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Numerical study of natural convection in a square cavity with partitions utilizing Cu-Water nanofluid

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ABSTRACT: In the present study, natural convection heat transfer in a partitioned square cavity utilizing nanofluids is studied. The vertical left and right walls are considered cold walls, and the partitions assumed to be hot. The nanoparticules used in this study is Cu with the volume fraction of 10%. The influence of different parameters such as Rayleigh number (Ra=10^3, 10^4 and 10^5), distance from the cold wall of the partition (d=0.3H-0.7H) are studied. According to the results, Rayleigh number and location of the partition are important factors that extremely affect the streamlines and isotherms. In this case we founded that the increase in Rayleigh number, increases the average Nusselt number for all the nanofluid volume fractions. The increment in average Nusselt number is strongly dependent on the location of the partition.

KEYWORDS: Nanofluid, Heat transfer, Natural convection, Partition, Rayleigh Number.

I. INTRODUCTION

The heat transfer is a process of great importance in the field of industry and technology. Although it manifests itself in various forms (radiation, conduction and convection), the latter is the most referred in clearly specified areas such as cooling of processors and electronic components, radiators and heat exchangers for industrial processes, etc. The improved heat transfer by convection is the main subject of several studies, and to do so, many researchers conducted a variety of numerical and experimental tests of the description of the phenomena manager convection, the Indeed the nature of the systems in which it takes place (especially geometry), and properties of the fluids involved (physic-chemical properties). Chronologically, although improvement ideas have mainly affected the geometry of the systems and the physicochemical nature of convective environments, work only affected the macroscopic order or sometimes microscopic process. But with the emergence and rapid development of nano-sciences and nanotechnologies in the second half of the 20th century, convection took a large share of this new wealth, and took another improvement aspect: it is the nano level material convective environment that recent work has been concentrated. The nanofluids are then one of the fruits of such wealth. Endowed with particular and interesting physicochemical properties such as their high thermal conductivity, the nanofluids provide a coefficient of thermal beat transfer by the other heat transfer. Studies in this new direction have provided an extensive bibliography, but very varied, although most are quite positive. Finally, finely understand the behavior of nanofluids, and provide universal formulas or correlations that describe, make possible to integrate them in different kinds of heat exchangers in various technological and industrial sectors, always for better efficiency.

II. RELATED WORK

The natural convection problem in a differentially heated square cavity is numerically simulated by Khanafer et al [1] considering the dispersion effect. In their methodology, the dispersion constant ‘‘C’’ is to be determined by experimental data observation. Lee et al. [2], measured the thermal conductivity of water and Cu–water nanofluids,

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and indicated that the thermal conductivity of nanofluids increases with solid volume fraction. He concluded that any new models of nanofluids thermal conductivity should contain the effect of surface area and structure dependent behavior as well as the size effect. Xie et al. [3] added spherical and cylindrical shaped nano sized SiC particules to water and ethylene glycol, separately and found that cylindrical nanoparticles increase thermal conductivity more than spherical ones. The dependence of thermal conductivity of nanoparticles–fluid mixture was estimated by Xie et al. [4]. A lot of researches have been conducted to study the performance of the nanofluids. Polidori et al. [5] investigated the natural convection heat transfer of Newtonian nanofluids (γ -Al_2O_3/water) in a laminar external boundary-layer from the integral formalism approach. Khanafer et al.[6],studied heat transfer enhancement in a two-dimensional enclosure utilizing nanofluids for various pertinent parameters. They found that the suspended nanoparticles substantially increase the heat transfer rate at any given Grashof number. In addition, the results illustrated that the nanofluid heat transfer rate increases with an increase in the nanoparticles volume fraction. The thermal characteristics of natural convection in a rectangular cavity heated from below with Al_2O_3nanofluids with Jang and Choi's model for predicting the effective thermal conductivity of nanofluids and various models for the effective viscosity was investigated by Hwang et al. [7]. The results showed that water-based Al_2O_3nanofluids is more stable than base fluid in a rectangular cavity heated from below as the volume fraction of nanoparticles increases, the size of nanoparticles decreases, or the average temperature of nanofluids increases. Abu-nada and chamkha [8] studied the natural convection heat transfer characteristics in a differentially heated enclosure filled with a CuO-EG-Water nanofluid for different variable thermal conductivity and variable viscosity models. According to the results, the effects, the viscosity models are predicted to be more predominant on the behavior of the average Nusselt number than the influence of the thermal conductivity models. In another study, Abu-nada and Oztop [9] numerically analyzed the effect of the inclination angle on the natural convection heat transfer and fluid flow in a two-dimensional enclosure filled with Cu nanofluid. They showed that at high Rayleigh numbers, the percentage of the heat transfer enhancement decreased. Furthermore, they proved that the inclination angle can be a control parameter for nanofluid filled enclosure.

Eastman et al. [10], experimentally observed that Al_2O_3/water and CuO/water with 5% nanoparticle volume fractions increased the thermal conductivity by 29% and 60%, respectively. Sivasankaran et al.[11], proved that the type of nanoparticles considered is very important on the convective heat transfer application. Although a lot of studies have been carried out to investigate the role of nanofluids in cavities, most of them have considered cavities without partition. To our knowledge very little work has been done on partitioned cavities utilizing nanofluids. Zaydan and Sehaqui [12], apply the High-order compact (HOC) formulations for the natural convection in a cavity filled with nanofluid. The momentum and energy equations are discredited according to a finite difference scheme of order four. Numerical simulations are performed for a Prandtl number (Pr=6.2), the Rayleigh number varies in the range of 103 $\leqslant$Ra$\leqslant$105 and for different solid volume fractions χ of nanoparticles (Cu and TiO_2) is varied 0%$\leqslant \chi \leqslant$20%, as taking water as a base fluid.

