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Numerical study of mixed convection of nanofluid in lid-driven enclosure partially heated

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***Abstract*— We present a numerical study of unsteady mixed convection in a rectangular cavity partially heated and filled with a nanofluid (Water/S_iO_2). Two heat sources of dimension (b) are placed opposite in the vertical walls, the remainder of these walls is maintained adiabatic, horizontal walls are cooled and driven at a constant speed U_0. The equations governing the flow are solved with a house code "NASIM" based on a finite volume method that uses the multigrid solver. The effect of the training walls on the hydrodynamic structure of the flow and heat transfer rate is predicted and analyzed. Two cases are considered as the training set into play horizontal walls. It has been shown that the structure of the flow, the heat transfer rate and the maximum temperature heat sources vary significantly as a result of training.**

***Keywords*— Nanofluid; Mixed convection; Partially active walls; Volume fraction; Driven cavity**

I. Introduction

Accurate prediction of the flow conditions in the driven cavity is of outmost importance for a number of technological applications, such as coating and polishing processes in microelectronics, passive and active flow control using blowing/suction cavities and riblets. A major limitation against increasing the heat transfer in such engineering systems is the inherently low thermal conductivity of the commonly used fluids, such as, air, water, and oil. The idea is to insert within the fluid, metallic particles of nanometer size hope to increase the effective thermal conductivity of the mixture. In fact, the presence of the nanoparticles in the fluids increases appreciably the effective thermal conductivity of the fluid and consequently enhances the heat transfer characteristics [1-2]. The term nanofluid was then introduced by Choi et al. [1] and is commonly used to characterize this type of colloidal suspension. Because the prospect of nanofluids is very promising, several studies of convective heat transfer in nanofluids have been reported in recent years.

Most of the studies considering the heat transfer performance using nanofluids in natural convection were investigated based on rectangular enclosures in the last decades [3–5]. However, little work has been carried out for mixed convection of nanofluid in cavity. Tiwari and Das [6] studied numerically the mixed convection in two-sided lid-driven differentially heated square cavity filled with nanofluid. They showed that the additions of nanoparticles in a fluid are capable of increasing the heat transfer capacity of base fluid. As solid volume fraction increases, the effect is more pronounced. Sebdani et al [7] numerically studied the effects of Al_2O_3/water nanofluid on mixed convection heat transfer in a square cavity with a heat source on the bottom wall and moving downward cold side walls. They prove that when the Reynolds number increases, while the Rayleigh number is keeping constant, the forced convection becomes stronger that causes the heat transfer rate to increases. When the Rayleigh number increases, while the Reynolds number is kept constant, the heat transfer rate increases. The presence of nanoparticles causes increase in heat transfer rate only at Ra = 10^3 while at Ra = 10^4 and 10^5 the rate of heat transfer decreases with increase in nanoparticles volume fraction. The main objective of this study is to investigate the effect of the training walls on the hydrodynamic structure of the flow and heat transfer rate in a rectangular cavity partially heated and filled with a nanofluid (Water/SiO_2). The results is obtained for a range of $0.001 \le Ri \le 10$, and $0 \le \phi \le 0.2$.

II. Model description

A schematic diagram of the considered model is shown in Fig.1 with coordinates. It is a two-dimensional square enclosure of height H and filled with SiO2-water nanofluid. A two heat sources with constant heat flux q" and dimensionless length B (B=H/2) are embedded in the two sidewalls. The top and bottom walls are kept at a maintained constant temperature TC; for the case 1 only the top wall is sliding at a constant speed U0, for the second case the bottom wall is sliding too at a constant speed -U0. The remaining boundary parts of the enclosure are adiabatic. The SiO2-water nanofluid is assumed to be Newtonian, in thermal equilibrium, and the nanoparticles are kept uniform in shape and size.

III. Mathematical modelling

The thermo-physical properties of the cu-water nanofluid, presented in Table 1, are considered to be constant with the exception of its density which varies according to the Boussinesq approximation.
Using the following dimensionless variables:

TABLE I
THERMO-PHYSICAL PROPERTIES OF WATER AND SILICA (SiO_2 NANOPARTICLES). [8]

	Pure water	SiO_2
$\rho(kgm^{-3})$	997.1	3970
$\beta(K^{-1})$	21×10^{-5}	0.63×10^{-5}
$k(Wm^{-1}K^{-1})$	0.613	36
$C_p(Jkg^{-1}K^{-1})$	4179	765

$$X=\frac{x}{H}\ ,\ Y=\frac{y}{H}\ ,\ U=\frac{u}{U_0}\ ,\ V=\frac{v}{U_0}\ ,\ P=\frac{p}{\rho_{nf}U_0^2}$$

$$\theta=\frac{T-T_C}{\Delta T}\ ,\ \Delta T=\frac{q''H}{k_f}\ ,Gr=\frac{g\beta_f\Delta TH^3}{\upsilon_f^2}$$

