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Enhancing Heat Transfer: Unraveling the Dynamics of Mixed Convection in a Vertical Porous Cavity

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ABSTRACT

Mixed convection inside a vertical square porous cavity is investigated numerically, with special attention given to left-side heating. In the study, the importance of the problem, methods used, main results, conclusions, and novelty of the work are discussed. The Brinkmann-Forchheimer-Darcy coupled momentum equations and the Local Thermal Equilibrium energy (LTE) equation are solved using the finite volume technique. Several elemental parameters are investigated, including Reynolds number ($50 \times Re \times 300$), Darcy number ($0.01 \times Da \times 100$), Richardson number ($0 \times Ri \times 30$), and porosity ($0.5 \times \varepsilon \geq 0.95$). The thermal conductivity ratio of solid to liquid was ($1<Kr<10^5$), and the inlet port width between ($0.05<d/H<0.3$). Results show that Nusselt number increases with Reynolds number, Richardson number, and pore thermal conductivity. With increasing porosity, an inverse relationship was observed. Nusselt number was maximized by finding the optimal inlet aperture size ($d/H = 0.25$). It provides valuable insights into how different parameters affect mixed convection in vertical square porous cavities. Heat transfer processes in porous media are improved by these results. This work goes beyond previous efforts in the literature in exploring a wide parameter space and determining the optimal inlet aperture size.

1. Introduction

Many thermal applications depend on the intricate dynamics of heat transfer due to burgeoning energy demands and the need for sustainable solutions. In a variety of applications, including small heat exchangers like microspheres or micro-pin-fin devices designed to cool electronic components, forced and natural convection flows can be combined within square or rectangular cavities filled with porous materials. Understanding the underlying complexities of these systems is crucial because they are nexus points between energy requirements and convective heat transfer [1,2].

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Flows driven by differential heating are characterized by two predominant driving forces: mechanical shear forces at the inlet and buoyancy forces. There are distinct phenomena such as buoyancy-aiding and buoyancy-opposing flows within the cavity, due to the interplay between these forces [3,4]. Energy transport and thermal management are significantly affected by this intricate dance of forces, necessitating a comprehensive investigation.

The study of mixed convection within lid-driven cavities filled with porous media by Al-Amiri [5], Khanafer and Chamkha [6], and Khanafer and Vafai [7] demonstrated the powerful impact of porous media on flow currents. Using top lid-driven porous cavities to transport convection energy, Oztop [8] explored optimal heater positioning in a porous cavity. According to Jeng and Tzeng [9], parameters like Grashof number, Reynolds number, and porosity affect the performance of mixed convection in cavities filled with aluminium foam.

Several studies have been conducted, including those by Vishnuvardhanarao and Das [10], Basak *et al.,* [11], Ahmed [12], Wang and Qin [13], and Wang *et al.,* [14], each with unique views on mixed convection inside cavities filled with porous materials. Various configurations are examined in these investigations, including the Richardson number, Darcy number, and the arrangement of heating elements. Using these findings, we can develop innovative solutions for improving energy efficiency by better understanding convection flows within porous cavities.

Conversely, researchers have explored mixed convection within porous cavities that are vented. In a vented porous cavity, Mahmud and Pop [15] studied steady Darcian mixed convection with the left vertical boundary heated, while the remaining boundaries were insulated. Researchers observed shifts in flow patterns from unicellular to multicellular based on Rayleigh, Peclet, and vent width measurements. According to Murthy and Kumar [16], injections and suctions can lead to non-Darcian convection within porous cavities, resulting in the emergence of multicellular flows within the cavity as a result of variations in critical parameters such as Grashof number, injection/suction opening width, and injection/suction inlet velocity.

By elevating the Rayleigh number or enlarging injection/suction ports, Kumar and Murthy [17] and Murthy and Kumar [18] respectively observed that Nusselt number could be increased in Darcian and non-Darcian convection scenarios. A Darcian mixed convection experiment was carried out by Behzadi *et al.,* [19] involving spheres in a horizontal porous cavity with a constant heat flux from below, cooled from above, and insulated vertical walls. Nusselt numbers decreased as sphere sizes increased, according to their research.