Anilkumar and Jilani [13], studied the natural convective heat transfer in a partitioned cavity utilizing nanofluids for various pertinent parameters like the solid volume fraction, partition height, Rayleigh numbers and aspect ratio of the cavity. The results illustrated that the nano particle solid volume fraction, leads to the increase in nanofluid heat transfer rate. A. Ramiar and al (2012), studied the effect of concentration of Al_2O_3 nanoparticles in 60:40 EG/Water mixture will be studied from hydrodynamic and heat transfer point of view. Then, the effect of solid region type on axial conduction and also the effect of variable properties on thermal performance of the fluid will be considered.

Amin Habibzadeh and al.[14], studied the Al_2O_3 nanofluid-filled partitioned square cavity. The effect of different locations of the partition as well as the height of the partition is investigated in the different Rayleigh numbers and nanoparticles volume fraction. Corcione [16], and Garoosi et al. [17], investigated the natural convection of nanofluid at different geometries using the model proposed by Corcione [15], to estimate the effective viscosity and thermal conductivity of nanofluid. Their results show that there is an optimum volume fraction of nanoparticles, where the maximum heat transfer rate occurs. Kefayati et al. [18] , and Sheikholeslami et al. [19], studied the effects of magnetic field on natural convection flow in a cavity filled with nanofluid for different geometries. They used the Brinkman [20],and Maxwell-Garnett [21], models to estimate the effective viscosity and thermal conductivity of the nanofluid. They stated that the increase in volume fraction of nanoparticle and Ra enhances the heat transfer. Sheikholeslami et al. [22], have performed a numerical study of magnetic field effects on natural convection around a horizontal circular cylinder inside a square enclosure filled with nanofluid. They found that the heat transfer rate is an increasing function

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of nanoparticle volume fraction as well as the Rayleigh number, while it is a decreasing function of the Hartmann number (Ha). In addition, their results indicated that for Ha<20 the enhancement in average Nusselt number at Ra= 10^4 is greater than at other Rayleigh numbers. In a similar work, Sheikholeslami et al. [23], studied the effects of magnetic field on ferro-fluid flow and heat transfer in a semi annulus enclosure. They reported that increasing Magnetic number, Rayleigh number and volume fraction of the nanoparticles lead to augmentation of the heat transfer rate but the average Nusselt number decreases with increase of Hartmann number and Radiation parameter.Asmaa and al. [36], experimentaly studied Tin Oxide (SnO_2) nanoparticles powder have been synthesized by chemical precipitation method. Thesamples were characterized by X-ray diffraction, UV-Visible absorption and scanning probe Microscope SPM. The Xrayanalysis shows that the obtained powder is SnO_2 with tetragonal rutile crystalline structure and the crystalline sizein the range of 8-10nm. The SPM investigation reveals that the average particles size is 73nm. The optical band gapvalues of SnO_2 nanoparticles were calculated to be about 4.3eV in the temperature 550 ^{o}C, comparing with that of thebulk SnO_2 3.78eV, by optical absorption measurement.Kalteh et al. [24], investigated laminar mixed convection of nanofluid in a lid-driven square cavity with a triangular heat source and found that increasing the nanoparticle diameter leads to a decrease in the heat transfer rate at any Ri. Talebi et al. [25], studied mixed convection of nanofluids inside the differentially heated cavity (DHC). They showed that heat transfer rate has a direct relationship with Rayleigh number and nanoparticle concentration. They also showed that at a given Reynolds number the stream function increases with increasing volume fraction of nanoparticles, in particular at the higher Rayleigh number.However, experimental study of Wen and Ding [26], questions the validity of the single-phase assumption for nanofluids. Therefore, studying the performance of the nanofluid in baffled cavities needs to be investigated more. The aim of the present paper is to study the Cu/Water nanofluids filled baffled square cavity. The effects of Rayleigh number, volume fraction and partitions location on the average Nusselt number are studied.

III. MATHEMATICAL FORMULATION

We have considered the continuity, momentum and energy equations for a Newtonian. It is further assumed that radiation heat transfer among sides is negligible with respect to other modes of heat transfer. Under the assumption of constant thermal properties, the Navier–Stokes equations for an unsteady, incompressible, two-dimensional flow are:
Continuity equation:

$$\frac{\partial \mathrm{u}}{\partial x}+\frac{\partial \mathrm{v}}{\partial y}=0 \tag{1}$$

X -momentum equation:

$$\rho_{nf,0}\left(\frac{\partial \tilde{\mathrm{u}}}{\partial \tilde{t}}+\tilde{\mathrm{u}}\frac{\partial \tilde{\mathrm{u}}}{\partial \tilde{x}}+\tilde{\mathrm{v}}\frac{\partial \tilde{\mathrm{u}}}{\partial \tilde{y}}\right)=-\frac{\partial \tilde{\mathrm{P}}}{\partial \tilde{x}}+\mu_{nf}\Delta_2\breve{u} \tag{2}$$

Y-momentum equation:

$$\rho_{nf,0}\left(\frac{\partial \tilde{\mathrm{v}}}{\partial \tilde{t}}+\tilde{\mathrm{u}}\frac{\partial \tilde{\mathrm{v}}}{\partial \tilde{x}}+\tilde{\mathrm{v}}\frac{\partial \tilde{\mathrm{v}}}{\partial \tilde{y}}\right)=-\frac{\partial \tilde{\mathrm{P}}}{\partial \tilde{y}}+\mu_{nf}\Delta_2\tilde{\mathrm{v}}+\left[\chi\beta_s\rho_s+(1-\chi)\rho_f\beta_f\right]g.\tilde{T} \tag{3}$$

Energy equation:

$$\frac{\partial \tilde{\mathrm{T}}}{\partial \tilde{t}}+\tilde{u}\frac{\partial \tilde{T}}{\partial \tilde{x}}+\tilde{v}\frac{\partial \tilde{T}}{\partial \tilde{y}}=\alpha_{nf}\left(\Delta_2\tilde{T}\right) \tag{4}$$

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Where

$$\Delta_2 = \frac{\partial^2}{\partial \tilde{x}^2} + \frac{\partial^2}{\partial \tilde{y}^2} and \alpha_{nf} = \frac{K_{eff}}{(\rho C_p)_{nf}} \quad (5)$$

