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Numerical investigation of the effect of circular porous fins on natural heat transfer enhancement in an annulus cavity

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Abstract. The present asymmetric numerical study aims to evaluate the effect of using a certain type of fins, i.e., circular porous fins, on heat convection inside an annulus enclosure. The outer and inner walls are considered to be having a constant temperature condition. The porous fins are installed on the outer wall and other walls are insulated. In addition, the effects of different parameters including annulus inclination angle, annulus aspect ratio, Darcy number, Rayleigh number, thermal conductivity, and the position, number, and length of fins on heat transfer enhancement inside the annulus were investigated. The obtained results revealed that increasing Darcy number over a particular value would dramatically increase the average Nusselt number at both aspect ratios; however, the annulus with an aspect ratio of 3:1 had a higher value of the average Nusselt number than those with an aspect ratio of 2:1. As observed, the application of relatively low solid-to-fluid phase thermal conductivity nullified the effect of increasing the number of porous fins on heat transfer enhancement; however, upon increasing the relative thermal conductivity to $K_e = 100$, installing four porous fins, compared to using only one fin, on the inner cylinder could raise the value of the average Nusselt number up to 7%.

1. Introduction

One of the challenging problems researchers are facing is how to increase heat transfer efficiency and modify the performance of thermal systems. Their extensive applications in industry as well as the necessity of reducing energy consumption have encouraged researchers to study different methods and different aspects of such underexplored systems. Over the past years, a number of studies have been conducted on heat transfer within internal heat exchangers. In this respect, heat convection inside the space between two concentric annuli is one of the issues that has been considered with different configurations and effective parameters. Many efforts have been made to enhance heat transfer enhancement; however, use of porous mediums, especially circular fins, has been less studied.

Davis and Thomas [1] numerically investigated natural convection between concentric vertical cylinders. Their results showed that heat transfer rate de-

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pended on Rayleigh number, aspect, and radius ratio. They also proposed a correlation based on the studied parameters to predict the average Nusselt number. In a numerical study, Schwab and Witt [2] analyzed the natural convection between two vertical coaxial cylinders. The results indicated that a fully developed boundary layer formed at Rayleigh number greater than 5000. In addition, there were a uniform vertical temperature gradient and a unicellular flow pattern between the cylinders. Powe et al. [3] employed a finite difference technique called "central difference" to predict airflow behavior inside the gap between two concentric cylinders for natural convection. A comparison between the results obtained from previous experimental works and their numerical results proved that their method would be able to predict the flow behavior in the transient region as well as steady laminar flow. Bejan and Chang-Lin [4] investigated the natural convection within a long horizontal space between two concentric cylinders. Two different cases were considered in the intended subject: (a) An incompressible fluid as the first case was used inside the annulus space; (b) Porous material saturated with fluid was used as the second case. Different temperatures at the end of the cylinders created a counter flow inside the gap. The results of their study indicated that the Nusselt number was enhanced by increasing the Rayleigh number. Havastad and Burns [5] numerically investigated the convective heat transfer in the vertical annuli filled with porous media. They reported that the radius ratio, aspect ratio, modified Rayleigh number, and Biot number were the effective nonlinear parameters. In addition, they suggested a critical height for maximum heat transfer due to the limitation of conduction in certain inner areas and it increased slowly beyond that limit. Hickox and Gartling [6] conducted a 2D numerical study on a vertical annular. The gap between cylinders was filled with a fluid-saturated porous medium whose permeability changed radially. The constant temperature was considered as the boundary condition for vertical walls ($T_i > T_o$). According to the results, natural convection exhibited 2.6 times more heat transfer rate than thermal conduction. In addition, application of anisotropic permeability for the porous region had a different influence on convection, considering different values for involved parameters. Prasad and Kulacki [7] numerically studied natural convection in a vertical porous annulus. They observed that the curvature had a considerable impact on temperature and velocity fields, disturbing the symmetry of the temperature and velocity distribution in the cavity. Moreover, heat transfer was found to be much greater at the top edge of the cold wall due to the higher convective velocities. Further, heat transfer followed an increasing trend with an increase in the radius ratio. Prasad [8] performed numerical simulations to determine the impact of applying constant heat flux of a vertical annulus on temperature distribution. According to the findings, upon increasing the Rayleigh number, the heated wall temperature would increase. In addition, an increase in radius ratio would in turn increase the overall Nusselt number. Moreover, they determined the better performance of heat transfer under constant heat flux boundary conditions than that under isothermal boundary conditions. Chai and Patankar [9] studied laminar natural convection in a horizontal annulus with fins. They found that installing fins had considerable effects on the flow patterns, temperature distributions, and local Nusselt number; however, the effect of the direction of the fins on the average Nusselt number was negligible. Hasnaoui et al. [10] studied natural heat transfer within a vertical porous annulus both analytically and numerically. The constant heat flux and adiabatic boundary conditions were considered in the inner wall and other ones, respectively. The results revealed that both analytical and numerical methods could accurately predict the flow behavior and heat transfer characteristics. Marpu [11] investigated natural convection in a vertical porous annulus using Forchheimer and Brinkman's extended Darcy flow model. The results demonstrated a higher rate of reduction in the average Nusselt number due to Brinkman's viscous terms compared to Forchheimer inertial terms. They also found that the non-Darcy flow model was in close agreement with the experimental data of the high permeability porous medium. Aboubi et al. [12] addressed a 2D gap between concentric cylinders filled with an anisotropic porous medium. To this end, they took into account the effects of different parameters such as angular position, permeability, and Rayleigh number. According to their findings, the flow behavior and the temperature distribution were not symmetrical while using an anisotropic porous medium. In case the inclination of the axis in the asymmetric mode is around 40 degrees, strong circulation occurs in the vicinity of the annulus. They also realized that the effect of using porous medium on the heat transfer enhancement was the same as that of using a magnetic field. Abu-Hijleh [13] studied the natural convection in a cylinder with high conductivity permeable fins. The effect of fin geometry on the average Nusselt number was analyzed and the results confirmed the better heat transfer enhancement of porous fins than that of solid ones. In addition, a certain number of porous fins were found beyond which no more heat convection enhancement occurred. Kiwan and Zeitoun [14] studied a horizontal annulus with porous fins. They also evaluated the effects of Darcy number, fin conductivity ratio, and Rayleigh number. Use of porous fins resulted in 75 and 100% rates of heat transfer enhancement compared to solid fins and bare annuls, respectively. Furthermore, heat transfer exhibited a decreasing pattern with an increase in the inclination angle of the porous fin. Chen et al. [15] investigated natural convection in a vertical annulus through the lattice Boltzmann method. The vertical walls were considered to be having a constant temperature condition. Their obtained results indicated that at Rayleigh number over 10^6, the flow became unsteady, while for $Ra < 10^6$, the flow was laminar. Upon increasing the Rayleigh number, the vortex moves toward the outer wall. While heat convection is the dominant heat transfer regime at $Ra \geq 10^5$, heat conduction is predominant for $Ra < 10^5$. According to their findings, increasing the Prandtl number would increase the total entropy generation monotonously. Sankar et al. [16] studied the natural convection of a vertical annulus filled with a porous medium. The effects of size and location of the heater were investigated, the results of which revealed that using small heaters or installing them in the middle of the gap would cause a more heat transfer rate. Togun et al. [17] reviewed the forced, natural, and mixed convection heat transfers in the annular gap. Different configurations for annular passage and boundary conditions were also taken into account. They concluded that the application of iso-flux heat as a boundary condition could provide better efficiency than that of isothermal walls. Chikh and Allouache [18] installed a layer of porous materials over the internal cylinder and analyzed the effect of the thickness of the porous layer and its permeability. It was observed that installation of a porous layer could reduce the total entropy generation inside the cavity. In addition, the highest heat transfer occurred on condition that the thickness of the layer would not exceed more than 40% of the gap. Jha and Yusuf [19] analytically investigated the transient natural convection in a porous annulus. They discovered the considerable impact of different heating boundary conditions as well as the gap between the cylinders on both velocity and temperature fields. Hatami [20] numerically investigated the natural convection within a nanofluid filled rectangular cavity with heated fins and evaluated the effect of different types of nanofluids, volume fraction, and heights of fins. The obtained results indicated higher average and local Nusselt numbers for TiO_2 nanofluid than those for Al_2O_3, especially in moderate volume fractions. Stark et al. [21] experimentally and analytically studied the degrees of thermal efficiency and resistance of porous metal foam-fins within a circular tube and developed analytical expressions to predict them in terms of convection and conduction. Nimvari and Jouybari [22] evaluated the turbulence impacts in a porous layer on the heat transfer performance in a partially filled pipe. They found that upon increasing the thickness of the porous layer, the turbulence effects intensified even more. They also highlighted the importance of the turbulence effects even at a pore base Reynolds number lower than the critical Reynolds number in the porous media. Bondareva and Sheremet [23] numerically examined the electronic cooling performance of PCM materials in the copper-finned heat storage system. They also evaluated the effect of the fin profile on heat convection and melting time of the PCM materials and observed an increasing Nusselt number trend following an increase in the fin's width due to the formation of effective circulation zones between the fins. In addition, they reported the same melting start time for different fin heights. Wang et al. [24] utilized the lattice Boltzmann method for studying laminar natural convection in a vertical annulus with a wall covered by a porous layer. They found a critical thickness value of the porous layer for maximum and minimum heat transfer. The mentioned critical value decreased by decreasing Darcy number and increasing Rayleigh number. Jha and Oni [25] analytically applied the time-periodic thermal boundary condition for the vertical annular cavity. They found that the periodic temperature and velocity followed a decreasing trend with Prandtl and Strouhal numbers, an increase which would enhance heat transfer. In addition, they found that for small Strouhal numbers, the temperature distribution was the same for different configurations of the cavity. Nada and Said [26] evaluated the effects of arrangements and dimensions of fins on natural heat convection inside horizontal annuluses. According to the obtained results, installing fins without consideration of the arrangement of fins or numbers would enhance heat convection inside the gap; however, further analysis revealed that longitudinal fins outperformed annular fins. Wanga and Daib [27] numerically investigated natural convection in a nanofluid-filled square cavity with non-uniform boundary conditions. They reported an increase in heat transfer with an increase in the inclination angle and dependence of the heat transfer augmentation on Rayleigh number resulting from an increase in the nanoparticle volume fraction. Mesgarpour et al. [28] numerically examined the heat transfer performance of porous tapered fins. Compared to the case of rigid fins in laminar flow, in their study, the heat transfer rate increased by 33% and pressure drop decreased by 9%. Tayebi et al. [29] investigated free convection in an annular space between confocal elliptic cylinders. The distribution pattern of the stream functions was symmetric, the upper part of which showed the isotherm lines about the vertical line and high temperature. Jing et al. [30] investigated the effect of heated fins with different geometries on heat transfer and entropy generation in a rectangular-shaped cavity filled with non-Newtonian nanofluid. They also evaluated the impact of Hartman number, nanoparticles type, and volume fraction. The obtained results indicated that increasing the fin length would result in enhancement in both local and average