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

$$\frac{\partial U}{\partial t}+U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{\mathrm{Re}}\frac{\mu_{eff}}{\upsilon_f\rho_{nf}}\left(\frac{\partial^2U}{\partial X^2}+\frac{\partial^2U}{\partial Y^2}\right)$$

$$\frac{\partial V}{\partial t}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{\mathrm{Re}}\frac{\mu_{eff}}{\upsilon_f\rho_{nf}}\left(\frac{\partial^2V}{\partial X^2}+\frac{\partial^2V}{\partial Y^2}\right)+\frac{(\rho\beta)_{nf}}{\rho_{nf}\beta_f}\mathrm{Ri}\theta$$

and

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

Where:

$$\mathrm{Re}=\frac{U_0H}{\upsilon_f}\ ,\ \mathrm{Ri}=\frac{\mathrm{Gr}}{\mathrm{Re}^2},\ and\ \mathrm{Pr}=\frac{\nu_f}{\alpha_f}$$

The boundary conditions consist of:

❖ Left and right walls

- $U=V=0$
- $\frac{\partial\theta}{\partial Y}=0\ \ for\begin{cases}X=0\ \ and\ \ 0\le Y\le(D-0.5B)\\ X=1\ \ and\ \ 1\ge Y\ge(D+0.5B)\end{cases}$
- $\frac{\partial\theta}{\partial Y}=-\frac{k_f}{k_{nf}}\ \ for\ X=0;1\ \ and\ \ (D-0.5B)\le Y\le(D+0.5B)$

❖ Top wall

- $U=U_0\ ;V=0$
- $\theta=0$

❖ Bottom wall

- $V=0\ ;\begin{cases}U=0\ \ for\,case1\\ U=-U_0\ \ for\,case2\end{cases}$
- $\theta=0$

The local Nusselt numbers on the heat source surface can be defined as:

$$Nu_s=\frac{hH}{k_f}$$

where h is the convection heat transfer coefficient:

$$h=\frac{q''}{T_s-T_c}$$

Rewrite the local Nusselt number by using the dimensionless parameters:

$$Nu_s(Y)=\frac{1}{\theta_s(X)}$$

Nu_s along the heat source $Nu_m=\frac{1}{B}\int_{D-0.5B}^{D+0.5B}Nu_s(X)\,dX$

IV. NUMERICAL APPROACH

The governing equations were numerically solved using the classical projection method [9]. A finite volume method on a staggered grid system have been implemented to disctretise the dimensionless equations and the QUICK scheme of Hayase et al. [10] is employed to minimize the numerical diffusion for the advective terms. The Poisson equation with homogeneous boundary conditions is solved with the help of an accelerated full multigrid method [11] whoever, the momentum equations are computed by a red black successive over-relaxation method.

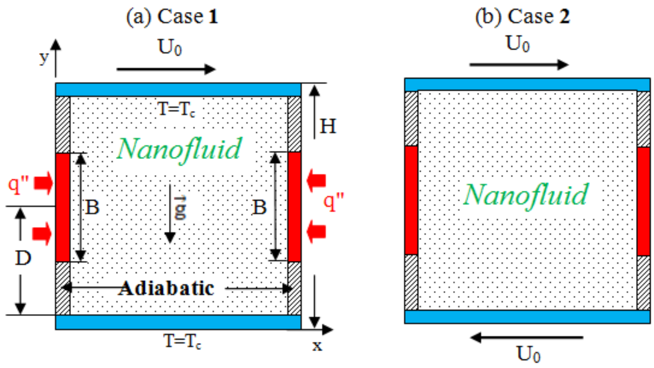

Fig. 1 Physical model and boundary conditions for the square cavity with active side walls.

Finally, the convergence of the numerical 2D velocity field is established at each time step by controlling the L2-residuals norm of all equations to be solved by setting its variation to less than 10−8. In order to secure the steady state conditions, the following criterion has to be satisfied: $\sqrt{\sum_{i,j}(x_{i,j}^n-x_{i,j}^{n-1})^2}<10^{-8}$

Here the superscript n indicates the iteration number and the subscript sequence (i, j) represents the space coordinates x and y. This numerical method was implemented in a FORTRAN home code named «NASIM» [12].