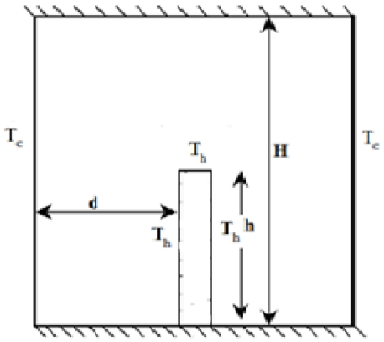

Fig. 1. Schematic of the partitioned square cavity

The viscosity of the nanofluid can be estimated with the existing relations for the two-phase mixture. The equation given by Brinkman [20] has been used as the relation for effective viscosity in this problem, as given by Xuan and Li [27]they have experimentally measured the apparent viscosity of the transformer oil–water nanofluid and of the water–copper nanofluid in the temperature range of 20–50 °C. The experimental results reveal relatively good agreement with Brinkman's theory. The effective density of the nanofluid at reference temperature is:

$$\rho_{nf} = \chi\rho_s + (1-\chi)\rho_f \quad (6)$$

The heat capacitance of the nanofluid is expressed as Abu-Nada, (2009) and Khanafer et al. [28]:

$$(\rho C_p)_{nf} = \chi(\rho C_p)_s + (1-\chi)(\rho C_p)_f \quad (7)$$

The effective thermal conductivity of the nanofluid is approximated by the Maxwell–Garnetts model[21]:

$$\frac{\kappa_{nf}}{\kappa_f} = \frac{(\kappa_s + 2\kappa_f) - 2\chi(\kappa_f - \kappa_s)}{(\kappa_s + 2\kappa_f) + \chi(\kappa_{f-}\kappa_s)} \quad (8)$$

Table 1. Thermo-physical properties of water and nanoparticles:

Physical properties	Fluid phase	(Cu)
$C_p(J/kg.K)$	*4179*	383
$\rho(kg/m^3)$	997.7	8933
$K(W/m.K)$	0.613	400
$\boldsymbol{\beta(K^{-1})}$	$\mathbf{2.1 \times 10^{-4}}$	$\mathbf{1.67 \times 10^{-5}}$

Eqs. (1)– (4) can be converted to the dimensionless forms by definition of the following parameters as:

$$\tilde{x} = \frac{x}{H} \quad ,\tilde{y} = \frac{y}{H} \quad ,\tilde{\mathrm{u}} = \frac{\mathrm{u}H}{\nu_f} \quad ,\tilde{\mathrm{v}} = \frac{\mathrm{v}H}{\nu_f} \quad ,\tilde{P} = \frac{\mathrm{p}}{\frac{\rho_f \nu_f{}^2}{H^2}} \quad , \qquad \tilde{t} = \frac{\mathrm{t}}{\frac{H^2}{\nu_f}} \quad ,\tilde{T} = \frac{T - T_c}{T_h - T_c}$$

The governing equations can now be written in dimensionless form as follows:
Continuity equation:

$$\frac{\partial \mathrm{u}}{\partial x} + \frac{\partial \mathrm{v}}{\partial y} = 0 \tag{9}$$

X-momentum equation:

$$\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\rho_f}{\rho_{nf,0}}\frac{\partial P}{\partial x} + \frac{\mu_{nf}}{\nu_f \rho_{nf,0}}\Delta_2 u \tag{10}$$

Y-momentum equation:

$$\left(\frac{\partial \mathrm{v}}{\partial t} + \mathrm{u}\frac{\partial \mathrm{v}}{\partial x} + \mathrm{v}\frac{\partial \mathrm{v}}{\partial y}\right) = -\frac{\rho_f}{\rho_{nf,0}}\frac{\partial \mathrm{P}}{\partial y} + \frac{\mu_{nf}}{\nu_f \rho_{nf,0}}\Delta_2 \mathrm{v} + \frac{\left[\chi\beta_s\rho_s + (1-\chi)\rho_f\beta_f\right]}{\beta_f \rho_{nf,0}} GrT \tag{11}$$

Energy equation:

$$\frac{\partial T}{\partial t} + \mathrm{u}\frac{\partial T}{\partial x} + \mathrm{v}\frac{\partial T}{\partial y} = \frac{\alpha_{nf}}{\alpha_f P_r}\Delta_2 T \tag{12}$$

Where:

$$\mathrm{Gr} = \frac{\beta_f g H^3 (T_h - T_c)}{\nu_f{}^2} \quad ; \mathrm{Ra} = \frac{\beta_f g H^3 (T_h - T_c)}{\nu_f \alpha} \quad ;$$

$$Pr = {}^{\nu_f}/_{\alpha_f}$$

Finally the dimensionless system of equations describing the convective flow is:

$$\nabla.\vec{V} = 0 \tag{13}$$

$$\frac{\partial \vec{V}}{\partial t} + (\vec{V}.\vec{\nabla})\vec{V} = -\frac{\rho_f}{\rho_{nf,0}}\vec{\nabla}P + \frac{[\chi\beta_s\rho_s + (1-\chi)\rho_f\beta_f]}{\beta_f \rho_{nf,0}} GrT\vec{e}_n + \frac{\mu_{nf}}{\nu_f \rho_{nf,0}}\Delta_2 \vec{V} \tag{14}$$

$$\frac{\partial T}{\partial t} + (\vec{V}.\vec{\nabla})T = \frac{\alpha_{nf}}{\alpha_f P_r}\Delta_2 T \tag{15}$$

Where the radial and tangential velocities are given by the following relations respectively:

$$u = \frac{\partial \psi}{\partial y} \quad , \quad \mathrm{v} = -\frac{\partial \psi}{\partial x} \tag{16}$$

In order to eliminate the pressure term that is not of importance in natural convection, the system of equations to be solved is formulated in terms ofdimension less vorticity ω and stream function ψ asfollows:
Stream function:

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$$\frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} = -\omega \tag{17}$$

