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Effect of Adiabatic Circular Cylinder on the Natural Convection Heat Transfer Characterizes in a Porous Enclosure

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The present work studies numerically natural convective flow between a circular adiabatic cylinder located in a square porous enclosure using finite element method. Darcy–Forchheimer model is used in solving the dimensionless governing equations including; continuity, energy and momentum of the fluid along with Bousseinesq approximation. The enclosure is heated from bottom and cooled at isothermal temperature for the vertical walls. The top wall and obstacle are assumed adiabatic. The considered parameters are $10^3 \le Ra \le 10^6$, $10^{-5} \le Da \le 10^{-3}$, obstacle vertical location $0.25 \le h \le 0.75$ and cylinder radius $0.1 \le D \le 0.9$ with Pr = 0.7. It is obtained that as the Rayleigh and Darcy numbers increase, both streamlines; Nusselt numbers will increase leading to increase the rate of heat transfer. The results show that the heat transfer rate is significantly dependent on the diameter of the circular cylinder and the location of the cylinder. It is found the maximum heat transfer rate obtained at D=0.1 and when the cylinder moves vertically upward at h=0.3

1. Introduction

Due to its significant concernment in extensive area of applications in engineering, the peculiarity of natural convection in porous enclosure had been received a considerable attention. A portion of these applications are nuclear reactor, heat exchanger, solar collector. A lot of investigations dealing with the natural convection in permeable cavities (Bin Kim 2001; Basak et al., 2006; Sathiyamoorthy et al., 2007; Abdulkadhim et al., 2018; Raju 2018; Al-Farhany et al., 2018). Then again presence of a body inside permeable enclosure in area had been analyzed by numerous specialists and their decisions demonstrated indicated that the enclosure thermal boundary conditions, the body presence within its cavity, the cavity shape, and the position of the inner body within the cavity effects strongly on the natural convective flow. Regarding the body inside an enclosure, numerous researchers investigated that. They examined different inner cylindrical body shapes, like as a circular, sphere, square, or triangular and 3D square. (Lee et al., 2005) demonstrated the inner square cylinder impact on the characteristics of heat transfer inside an enclosure under different values of Rayleigh numbers. They introduced a helpful comparison between results of the natural convection inside an enclosure contains inner square cylinder with that for the enclosure does not contain an inner body and their main conclusion were its existence influences on fluid flow the strength and the characteristics of the heat transfer in the cavity. Many researchers considered different shapes and sizes of inner bodies to decide how the inward body influences on the natural convection in an enclosure (Moukalled et al., 1996; Asan 2000; Shu et al., 2002; Angeli et al., 2008; Xu et al. 2010; Ali et al., 2018). The enclosure shape impacts on the characteristics of the natural convection and the researchers studied this parameter either by changing the aspect ratio of enclosure or changing the cross-section. (Warrington et al., 1985) have examined some experiments to illustrates the effect of concentrically mounted bodies at low Rayleigh numbers for the shapes of the spherical and cubical enclosures. They acquired that the shape of the enclosure impacts profoundly on the temperature appropriation and attributes of heat transfer. (De et al., 2006) analyzed the impact of cavity aspect ratio and obtained that the thermal layer and the fluid flow is highly dependent on the enclosure aspect ratio. Some of researchers examined like (Humaira et al., 2002; Shu et al., 2001; Ding et al., 2005; Kim et al.,

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2008; Hussain et al., 2010; Lee et al., 2010; Park et al., 2013; Karimi et al., 2014; Majdi et al., 2019) investigated the effect of the position for the internal body on the heat transfer by the natural convection within the enclosure. Two main position of inner body in the enclosure had been taken by the researchers either changing the position vertically and horizontally. They have found that the characteristics of the flow of the fluid and heat transfer as a function of the inner body position as well as the Rayleigh number. Recently, many investigations focused on this problem (Feldman 2018; Cho et al., 2018; Jha and Yusuf, 2018; Selimefendigil et al., 2018).