Identifying the novel aspects of the study is as follows; it emphasizes the significance of directional thermal influences and distinguishes itself from generic investigations due to its focus on left-side heating within a vertical square porous cavity. In this study, Reynolds number, Darcy number, Richardson number, porosity, thermal conductivity ratio of solid to liquid, and inlet port width are examined in detail. As a result of this extensive parameter exploration, it is possible to gain an understanding of how the system will behave under various conditions. An innovative methodological approach is employed to solve the coupled Brinkmann-Forchheimer-Darcy momentum equations and the Local Thermal Equilibrium (LTE) energy equation using the finite volume technique. By using a numerical approach, the results are more precise and contribute to the advancement of computational methods for studying mixed convection. Nusselt number can be maximized by identifying the optimal inlet aperture size (0.25 d/H). An important contribution to the field is provided by this new insight into optimizing the convective heat transfer in vertical square porous cavities [20,21].

The purpose of this study is to improve heat transfer processes in porous media by using the results obtained from the study. Considering real-world applications and potential advancements in thermal management, this practical application adds further value to the research. A wide parameter space is explored in the study, which goes beyond previous studies. As a result, a more comprehensive understanding of mixed convection phenomena has been achieved, surpassing the limitations of narrower studies. The study is unique in that it is nuanced in its focus on left-side heating, thoroughly explores various parameters, uses the finite volume technique, determines the optimal size of an inlet aperture, explores a wide parameter space, and offers practical implications for heat transfer in porous media. A combination of these elements distinguishes the research and advances knowledge about mixed convection within vertical porous cavities.

2. Mathematical Formulation

Figure 1 shows a schematic diagram of the current case under study. It consists of a square cavity filled with a porous substrate, heated isothermally at constant temperature (T_h), from its left boundary, and adiabatically along its remaining boundaries. Through an inlet vent with a diameter of (d), and an opposite slot with the same diameter, the cavity is ventilated vertically from below by a cold jet of air at constant velocity (v_o) and temperature (T_o),. During the study, the jet width is adjustable while the cavity length (H), is fixed. In order to describe the current problem, Khdher *et al.,* [22] and Nield and Bejan [23] provide the following dimensional equations:

$$\nabla.\boldsymbol{\acute{u}} = 0, \tag{1}$$

$$\frac{\rho}{\varepsilon}\left(\frac{\partial \boldsymbol{\acute{u}}}{\partial \acute{t}}\right) + \frac{\rho}{\varepsilon^2}(\boldsymbol{\acute{u}}.\nabla\boldsymbol{\acute{u}}) = -\frac{\mu}{K}\boldsymbol{\acute{u}} - \frac{\rho C_f \varepsilon}{\sqrt{K}}\left|\vec{\acute{u}}\right|\boldsymbol{\acute{u}} + \frac{\mu}{\varepsilon}(\nabla^2\boldsymbol{\acute{u}}) - \nabla\acute{P} - \rho g\beta\Delta\acute{T}, \tag{2}$$

$$(\rho c_p)_m\left(\frac{\partial \acute{T}}{\partial \acute{t}}\right) + (\varepsilon\rho c_p)_f\left(\boldsymbol{\acute{u}}.\nabla\acute{T}\right) = k_{f.eff}\left(\nabla^2\acute{T}\right), \tag{3}$$

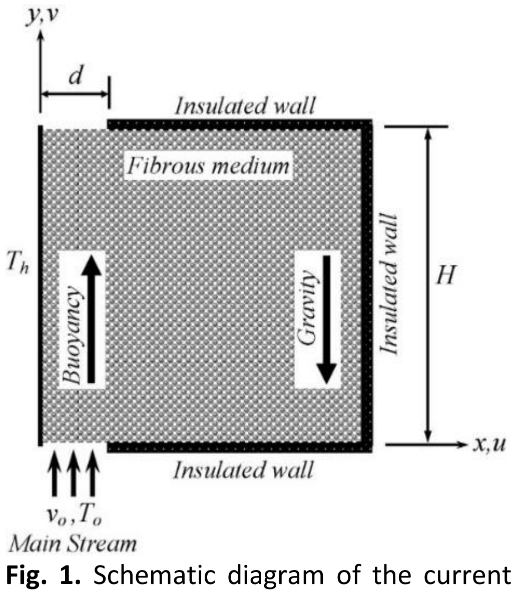

Fig. 1. Schematic diagram of the current studied case

Here, $(\rho c_p)_m = \varepsilon(\rho c_p)_f + (1-\varepsilon)(\rho c_p)_s$, (ε) is the porosity, $(\boldsymbol{\acute{u}})$ is the velocity vector, (C_f) is the inertia coefficient, $(k_{f.eff})$ is the thermal effective conductivity of fluid, (K) is the permeability. The following non-dimensional parameters:

