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INFLUENCE OF CORNER FLOW ON MIXED CONVECTION IN A CAVITY

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Abstract: The external flow based mixed convection has wide uses in many applications. The present numerical investigation is carried out on a ventilated porous cavity providing fluid flow diagonally. Two heating elements and cooling elements are placed in the middle of the walls of the cavity. The simulation is carried out using an in-house code for a range of parameters such as different flow regime (using Richardson number Ri= 0.120), Flow velocity (using Reynolds number Re= 10200), permeability (using Darcy number Da= $10^{-7}10^{-3}$) and porosity (ε = 0.30.8) of the porous media. The results are presented using isotherms, streamlines and average Nusselt number. The obtained results show a strong parametric dependence of fluid flow and associated heat transfer phenomena. An exception to the usual trends of heat transfer characteristics of clear and porous domain, the heat transfer of porous domain increases due to corner flow.

Keywords: Mixed convection; Porous cavity; Corner flow; Heat transfer.

NOMENCLATURE

Da	Darcy Number
g	gravitational acceleration (m/s)
Gr	Grashof number
K	permeability (m^2)
L	length of the cavity (m)
Nu	average Nusselt number
p	dimensional pressure (Pa)
P	dimensionless pressure
Pr	fluid Prandtl number
Re	Reynolds number
Ri	Richardson number
T	dimensional temperature (K)
u,v	dimensional velocities (m/s)
U,V	dimensionless velocities
V_i	inflow velocity (m/s)
x,y	dimensional Cartesian coordinates (m)
X,Y	dimensionless coordinates
	Greek symbols
α	thermal diffusivity (m^2/s)
β	volumetric thermal expansion coefficient of fluid (K^{-1})
ε	Porosity
θ	dimensionless temperature
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
ψ	dimensionless stream function
	Subscripts
a	ambient
c	cold wall
h	hot wall

1. INTRODUCTION

Earlier people used to spend more time outside but with the advancement of technology men have confined themselves within the four walls. With this rapid progress of mankind almost everything has come within his reach. However, with this bloom the concern for the betterment of the indoor environment has been on the cards from the past century. Of the many variables affecting the indoor lifestyle, ventilation or the adequate supply of fresh air has been the most challenging issue of the past decade and is the most widely exploring field as of now. However, with ventilation comes great responsibility in tackling the energy consumption management and qualifying comfort level in the day-to-day lives. Maintaining proper condition of air might be the sole objective of ventilation in air-conditioning systems used in domestic as well as in medical, coolers, chillers, heat exchangers and various other equipments that moderate the environment for indoor machines. The principle behind ventilation is basically mixed convection comprising of forced convection from the inlet air and natural convection inside the square enclosure. In the present work, authors have studied the variation of parameters on a working model. In the following texts authors mention some research works pertaining to this arena.

In the above context, combination of free or natural and forced convection, which is also termed as mixed convection, has wide spread engineering applications in domestic, medical science, and industry for cooling, drying, coating, manufacturing, etc. Several researchers have focused on studying the mixed convective heat transfer in cooling application and a considerable numbers of research articles have published and still continuing as on date. Numerically Raji and Hasnaoui [1] examined the mixed convective heat transfer in a cavity with opening and subjected to a constant heat flux along with radiation effect using air as a working medium. In this study, they observed that maximum temperature within the cavity reduces significantly due to the better mixing by the radiation effect. Laminar mixed convective heat transfer in a differentially heated rectangular cavity having inlet and outlet openings has been numerically studied by Singh and Sharif [2]. Their study revealed that, cooling efficiency is influenced markedly by changing the position of the slots. On the other hand, Omri and Nasrallah [3] investigated the mixed convective heat transfer in a rectangular cavity with left-hot right cold walls and having inflow/outflow openings in the vertical walls. They reported that better cooling efficiency can be achieved with the inlet opening located at the lower part of the heated wall. Rahman et al. [4] numerically investigated the mixed convective heat transfer in a ventilated cavity under different influencing factors like Pr, Re, Ri. They found that enhanced heat transfer (in terms of Nusselt number) is achieved with a fluid of higher Pr. A numerical study of free convection in an isothermal open cubical cavity has been carried out by Hinojosa and Gortari [5]. Rodríguez et al. [6] numerically studied the heat transfer from a ventilated cavity under different flow regime varying Ra and Re. In their study, they found that natural convection mode dominate the fluid flow and associated heat transfer. Thereafter, utilizing similar geometry as in Ref. [6], Papanicolaou and Jaluria [7] investigated the mixed convective heat transfer adopting the conducting walls along with three sets of localized heating elements having variable lengths. It is found that, thermo-fluid flow is influenced significantly under different parametric variations and outcome of the study can be applied in the electronic equipment cooling system. Natural convection heat transfer in an enclosure subjected to heating at the one side and cooled at the ceiling has been investigated by Aydin et al. [8]. Churchill [9] has studied the natural convective heat transfer in a rectangular cavity varying the heater size, its location, aspect ratio, etc. Similarly, Aswatha [10] studied the natural convective heat transfer characteristics in a cavity under different heating conditions such as uniform, non-uniform (sinusoidal profile) and linearly varying temperature at the bottom wall and isothermally cooled at the sidewalls for a wide range of Rayleigh numbers, cavity aspect ratios.

