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HEAT TRANSFER BY NATURAL CONVECTION FROM A HEATED SQUARE INNER CYLINDER TO ITS ELLIPTICAL OUTER ENCLOSURE UTILIZING NANOFLUIDS

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In this paper a numerical study of natural convection of stationary laminar heat transfers in a horizontal ring between a heated square inner cylinder and a cold elliptical outer cylinder is presented. A Cu-water nanofluid flows through this annular space. Different values of the Rayleigh number and volume fraction of nanoparticles are studied. The system of equations governing the problem was solved numerically by the fluent calculation code based on the finite volume method and on the Boussinesq approximation. The interior and exterior surfaces are kept at constant temperature. The study is carried out for Rayleigh numbers ranging from 10^3 to 10^5. We have studied the effects of different Rayleigh numbers and volume fraction of nanoparticles on natural convection. The results are presented as isotherms, isocurrents, and local and mean Nusselt numbers. The aim of this study is to study the influence of the thermal Rayleigh number and volume fraction of nanoparticles on the heat transfer rate.

Key words: natural convection, Rayleigh numbers, nanofluid, volume fraction, square elliptical.

1. Introduction

Natural convection of nanofluids has attracted attention of researchers due to its occurrence in nature and applications in the fields of engineering such as solar basins, room ventilation solar basins, and insulation of the reactor. Improved heat transfer through convection is the main object of several tasks. For this reason, a large number of researchers have carried out a large number of numerical and experimental tests. Leongand and Lai [1], studied the flow induced by buoyancy inside a porous cavity. A parametric investigation was carried out to study the influence of the Darcy number, relative thermal conductivity and the Rayleigh number on heat transfer. Zhou *et al*. [2] examined natural convection in a rectangular layer by the method of Boltzmann.

The results show that the solid size portion of nanoparticles and Rayleigh number affect the improvement of heat transfer of nanofluids. Al-Asad *et al*. [3] studied the influence of a rectangular heat source in a closed space on natural convection. Hossain *et al*. [4] investigated an opened squarish cavity

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containing a centrally heated circular cylinder and heat transfer by magnetohydrodynamic free convection (MHD) flow. The studies are carried out for various values of the Hartmann number (Ha) and the Rayleigh number (Ra). Hossain *et al.* [5] investigated MHD magnetohydrodynamic natural convection within a trapezoidal cavity for various heated triangular obstacle aspect ratios $\left(1/3\ AR \le 1\right)$. The top wall was thermally insulated and both vertical walls were maintained at a cold temperature. Ammar [6] studied heat transfer by natural convection betwixt a heated interior circular cylinder housed in a cooled oval vessel filled by a copper nanofluid and water by interior heat generation/absorption with a horizontal magnetic field. The system was solved digitally without dimension with the finite element method. Hadiand Salah [7] studied numerically the flow of a nanofluid and heat transfer by natural convection caused by the temperature inequality betwixt a heated internal cylinder with various scanning devices (i.e. triangular, circular, oval, and rhombic) and a cold exterior square container. Ellahi *et al.*[8] studied the effects of heat transfer by natural convection into a cold outer circular container containing an inside heated elliptical cylinder. The liquid in the cylinder was a water-copper nanofluid. Abdelraheem [9] performed a digital study on natural convection from a hot square shape into a round vessel filled with a nanofluid. Abeer[10] performed numerical calculations for the normal load induced by the variation in temperature within a hot inside square cylinder and a cold outside square cylinder. The solution was in a two-dimensional system for an unstable normal load, using the submerged limit method (IBM)(Abdallaoui [11]). The study was carried out on silver water and pure water numerically using the Boltzmann lattice method for normal convection for an eccentric triangular cylinder located within a square cylinder, where the Rayleigh number did not exceed 10^7, while the volume fraction of nanoparticles was confined between 0 and 0.1. Dogonch [12] studied numerically the effects of nanoparticle form on natural convection within fluid-saturated porous ring developed betwixt the elliptical cylinder and the square vessel. Matin *et al.* [13] studied numerically the combined heat transfer and convection concerning Al_2O_3-water nanofluid inside a horizontal eccentric annulus. Bouras *et al.* [14-17] studied numerically natural convection of stationary laminar heat transfers in an annular space containing a Newtonian liquid. He studied the changes in the Nusselt number and temperature for different Rayleigh numbers. He solved the system of equations that govern the problem digitally by fluid arithmetic symbol based on the specific volume method and using the Businesque approximation. Taloub *et al.* [18] studied natural convection of stationary laminar heat transfers in a semi-elliptical inclined cavity.