Most of studies dealt with porous enclosure without obstacle or dealt with enclosure containing obstacle located in non-porous enclosure. Also, most of the studies deal with the inner body as a hot body not as an insulated body. Thus, the main scope of the present work is to close this leak for natural convective flow in a porous enclosure but within a fixed circular cylinder. The considered dimensionless parameters are Ra, Da numbers, different vertical wall locations, and obstacle diameter. The dimensionless governing equation will be solved numerically using finite element method along with Darcy-Forchheimer model

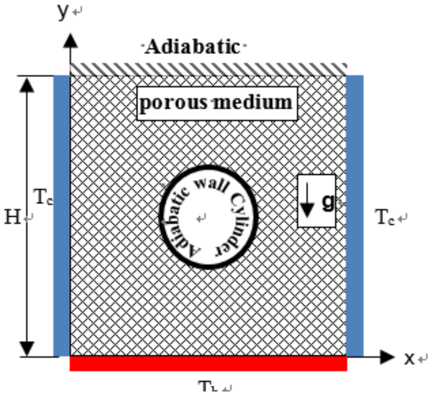


Figure. 1: Geometry of the circular cylinder inside porous enclosure

2. Mathematical formulation

The present article describes, characteristics natural convection fluid flow in a laminar nature under two-dimensional, between an insulated circular cylinder located inside square enclosure. It has been examined numerically utilizing finite element approach heated from the bottom and the two vertical side's walls are kept at an isothermal cold temperature. The top wall and the cylinder are considered adiabatic as shown in Fig. 1. At the enclosure centre, the circular cylinder is located with radius R = 0.2 and move vertically downward and upward. For solving the governing dimensionless equations of continuity, energy and momentum of the fluid, the Darcy-Forchheimer model without the Forchheimer inertia along with Boussenisq approximation is utilized which they are becomes (Basak et al., 2006):

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

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\Pr\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)-\frac{\Pr}{Da}U \tag{2}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)-\frac{\Pr}{Da}V+\mathrm{Ra}\Pr\theta \tag{3}$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2} \tag{4}$$

Where

$$\mathrm{Y}=\frac{\mathrm{y}}{\mathrm{H}},\ \mathrm{X}=\frac{\mathrm{x}}{\mathrm{H}},\ \mathrm{U}=\frac{\mathrm{uH}}{\alpha},\ \mathrm{V}=\frac{\mathrm{vH}}{\alpha},\ \theta=\frac{\mathrm{T-Tc}}{\mathrm{Th-Tc}},\ \Pr=\frac{\nu}{\alpha},\ \mathrm{P}=\frac{pH^2}{\rho\alpha^2},\ \mathrm{Da}=\frac{K}{H^2},\ \mathrm{Ra}=\frac{g\beta_f(\mathrm{Th-Tc})L^3}{\nu\alpha} \tag{5}$$

3. Code validation

In order to investigate if the results obtained numerically are acceptable, a validation is presented with the (Hussain et al., 2010) results in Fig. 2 for streamlines and isotherms for various Rayleigh number.

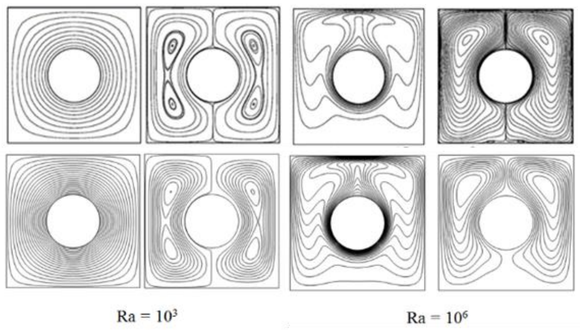
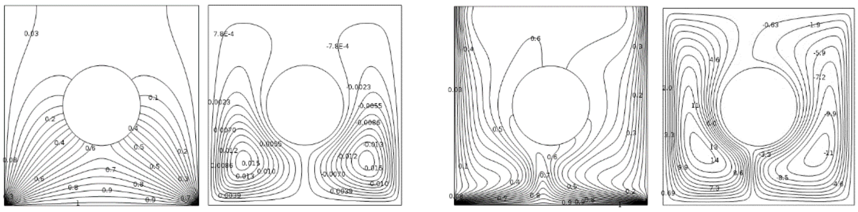


Figure 2: Validation of the present (bottom) work with Hussain and Hussein (Hussain and Hussein 2010) (top) of isotherms (left) and streamlines (right) for various Rayleigh numbers

