Journal of Mines, Metals and Fuels, *71(12A) : 390-393; 2023. DOI: 10.18311/jmmf/2023/43055*

Print ISSN : 0022-2755

Journal of Mines, Metals and Fuels

Contents available at: www.informaticsjournals.com/index.php/jmmf

Thermal Convection in Porous Cavity with Heated Block

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Abstract

This work investigates mixed convection numerically using a heated square cylinder at the middle of a vented cavity filled with fluid-saturated porous media. External cold fluid enters into the cavity through the opening located either at the bottom wall or at the bottom and side walls, whereas the hot plume is vented through an opening at the top wall. All the walls of the cavity are insulated. The governing equations are solved by in-house code. Flow through porous medium has been modeled using Brinkman-Forchheimer-Darcy Model (BFDM). The influence of flow parameters (Reynolds number Re = 10–500, Richardson number Ri = 0.1–100) and porous-medium parameters (Darcy number Da = 10^{-7}–10^{-3}, porosity ε = 0.1–1) and inlet openings are investigated systematically. It is observed that the heat transfer increases as Ri and Re increase.

Keywords: *Heated Block, Heat Transfer, Mixed Convection, Multiple Inlet, Porous Cavity.*

1.0 Introduction

Numerous technological processes, including cooling of electronic and electrical equipment, geothermal energy systems, solar heating, storing of nuclear waste, oil and gas production, grain and food processing, have practical applications for understanding flow-dynamics and heat transport phenomena in a ventilated cavity filled with fluid-saturated porous medium under mixed convection. The principles of convective flow in porous media have been extensively reviewed[1,2].

The study of mixed convection in a ventilated enclosure in the presence of barriers, partitions, or fins affects the convection flow phenomena, according to the literature review. Study on this subject has been initiated many years ago. When there is a porous material present, flow complexity increases. According to Chamkha *et al.*,[3] mixed convection heat transfer of air within a square cavity with a heated horizontal square cylinder has been studied. Rahman *et al.*[4] examined the effects of Reynolds and Prandtl numbers on mixed convection in a vented cavity containing a heat-generating solid circular block. Mixed convection in a vented cavity with a heat-conducting circular cylinder was studied by Gupta *et al.*[5]. Very recently, Chamkha *et al.*[6] have investigated mixed convection in a partially layered porous cavity with an inner rotating cylinder and found that the local and averaged heat transfer enhances/ decreases as the cylinder moves from its center location.

The current work's objective is to evaluate the thermal performance under mixed convection in porous square cavity with a heated square cylinder for different inlet port configurations. The present study focuses on the impact of multiple inlet openings on heat transfer. The effects of various pertinent parameters on flow field and heat transfer are presented in terms of the average Nusselt number, and contours of streamlines, and isotherms.

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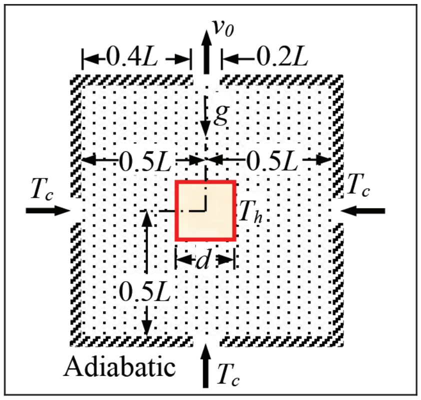


Figure 1. The problem's physical description and boundary conditions.