In this investigation, we studied natural convection of stationary laminar heat transfers into a horizontal ring within a heated square interior cylinder and a cool elliptical exterior cylinder. A water – copper nanofluid flow is studied in a laminar and permanent.

We investigated the influence of the nanofluid fraction and Rayleigh number on the flow structure, current function, and temperature variations as well as the heat transfer rate, the latter being defined by the local and mean Nusselt numbers.

2. Formulation of the problem

The geometry of the considered problem is presented in Fig.1. It is a square cylinder enclosure, placed within an elliptical cylinder fence where the eccentricity equals 0.7. The wall of the elliptical outer fence was kept at constant temperature T_F, and the square cylinder characterized by side length L, is kept at a constant elevated temperature Tc.

The nanofluidthermophysical properties are constant, except for the variation in density, which is estimated by the Boussinesq approximation. The viscous dissipation is inconsiderable, the flow is two-dimensional, stationary, and laminar.

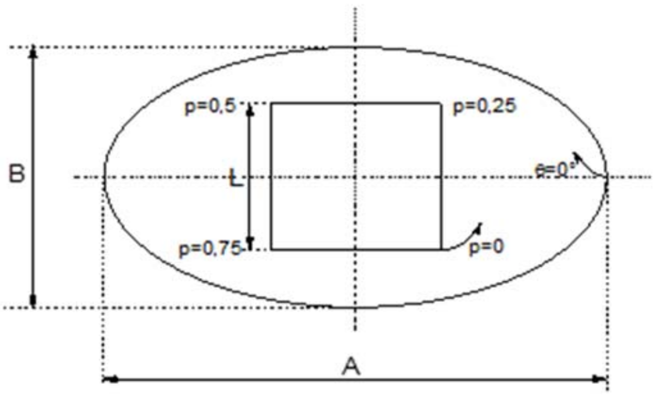


Fig.1. Schematic presentation of the physical model.

2.1. Mathematical model

The governing equations inside the square fence with a heated elliptical cylinder are described by the energy, and Navier-Stokes equations. The governing equations are transformed to dimensionless forms by the following non-dimensional variables:

$$X = x/L, \quad Y = y/L, \quad U = \frac{u}{\left(\frac{\alpha_f}{L}\right)}, \quad V = \frac{v}{\left(\frac{\alpha_f}{L}\right)}, \quad \theta = \frac{T - T_c}{T_h - T_c},$$

$$P = \frac{p}{\rho\left(\frac{\alpha_f}{L}\right)^2}, \quad \Pr = \frac{\vartheta_f}{\alpha_f}, \quad Ra_t = \frac{g\beta_f\left(T_h - T_c\right)}{\alpha_f \vartheta_f} D_h^3.$$

By introducing the dimensionless quantities into the conservation equations of mass, motion and energy, we obtain, respectively:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0, \tag{2.1}$$

$$U\frac{\partial u}{\partial x} + V\frac{\partial v}{\partial y} = -\frac{\partial p}{\partial x} + \frac{\mu_{nf}}{\rho_{nf}\alpha_{nf}}\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial x^2}\right), \tag{2.2}$$

$$U\frac{\partial v}{\partial x} + V\frac{\partial u}{\partial y} = -\frac{\partial p}{\partial y} + \frac{\mu_{nf}}{\rho_{nf}\alpha_{nf}}\left(\frac{\partial^2 v}{\partial\ x^2} + \frac{\partial^2 v}{\partial\ y^2}\right) + \frac{(\rho\beta)_{nf}}{\rho_{nf}\beta_f} Ra_t \Pr\theta, \tag{2.3}$$

$$U\frac{\partial \theta}{\partial x} + V\frac{\partial \theta}{\partial y} = \frac{\alpha_{nf}}{\alpha_f}\left(\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2}\right). \tag{2.4}$$