Nusselt numbers. As reported earlier, Fe_3O_4 nano particle outperformed Cuo and Al_2O_3. Dogonchi et al. [31] studied the natural convection of nanofluid within the porous annulus. Results confirmed the significant role of the aspect ratio of cavity in heat transfer enhancement; in other words, the larger the aspect ratio, the greater the heat transfer. Mebarek-Oudina et al. [32] studied magnetized nanofluid natural convection in the annular space between vertical cylinders filled with a porous medium. They evaluated the effect of the discrete heat source and found that an increase in the length of the heat source would lead to a lower heat transfer rate. Astanina et al. [33] numerically investigated natural convective heat transfer in a porous chamber with a finned heat sink. They studied the effect of Darcy and Ostrogradsky numbers as well as the geometric features of the radiator and reported better heat transfer as a result of increasing Darcy number, porous layer permeability, and temperature-dependent viscosity. Moreover, they found the optimum heat transfer with fin numbers 3 and 4. Lakshmi et al. [34] analyzed the free convection of nanofluid in a low-porosity cylindrical annulus. In their analytical approach, the maximum and minimum heat transfer occurred in the lower and upper parts of the annuli, respectively. Moreover, they identified the porous region as a heat storage system due to its low porosity, thus leading to lower thermal conductivity in the porous part. Belabid and Allali [35] evaluated the impact of temperature modulation on heat transfer in a horizontal porous annulus with a time-dependent periodic constant temperature boundary condition. They reported stability of heat convection in moderate values of the modulation frequency. In their three-dimensional numerical investigation, Yousif et al. [36] studied the impact of porous media on forced heat convection in a concentric vertical annular tube. They also considered different sizes of porous media, porosity, and Reynolds number and reported an increase in heat transfer value with decreasing porosity. Kiwan et al. [37] experimentally studied natural heat convection in a vertical cylinder using porous fins. They evaluated the effect of the number and thickness of fins and reported the significant dependence of Nusselt number of fins' permeability. They also found that increasing the fin thickness would produce greater heat transfer. Esfe et al. [38] numerically studied natural convection of a nanofluid filled cubical cavity with porous fins. They identified uniform isotherms at low Reynolds numbers created by weak buoyancy force. Mebarek-Oudina et al. [39] studied buoyancy convection in an annulus filled by nanofluid with a heat source on the inner wall. Different locations of the heater were taken into account to determine the optimum location for minimum and maximum heat transfer rates. They found that the mentioned parameter totally altered the fluid field and temperature distribution within the cavity and concluded that the heater location could be used as a control mechanism of heat transport. Kiran et al. [40] theoretically analyzed the thermal pattern and natural convection in an annular cavity. According to their findings, increasing the thermal conductivity ratio or thickness of the cylinder would result in lower heat convection.

Despite the numerous applications of annulus cavities as a common configuration, few studies have been conducted on heat transfer enhancement using fins, especially circular porous fins. To fill the mentioned gap, the current study evaluated the effect of using circular porous fins in an annulus cavity under axisymmetric boundary conditions. To the authors' best knowledge, the application of circular porous fins in the annular cavity has not been analyzed yet.

2. Problem definition

The present study investigated a laminar axisymmetric steady-state natural convection inside a cylindrical annular enclosure with circular porous fins installed on the hot wall of the cavity. More details of geometry and boundary conditions are given in Figure 1 where D and H represent the width and height of the cavity, respectively. The inner wall of the annulus on which the circular porous fins were installed was kept at a constant higher temperature, while the outer wall was maintained at a constant lower temperature. The horizontal walls of the cavity were assumed to be adiabatic. In addition, the fluid within the cavity was assumed to be Newtonian and incompressible. Regardless of the small temperature variations from the inner wall to the outer one, all other properties except density were considered to be constant. In line with the investigation of natural convection, Boussinesq approximation was also considered for this term. In addition, compared to conduction and convection effects, the effect of heat dissipation was neglected. Here, porous fins were assumed to be isotropic, and volume-averaging method and Darcy-Forchheimer model were used for porous fins modeling.

3. Governing equations

The dimensionless forms of the governing equations including continuity, momentum, and energy were taken into account in this study. In order to combine the governing equations for porous and impermeable regions inside the annular enclosure, the binary parameter δ was introduced which was either one or zero for porous fin and clear region, respectively [20,41].

$$\frac{1}{R}\frac{\partial(RU)}{\partial R}+\frac{\partial W}{\partial Z}=0, \tag{1}$$

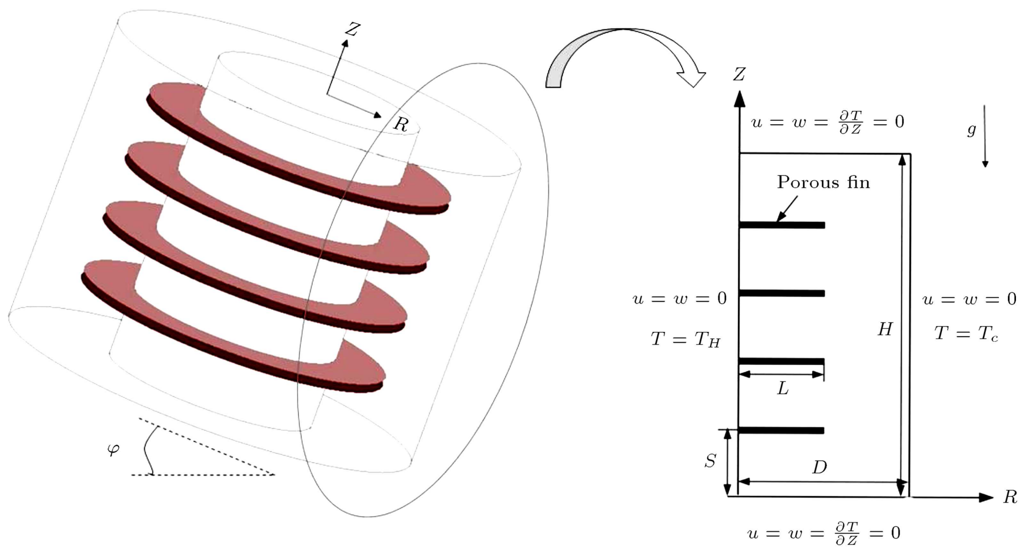


Figure 1. The details of the annular enclosure, boundary conditions, and coordinate system.