$x = \frac{\acute{x}}{H}, y = \frac{\acute{y}}{H}, \boldsymbol{u} = \frac{\acute{\boldsymbol{u}}}{u_o}, P = \frac{\acute{P}}{\rho u_o}, t = \frac{\acute{t}}{u_o H}, T = \frac{(\acute{T} - T_o)}{(T_h - T_o)},$

Were used to formulate the following dimensionless equations:

$$\nabla . \boldsymbol{u} = 0 \tag{4}$$

$$\left(\frac{\partial \boldsymbol{u}}{\partial t}\right) + \frac{1}{\varepsilon}(\boldsymbol{u}.\nabla \boldsymbol{u}) = -\frac{\varepsilon}{\mathrm{Re.Da}}\boldsymbol{u} - \frac{C_f \varepsilon^2}{\sqrt{\mathrm{Da}}}|\vec{\boldsymbol{u}}|\boldsymbol{u} + \frac{1}{\mathrm{Re}}(\nabla^2 \boldsymbol{u}) - \varepsilon \nabla P - \varepsilon \mathrm{Ri} \Delta T \tag{5}$$

$$\left(\frac{\partial T}{\partial t}\right) + \frac{\varepsilon}{C}(\boldsymbol{u}.\nabla T) = \frac{k_{f.eff}}{C.\mathrm{Re.Pr}}(\nabla^2 T) \tag{6}$$

Here, $C = \varepsilon + (1 - \varepsilon)\left(\frac{k_r}{\alpha_r}\right)$. (α_r) and (k_r) are thermal ratios of conductivity and diffusivity, respectively. (Da), (Pr), (Ri), (Re) refer to the non-dimensional group of Darcy number, Prandtl number, Richardson number and Reynolds number. The rate of heat transport from the heat source to the cooling air is estimated by means of Nusselt number as follows:

$$Nu_m = \frac{1}{H}\sum \int_0^H \frac{h_x . y}{k_f} = \frac{-k_{f.eff}}{k_f} . \left.\frac{\partial T}{\partial x}\right|_h . dx \tag{7}$$

3. Numerical Approach

A computational technique of discretization is used to solve the highly connected non-dimensional governing Eq. (4) to Eq. (6). The finite-volume method described by Patankar [24] was used to calculate the temperatures and velocities inside the cavity by solving the controlling equations. According to Figure 2, the cavity is first divided into unequal mesh lines perpendicular to the horizontal axis. A mesh node is formed when two lines intersect. Using the Hybrid scheme outlined in Ferziger and Peric [25], the Eq. (4) to Eq. (6) are integrated throughout the whole node to generate nonlinear algebraic formulations. Following that, using the alternating direction implicit technique (ADI), the SIMPLE algorithmic rule is used to link the momentum and continuity equations throughout the domain.

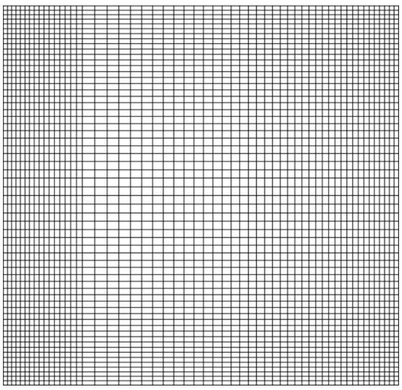

Fig. 2. The computational mesh

It was determined that mesh independency was necessary for accurate solutions in a mesh independency study. The following variant grid types are being used in Table 1: *G1, G2, G3, G4, G5,* and *G6*. Figure 3 shows that the results of the study when *Re*=250, *Ri* = 30, and d/H = 0.05. For mesh size *G4*, $(\Delta x \times \Delta y = 73 \times 73)$ was chosen to allow for a relevant minimum error < 0.1%.