4. Results and dissuasion

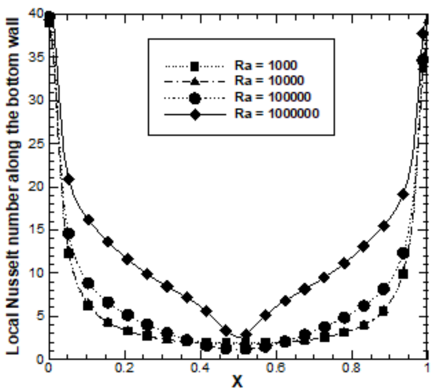
4.1 Modified Rayleigh number

The present paragraph illustrates the Rayleigh number impact on the fluid flow strength and isotherms are presented in Fig.3 at [Da=10^{-3}, Pr=0.7, R= 0.2]. It is obtained that the higher the Rayleigh number, the higher maximum stream function value. For example, stream function at its maximum value will increase from ψ_{max}=0.015 to ψ_{max}=14 for the Rayleigh number increases from Ra = 10^3 to Ra = 10^6, respectively. Figs 4 and 5 present the impact of Rayleigh and Darcy numbers on heat transfer rate in terms of local Nusselt number along the base wall. It can be seen that the local Nusselt number increments as Rayleigh and Darcy number increment for both base and vertical side dividers.



Ra=10^3 Ra=10^6

Figuer 3: Isotherms (left) and streamlines (right) for various Rayleigh numbers at Pr=0.7, Da=10^{-3}, R=2



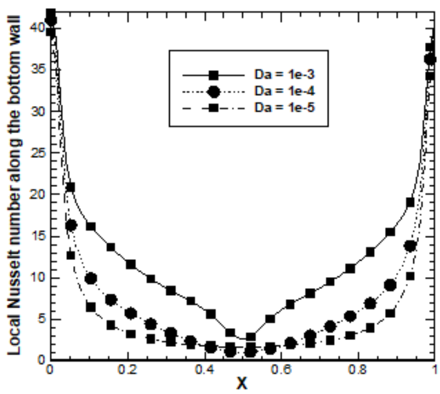
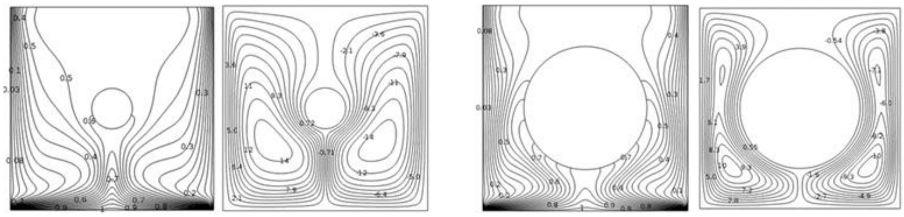


Figure 4: Local Nusselt number for various Rayleigh numbers

Figure 5: Local Nusselt number for various Darcy numbers

4.2 Effect of cylinder diameter

The impact of cylinder diameter on the heat transfer and fluid flow characteristics are presented in Fig.6 at [Ra=10^6, Da=10^{-3}, Pr=0.7]. It is may be noticed from Fig.6 that as the cylinder diameter increases, the maximum stream function decreases. For example, when the cylinder diameter increases from D=0.2 to D=0.5, the maximum stream function decreases from $\psi_{max} = 14$ in $\psi_{max} = 10$ because when the cylinder diameter increases, the conduction mode will be dominated leading to reduce the flow strength. Fig.8 (a) illustrates the influence of the diameter of the circular adiabatic cylinder on the rate of heat transfer, which is presented in terms of average Nusselt number for different values of Rayleigh numbers. It is obtained that the relation between the diameter of circular cylinder and the Nusselt number will be inverse relation the i.e., the increasing in the cylinder diameter decreases the average Nusselt number leading to change the heat transfer mechanism from the natural convection mode into conduction mode leading to reduce the heat transfer rate. It can be noticed that the maximum heat transfer rate is obtained when the cylinder diameter is D=0.1 at Ra=10^6 and Da=10^{-3}, hence we will consider this diameter to investigate the effect of vertical wall location on the heat transfer rate.