Vorticity:

$$\left(\frac{\partial \psi}{\partial y}\right)\left(\frac{\partial \omega}{\partial x}\right) - \left(\frac{\partial \psi}{\partial x}\right)\left(\frac{\partial \omega}{\partial y}\right) = \frac{\mu_{nf}}{v_f \rho_{nf,0}}\left(\frac{\partial^2 \omega}{\partial x^2} + \frac{\partial^2 \omega}{\partial y^2}\right) + \eta\left(\frac{\partial T}{\partial x}\right) \tag{18}$$

Where:

$$\eta = \frac{[\chi \beta_s \rho_s + (1-\chi)\rho_f \beta_f] * Ra}{P_r \beta_f \rho_{nf,0}}$$

Energy:

$$\left(\frac{\partial \psi}{\partial y}\right)\left(\frac{\partial T}{\partial x}\right) - \left(\frac{\partial \psi}{\partial x}\right)\left(\frac{\partial T}{\partial y}\right) = \frac{\alpha_{nf}}{\alpha_f P_r}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{19}$$

The boundary conditions, used to solve the equations are as follows:

The right wall :T = u = v = 0 at x = 1 and $0 \le y \le 1$

The left wall :T = u = v = 0 at x = 0 and $0 \le y \le 1$

The high wall :$\frac{\partial T}{\partial Y}$= u = v = 0 at y = 1 and $0 \le x \le 1$

The low wall: $\frac{\partial T}{\partial Y}$= u = v = 0 at y = 0 and $0 \le x \le 1$

On the partition: T = 1

Nusselt number(Nu) is a dimension less number to characterize the ratio of heat transfer by convection and heat transfer by conduction. This quantity is used to define the law of heat exchange as a result:

$$Nu = \frac{hH}{k_f} \tag{20}$$

The heat transfer coefficient is expressed as:

$$h = \frac{q_w}{(T_h - T_c)} \tag{21}$$

The thermal conductivity is expressed as

$$k_{nf} = -\frac{q_w}{\partial T / \partial x} \tag{22}$$

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By substituting Eqs. (21), (22), and (8) into Eq. (20), and using the dimensionless quantities, the local Nusselt number is written as:

$$Nu = -\left(\frac{k_{nf}}{k_f}\right)\frac{\partial T}{\partial x} \tag{23}$$

The average Nusselt number is defined as

$$\overline{Nu} = \int_0^1 Nu(x)\,dx \tag{24}$$

IV. GRID TESTING AND NUMERICAL METHOD

Eqs. (10) to (12) and (17), along with the related boundary conditions, are solved using an efficient finite difference method. The central difference method is applied for discretization of the equations. The solution of equations was performed using the ADI method (Alternating Direct Implicit). And the convergence criteria 10^6 is chosen for all dependent variables. A regular grid is used in the study and to investigate the grid independency, the mesh sizes are increased from 41 × 41 to 101 ×101. The optimum grid dimension is chosen according to the variation of the velocity V. Thus, it is decided to select the 81 × 81 grid dimensions in this study (fig. 2):

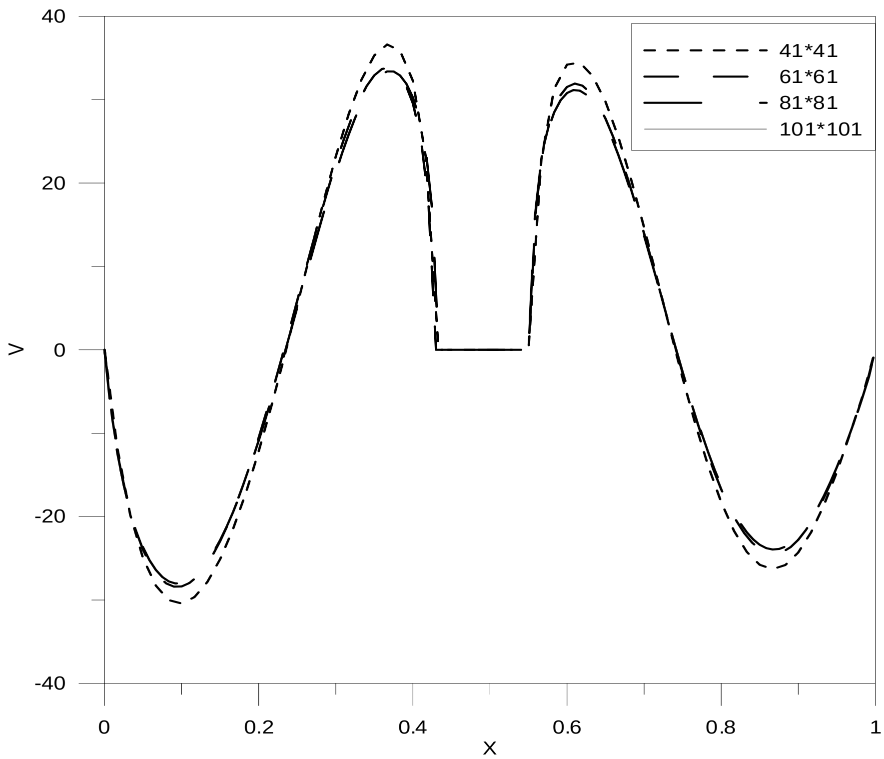

Fig. 2. Vertical velocity of the cavity for different uniform grids.

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V. CODE VALIDATION

To use the numerical code developed with more confidence, it is essential to validate with existing results in the literature.Thus, we compared our results with success with those obtained in a heated from the side in the case of pure fluid cavity (χ=0).The numerical simulations are performed for Prandtl number (Pr=6.2) the Rayleigh numbers varying between $10^3 \leq Ra \leq 10^5$ and for different volume fractions for the pure fluid (χ =0) and for nanoparticules 0%$\leqslant \chi \leqslant$10%.

