. Dimensionless temperature field for Re=10 and Gr=1x10^5.

such as solar heating, energy conservation in buildings, refrigeration of electronic equipment and other systems where heat transfer occurs by force 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.

Nomenclature: g, gravitational acceleration $m.s^{-1}$; $\frac{\partial p}{\partial y}$, pressure gradient in y direction $Pa.m^{-1}$; **Gr**, Grashof number; **D**, width of the open cavity m; **H**, height of the open cavity m; **p**, pressure Pa; $\mathbf{p}^*$, modified pressure Pa; **P**, dimensionless pressure; **Pr**, Prandtl number; **q**, surface heat flux W; **Re**, porous wall Reynolds number; **T**, temperature K; $\mathbf{T_p}$, temperature of the fluid inlet at the porous wall K; **x**, Cartesian coordinate m; **y**, Cartesian coordinate m; **U**, dimensionless velocity in x direction; **V**, dimensionless velocity in y direction.

Greek symbols: β, coefficient of thermal expansion K^{-1}; ∇^2, laplace operator in Cartesian coordinates; **ρ**, specific mass $kg.m^{-3}$; ρ_p, specific mass of the fluid at the porous wall $kg.m^{-3}$; ν, kinematics viscosity $m^2.s^{-1}$; θ, dimensionless temperature; θ_w, dimensionless heated wall temperature.

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