Table 1
The mesh dependency study's mesh sizes

Grid	Size ($\Delta x \times \Delta y$)
G1	(43×43)
G2	(53×53)
G3	(63×63)
G4	(73×73)
G5	(83×83)
G6	(93×93)

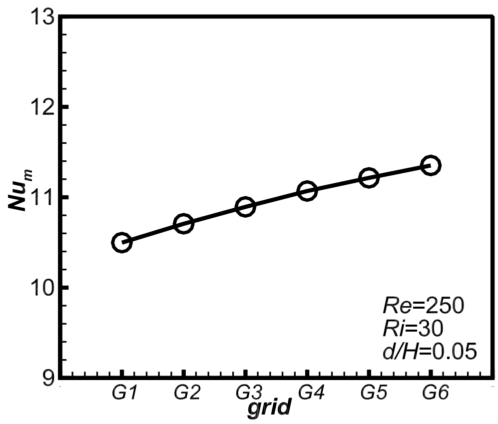

Fig. 3. Mesh dependency study

4. Results and Discussion

Using Porosity, Nusselt Number, and Darcy number as constant properties ($d/H = 0.2$), with constant porous properties such as thermal conductivity ratio ($Kr = 1000$), porosity ($\varepsilon = 0.95$), Darcy number ($Da = 0.1$). Figure 4 displays the relationship between Nusselt number and Reynolds number for various Richardson numbers. It can be noticed when Reynolds or Richardson numbers increase, the Nusselt number increases considerably for various Re numbers. It becomes increasingly important for Richardson number to be taken into account when *Re* increases. Since thermal dispersion produces more currents of convection in the cavity with greater *Re* or *Ri,* it is a result of the effect of thermal dispersion. Furthermore, heat is transferred more rapidly via convection due to these currents inducing thermal blending of fluids.

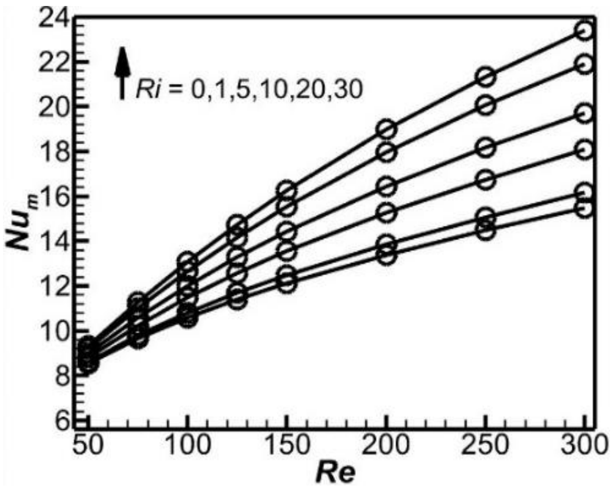

Fig. 4. Impact of *Re* on Nu_m for Various *Ri*, at *d/H* = 0.2, $Kr = 1000$, $Da = 0.1$, ε = 0.95.

The Darcy number expresses the effect of permeability on the average Nusselt number, for different *Re* and *Ri* numbers, at a constant thermal conductivity ratio (Kr = 1000) and porosity (ε = 0.95) [26]. As shown in Figure 5, there is a clear improvement in convective heat transfer (Nu_m) as the Darcy number (*Da*) increases; this improvement increases with larger *Re* and *Ri*. Due to the increased activity and flows in streams, the activity improvement leading in attributed to the growth in stream activity. Thermal energy is transferred from the hot surface at a faster rate when Da is increased.