2.2. Calculation of the Nusselt number

The local and average Nusselt number for the outer and inner cylinders are calculated as follows [19, 20]:

$$Nu_h = -\frac{k_{nf}}{k_f}\left(\frac{\partial T}{\partial y}\right)_{y=0}, \tag{2.5}$$

$$Nu_c = -\frac{k_{nf}}{k_f}\left(\frac{\partial T}{\partial y}\right)_{y=1}, \tag{2.6}$$

$$\overline{Nu_{avg}} = -\frac{1}{D_h}\int_0^{Dh} Nu_x dx. \tag{2.7}$$

2.3. Nanofluid properties

The thermophysical characteristics of nanoparticles (copper) and liquid (water) are provided in Tab.1. The properties of the nanofluids can be obtained from the properties of the nanoparticles, and the liquid. The density and heat capacity of the nanofluids are evaluated based on the recommendations of Ramiar and Ranjbar [21]

Table 1.Thermophysical properties of the base fluid and Cu nanoparticles [22]

	$C_p\left(Jkg^{-1}K^{-1}\right)$	$\rho\left(Kgm^{-3}\right)$	$\mu(Ps.s)$	$K\left(Wm^{-1}K^{-1}\right)$	$\beta\left(K^{-1}\right)$
Copper(Cu)	385	8933	-	401	1.67×10^{-5}
Pure Water	4179	997.1	0.000891	0.613	21×10^{-5}

The thermal diffusivity, the heat capacitance, the thermal expansion coefficient, and the effective dynamic viscosity of the nanofluid and the effective density determined through Brinkman [21] are as follows:

$$(\rho\beta)_{nf} = \varphi(\rho\beta)_p - (\varphi-1)(\rho\beta)_f, \tag{2.8}$$

$$(\rho Cp)_{nf} = -(\varphi-1)(\rho Cp)_f + \varphi(\rho Cp)_p, \tag{2.9}$$

$$(\alpha)_{nf} = \frac{k_{nf}}{(\rho Cp)_{nf}}, \tag{2.10}$$

$$(\rho)_{nf} = \varphi\rho_p + (1-\varphi)\rho_f. \tag{2.11}$$

The effective dynamic viscosity of the nanofluid determined by Brinkman [21] is

$$(\mu)_{nf} = \frac{\mu_f}{(1-\varphi)^{2.5}_{\ n}}. \tag{2.12}$$

With respect to effective thermal conductivity of the nanofluid, Maxwell [21] presented the following model for low-density mixtures with micro-sized spherical particles,

$$K_{nf} = K_f \frac{\left(k_p + 2k_f\right) - 2\varphi\left(k_f - k_p\right)}{\left(k_p + 2k_f\right) + \varphi\left(k_f - k_p\right)}. \tag{2.13}$$

2.4. Boundary condition

For solving the system of equations we got earlier we must determine the boundary conditions. The temperature of the square cylinder is constant and equal to T_F. The hot part of the elliptical cylinder has a constant temperature equal to T_C.
These different boundary conditions in dimensional form can be summarized as follows.
The initial conditions are:

$$V = U = 0, \tag{2.14}$$

$$\theta(x, y) = 0. \tag{2.15}$$

Square cylinder

$$V = U = 0, \tag{2.16}$$

$$\theta(x, y) = 1. \tag{2.17}$$

Elliptical cylinder

$$V = U = 0, \tag{2.18}$$

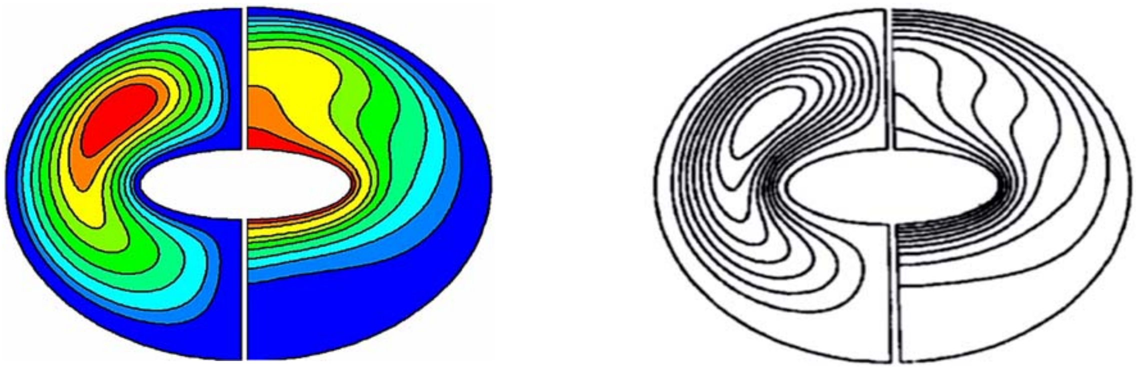
$$\theta(x, y) = 0. \tag{2.19}$$

2.5. Validation of results

To verify the correctness of the numerical results, digital validation code was made by taking into account certain numerical studies available in the literature. We study the influence of various values of the Rayleigh number on the isotherms, isocurrents and local and mean Nusselt numbers. The results obtained by Elshamy [23] have been selected for the validation of that research. Validations have been presented under forms of isocurrents and isotherms concerning a various Rayleigh number (Fig.1a). In addition, the local Nusselt number has been compared to the reference [23] for different Rayleigh numbers between 10^4 and 2×10^5 as shown in the Fig.1b.

The case for the two eccentricities which correspond to 0.9 for the internal wall and 0.4 for the external wall and for two Rayleigh numbers is shown in Fig.2b. By comparing the results obtained with the results presented in [23], we find that there is an acceptable agreement.

a)



Present travail Elshamy [23]

Fig.1a. Qualitative comparison of isocurrents and isotherms for $Ra = 10^4$ with reference [23].

b)

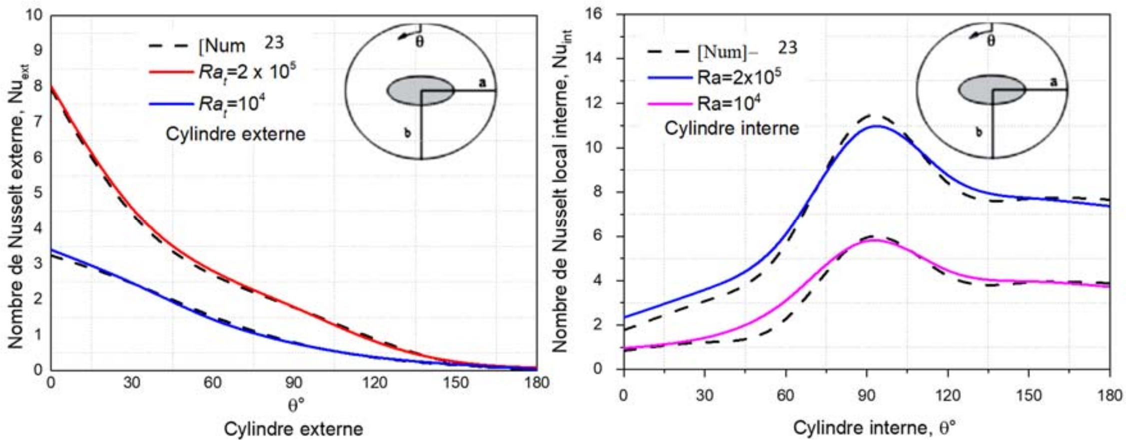


Fig.2b. Quantitative comparison of local Nusselt number values on the exterior and interior walls with reference [5].

2.6. Meshing choice

In this study several meshes were used arbitrarily for $Ra_T = 10^5$ to see their effects on the results (Tab.1). The 130×230 mesh was accepted in all simulations, because the relative error in this mesh is less than *0.1272*.

Table 1. Effect of the mesh on the mean Nusselt number for $Ra_T = 10^5$.

NN × NI	80×180	100×200	110×210	120×210	130×230	140×240
Nu (mean)	5.883	5.650	5.582	5.496	5.432	5.425
Relative error (%)		4.142	1.215	1.570	1.171	0.127

3. Results and discussion

Figure 3 present the current lines and isotherms for various values of the Rayleigh number. It is noted that those isocurrents and isotherms are symmetric compared to the imaginary vertical midplane. The flow regime is single-cell, within the left side from the symmetry plane, the flow rotates counterclockwise and within the right side, in the opposite direction, the fluid near the heated interior wall rises under the influence of Archimedes' thrust, then falls again after its temperature decreases. From $Ra = 10^3$ the current is getting organized in two main cells that rotate well leisurely to the other direction. The isotherms are closed and concentric curves which follow the profiles of the walls fairly well, so convection is weak. The temperature values, in this case, are simply decreasing from the square cylinder to the elliptical cylinder. We can say that convection is relatively weak because most of the heat transfer takes place by conduction at the level of the internal cylinder. Also, for the values of the current function, its value is very small. However, for $Ra = 10^4$, the isothermal lines transform symmetrically around the vertical axis and change appreciably, and in addition, we note that the values of the current function shown in the same figure also increase appreciably, Through the shape of the isometric lines we notice that there is an increase in thermal convection, but this increase remains relatively small.