D=0.2 D=0.5

Figure 6: Isotherms and Streamlines for various cylinder diameter at Ra=10^6, Da=10^{-3}, Pr=0.7

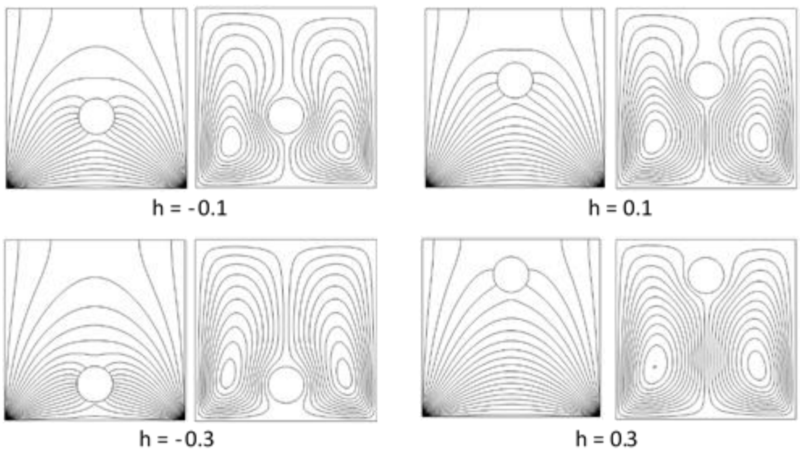


Figure 7: Isotherms and Streamlines for different vertical location of adiabatic cylinder at Ra=10^4, Da=10^{-3}

4.3 Effect of vertical location

Figs 7 show the isotherms and streamlines for various vertical locations of adiabatic cylinder. It can be seen that the vertical cylinder location influences on the fluid flow and isotherms contours and the two inner cells formed below the insulated cylinder when it moves upwards vertically. In the other hands, the two inner cells formed above the cylinder as it moves downwards vertically for various Rayleigh numbers. Fig. 8(b) illustrates the effect of vertical location movements on the heat transfer enhancement, which is presented in terms of average Nusselt number under different values of Rayleigh numbers. The obtained results indicate that the maximum heat transfer rate is for the insulated cylinder that moves vertically upward at (h=0.3) while the minimum heat transfer rate at h=-0.3 (i.e., when the cylinder moves vertically downward).

5. Conclusions

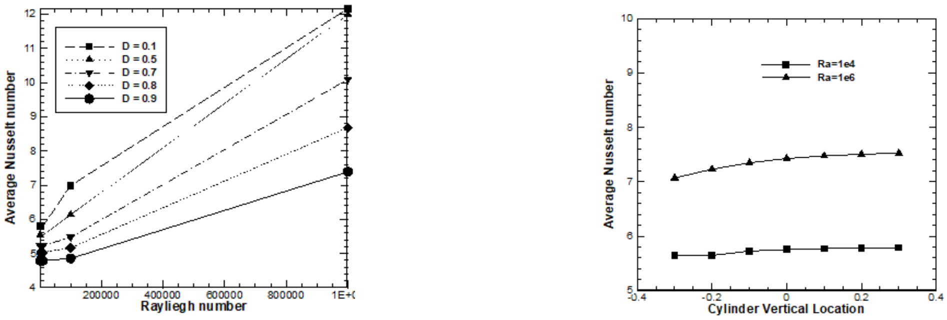
The main results that obtained from the present work can be summarized as follow:
The heat transfer mechanism is converted from conduction mode into convection as the Rayleigh and Darcy number increases.

It was obtained that the cylinder size has a sturdy impact on the fluid flow strength and heat transfer characteristics because it shortens the gap for fluid flow between the circular cylinder and the enclosure walls, giving rise in the conduction heat mode and reducing the average Nusselt number leading to decrease the convection heat transfer rate.

The diameter of the circular cylinder influence dramatically on the fluid flow strength and the heat transfer rate. It is found that as the diameter increases, both stream function and Nusselt number decreases, leading to reduce the heat transfer rate.

It is found that the maximum heat transfer rate is obtained when the cylinder diameter is D=0.1.

It is recommended for better heat transfer rate; the cylinder should move vertically upward at h=0.3



(a) (b)

Figure 8: Variation of Average Nusselt number versus (a) Rayleigh numbers, (b) for various vertical locations

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