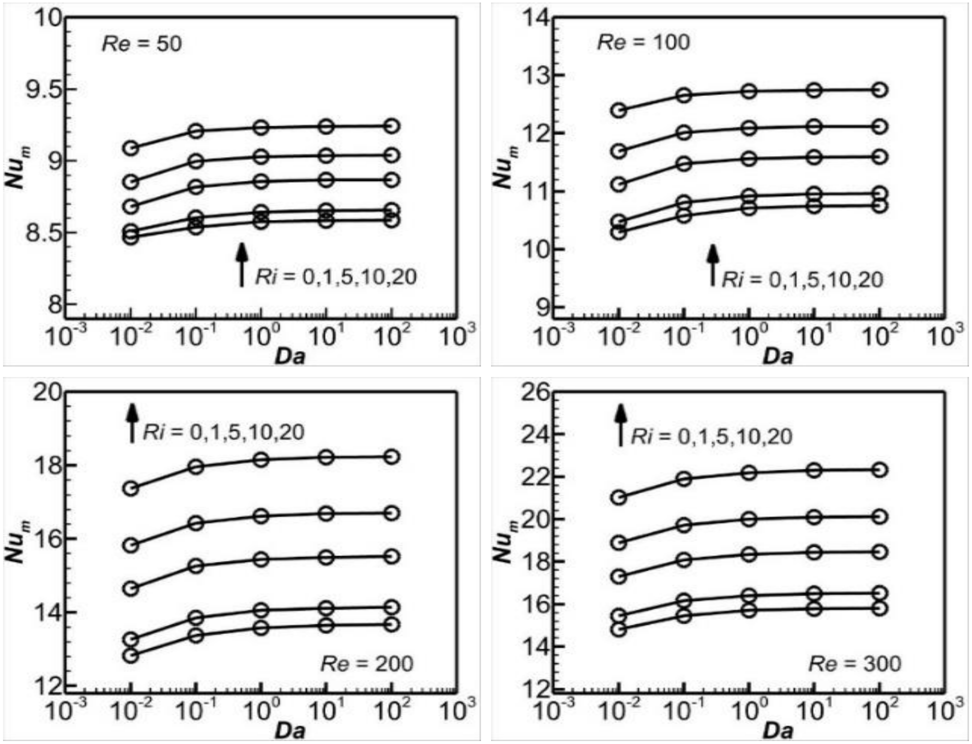

Fig. 5. Impact of *Da* on Nu_m for various *Ri*, *Re*, at *d/H* = 0.2, $Kr = 1000$, ε = 0.95

Figure 6 shows the effect of streamline patterns on the flow behaviour throughout the cavity for several *Re*, *Ri*, and *Da*, with constant values of (Kr = 1000) and (ε = 0.95). When *Re* = 50, *Ri* = 1.0, and *Da* = 0.1, no secondary flows develop inside the cavity, and a very smooth creeping flow results. The density of the vortex increases as Da increases inside the cavity. A significant increase in *Ri* from 1.0 to 20 generated two major secondary vortices, whereas a significant increase in *Re* from 50 to 300 produced three complex cells caused by the connection between flow-induced hydrodynamic forces and buoyancy forces inside the cavity.

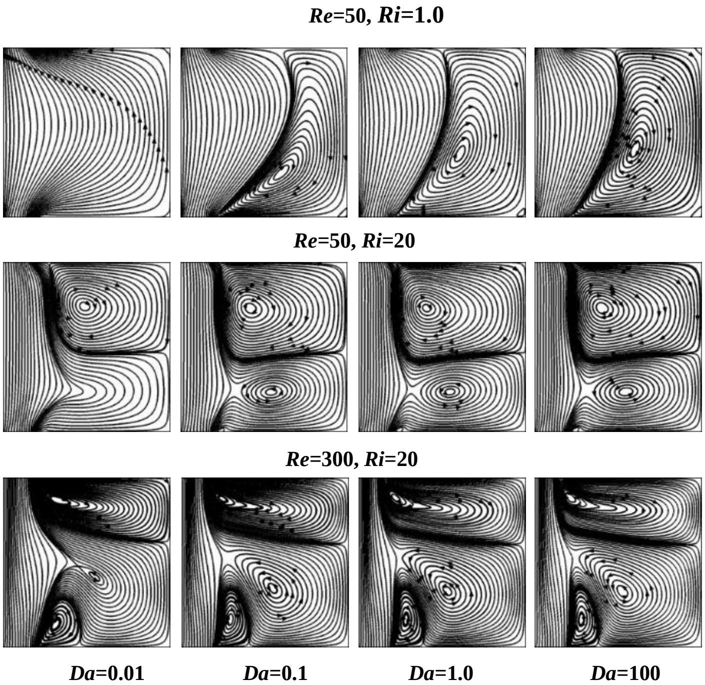

Fig. 6. Distribution of streamlines for various *Da, Re*, *Ri*, at *d/H* = 0.2, $Kr = 1000$, ε = 0.95