V. RESULTS AND DISCUSSIONS

Mixed convection flow and temperature fields in lid-driven square cavity are examined. The governing parameters in this problem is Richardson number, Ri $=Gr/Re^2$, which characterizes the relative importance of buoyancy to forced convection. To vary Richardson number, Grashof number is fixed at Gr=1,613.103 while changing Reynolds number through the plate velocity U_0. Investigations through the cavity are made for ranges of the Richardson number from 0.001 to 10. Lid-driven cavity is analyzed according to the number of horizontal moving plate in two cases shown in Fig.1. The results for each will be presented next:

Case 1: The top wall is moving from left to right. It is noted that forces due to moving lid and buoyancy act in opposite directions at the right wall but they force are in same direction at the left wall. Streamlines and isotherms for the SiO_2-water nanofluid ($\phi = 0.2$) for the case1 at Ri =0.001 and Ri=10 are shown in Fig.2. For the sake of comparisons, the streamlines and the isotherms for pure water ($\phi = 0$) are also shown in this figures. For Ri = 0,001, Fig. 2a and b shows that the forced convection plays a dominant role and the recirculation flow is mostly generated only by moving lids.

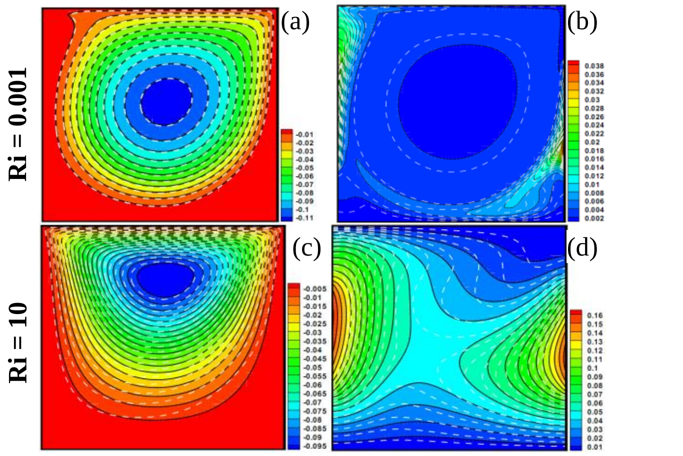

Fig. 2 Streamlines (on the left) and isotherms (on the right) for the enclosures filled with SiO2–water nanofluid, (line solid black), and pure water (line dashed white) at Ri =10, Ri=0.001, Gr=1,613.10^3 (D = 0.5 and B = 0.5) for the Case1

As it is seen from Fig. 2a the recirculation is clockwise and some perturbations are seen in streamlines in the upper left corner due to impingement of nanofluid to the vertical top wall.

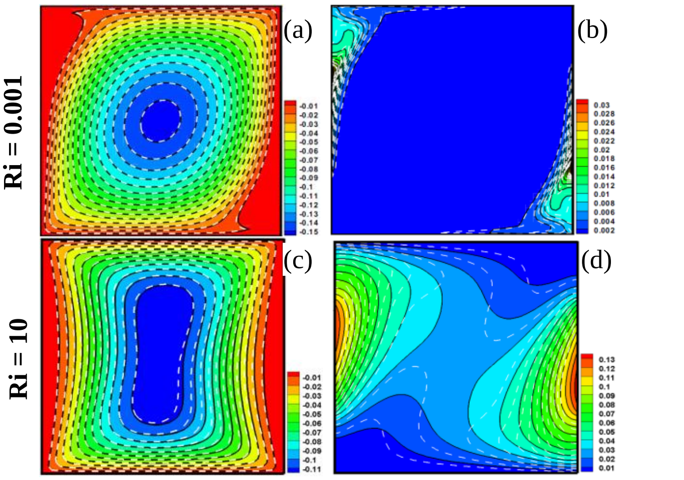

Fig. 3 Streamlines (on the left) and isotherms (on the right) for the enclosures filled with SiO2–water nanofluid, (line solid black), and pure water (line dashed white) at Ri =10, Ri=0.001, Gr=1,613.10^3 (D = 0.5 and B = 0.5) for the Case2

Fig.2a shows that the horizontal temperature gradient is disappeared. For this Richardson number addition of nanoparticles does not have influence on the structure of the flow but they increases the heat transfer near the two heat sources. Even if the observed perturbations at the upper right and lower left corners are ignored, we can see that they are too different from streamlines and isotherms observed in a partially heated cavity. When Ri = 10, the effect of natural convection is far more compared to the forced convection effect (Fig. 2c, 2d) ,hence the core of eddy moves upward and the flow pattern and temperature distribution is changed totally.

Case 2: In this case, the top wall is kept moving from left to right and the bottom wall is moving in the contrary direction. Moving the bottom wall increase moreover the forces due to moving lid. Streamlines and isotherms for the SiO2-water nanofluid ( ) for the case2 at Ri =0.001 and Ri=10 are shown in Fig.3. For the sake of comparisons, the streamlines and the isotherms for pure water ( ) are also shown in this figures. For Ri = 0,001, when the forced convection dominated regime, we can see from Fig. 3a a clockwise recirculation and some perturbations are seen in streamlines in the upper left and lower right corners due to impingement of nanofluid to the vertical wall and the core of cell is located in the center of the cavity.