$$\left[\frac{\delta}{\varepsilon^2} - (\delta - 1)\right]\left(U\frac{\partial U}{\partial R} + W\frac{\partial U}{\partial Z}\right) = -\frac{\partial P}{\partial R} + \Pr\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial U}{\partial R}\right) - \frac{U}{R^2} + \frac{\partial^2 U}{\partial Z^2}\right] - (\delta)\frac{\Pr}{Da}U + Ra\Pr\theta\sin\varphi, \quad (2)$$

$$\left[\frac{\delta}{\varepsilon^2} - (\delta - 1)\right]\left(U\frac{\partial W}{\partial R} + W\frac{\partial W}{\partial Z}\right) = -\frac{\partial P}{\partial Z} + \Pr\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial W}{\partial R}\right) + \frac{\partial^2 W}{\partial Z^2}\right] - (\delta)\frac{\Pr}{Da}W + Ra\Pr\theta\cos\varphi, \quad (3)$$

$$U\frac{\partial \theta}{\partial R} + W\frac{\partial \theta}{\partial Z} = \left[\delta\left(\frac{K_{eff}}{K_f} - 1\right) + 1\right]\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial \theta}{\partial R}\right) + \frac{\partial}{\partial Z}\left(\frac{\partial \theta}{\partial Z}\right)\right], \quad (4)$$

where ε and K_{eff} are porosity and effective thermal conductivity, respectively. Further, K_{eff} is defined as follows [42]:

$$K_{eff} = \varepsilon K_f + (1 - \varepsilon)K_s, \quad (5)$$

where K_f and K_s represent the thermal conductivity of the fluid and solid phases, respectively.

The considered dimensionless parameter in the above equations is as follows:

$$R = \frac{r}{D}, \qquad Z = \frac{z}{D}, \qquad U = \frac{uD}{\alpha}, \qquad W = \frac{wD}{\alpha},$$

$$P = \frac{pW^2}{\rho\alpha^2}, \qquad Ra = \frac{g\beta(T_H - T_C)D^3}{\nu\alpha},$$

$$Da = \frac{K}{D^2}, \qquad Pr = \frac{\nu}{\alpha}, \qquad \theta = \frac{(T - T_C)}{(T_H - T_C)}, \quad (6)$$

where P is the dimensionless pressure; Ra, Da, and Pr are the Rayleigh, Darcy, and Prandtl numbers, respectively. The dimensionless form of the boundary conditions is as follow:

Inner wall:

$$U = W = 0, \qquad \theta = 1,$$

Outer wall:

$$U = W = 0, \qquad \theta = 0,$$

Bottom and top walls:

$$U = W = \frac{\partial \theta}{\partial Z} = 0. \quad (7)$$

At the porous and fluid interface, we have:

$$U_{fluid} = U_{porous}, \qquad W_{fluid} = W_{porous},$$

$$\mu_f \left.\frac{\partial U}{\partial n}\right|_{fluid} = \mu_{eff} \left.\frac{\partial U}{\partial n}\right|_{porous},$$

$$\mu_f \left.\frac{\partial W}{\partial n}\right|_{fluid} = \mu_{eff} \left.\frac{\partial W}{\partial n}\right|_{porous},$$

$$\theta_{fluid} = \theta_{porous},$$

$$K_f \left.\frac{\partial \theta}{\partial n}\right|_{fluid} = K_{eff} \left.\frac{\partial \theta}{\partial n}\right|_{porous}. \quad (8)$$

Eq. (9) is used to calculate the average Nusselt number on the cold wall:

$$\overline{Nu} = \int_0^H \frac{\partial \theta}{\partial Z} dR. \quad (9)$$

4. Numerical method

Ansys Fluent software was utilized with the accuracy of 2 ddp to solve the governing equations. Moreover, a separable solver was applied to solve the governing equations and the SIMPLE algorithm was used to couple the momentum and pressure equations. The PRESTO method was selected to discretize the pressure term. For the momentum and energy equations, the second-order Upwind scheme was used. The convergence criteria used for the continuity and momentum equations and energy equation were set at 10^{-4} and 10^{-6}, respectively. In addition, Table 1 demonstrates the different generated grids to detect the variations in the average Nusselt number within the enclosure through different mesh sizes and aspect ratios.

Experimental and numerical validation was performed to test the present numerical approach. Table 1 presents a comparison between the average Nusselt number in the present study and that in the experimental study conducted by Prasad et al. [43]. Different Rayleigh numbers were in good agreement. In addition, to check the selected model for porous fins, additional validation was carried out by comparing the average Nusselt number, streamlines, and isotherms in the present study with those in the numerical study of Keyhani-Asl et al. [44]. Tables 2 and 3 and Figure 2 present the results that are in good agreement.

Table 1. Comparison of the average Nusselt number for various grids.

$AR = 2:1$		$AR = 3:1$	
Grid size	$\overline{Nu}$	**Grid size**	$\overline{Nu}$
60 × 100	17.41367	**60 × 150**	23.61549
90 × 200	17.49256	**90 × 300**	23.90658
120 × 300	17.51087	**120 × 450**	23.91671
170 × 400	17.51087	**120 × 450**	23.91671

Table 2. Comparison of the average Nusselt number between the present study and that of Prasad et al. [43].

Ra^*	Nu		
	Prasad et al. [43]	**Present work**	**Error**
400	13.73	13.71	0.14%
1000	21.51	21.69	0.82%
5000	50.27	50.13	0.27%

Table 3. Comparison of the average Nusselt number for different Rayleigh and Darcy numbers between the present study and that of Keyhani-Asl et al. [44].

Ra	Da	Keyhani-Asl et al. [44]	Present	Error
10^5	10^{-2}	4.5	4.49	0.22%
10^5	10^{-4}	4.27	4.25	0.46%
10^4	10^{-2}	2.49	2.46	1.2%
10^6	10^{-2}	8.31	8.22	1.08%

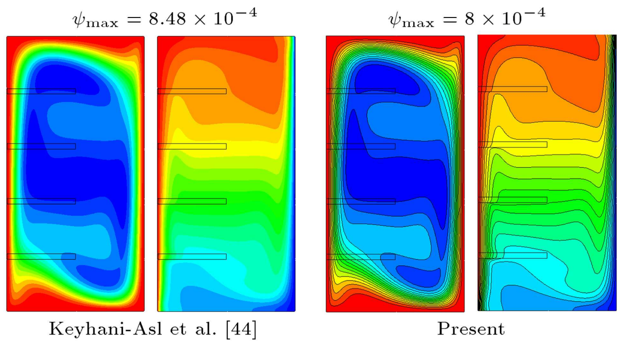


Figure 2. Comparison of the streamlines and isotherms of the present study and that of Keyhani-Asl et al. [44].

5. Results and discussion

5.1. Aspect ratio and fin number

Figure 3 shows the effect of Rayleigh number and number of porous fins installed on the hot wall of the annulus cavity with different aspect ratios on the average Nusselt number. Obviously, increasing the Rayleigh number would result in a higher average Nusselt number in all the considered cases. This phenomenon is the result of the dominant convection to

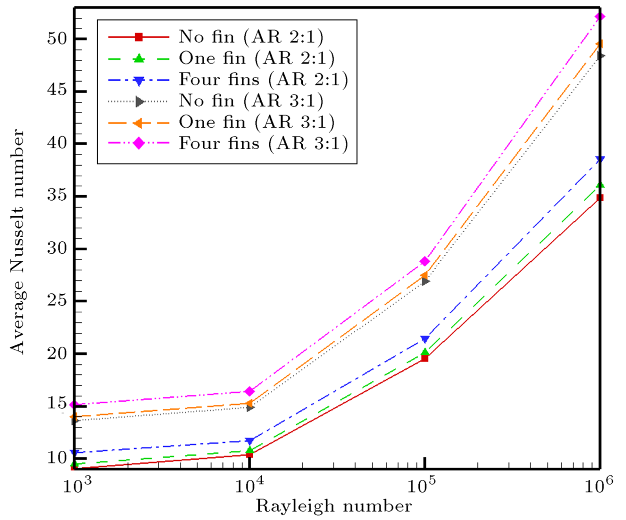


Figure 3. Comparison of the average Nusselt number between annular enclosure with different aspect ratios and different fin numbers at various Rayleigh numbers ($Da = 10^{-2}, Ke = 100, L = 0.5, \varepsilon = 0.9, \varphi = 0$).