The ratio ($Kr = k_s/k_f$) expresses the change in thermal conductivity of the porous substrate, noting that the air conductivity was considered constant in the current study, as shown in Figure 7. When (Da = 0.1) and (ε = 0.95) are fixed, the value of this ratio is evaluated for values number of *Re* and *Ri*. For all heating and inflow conditions, such as *Ri* and *Re*, using porous materials with higher thermal conductivity increases Nu_m considerably. In fact, the porous substrate enhances the thermal conductivity inside the cavity, encouraging Nu_m to grow. Additionally, higher porous thermal conductivity increases flow speed near the vertical hot surface, reducing thereby the hydro-dynamic and thermal boundary layers, leading to higher temperature gradients, resulting in higher numbers.

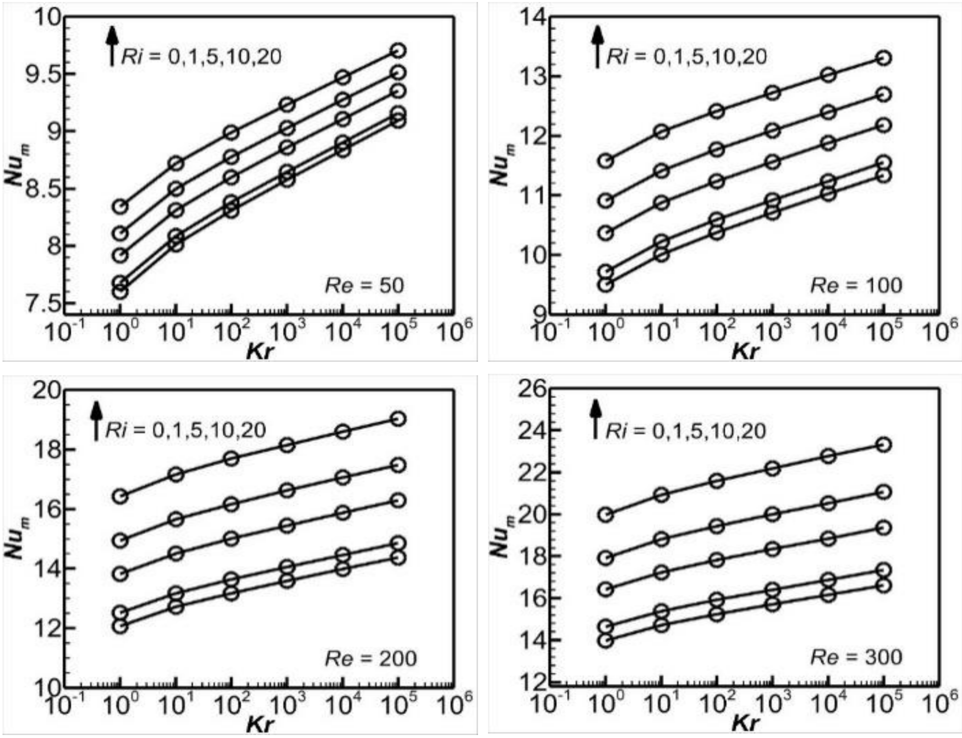

Fig. 7. Effect of thermal conductivity on mean Nusselt *Nu$_m$*ber for Various *Ri, Re*, at *d/H* = 0.2, $Da = 0.1$, ε = 0.95

Figure 8 shows the changes in *Nu$_m$* with porosity for different *Ri* and *Re*, at (Kr = 1000) and (Da = 0.1). Due to their lower flow resistance and increased flow speed near hot walls, high-porosity materials have a lower Nu_m. More convection heat is transferred. Moreover, at higher porosities, *Ri* becomes more significant, whereas at lower porosities, it disappears completely.