	$Ra = 10^3$	$Ra = 10^4$	$Ra = 10^5$	$Ra = 10^6$
G.de Val Davis (1983)	1.118	2.243	4.519	8 .799
Zaydan and Sehaqui (2013)	*1.127*	*2.253*	*4.546*	*7.980*
Markatos and perikleous (1984)	*1.108*	*2.201*	*4.430*	*8.754*
G.V.Hadjisophcleous et al (1998)	*1.141*	*2.290*	*4.964*	*10.390*
R.K.Tiwari, M.K.Das (2007)	*1.141*	*2.290*	*4.964*	*10.390*
B.Ghasemi,S.M.Aminossadati (2010)	*1.118*	*2.248*	*4.547*	*8.980*
I. El Bouihi, R. Sehaqui (2012)	*1.042*	*2.024*	*4.520*	*8.978*
Ean Hin Ooi, V. Popov (2013)	*1.107*	*2.232*	*4.577*	*---*
A.Habibzadeh, A.Zehforoosh, A. Khojaste (2011)	*---*	*2.241*	*4.526*	*8.919*
Present work	*1.124*	*2.291*	*4.801*	*9.622*
Difference with R.K.Tiwari, M.K.Das %	*1.48*	*0.04*	*3.28*	*7.39*

Table 2: Comparison between the present work and other studies for (Nu).

VI. RESULTS AND DISCUSSION

A numerical study is performed to investigate the natural convection in a nanofluid-filled (Cu-Water) partitioned square cavity shown in fig.1.The results discussed here are for three partitions locations of d= 0.3, 0.5 and 0.7, a range of Rayleigh number from 10^3 to 10^5 and the volume fractions of the nanoparticles ranging from 0 to 0.1. The results are presented in terms of streamlines and isotherms inside the enclosure and the average Nusselt number of the hot partition.

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$Ra = 10^3$ $Ra = 10^4$ $Ra = 10^5$

(a) (b) (c) (d) (e)

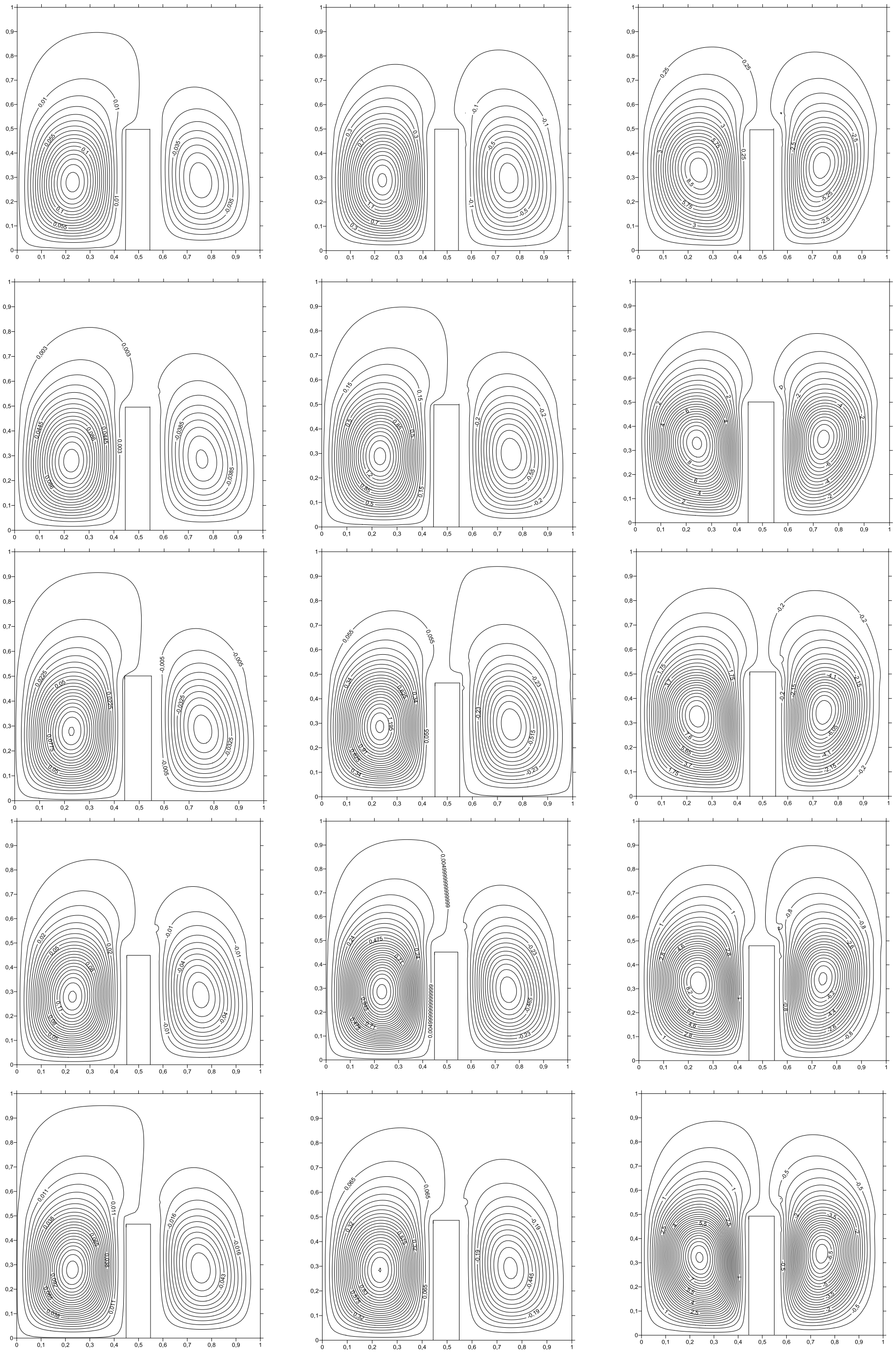

Fig. 3. Stream functions for water- Cu, $(\boldsymbol{a})\chi = \mathbf{0}\%$, $(\boldsymbol{b})\chi = \mathbf{3}\%$, $(\boldsymbol{c})\chi = \mathbf{5}\%$, $(\boldsymbol{d})\chi = \mathbf{7}\%$, $(\boldsymbol{e})\chi = \mathbf{10}\%$.