2.0 Problem Description

Figure 1 presents a schematic representation of the problem's physical description in detail. A 2-D square cavity (of side L) that is air-saturated and filled with porous materials (Pr = 0.71). In the middle, a square solid cylinder (with a side of 0.2L) is heated evenly to temperature T_h. External cold fluid enters through the openings (of size 0.2L) located either at the middle of bottom wall or at the middle of bottom and side walls and leaves through the exit port (of size 0.2L) located at the middle of the top wall. All the walls are well insulated. It is assumed that the flow is laminar, Newtonian, stable, and incompressible. Except for the density fluctuation in the buoyancy component, which is determined by the Boussinesq approximation, all of the fluid's thermo-physical characteristics are taken to be constant. Assuming isotropic, homogeneous fluid-saturated porous medium, the Darcy-Brinkman-Forchheimer Model (BFDM) and local thermal equilibrium between porous material and fluid have been used. The non-dimensional governing equations for mass, momentum, and energy in a two-dimensional Cartesian coordinate system are formulated as follows using the aforementioned premises:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{1}$$

$$\frac{1}{\varepsilon^2}\left(U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}\right) = -\frac{\partial P}{\partial X}+\frac{1}{\text{Re}}\frac{1}{\varepsilon}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) -\frac{1}{Da\,\text{Re}}U-\frac{F_C\sqrt{U^2+V^2}}{\sqrt{Da}\;\varepsilon^{3/2}}U \tag{2a}$$

$$\frac{1}{\varepsilon^2}\left(U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}\right) = -\frac{\partial P}{\partial Y}+\frac{1}{\text{Re}}\frac{1}{\varepsilon}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right) -\frac{1}{Da\,\text{Re}}V-\frac{F_C\sqrt{U^2+V^2}}{\sqrt{Da}\;\varepsilon^{3/2}}V + Ri\,\theta \tag{2b}$$

$$\left(U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}\right) = \frac{1}{\text{Re Pr}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{3}$$

These equations subject to following dimensionless quantities:

$$\text{Pr}=\frac{\nu}{\acute{a}},\ \text{Da}=\frac{K}{L^2}$$

$$\text{Re}=\frac{v_0 L}{\nu},\ Ri=\frac{Ra_m}{Da\,\text{Pr}\,\text{Re}^2},$$

$$Ra_m=\frac{g\beta(T_h-T_c)LK}{\nu\alpha}$$

$$\text{Pr}=\frac{\nu}{\acute{a}},\ \text{Da}=\frac{K}{L^2}$$

$$\text{Re}=\frac{v_0 L}{\nu},\ Ri=\frac{Ra_m}{Da\,\text{Pr}\,\text{Re}^2},$$

$$Ra_m=\frac{g\beta(T_h-T_c)LK}{\nu\alpha}$$

Pr, Da, Re, Ra_m and Ri are, successively Prandtl number, Darcy number, Richardson number, Reynolds number, Darcy–Rayleigh number and Richardson number. Forchheimer coefficient is given by $F_C = 1.75/\sqrt{150}$.

For obtaining a solution, the following boundary conditions are used:

$U=V=0,\ \partial\theta/\partial X=\partial\theta/\partial Y=0$ on the adiabatic walls

$U=V=0,\ \theta=1$ on the heated walls (block)

$\theta=\ 0,\ P=0$ and zero velocity gradient at inlet port (s)

$U=0,\ \ V=1,\ \partial\theta/\partial Y=0$ at outlet port

The method used to compute the average Nusselt number at the heated surface is $Nu_{avg}=-\int_{\varepsilon}\frac{\partial\theta}{\partial n}d\varepsilon/\int_{\varepsilon}d\varepsilon$, where ε is the whole length of the block's heated wall and n denotes the heated wall's surface's outward normal direction.

For the governing equations (1)–(4) with boundary conditions, numerical simulations are carried out using custom-made code and the finite-volume computational method using the SIMPLE algorithm[7], TDMA solver, and ADI sweep. The maximum values of residuals and the continuity mass-defect criteria are chosen at 10^{-8} and 10^{-10}, respectively, to achieve the convergence of solutions repeatedly. In the context of our past study, the identical code was applied[8,9]. In the current study, air (Pr = 0.71), a porous medium (Darcy number Da = 10^{-4}, porosity ε = 0.6, Richardson number and Reynolds number in the ranges of 1-100 and 50-200, respectively), is utilised as the working fluid.