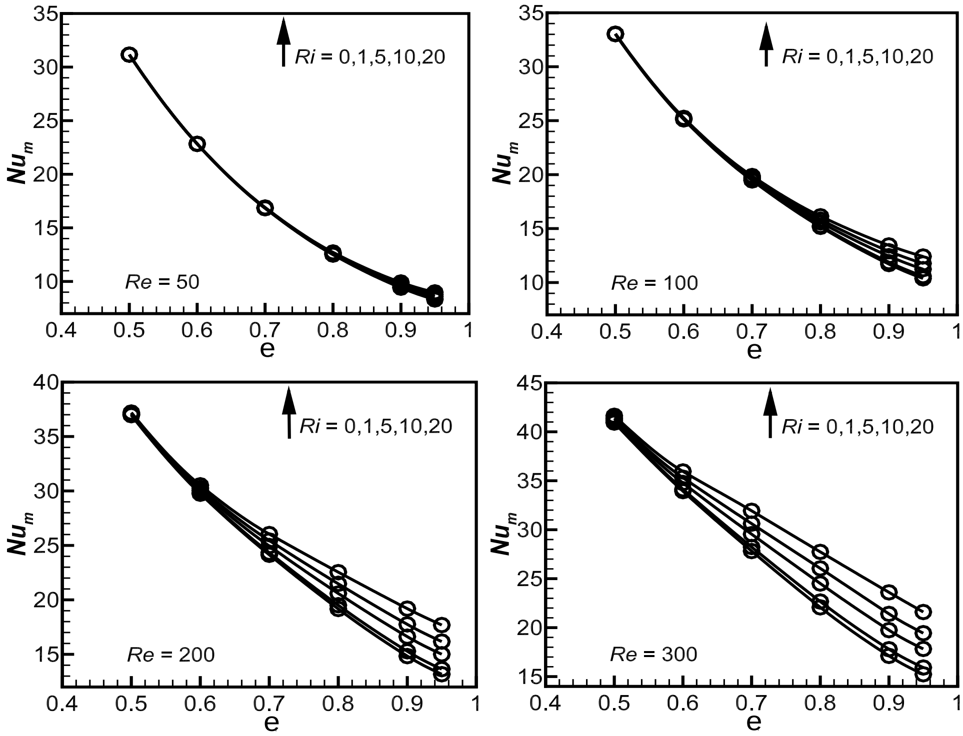

Fig. 8. Effect of porosity on Nusselt *Nu$_m$*ber for various *Ri, Re*, at *d/H* = 0.2, $Kr = 100$, $Da = 0.1$

Figure 9 manifests the temperature distribution within the cavity for different *Re, Ri*, and ε, at fixed (Kr=100), and (Da=0.1). When *Re, Ri*, and are low, the system is more conductive. When any of these parameters are increased, the convective system becomes more active and the thermal boundary layer of the fluid on the heated vertical left surface becomes thinner. As a result, temperature gradients are higher, and Nu_m values are higher.

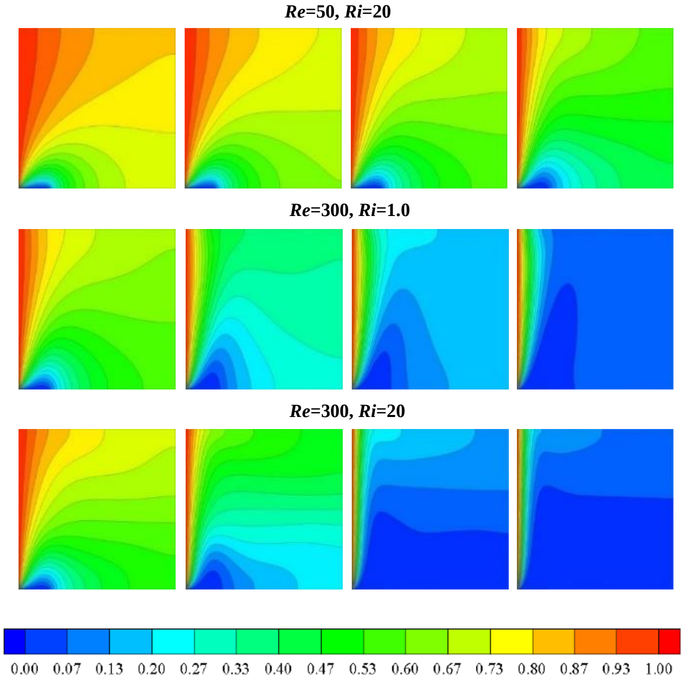

Fig. 9. Isotherm patterns for various *ε, Ri, Re*, at *d/H* = 0.2, $Kr = 100, Da = 0.1$

A Nu_m profile with the inlet port width varying between (d/H = 0.05–0.3) is illustrated in Figure 10. The results showed that the Nu_m is significantly increased for all ranges of Re and *Ri* when the inlet port width is increased. Additionally, wider ports have a greater impact on *Ri* at higher levels. Using the plots, it can be seen that d/H = 0.25 is the optimal port width to get the greatest values of Nu_m [27].