It all signifies the importance of mixed convection heat transfer in a ventilated cavity. Accordingly, a ventilated cavity with corner flow arrangement is opted in this work to explore deeper insight of heat transfer process in both clear and porous domains packed in a square cavity with heating elements at the middle position of the left and bottom walls of the enclosure. Similarly opposite walls of the cavity is cooled. To the authors best knowledge such study has not been reported in the open literature rill date. This motivates the authors to conduct the present study.

2. PROBLEM DESCRIPTION AND SOLUTION APPROACH

The schematic description of the partially differentially heated ventilated cavity with the corner flow arrangement is depicted in Fig.1. The length and height of the cavity are *L* and *H* respectively (where *L* = *H*) and the size of corner opening is 0.1*L*×0.1*L*, which are positioned diagonally. The flow enters through the left-bottom corner opening and exits through the right-top corner. The heating elements of length 0.5*L* are placed on the middle position of the left sidewall and bottom wall respectively. Two cooling elements of length of 0.5*L* are placed exactly opposite to the heating elements in the remaining two walls of the cavity.

In this work, the steady laminar, incompressible, Newtonian flow is analyzed in a two dimensional Cartesian co-ordinate systems adopting the Boussinesq approximation for the density variation. Brinkman-Forchheimer -Darcy model (BFDM) is used for the porous matrix formulation and local thermal equilibrium (LTE) condition is taken between the fluid part and porous body. Thus, the governing equations are framed considering thermo-physical properties (of air and porous media) constant. These equations in non-dimensional forms are given by

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

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

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

$$\left(U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}\right)=\frac{1}{\mathrm{Re\,Pr}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{4}$$

Here, all the symbols are given in dimensionless forms. They are dimensionless coordinates ($X = x/L$, $Y = y/L$), velocities ($U = u/V_i$, $V = v/V_i$), temperature ($\theta = (T-T_c)/(T_h-T_c)$), pressure ($P = ((p+\rho g y) - p_a)/\rho V_i^2$) and porosity ($\varepsilon$). The Forchheimer coefficient F_C equals to $1.75/\sqrt{150}$. The evolved dimensionless numbers in the governing equations Eqn. (1)–(4) are Re, Pr, Ri, and Da (Reynolds, Prandtl, Richardson and Darcy numbers, respectively), which are defined by:

$$\mathrm{Pr} = \frac{\nu}{\alpha},\ \mathrm{Da} = \frac{K}{L^2},\ \mathrm{Re} = \frac{V_i L}{\nu},\ \mathrm{Ri} = \frac{g\beta(T_h - T_c)L^3/\nu^2}{(V_i L/\nu)} \qquad (5)$$

The boundary conditions for the present configuration are applied considering fluid inflow along the diagonal of the cavity. However, for the ease of simulation, the imposed conditions are taken as $U = V = 1$, $\theta = 0$ at the inlet and zero gradients of velocity and temperature at the outlet. Whereas, at the heated walls $U = V = 0$, $\theta = 1$ and at the cold walls $U = V = 0$, $\theta = 0$ are considered.