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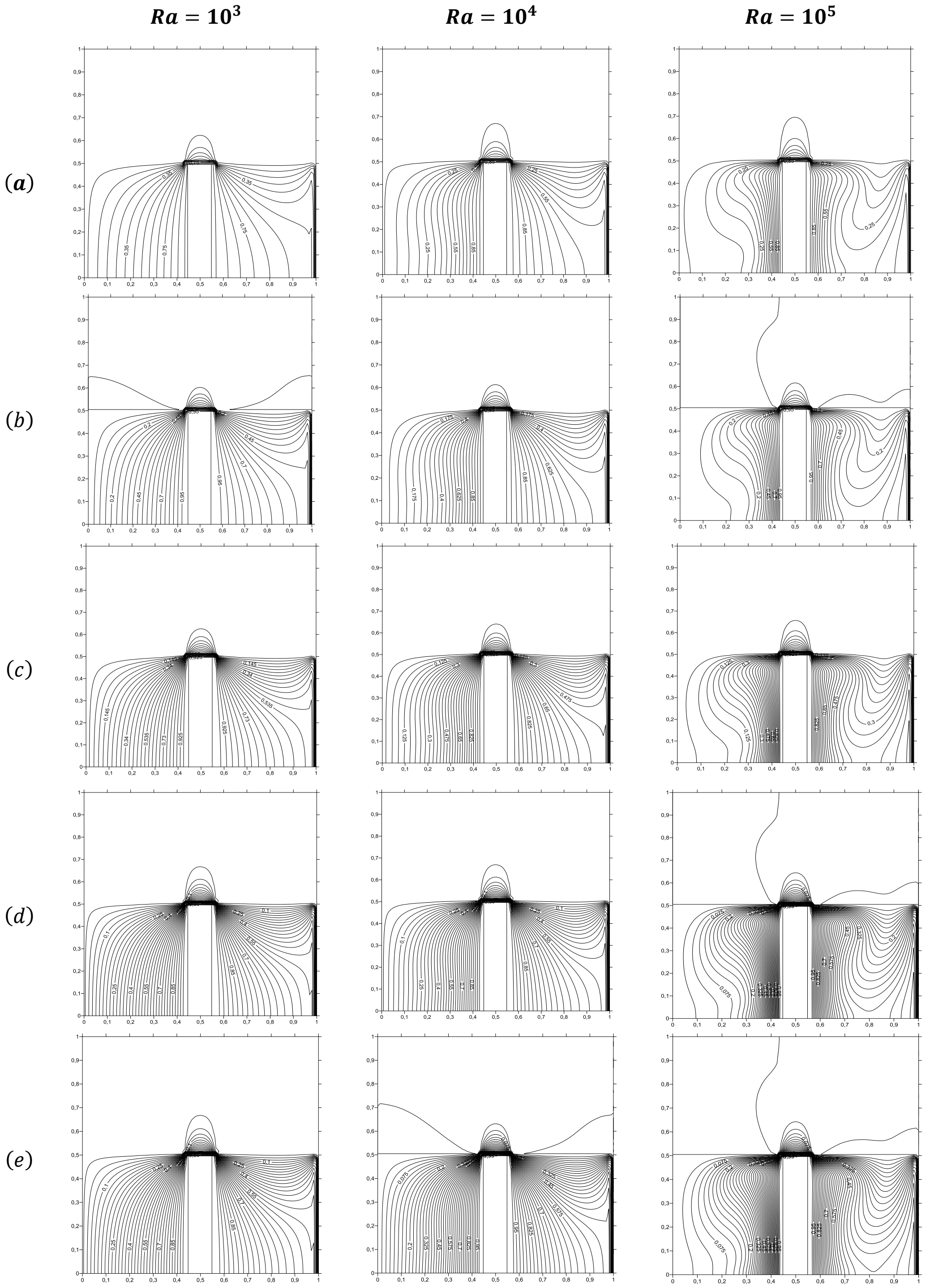

Fig. 4. Isotherms for water- Cu,$(a)\chi = 0\%$, $(b)\chi = 3\%$, $(c)\chi = 5\%$, $(d)\chi = 7\%$, $(e)\chi = 10\%$.

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Fig. 3. Displays stream lines at Ra=10^3, 10^4, 10^5 for both pure fluid and with different values of nanofluid. The fluid close to the hot bloc absorbs heat, so its buoyancy force augments and moves toward up. It was replaced by the cold fluid that is near the right and the left wall. At Ra=10^3 the weak buoyancy force leads to a weak flow. With increasing the Rayleigh number, the flow augments and position of vortex moves one toward right horizontal wall and the other one to the left. The presence of nanoparticles in the water enhances the effective thermal conductivity and leads to the increase in intensity of flow as a result of the augmentation in the buoyancy force, so that at Ra=10^3 with 10% increase in solid concentration, the value of stream function in the centre of the vortex increases and at Ra=10^5 it increases to like as seen.

Fig.4. Shows the isotherms for Cu–water nanofluid for the entire range of the partitioned square and all the considered values of Ra, the isotherms near the hot partition are approximately parallel to each other. This indicates that the heat transfer near the hot partition is due to conduction. The effect of convection on heat transfer in the core region of the cavity is evident as the isotherms. Are irregular and distorted in this region. As the Rayleigh number Increases, the isotherms are largely distorted at the central part of the cavity due to a high convective flow. This implies that the thermal mixing is higher specially when Ra=10^5, with the increase of nanoparticles volume fraction 0.1, the thermal gradient near the hot partition are larger than the pure fluid. This is because the addition of nanoparticles increases the thermal conductivity, which causes the heat to penetrate much deeper into the nanofluid before being carried away by the convection.