For $Ra = 10^5$ and $Ra = 5\times10^5$ the isotherms switch and finally hold the form of a mushroom. The temperature apportionment decreased from the square wall to the cylindrical wall. The sense of deformity of the isotherms corresponds to the sense of rotation of the isocourants. We also note that there is a significant increase in the values of the current functions, which means that convection is very important.

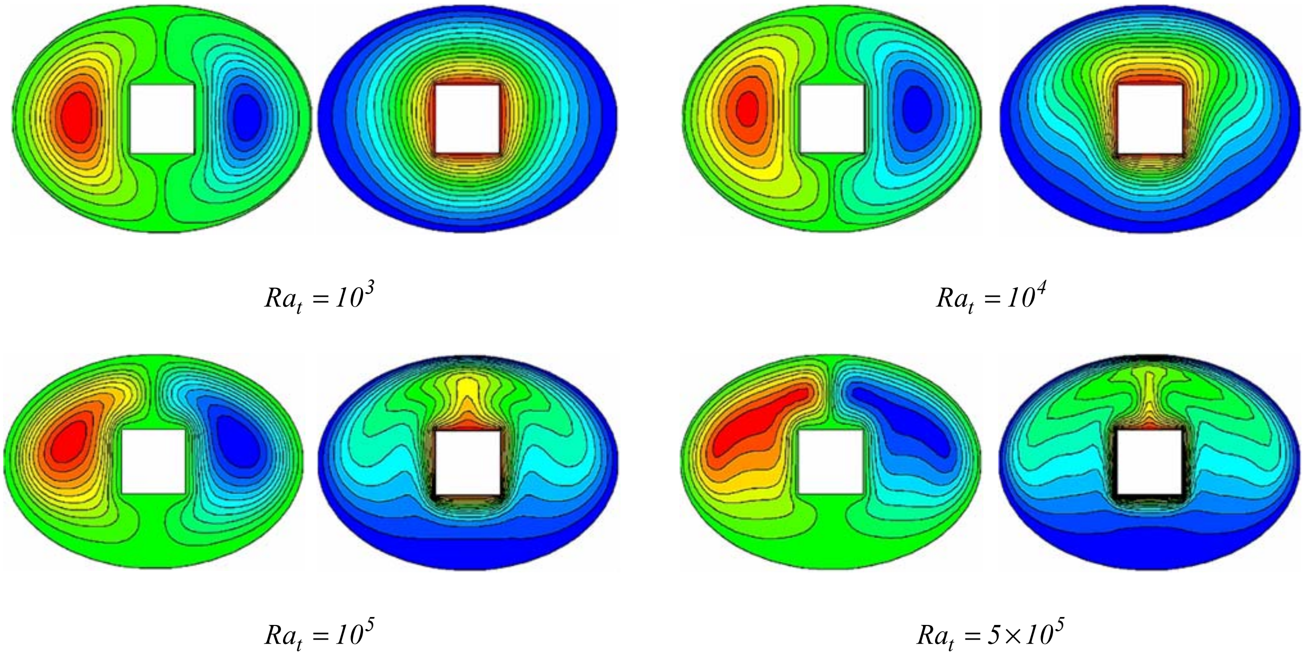


Fig.3. The effect of thermal Rayleigh number on isotherms and streamlines at $Ø = 0$.

We observed that the Nusselt number, which is insensitive to the volume fraction for low Rayleigh numbers, becomes strongly affected when the Rayleigh number is greater than 10^4.

But we see that whatever the Rayleigh number, the increase in the value of the volume fraction always leads to an increase in the Nusselt number, but we see that whatever the Rayleigh number, the increase in the value of the volume fraction always leads to an increase in the Nusselt number, which leads to a marked improvement in the convective transfer rate.

For a more useful comparison, Fig.4 shows the change in mean Nusselt numbers with the change in volume fraction of nanoparticles for various Rayleigh numbers. It is observed that the mean Nusselt number increases with the increasing volume fraction of nanoparticles. So, the heat transfer coefficient changes with the change in the value of the Rayleigh number and the volumetric fraction of nanoparticles (Fig.5).