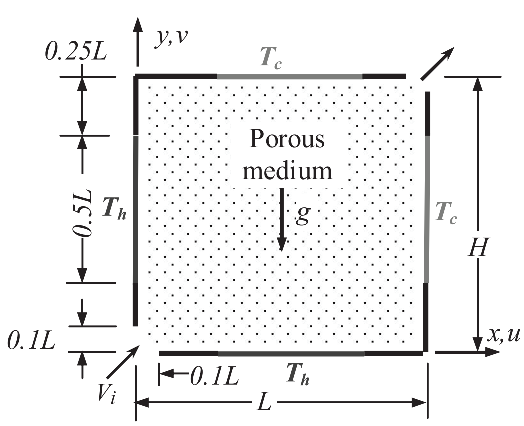

Fig. 1. Physical system of the heating and cooling elements of the studied problem.

The dimensionless flow governing Eqn. (1)–(4) applying appropriate boundary conditions are solved numerically using a well-validated in-house code [11–16] based on the finite-volume approach. Thus for brevity, no other validation study is not presented here. However, for selecting the appropriate grid size, a grid independence study is carried out for the studied problem considering five different grid sizes (150×150, 180×180, 210×210, 240×240). Non-uniform grids with finer grids closer to the walls are distributed and finest grids 0.001 are taken near the walls. The corresponding results of estimated average Nusselt number (at the heated wall) at Re = 200 for the range of Ri are indicated in Table 1. The bracketed quantity indicates relative error in % with respect to the immediate coarser grid, which show very small error < 1%. Finally, the grid size of 210×210 is taken into consideration for the present numerical simulation.

Table 1: Grid independence study at Re = 200, Da = 10^{-4} and $\varepsilon = 0.6$ with varying Ri

Ri	**Nu** (% error)			
	150×150	**180×180**	**210×210**	**240×240**
0.1	6.058	6.043 (0.253)	5.999 (0.73)	6.043 (0.05)
1	6.511	6.487 (0.372)	6.439 (0.74)	6.487 (0.17)
10	10.855	10.866 (0.104)	10.753 (1.04)	10.817 (0.60)
20	12.901	12.972 (0.506)	12.951 (0.162)	12.940 (0.085)

3. RESULTS AND DISCUSSION

The results of the present analysis are arranged systematically to address the parametric effects of Richardson number, Reynolds number, Darcy number and porosity. These results are discussed in different subsections.

3.1. Effect of Different Richardson Number (Ri)

The progression of fluid flow structure, temperature allotment and associated heat transfer characteristics with varying Richardson number (Ri) are presented below. For 0 < Ri < 1 the convection is of forced type and for 1 < Ri <∞, the convection is of free type. Results has been obtained for Re = 200, and Ri = 0.1, 1, and 10.

The results pertained to clear domain is indicated in Fig. 2. It is seen that the isotherms for Ri = 0.1 and 1 are almost similar, besides this it can also be seen that the streamlines for the two cases are also almost similar in nature, i.e. passing through the center diagonally from the entry point of the fluid at the left-bottom corner to the right bottom corner. But distinctive changes can be observed for the case of Ri = 10 in both isotherms and the Streamlines. It is observed that for Ri = 0.1 and Ri = 1 the temperature (non-dimensional) increases substantially from the entry point to the center of the cavity but from the center to the exit point the increase in temperature is very less. Also for Ri = 10 it is observed that there is a distinctive increase in temperature from the center to the exit point and also the distribution of temperature at the center is more uniform for Ri = 10. From the streamlines one can observe that for Ri = 10 the stream lines initially are straight lines at the entry point but as it moves towards the center the stream lines spread out towards the vertical and horizontal walls, forming vortices at the center, finally reuniting as straight lines at the exit point.

Finally, it can be observed that the magnitude of the heat transfer rate is increases with the increasing Ri value, which can be inferred from the increasing *Nu* values, as the fluid in the first two cases enters the cavity and directly passes through it without spreading much, and hence the heat transfer is less in first two cases compared to the third case where there is more spreading of the fluid leading to more heat transfer.

For the porous domain, the flow structures are shown in Fig. 3. It is observed that for both Ri = 0.1 and 1, there is a considerable increase in Nu values from that of the clear domain. This is because in the porous domain, the fluid facing more resistance to flow spreads out more compared to clear domain as can be observed from the streamlines of the corresponding isotherms. Due to more spreading of the fluid there is a better heat transfer, as indicated by Nu values. In case of Ri = 10, it is found that heat transfer rate is lower compared to that in clear domain. From the streamlines of both domains it is understandable that there is more spread of fluid in the clear domain compared to that of porous domain, because of the resistance to flow encountered in porous domain. The heat transfer marginally increases in the porous domain remaining almost constant with the increase in Ri values, showing that the change in the convection mode does not influence the rate of heat transfer much in a porous domain. Also the nature of isotherms and streamlines indicate that the heat transfer in all the cases is almost same.