3.0 Results and Discussion

The evolution of fluid flow and temperature distribution in the ventilated cavity is presented in Figure 2 for Ri =1, 10, 100 considering single inlet from the middle of the bottom wall (Figure 2a) and three inlets from the middle of the side walls and the bottom wall (Figure 2b). It is evident from the flow-field patterns in Figure 2a that neither the fluid nor the heat flow deviates much from symmetry about the mid-vertical plane of the hollow. Streamline distributes on the both sides of the hot block. Under three inlet configuration, significant changes in the flow fields as well as the thermal field is observed. It happens so due to ingress of external flow from three sides, which causes varying flow and temperature distribution on the periphery of the heated block. In Figure 3, the overall trend of heat transfer with the variation Re is presented in terms of Nu_{avg} for different Ri values for single inlet, two inlets and three inlets. It indicates a significant decrease in Nu_{avg} for in case of two inlet port configuration than single inlet port configuration, whereas for three inlet case heat transfer is slightly higher than two inlet port configurations. Higher rate of heat transfer is observed at higher Re and Ri values.

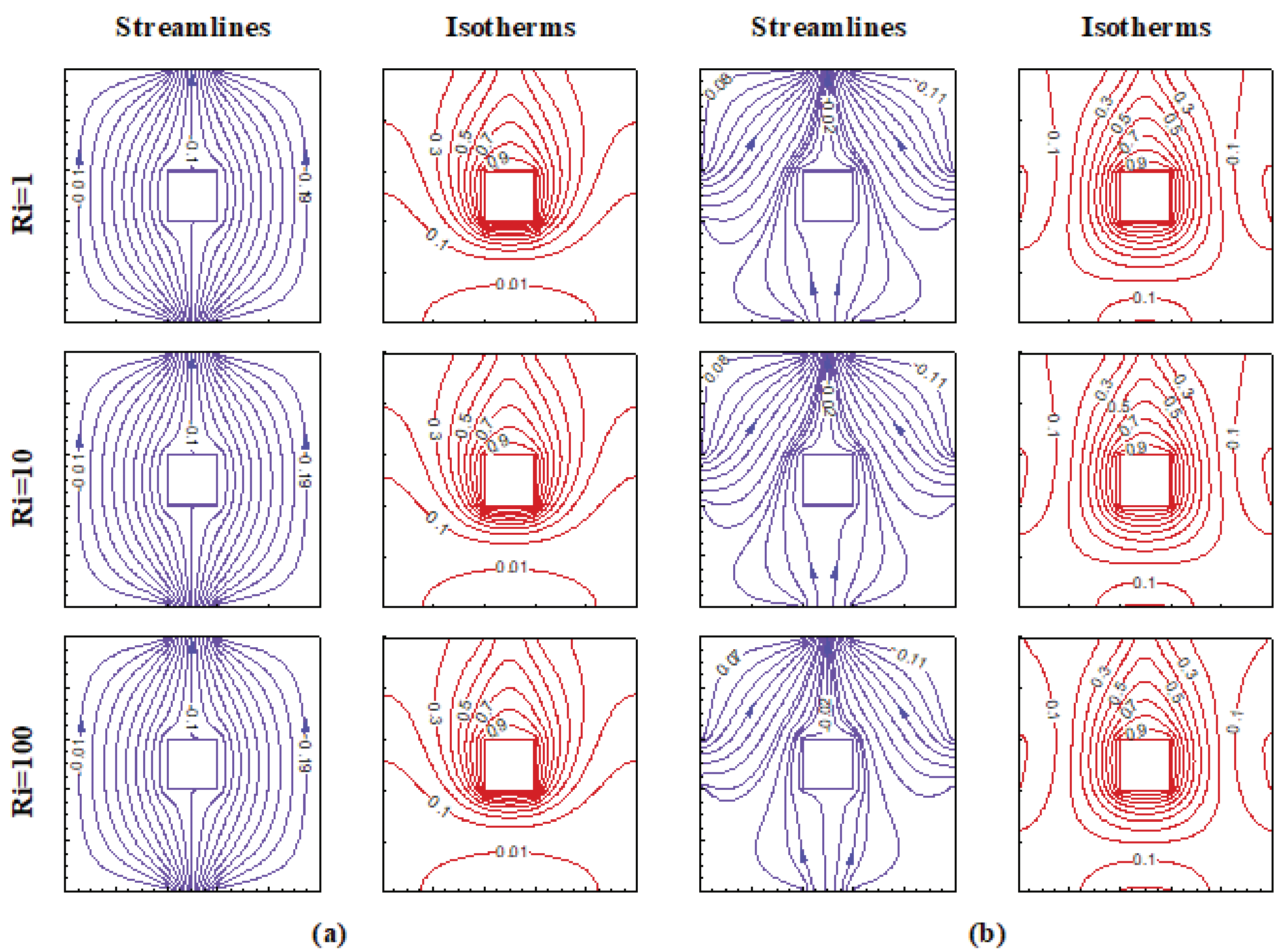
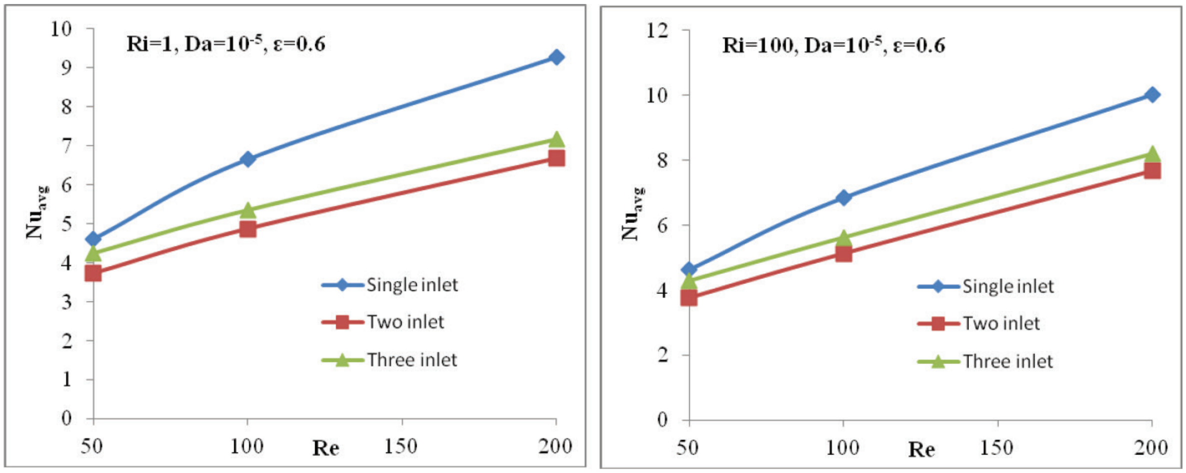


Figure 2. Visualization of flow-fields using streamlines and isotherms at Re = 100, Da = 10^{-5}, ε = 0.6 for different Ri values for single inlet (a), three inlets (b).



(a) (b)

Figure 3. Variation of average Nusselt number for different Re value: (a) Ri = 1 and (b) Ri = 100 at Da = 10^{-5}, ε = 0.6.

4.0 Conclusion

The impact of single inlet and multiple inlets on heat transfer is investigated considering isothermally hot block at the center of a porous cavity. The study highlights how Ri and Re values significantly affect the heat transfer mechanism. As Re increases Nu_{avg} increases. Similarly, heat transfer increases with Ri. The heat transfer is found to increase with single inlet, and decreases with multiple inlets. It is due to reduced local convection near the heated block. Through multiple inlet openings, the total flow is distributed over the cavity, thus fluid velocity passing over the heating block decreases significantly. However, with more numbers of openings fluid can be well distributed over the entire cavity area (or volume) and could results good mixing.

5.0 References

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