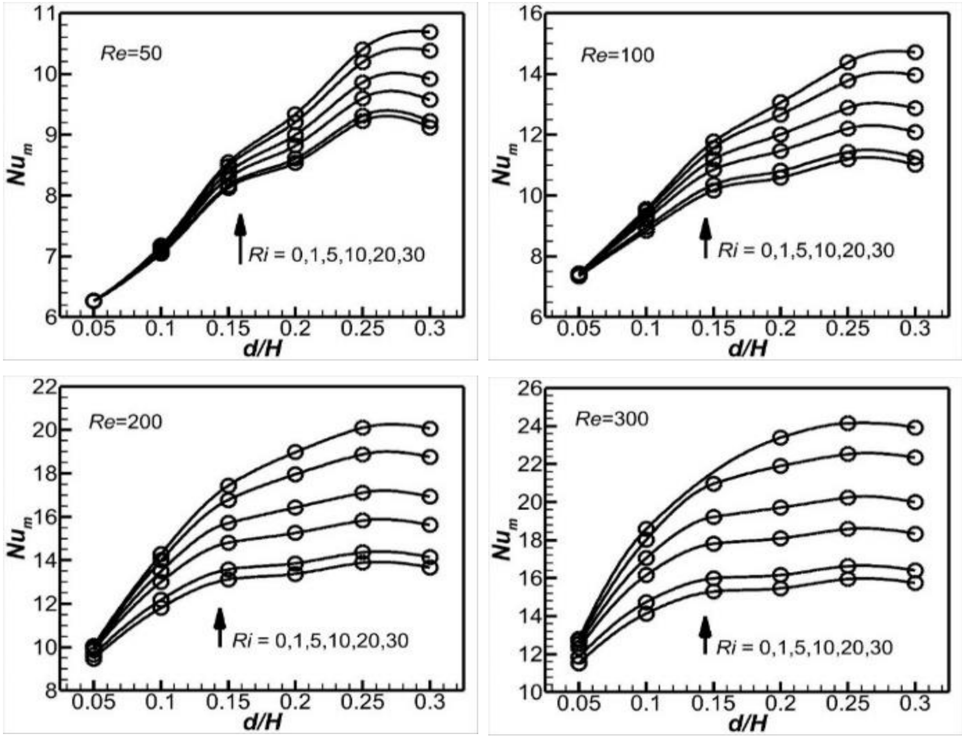

Fig. 10. Effect of inlet width on Nusselt number for various *Ri, Re*, at $Kr = 1000$, $Da = 0.1$, ε = 0.95

5. Conclusions

Transient mixed convection between porous cavities vented vertically by air, and heated from the sidewalls is explored by this numerical study. For predictions of non-Darican velocity and temperature fields, the current study use the Brinkman-Forchheimer-Darcy equation and LTE energy equation. A wide range of important parameters determine the inlet port width, including Reynolds number (Re=50-300), Darcy number (Da=0.01-100), Richardson number (*Ri*=0-30), porosity (ε=0.5-0.95), thermal conductivity ratio ($Kr = 1 - 10^5$), and inlet port width (d/H=0.05-0.3). One of the most important conclusions from this study is that: When Reynolds, Richardson, or Darcy numbers are raised; the Nusselt number is increased by an increasing percentage. As porous thermal conductivity increases, the average Nusselt number increased. When porosity increases, the Nusselt number also decreased. The average Nusselt number improved by increasing the inlet port width. However, there is an optimal size (d/H=0.25) for acquiring peak Nusselt numbers.

Conflict of interests

The authors confirm and declare that there is no conflict of interests regarding the publication of this paper.

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