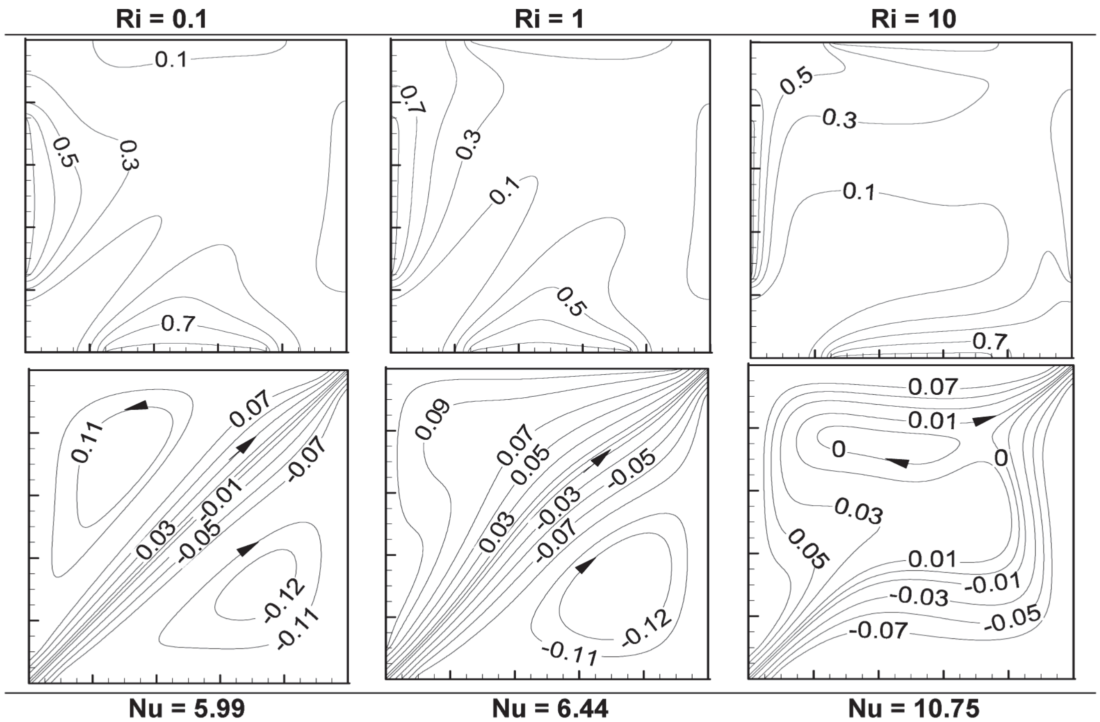

Fig. 2. Effects of different Richardson number (Ri = 0.1, 1, 10) on the isotherms (top row), streamlines (second row), and heat transfer rate (below isotherms) for clear domain at Re = 200.