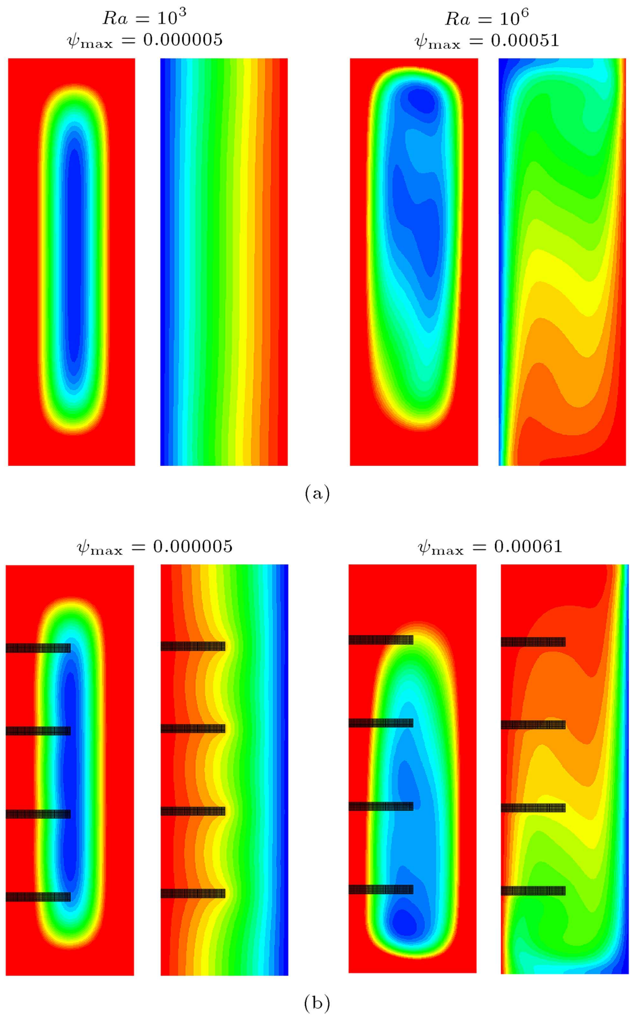


Figure 4. Effect of Rayleigh number on streamlines and isotherms: (a) No fin and (b) 4 fins ($AR =$ 2:1, $Da = 10^{-2}, Ke = 100, L = 0.5, \varepsilon = 0.9, \varphi = 0$).

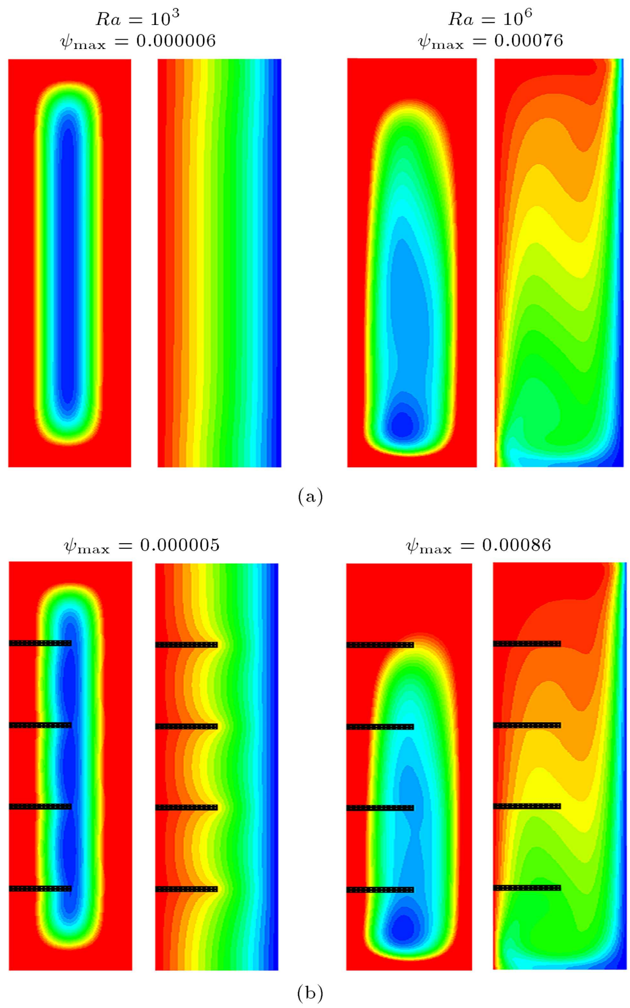


Figure 5. Effect of Rayleigh number on streamlines and isotherms: (a) No fin and (b) 4 fins ($AR = 3:1, Da = 10^{-2}, Ke = 100, L = 0.5, \varepsilon = 0.9, \varphi = 0$).

conduction at higher Rayleigh numbers. In addition, at both of the aspect ratios, use of circular porous fins had a considerable impact on heat transfer enhancement. Figure 3 demonstrates, on average, about 5 and 4% increases in the average Nusselt number in cavities with one porous fin compared to enclosures without any fins with aspect ratios of 2:1 and 3:1, respectively. Moreover, increasing the number of circular porous fins would yield approximately 19 and 14% growth in heat convection. A comparison of the streamlines in Figures 4 and 5 shows the effect of porous fins on the stream function value within the enclosure. As observed, upon increasing the Rayleigh number, the value of the stream function would increase. The shape of the vortices within the enclosure at both aspect ratios was uniform at the lower Rayleigh number; however, as the Rayleigh number increased, the areas affected by vortices with higher stream function value increased. As observed in the mentioned figures, the effect of adding porous fins is more evident at higher Raleigh numbers. As shown in Figures 4(b) and 5(b) at both aspect ratios, installing porous circular fins on the hot wall increases the number of locations captured by vortices with higher stream function values, and the vortices with less significant impact are forced under the lower circular porous fin. Moreover, a considerable increase in the maximum stream function was observed; in other words, increasing the Rayleigh number would overcome the impacts of the convection and vortices. In addition, at $Ra = 10^6$, the maximum stream function experienced a noticeable rise of about 23% compared to the cavity without any fins and aspect ratio of 2:1. The mentioned comparison at an aspect ratio of 3:1 is approximately 15%. Therefore, adding circular porous fins empowers the vortices within the cavity due to the penetration of the fluid into the porous region and prevention of the disturbance of the vortices, which in turn leads to better heat transfer. Figures 4 and 5 also show the effect of porous fins on temperature distribution. As observed in these

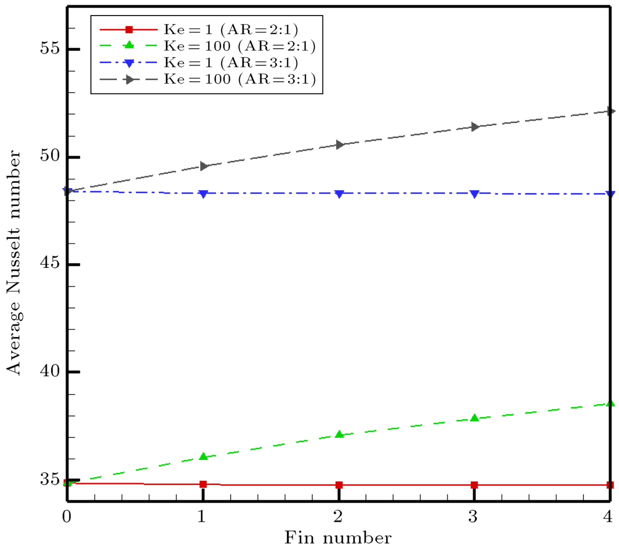


Figure 6. Effect of the circular porous fin number with different thermal conductivity ratios on average Nusselt number ($Da = 10^{-2}$, $Ke = 100$, $Ra = 10^6$, $L = 0.5$, $\varepsilon = 0.9$, $\varphi = 0$).

figure, similar to the distribution of streamlines, the temperature distribution is uniform at lower Rayleigh numbers. The presence of porous fins considerably affected the isotherms, and as the Rayleigh number increased, the thermal boundary layer would totally change. As demonstrated in Figures 4(b) and 5(b), the porous fins considerably interrupt the boundary layer, hence better heat transfer.

Generally, cavities with an aspect ratio of 3:1 outperform those with that of 2:1 in terms of heat transfer. As shown in Figure 3, this phenomenon is linked to the fact that in larger annular cavities, the vortices have the opportunity to provide better recirculation and are active in a larger area with a higher maximum stream function value.