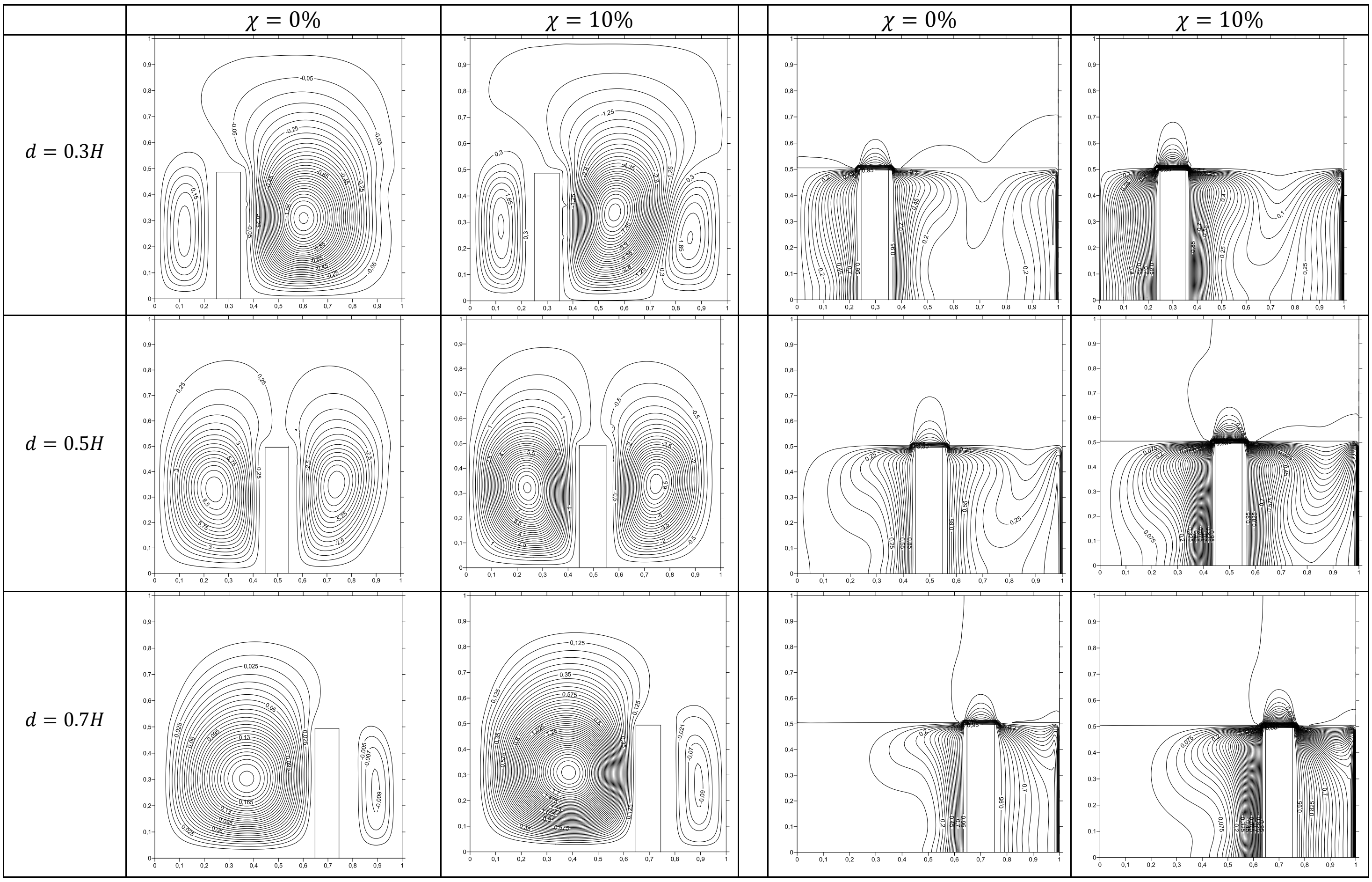

Fig. 5. Effect of solid volume fraction and partitions locations on stream functions and Isotherms for different values of partitions locations d= 0.3, 0.5 and 0.7 at Rayleigh number, Ra=10^5.

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Fig.5. Shows streamlines and isotherms for different partitions locations at Ra = $10^5, \chi = 10\%$, with d=0.3, 0.5, 0.7 and for h = 0.5. At d=0.3 there is a too weak flow between the left wall of the hot block and the vertical wall of the cavity, so conduction dominates and temperature rises in the left side. The long distance between the right wall of the hot block and the right vertical wall of the cavity causes the flow to have enough time to gain speed. So a stronger recirculation cell is made above the hot block that covers most parts of the cavity. It enhances convective heat transfer on the right of the hot block surface, and then its temperature reduces of the right side of the square cavity. With the increase of d=0.5, as can be seen clearly there is a transformation to get two cells, so that two separate vortexes form on the right and left of the hot block, so the intensity of flow recirculation is higher. For d=0.7 there is a stronger resistance against gaining the speed of the flow. Thus the Vortex in this condition especially with 10% increase in solid concentration is more efficient than the other states to become tree cells like as seen. The presence of nanoparticles enhances flow intensity due to the increase in energy transfer. Nanofluid has the most effect at d=0.3 and 0.7H, that conduction has an important role in heat transfer, so that the stream function at the center of the vortex increases with 10% increase in solid concentration, but it has the least effect at d=0.3.

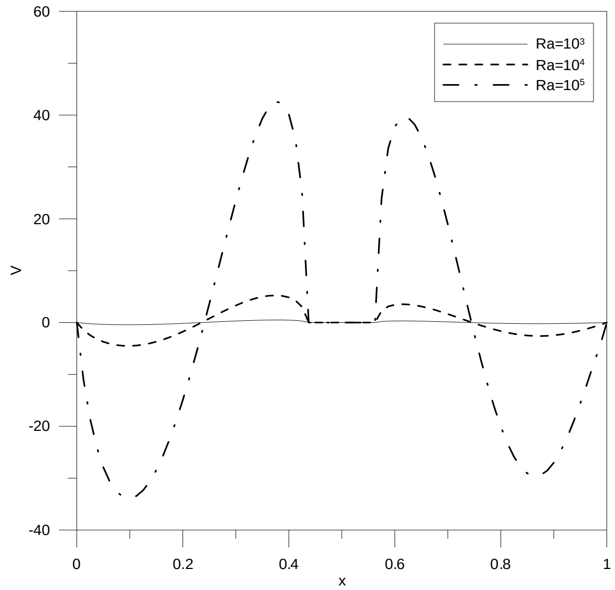

Fig. 6.V-velocity distributions of the cavity for different Rayleigh numbers and volume fractions $\chi = 0\%$.