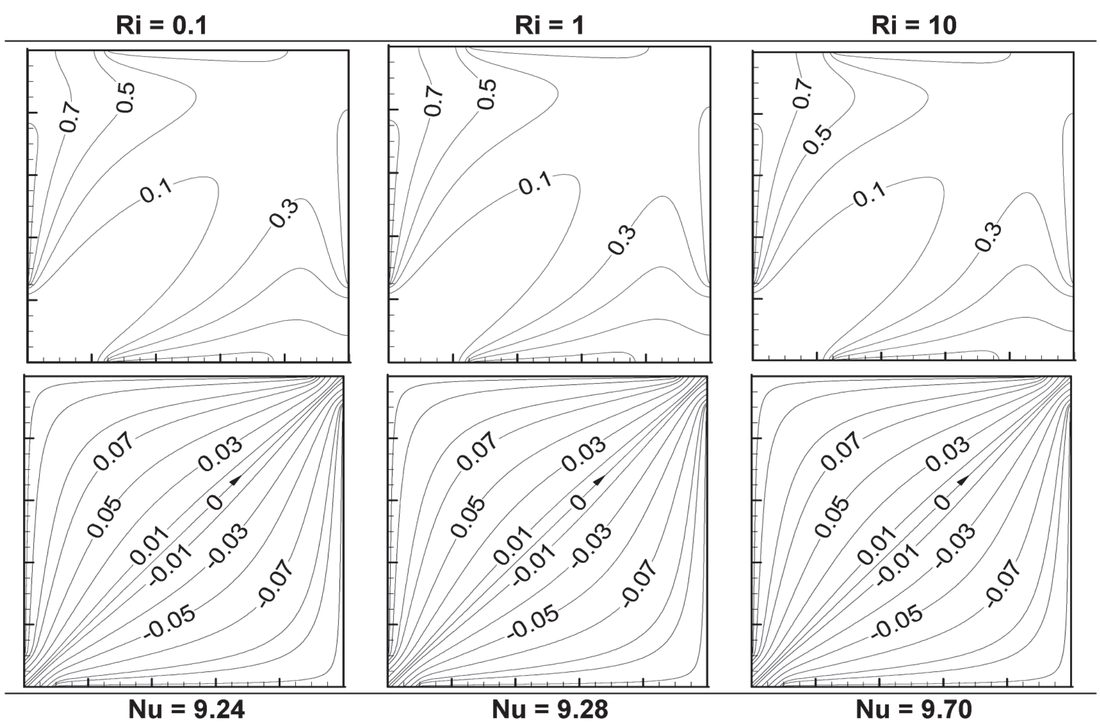

Fig. 3. Effects of different Richardson number (Ri = 0.1, 1, 10) on the isotherms (top row), streamlines (second row), and heat transfer rate (below isotherms) for porous domain at Re = 200, Da = 10^{-4} and ε = 0.6.