The impact of different numbers of circular porous fins with different thermal conductivity ratios is illustrated in Figure 6. At both of the studied aspect ratios, upon increasing the number of the fins while using fins with lower relative thermal conductivity ratios, the average Nusselt number remains constant. On the other hand, placing more fins with higher relative thermal conductivity on the hot wall results in a dramatic rise in the heat transfer. A comparison between the streamlines presented in Figures 7 and 8 at different relative thermal conductivity ratios reveals that the porous fins with lower thermal conductivity ratios have a negligible impact on the maximum streamline value. At both aspect ratios, upon the addition of more fins, the streamlines remain constant compared to those in the enclosure without any fins. However, at a highly relative thermal conductivity ratio, the streamline values experience an increase of about 15 and 10% on average, compared to those in the cavity without any fins at aspect ratios of 2:1 and 3:1, respectively. Figures 7 and 8 show the considerable impact of highly conductive porous fins on the isotherms and temperature distribution. As mentioned earlier, the isotherms remain the same while using less conductive porous fins; however, the highly conductive porous fins ultimately affect the thermal boundary layer and this effect is more considerable in the upper part of the porous fins, especially in the case of four circular porous fins.

5.2. Darcy number

Figure 9 demonstrates the variations of the Nusselt number as a result of Darcy number enhancement. According to the presented diagram for the annulus with an aspect ratio of 2:1, variation in the average Nusselt number is almost negligible in the range of $Da = 10^{-8}$ to $Da = 10^{-6}$. Further, according to the streamlines in Figures 10 and 11, the permeability of porous fins in the mentioned range of Darcy numbers is quite low, meaning that the utilized porous fins function as solid fins, thus preventing the penetration of the fluid flow and making the flow turn around the fins. Upon increasing the Darcy number over 10^{-6}, the permeability of porous fins would increase, hence an increase in the average Nusselt number. The annulus

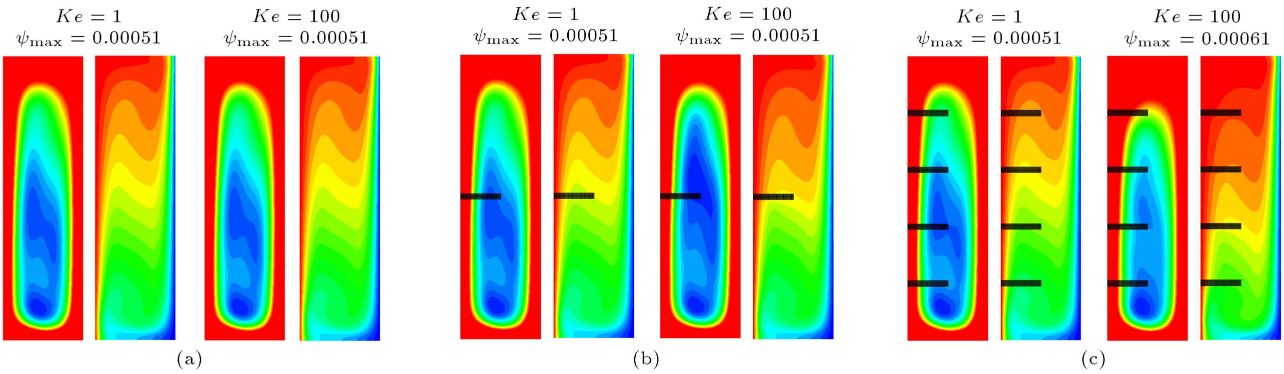


Figure 7. Effect of the number of porous fins with different relative thermal conductivity ratios on streamlines and isotherms: (a) No fin, (b) 1 fin, and (c) 4 fins ($AR = 2{:}1$, $Da = 10^{-2}$, $Ra = 10^6$, $L = 0.5$, $\varepsilon = 0.9$, $\varphi = 0$).

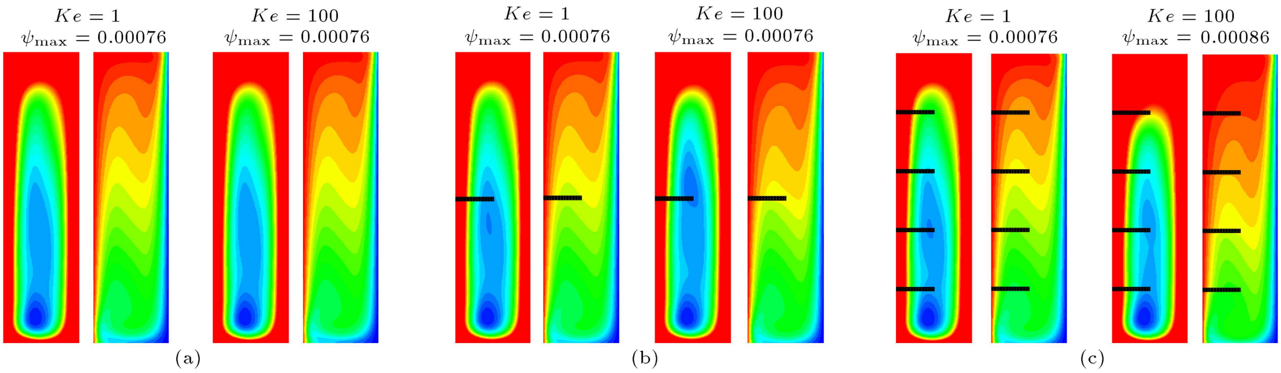


Figure 8. Effect of porous fins with different relative thermal conductivity ratios on streamlines and isotherms: (a) No fin, (b) 1 fin, and (c) 4 fins ($AR = 3:1, Da = 10^{-2}, Ra = 10^6, L = 0.5, \varepsilon = 0.9, \varphi = 0$).

with an aspect ratio of 3:1 outperforms that with an aspect ratio of 2:1 in terms of heat transfer. This improvement is the result of the highly extended height and substantial effect of buoyancy force within the annulus. Impenetrable properties of the porous fins within the cavity with an aspect ratio of 3:1 continued up to $Da = 10^{-5}$. Maximum variation in the Nusselt number occurred in the range of 10^{-5} to 10^6 for both of the annulus aspect ratios. To be specific, at values below this Darcy number range, the installed porous fins act as baffles against the circulation of the flow stream along with the whole domain and force the flow to circulate in smaller regions between fins. Further, the presented contours for temperature distribution in Figures 10 and 11 indicate that the flow with a higher temperature sticks at the top of the annulus, and upon increasing Darcy number, the distribution at high temperatures within the annulus occupies a larger region. According to different studies in the literature [42,44,45], $Da = 10^{-5}$ is a critical value for effective consumption of porous fins.

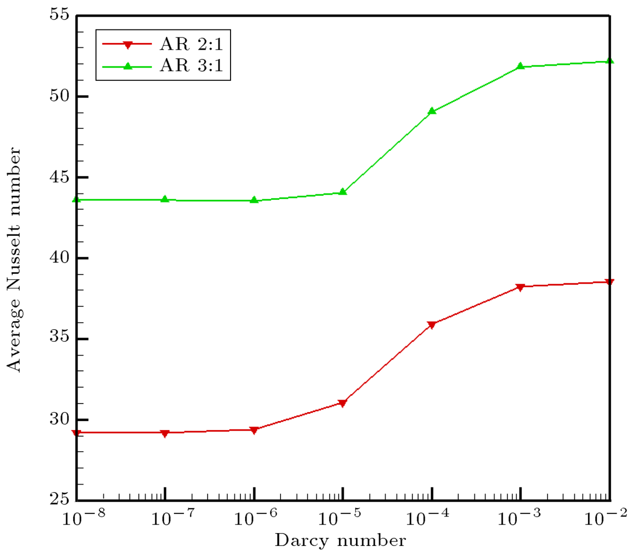


Figure 9. Effect of various Darcy numbers on average Nusselt number for cavities with different aspect ratios ($Ke = 100, Ra = 10^6, L = 0.5, N = 4, \varepsilon = 0.9, \varphi = 0$).