This result is consistent with the thermal profiles presented in Fig.5. It can as well be noted that the extreme values are affected by the presence of nanoparticles.
In fact, the existence of nanoparticles causes an accumulation of isotherms near the hot wall which means an improvement in the rate of heat transfer (Fig.6).

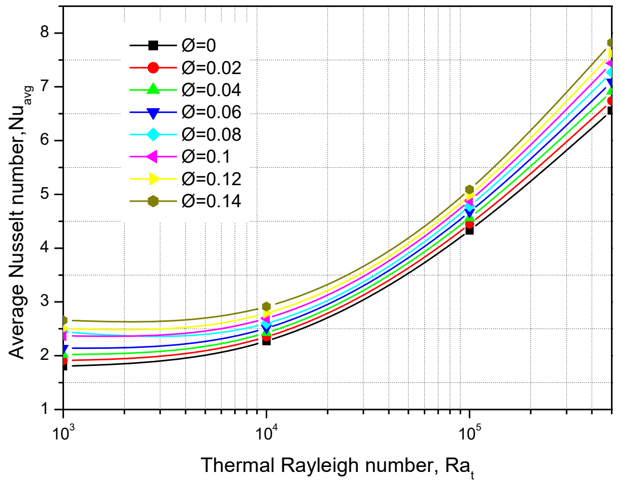


Fig.4.Effect of the thermal Rayleigh number on the mean Nusselt number for different values of volume fractions of nanoparticles.

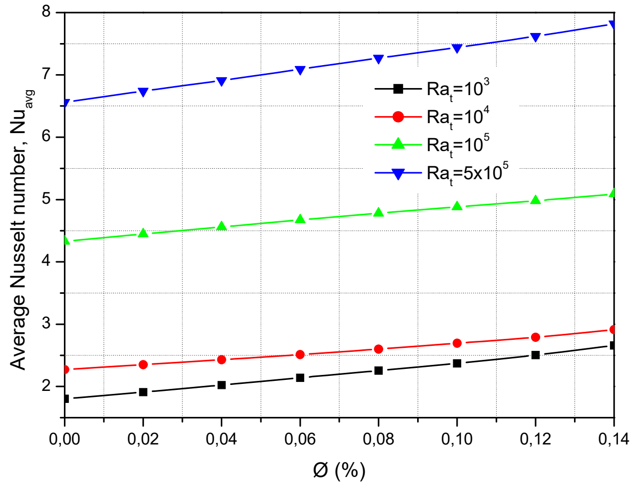


Fig.5. Effect of volume fractions on the mean Nusselt number for different thermal Rayleigh number values.