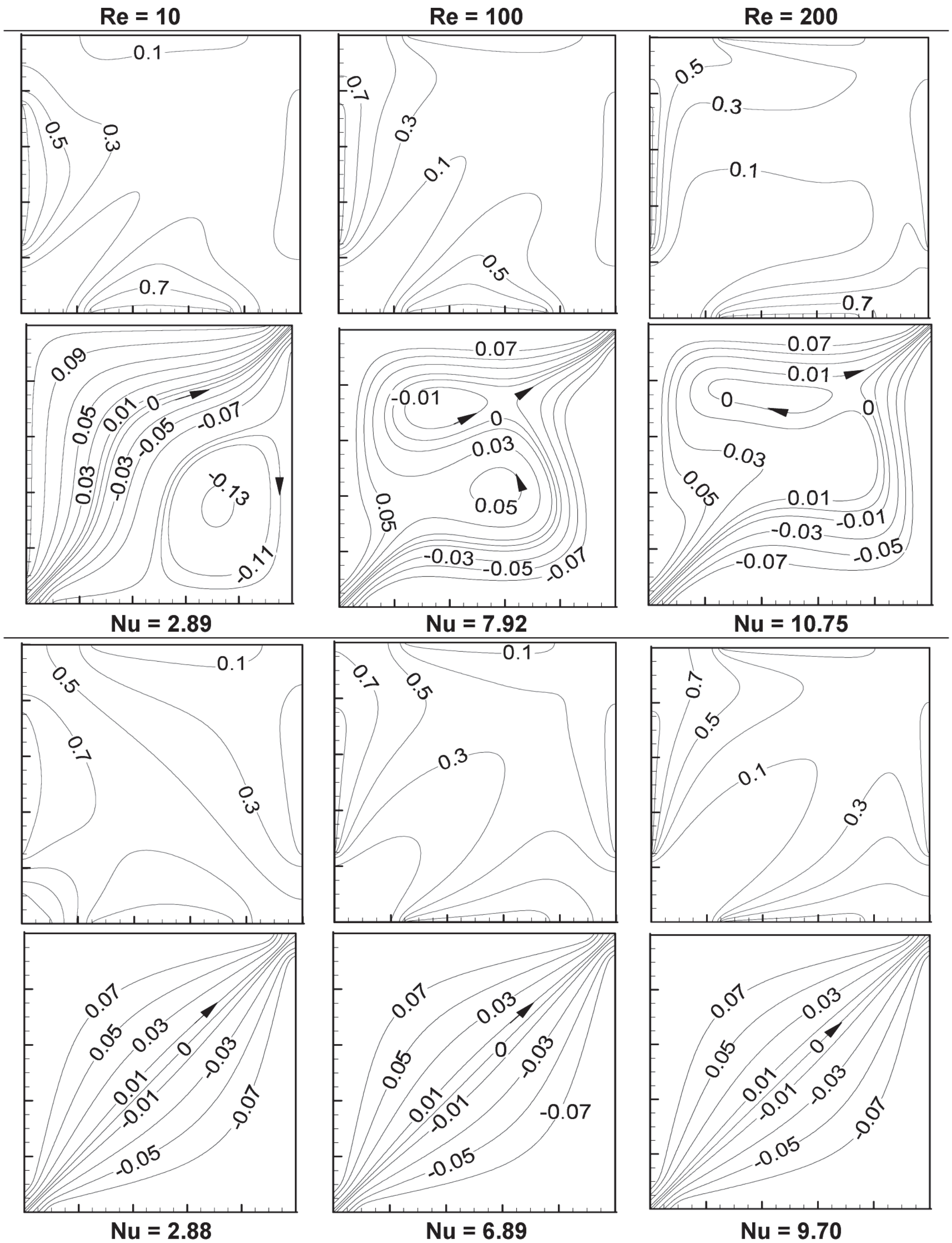

Fig. 4. Effects of different Reynolds number (Re = 10, 100, 200) on the isotherms (top row), streamlines (second row), and heat transfer rate (below isotherms) for (a) clear domain and (b) porous domain at Ri = 10, Da = 10^{-4} and ε = 0.6.

3.2. Effect of Corner Flow Velocity (Re)

The effect of different velocity of corner flow is indicated in Fig. 4 both for the clear domain (Fig. 4a) as well as porous domain (Fig. 4b). The stationary cavity walls lead to the formation of boundary layers (both velocity and thermal) following the no-slip condition in the fluid moving over them. The boundary layer thickness is indicated by the Re value. The increase in the Re value leads to increase in thickness of boundary layer and better heat transfer as seen by the increase in it can be visible from the streamlines that there is less spread of fluid in porous domain (more clearly for Ri = 100, 200). For Re = 10, the thickness of boundary layer is very less, as indicated by the very low values of Nu number for both the domains and the thickness increases significantly from Re =

3.3. Effect of Varying Darcy Number (Da)

The impact of Darcy number of the porous media on the flow pattern and isotherms is shown below in the Fig. 5. From the Nu value,