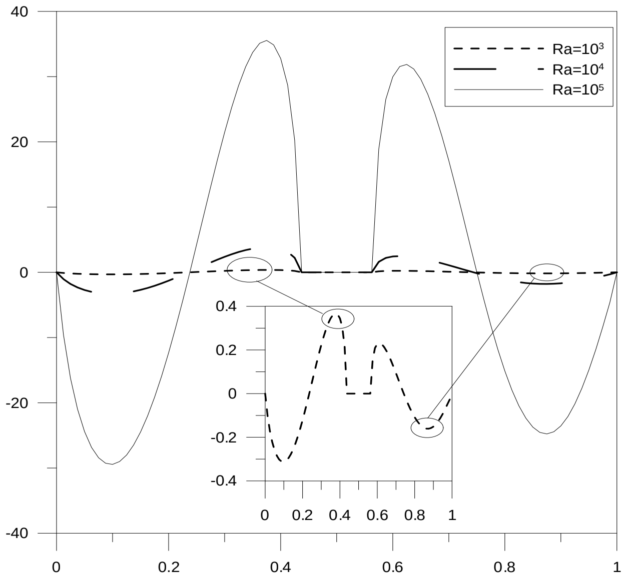

Fig. 7. V-velocity distributions of the cavity for different Rayleigh Numbers and volume fractions$\chi = 10\%$.

Fig. 6, Fig. 7. According to the results at $\chi = 0\%$ and 10%. Consequently, in this situation V-velocity does not have a symmetric profile. It is obviously recognized from the case of $\chi = 0\%$ and 10% in which natural convection is dominant that the vertical component of velocity has a symmetric manner. High values of χ cause the fluid becomes more viscous which causes the velocity to attenuate consequently.

Fig.8. Displays the effect of solid concentration on the variation of the average Nusselt number on the heat source surface for different values of Rayleigh number. With an increase of Rayleigh number, convective heat transfer and hence Nusselt number increases. It is observed that the Nusselt number increases with an increase in solid concentration, indicating the better heat transfer. The presence of nanoparticles has more effect at lower Number of Rayleigh, so that increase of solid concentration enhances the Nusselt number.**Fig.9.** Shows the variation of the average Nusselt number against the solid concentration on the heat source surface for the different value of h. So with increase of distance at d= 0.5 Nusselt number grows up that is more effective with respect to other values of d,the intensity of recirculation cell decreases on both on d=0.3 and d=0.7, so heat transfer and hence Nusselt decrease.

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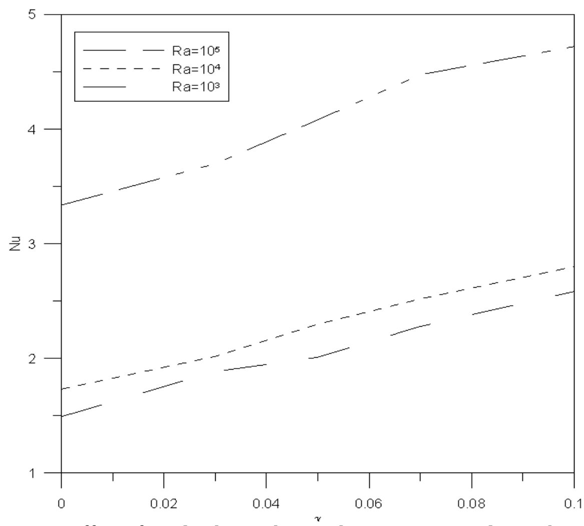

Fig .8. Effect of Rayleigh Number on the average Nusselt Number.

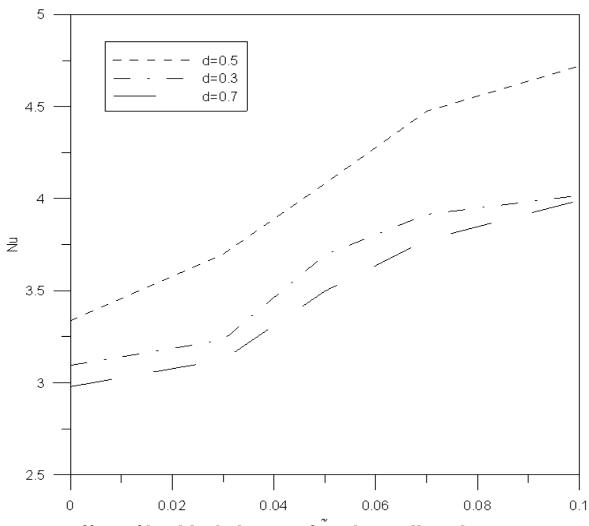

Fig. 9. Effect of hot block distance from hot wall on the average Nusselt Number.

VII. CONCLUSION

This paper presents a natural convective heat transfer in a partitioned square cavity utilizing nanofluids is studied. The vertical left and right walls are considered as cold walls, respectively and the partitions assumed to be hot. The nanofluid used in this study is Cu- Water with the volume fraction between 0% - 10%. Validation is conducted by comparing with different previously published data. The variation of different parameters, including Rayleigh number, partition distance from the cold wall, partition height and the volume fraction of the nanoparticles is studied. The main conclusions from this study at the defined ranges are as follows:

- The effect of nanofluid on convection is particularly evident at high Rayleigh number.
- Increasing the volume fraction of nanofluid promotes the heat transfer benefit.
- The hot block has a major influence on the structure of flow and heat generated within this geometry.
- The average Nusselt number is maximum when the partition is placed at the center (d = 0, 5).

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