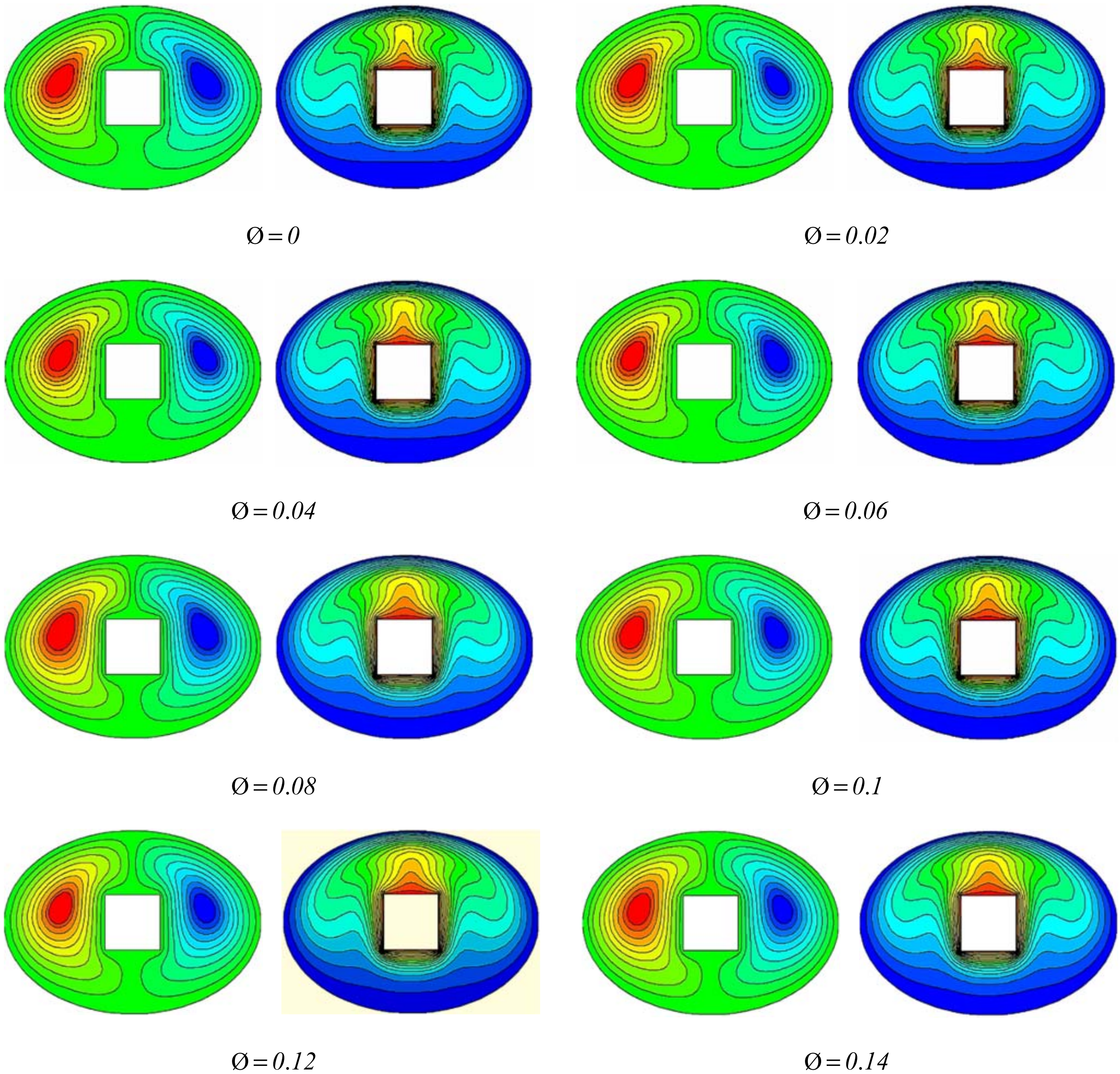


Fig.6. Streamlines and isotherms for different volume concentrations.

In Fig.7, we have illustrated the Nusselt variation with the variation of the Rayleigh number for various solid volume concentrations. From the figure, we can see that as the number of *Ra* augments, the average Nusselt number augments. This results in an amplification of the heat transfer rate, this increase is greater for the highest *Ra* numbers. We also note that the Nusselt values for the nanofluid are raised if we compare them to those for pure water. This is because of the increased conductivity of the nanofluid compared to that of pure water, which increases heat transfer by diffusion (conduction) through the bottom wall.

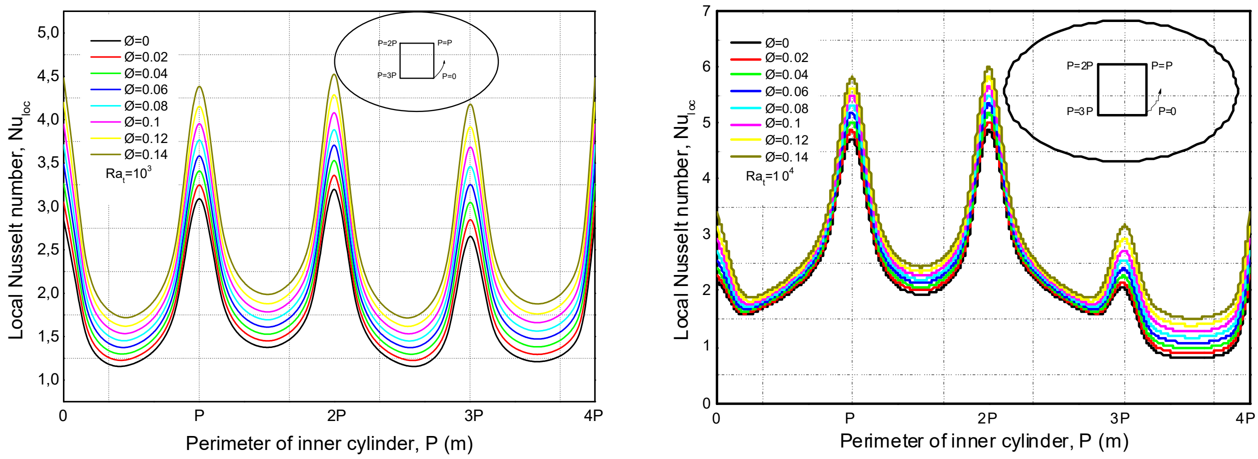


Fig.7. The values of the local Nusselt number on the square wall.