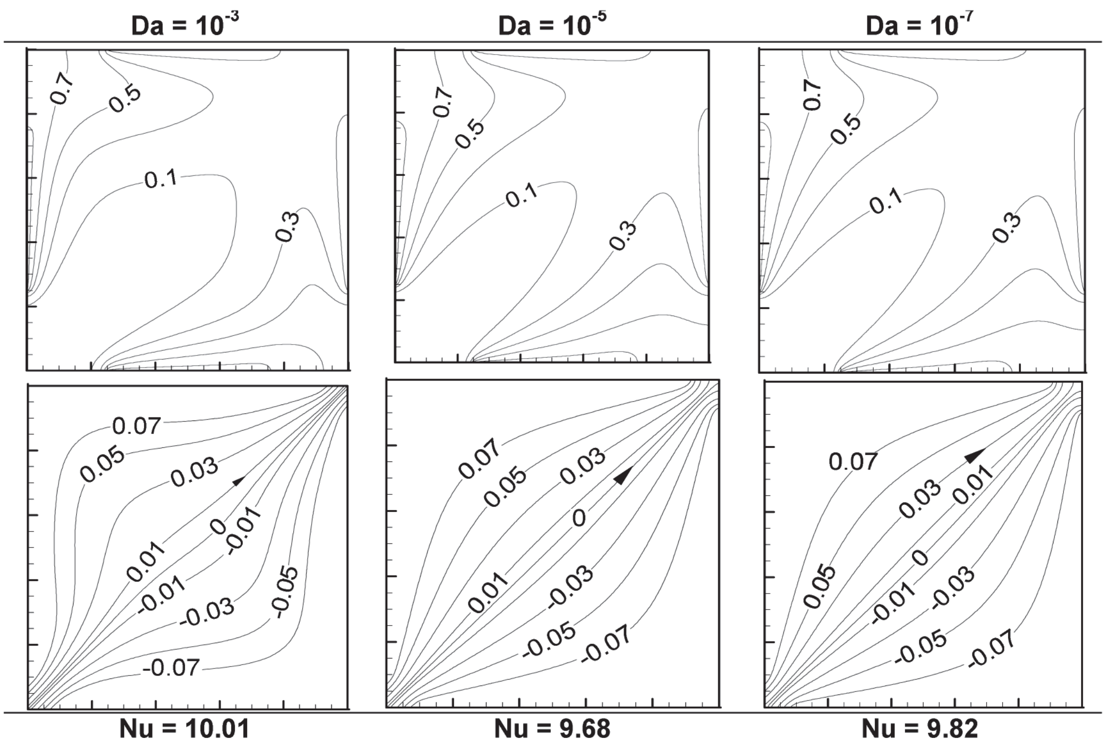

Fig. 5. Effects of permeability (Da = 10^{-3}, 10^{-5}, 10^{-7}) on the isotherms (top row), streamlines (second row), and heat transfer rate (below isotherms) at Re = 200, Ri = 10 and ε = 0.6.

Nu values for both the domains (clear and porous). But it can be seen that there is a decrease in the Nu values from the clear domain to Porous domain due to the resistance to flow in porous domain and also it is observed that there is a decrease in value of heat transfer from Da = 10^{-3} to 10^{-5} and again increases again from 10^{-5} to 10^{-7}. The better heat transfers in the case of Da = 10^{-3} is understandable from the flow pattern, where there is a better spread of fluid towards the heated walls owing to more resistance in flow in the cavity compared to that of Da = 10^{-5} and 10^{-7}. From Da = 10^{-5} to 10^{-7} the heat transfer increases following to the general trend that the rate of heat transfer increases with the decrease in Darcy number.

3.4. Effect of Varying Porosity (ε)

The presence of resistance to fluid flow is analyzed by changing the porosity value and associated isotherms and streamlines are presented in Fig. 6 with Re = 200, Ri = 10 and Da = 10^{-4}. The heat transfer is found to increase as the porosity value increases, it means decrease in the resistance of flow and thus allowing the fluid to expand more in the cavity, which can be observed from the streamlines, and resulting in better and more amount of heat transfer as indicated by the increase in Nu values with the increase in porosity.

3.5. Heat Transfer Characteristics

The heat transfer characteristics are shown in Fig. 7. From the Re variation plot it can be seen that for both porous and clear domain, the value of Nu increases monotonically with Re for Ri = 10, but for Ri variation, one can see that for clear domain the Nu remains almost constant while for porous domain it increases with Ri and there is a sudden increase of Nu from Ri = 1 to 10.