5.3. Relative thermal conductivity ratio

The effect of a change in the relative thermal conductivity on heat transfer enhancement inside the annulus with two different annulus aspect ratios was also evaluated, as shown in Figure 12. Accordingly, it was shown that using a relative thermal conductivity

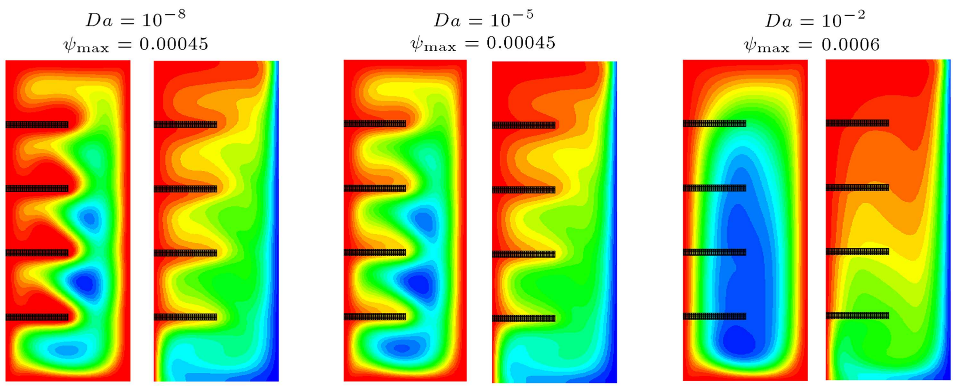


Figure 10. Effect of porous fins with different Darcy numbers on streamlines and isotherms ($AR = 2.1$, $Ke = 100$, $Ra = 10^6$, $L = 0.5$, $N = 4, \varepsilon = 0.9, \varphi = 0$).

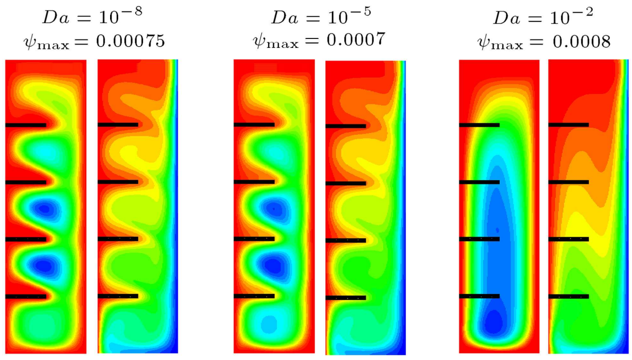
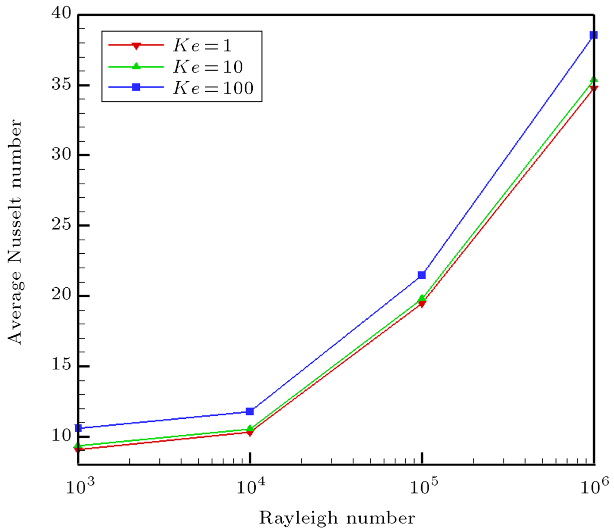
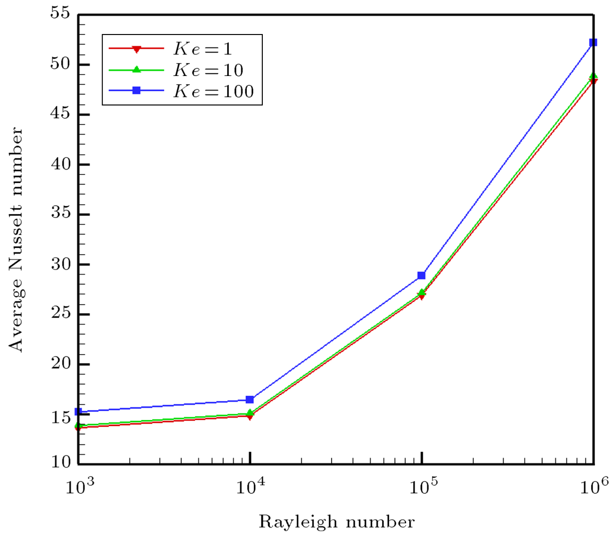


Figure 11. Effect of porous fins with different Darcy numbers on streamlines and isotherms ($AR = 3.1$, $Ke = 100$, $Ra = 10^6$, $L = 0.5$, $N = 4$, $\varepsilon = 0.9$, $\varphi = 0$).

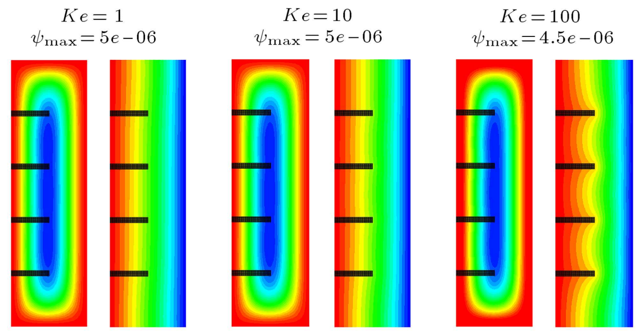


(a)

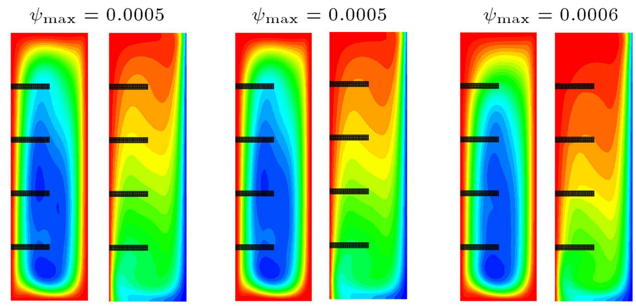


(b)

Figure 12. Effect of porous fins with various relative thermal conductivity ratios on average Nusselt number: (a) $AR = 2:1$ and (b) $AR = 3:1$ ($Da = 10^{-2}$, $L = 0.5$, $N = 4$, $\varepsilon = 0.9$, $\varphi = 0$).



(a)



(b)

Figure 13. Effect of porous fins with various relative thermal conductivity ratios on streamlines and isotherms: (a) $Ra = 10^3$ and (b) $Ra = 10^6$ ($AR = 2:1$, $Da = 10^{-2}$, $L = 0.5$, $N = 4$, $\varepsilon = 0.9$, $\varphi = 0$).

ratio of 100 could raise the average Nusselt number approximately by 12% compared to $Ke = 1$. In addition, the obtained results revealed that the average Nusselt number in an annulus at an aspect ratio of 3:1 had a 39% higher value than that at an aspect ratio of 2:1. The given streamlines in Figures 13 and 14 confirmed that the flow behavior inside the annulus for $Ke = 1$ and $Ke = 10$ was almost identical at both aspect ratios. Values could also approve this similarity in the flow behavior of $Ke = 1$ and $Ke = 10$ of Nusselt number in Figure 12, and the difference between these values was about 2%. Streamlines showed that the formed vortices at $Ra = 10^3$ were smaller and weaker than those at $Ra = 10^6$, and that they were located in a region between porous fins; however, in the case of $Ra = 10^6$, circulation occurs in the lower part of the annulus. According to the contours for temperature in Figures 13 and 14, the affected area with high-temperature gradient increased by enhancing the effective thermal conductivity at both aspect ratios.

5.4. Enclosure inclination angle

To determine the annulus inclination, six different angles of 0, $\pi/6$, $\pi/4$, $\pi/3$, $\pi/3$, and $2\pi/3$ were taken into account. As shown in Figure 15, the heat transfer rate decreased by increasing the inclination angle, mainly due to a reduction in the height of the

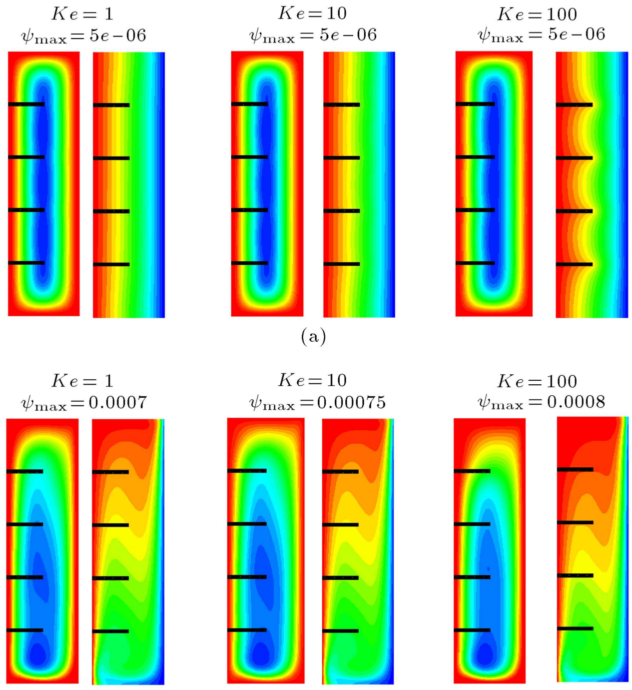


Figure 14. Effect of porous fins with various relative thermal conductivity ratios on streamlines and isotherms: (a) $Ra = 10^3$ and (b) $Ra = 10^6$ ($AR = 3$:1, $Da = 10^{-2}$, $L = 0.5$, $N = 4$, $\varepsilon = 0.9$, $\varphi = 0$).