Figure 8 illustrates the distribution of the local Nusselt number about the elliptical wall of the ring, this distribution allowed us to observe that the rise in the volume fraction creates a build-up concerning natural convection which indicates a rise in the values of local Nusselt numbers.

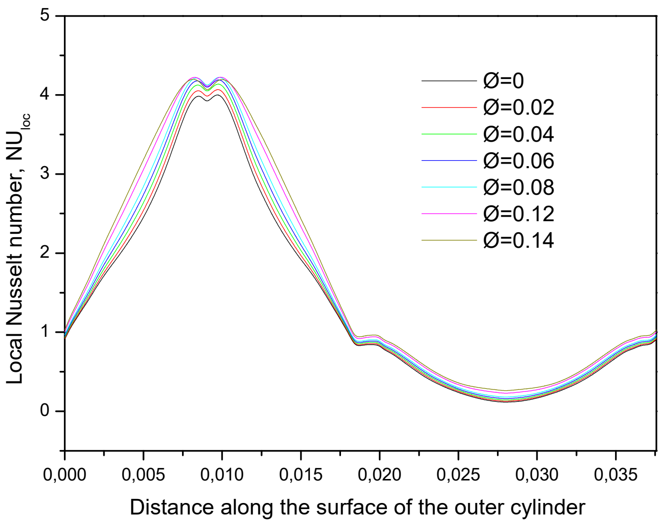


Fig.8. The values of the local Nusselt number on the cylindrical wall.

We also observe in this figure, that Nu is utmost in the top part at $\theta = 90^{\circ}$ and minimal in the lower part of the enclosure at $\theta = 270^{\circ}$ where the fluid is practically immobile.

4. Conclusion

Natural convection of stationary laminar heat transfers into a horizontal ring betwixt a heated square interior cylinder and a cool elliptical exterior cylinder was investigated numerically. This annular space is filled with a Cu-water nanofluid. The obtained results showed:

- The Nusselt values of the nanofluid are higher than those for pure water.

- The Nusselt number increased together with raising volume fraction φ, therefore the addition of nanosolid brings a significant improvement in the heat transfer coefficient.
- The heat transfer inside the ring is primarily controlled by the conduction process when the thermal Rayleigh number is low. On the other hand, when the thermal Rayleigh number is high, the mode of heat transfer by convection becomes predominating.
- The effect of the nanofluid on convection is particularly evident at a high Rayleigh number
- Our study is related to the study of natural convection within rectangular and elliptical cavities and requires further investigation:
- The impact of other main control parameters, particularly the Prandtl number (Pr)
- An experimental and numerical three-dimensional study of this phenomenon.
- The influence of stationarity and the flow regime.

Nomenclature

A – major axe of the elliptic cylinder, $[m]$

a – thermal diffusivity, $[m/s^2]$

B – minor axe of the elliptic cylinder, $[m]$

Cp – specific heat at constant pressure $\left[\frac{J}{kg\,K}\right]$

e – eccentricities of ellipse

g – gravitational acceleration $[m/s^2]$

k – thermal conductivity $\left[\frac{W}{m\,K}\right]$

Nu – local Nusselt number, $[-]$

Nua – average Nusselt number, $[-]$

P – pressure, $[N/m^2]$

Pr – Prandtl number, $[-]$

P – length or width of the square enclosure, $[m]$

ρ – local fluid density, $[kg/m^3]$

ρ_0 – characteristic density at reference temperature, $[kg/m^3]$

Ra – Rayleigh number, $[-]$

T – local temperature, $[K]$

T_0 – reference temperature, $[K]$

Tc, Th – cold and hot wall temperature, $[K]$

ΔT – temperature difference, $[K]$

U – velocity component in the x- direction, $[m/s]$

V – velocity component in the y- direction, $[m/s]$

x – Cartesian coordinate in the horizontal direction, $[m]$

v – Cartesian coordinate in the vertical direction, $[m]$

Ø – nanoparticle volume fraction

β – volumetric coefficient of thermal expansion, $[K^{-1}]$

Subscripts

c – cold
f – fluid
h – hot
nf – nanofluid
p – solid particles

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