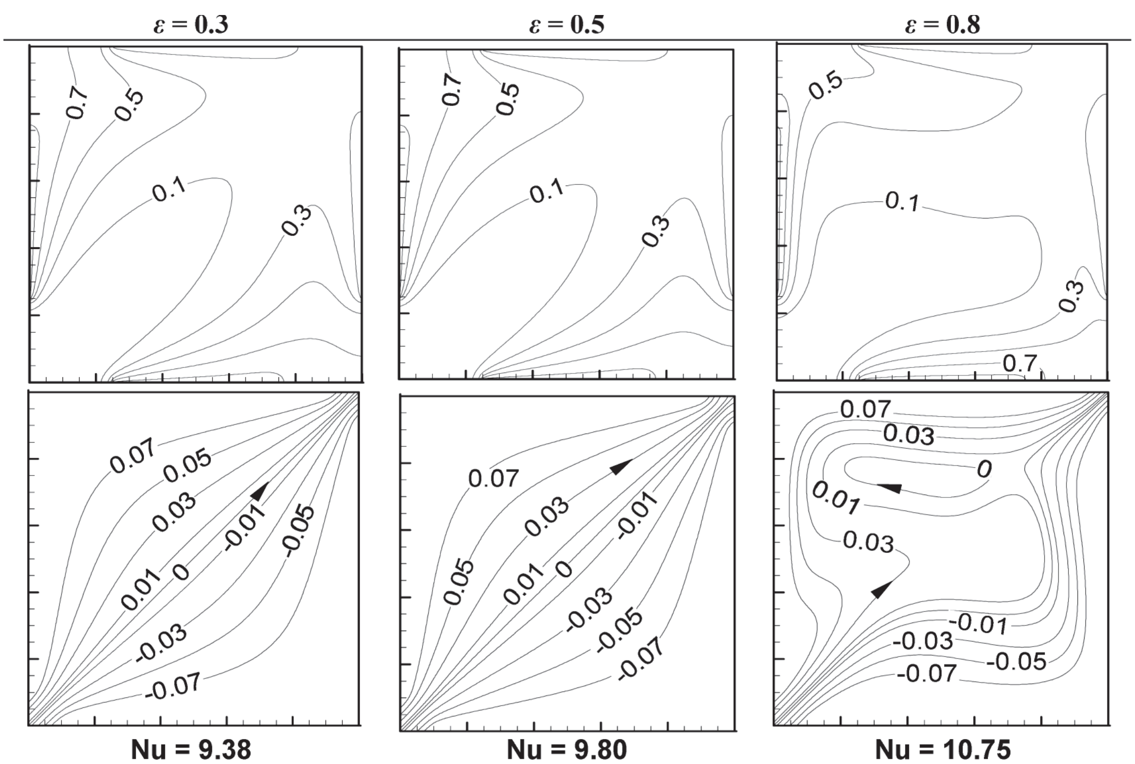

Fig. 6. Effect of porosity (ε = 0.3, 0.5, 0.8) on the isotherms (top row), streamlines (second row), and heat transfer rate (below isotherms) at Re = 200, Ri = 10 and Da = 10^{-4}.

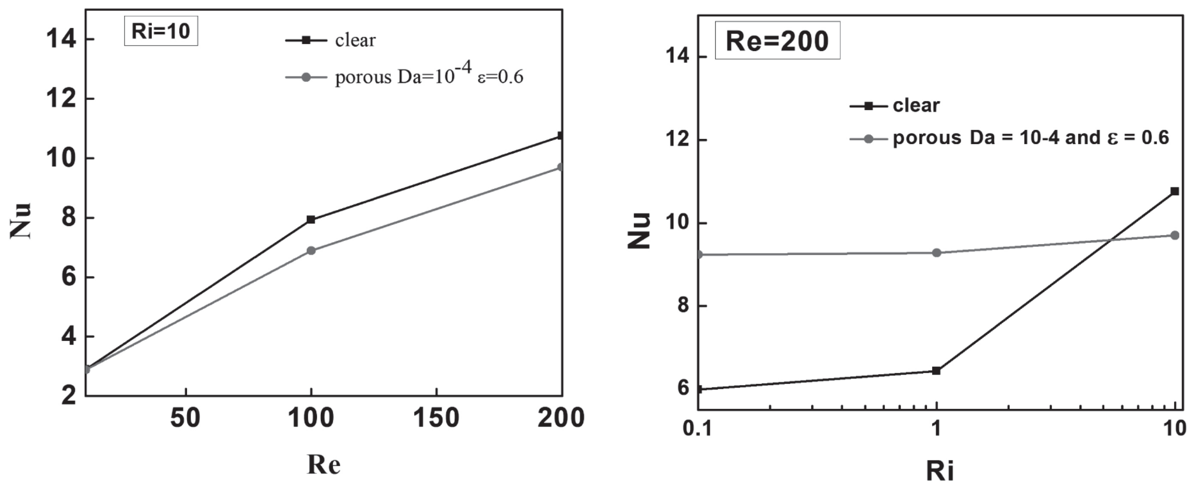

Fig. 7. Heat transfer characteristics: Nu vs Re for Ri = 10, and Nu vs Ri for Re = 200.

4. CONCLUSIONS

The mixed convection heat transfers characteristics in a ventilated enclosure are investigated numerically adopting the corner flow and with or without porous substance. This cavity is heated from the two adjacent walls, left and bottom. From the study, the major observations are as follows:

- It has been found out that the heat transfer is more in clear domain than in porous domain following the general rule, for any value of Re of the corner flow when Ri = 10.

- However, for fixed Re = 200, Richardson number has strong impact on Nu. For the present configuration it has been found that the heat transfer is less in clear domain than in porous domain defying the general rule when Ri < 10. The heat transfer is more in porous domain than that of clear porous domain. In this case more resistance offered by porous medium causes the flow to spread towards heater, that increases the heat transfer with porous medium.

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