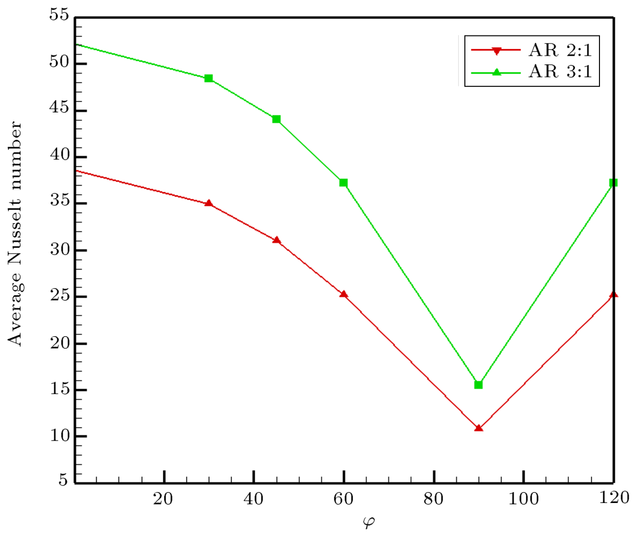


Figure 15. Effect of enclosure inclination angles on average Nusselt number at different aspect ratios ($Da = 10^{-2}$, $Ke = 100$, $Ra = 10^6$, $L = 0.5$, $N = 4$, $\varepsilon = 0.9$).

annulus by increasing its inclination, hence depowering the buoyance force. Nusselt number reached its lowest value at $\varphi = \pi/2$ at both annulus aspect ratios. Based on the streamlines in Figures 16 and 17, it can be concluded that upon increasing the inclination angle, the region in the middle of the annulus where circulation occurs gets smaller; further, at $\varphi = \pi/2$, the primary vortex disappears, and multiple vortices

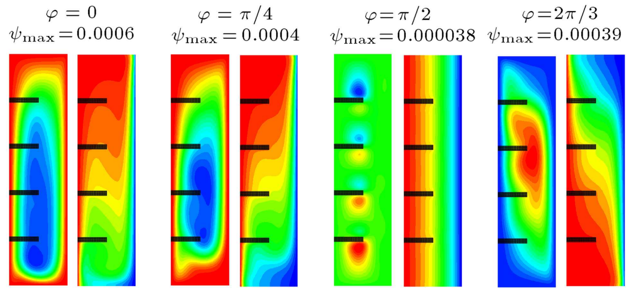


Figure 16. Effect of enclosure inclination angle on streamlines and isotherms ($AR = 2$:1, $Da = 10^{-2}$, $Ke = 100$, $Ra = 10^6$, $L = 0.5$, $N = 4$, $\varepsilon = 0.9$).

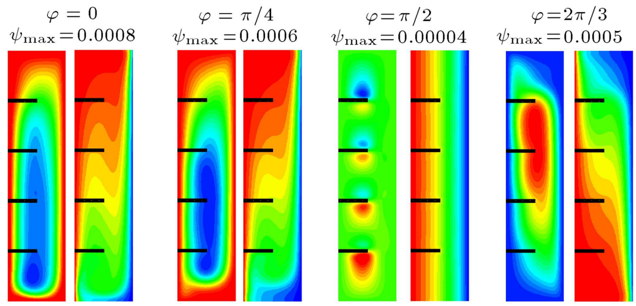


Figure 17. Effect of enclosure inclination angle on streamlines and isotherms ($AR = 3$:1, $Da = 10^{-2}$, $Ke = 100$, $Ra = 10^6$, $L = 0.5$, $N = 4$, $\varepsilon = 0.9$).

with weaker strengths are formed in a region close to the upper parts of the fins. With further increase in the inclination angle, the primary vortex appears again in the middle of the cavity; therefore, the heat transfer rate increases again at both aspect ratios. Figures 16 and 17 show the impact of a change in the inclination angle on temperature distribution. According to these tables, upon increasing the inclination angle at both aspect ratios, the dimensions of the region with higher temperature increase and a totally different temperature distribution can be observed once reaching the inclination angle value of $\pi/2$. A reversed thermal boundary growth appears beyond the inclination angle of $\pi/2$, and isotherm contours exhibit reverse behavior at $\varphi = 2\pi/3$, compared to $\varphi = \pi/6$ at which the fluid located at the top of the annulus is heated.

5.5. Circular porous fin length

Figure 18 shows the effect of the three dimensionless lengths of a single circular porous fins located in the middle of the hot wall of the enclosure at different aspect ratios. As observed, with an increase in the fin length with an aspect ratio of 3:1, the average Nusselt number remains almost constant. The same pattern is observed at an aspect ratio of 2:1 until certain dimensionless length of 0.5 beyond which heat transfer enhancement would occur declines by 6%. Figures 19 and 20 clearly demonstrate the mentioned effects of the fin length on streamlines and isotherms. As shown

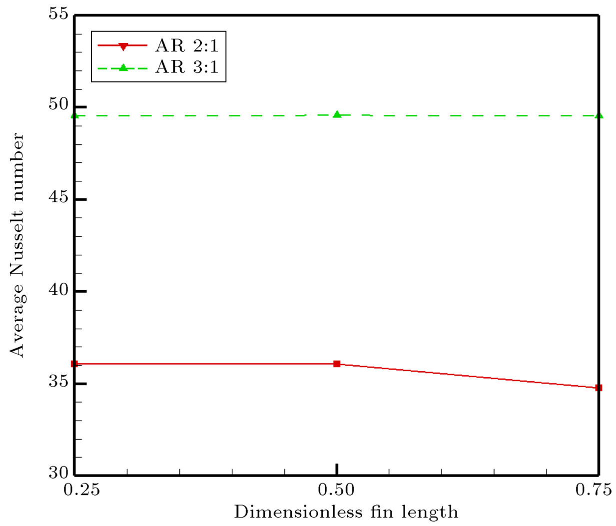


Figure 18. Effect of dimensionless fin length on average Nusselt number at different aspect ratios ($Da = 10^{-2}$, $Ke = 100$, $N = 4$, $RA = 10^6$,$\varepsilon = 0.9, \varphi = 0$).

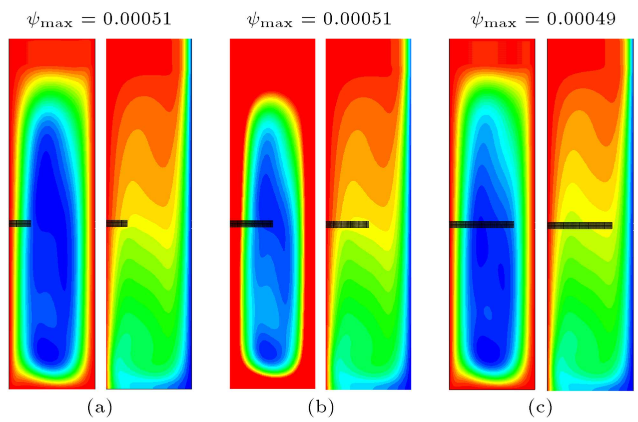


Figure 19. Effect of fin length on streamlines and isotherms: (a) $L = 0.25$, (b) $L = 0.5$, and (c) $L = 0.75$ ($AR = 2:1, Da = 10^{-2}, Ke = 100, N = 4, RA = 10^6, \varepsilon = 0.9, \varphi = 0$).

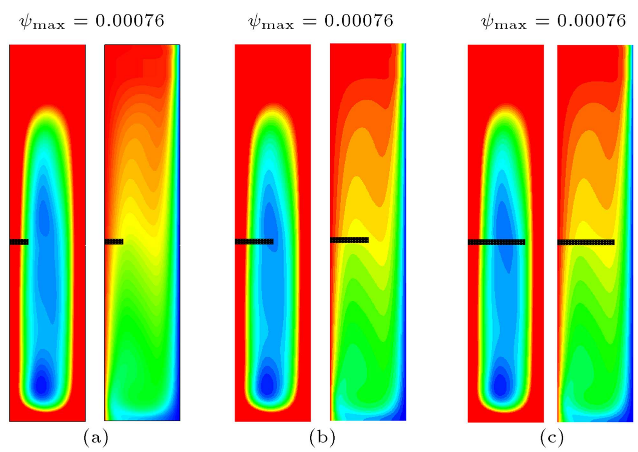


Figure 20. Effect of fin length on streamlines and isotherms: (a) $L = 0.25$, (b) $L = 0.5$, and (c) $L = 0.75$ ($AR =$ 2:1, $Ke = 100$, $Da = 10^{-2}$, $N = 4$, $RA = 10^6$, $\varepsilon = 0.9$, $\varphi = 0$).

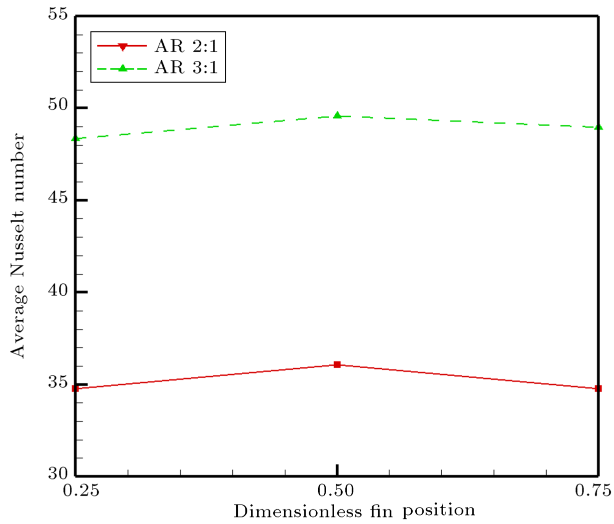


Figure 21. Effect of dimensionless fin position on average Nusselt number at different aspect ratios ($Da = 10^{-2}$, $Ke = 100$, $L = 0.5$, $N = 4$, $Ra = 10^6$, $\varepsilon = 0.9, \varphi = 0$).

in Figure 19, at a smaller aspect ratio of 2:1, the maximum stream function value and active vortices inside the cavity are almost the same; however, with a decrease in the dimensionless length below 0.75, the temperature distribution in the top part of the fins mostly affects the shorter porous fins. On the other hand, considering the effect of different porous fin lengths at a larger aspect ratio of 3:1 on streamlines and isotherms in Figure 20 reveals the same distribution. Therefore, as mentioned earlier, varying porous fin lengths at a larger aspect ratio has a negligible impact on heat transfer enhancement.

5.6. Circular porous fin location

The impact of the three dimensionless positions of a single circular porous fin on the enclosure's hot wall at different aspect ratios is depicted in Figure 21. At both aspect ratios, the optimum installation location is expected to be the middle of the cavity, which is more evident at a smaller aspect ratio of 2:1. Comparison of Figures 22 and 23 indicating streamline and isotherms for different circular porous fin locations reveals that the active vortex within the enclosure is mostly affected by the presence of the fin when located in the middle of the cavity. This phenomenon is observed to be valid at both aspect ratios and better natural convection occurs. Nevertheless, minor impacts are observed on the temperature distribution by changing the location of circular porous fins on the hot wall.

6. Correlation of average Nusselt number

Correlation of average Nusselt number is developed based on various parameters including Rayleigh number, Darcy number, Fin number, and annular cavity

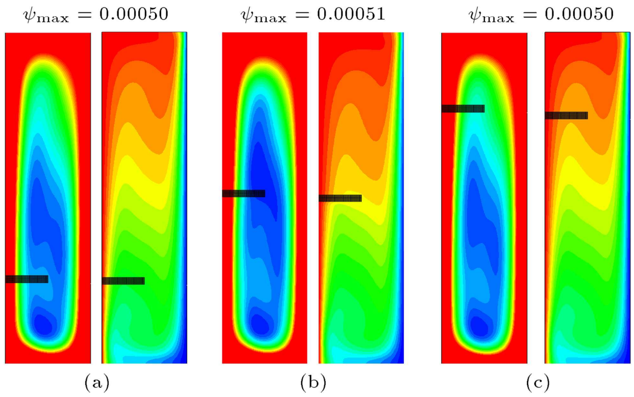


Figure 22. Effect of fin position on streamlines and isotherms: (a) $S = 0.25$, (b) $S = 0.5$, (c) $S = 0.75$ ($AR =$ 2:1, $Da = 10^{-2}$, $Ke = 100$, $L = 0.5$, $N = 4$, $Ra = 10^6$, $\varepsilon = 0.9$, $\varphi = 0$).

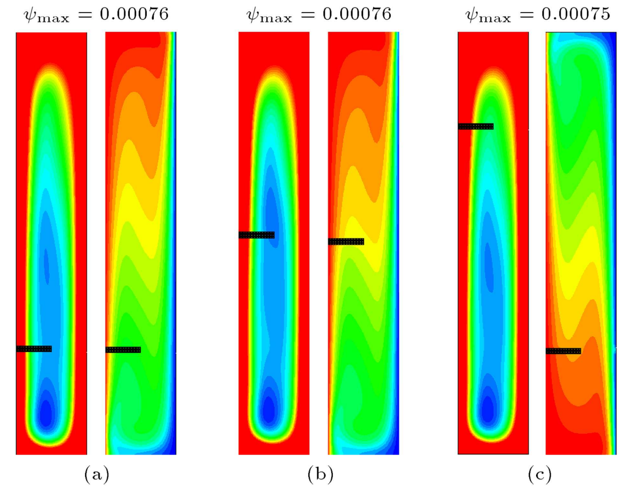


Figure 23. Effect of fin position on streamlines and isotherms: (a) $S = 0.25$, (b) $S = 0.5$, (c) $S = 0.75$ ($AR =$ 3:1, $Da = 10^{-2}$, $Ke = 100$, $L = 0.5$, $N = 4$, $Ra = 10^6$, $\varepsilon = 0.9$, $\varphi = 0$).

aspect ratios as follows:

$$Nu = 0.163 + 0.174\ Ra^{0.316}\ Da^{0.02},$$

$$N^{-0.097}\ AR^{0.901};\ R^2 = 97.65\%,$$

$$10^3 \leq Ra \leq 10^6 \quad \text{(Rayleigh number)},$$

$$10^{-8} \leq Da \leq 10^{-2} \quad \text{(Darcy number)},$$

$$1 \leq N \leq 4 \quad \text{(fin number)},$$

$$2 \leq AR \leq 3 \quad \text{(annular cavity aspect ratio.)} \tag{10}$$

The correlation demonstrates that the most effective parameter in the average Nusselt number is the annular cavity aspect ratio followed by the Rayleigh number, Darcy number, and fin number.

7. Conclusion

In the present investigation, natural convection inside an axisymmetric 2-D annulus with attached circular porous fins on the inner wall was studied numerically. Governing equations were discretized using the Finite Volume Method (FVM) based on the second-order upwind scheme. Effects of various parameters such as Rayleigh number, Darcy number, thermal conductivity, annulus inclination angle, annulus aspect ratio, the position, number, and length of fins were studied, and a summary of the achieved results is given below:

1. Increasing the Rayleigh number would enhance the average Nusselt number within the annulus due to dominant convection to conduction at higher Rayleigh numbers. It was also illustrated that utilizing low relative thermal conductivity nullified the effect of the increasing number of porous fins on heat transfer enhancement;
2. By increasing Darcy number to a particular value at both aspect ratios, the average Nusselt number remained almost constant and the utilized fins acted as solid fins, meaning that they would not allow fluid flow to penetrate through them. Impenetrable properties of porous fins at an aspect ratio of 3:1 continued up to $Da = 10^{-5}$. The maximum variation in Nusselt number occurred at 10^{-5} to 10^{-6} at both annulus aspect ratios;
3. The results indicated that the average Nusselt number had no noticeable change between $Ke = 1$ and $Ke = 10$; however, by utilizing porous fins with $Ke = 100$, the average Nusslet number increased approximately by 12% compared to $Ke = 1$. Also, it was understood that an annulus with an aspect ratio of 3:1 had 39% greater heat transfer than that at an aspect ratio of 2:1;
4. Results indicated that by increasing the annulus inclination angle, heat transfer declined due to the reduction of the annulus's height and depowering of buoyant force. At $\varphi = 90°$, that annulus was horizontal and the average Nusselt number was at its lowest value, and the primary vortex within the annulus disappeared;
5. Achieved results demonstrated that installing longer fins had no role in enhancing heat transfer inside the annulus than the shorter ones. Also, results illustrated that the highest heat transfer occurred when the fin was installed in the middle of the annulus.

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