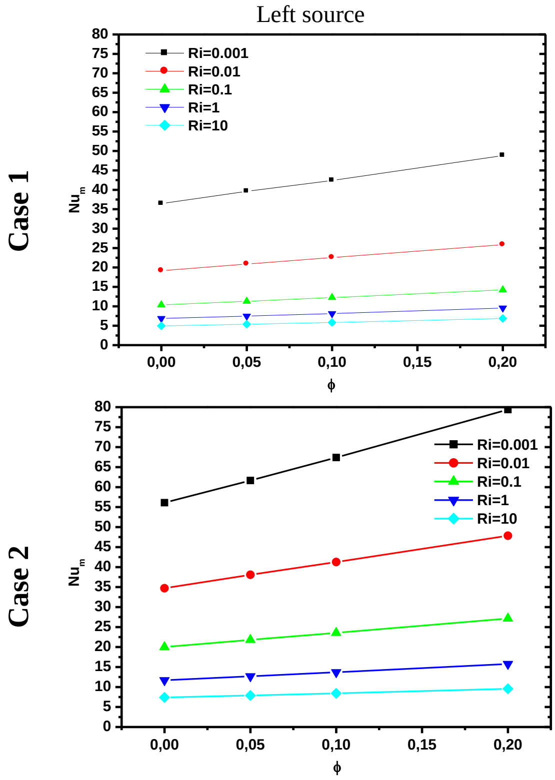

Fig.4. Variation of average Nusselt number on the left source with solid volume fraction ϕ at various Richardson numbers, Gr=1,613.10^3

The core region of cavity is isotherms while a steep temperature gradient occurs within a thin region near the side walls. By increasing Richardson number to Ri=10 a single large fully developed clockwise eddy is formed inside the cavity. In Fig.3, the variations of the average Nusselt number at the left and right heat sources with respect to the volume fraction of the nanoparticles for two investigated cases is presented at different Richardson numbers. The figure shows, for the all configurations investigated, that the heat transfer increases almost monotonically with increasing the solid volume fraction independently of the Richardson number. Moreover, it can be inferred from Fig.3 that the

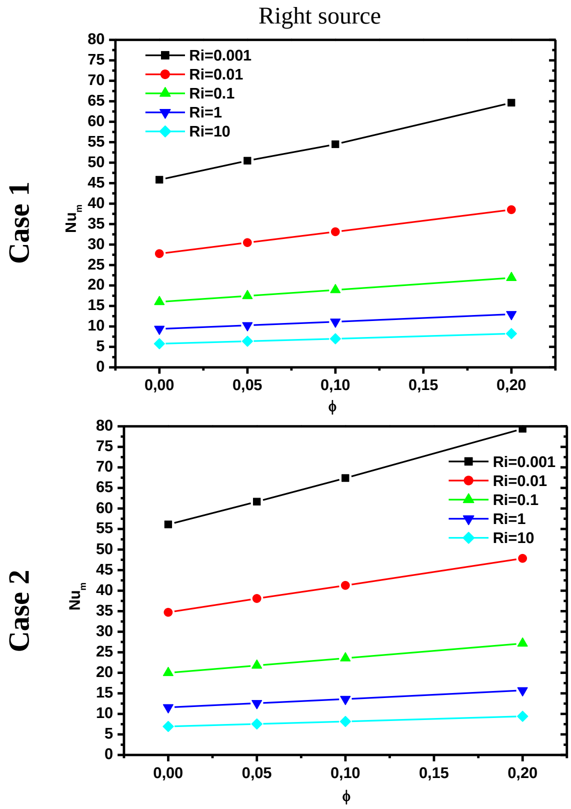

Fig.4. Variation of average Nusselt number on the right source with solid volume fraction ϕ at various Richardson numbers, Gr=1,613.10^3

increase of solid volume fraction tends to increase greatly the average Nusselt number and causes the heat source maximum temperature to decrease particularly at high Richardson numbers where forced convection is the main heat transfer mechanism. For all Richardson numbers, it can be conclude from Fig.3, that the maximum average Nusselt number is obtained for the case 2 precisely for Ri=0.001 and $\phi = 0.2$,Num=7.403 .

VI. CONCLUSIONS

This study has been concerned with the numerical modeling of mixed convection in lid-driven partially heated cavities. It has been performed for two different cases characterized by the number of movement of horizontal walls. The governing parameter is Richardson number, which characterizes the heat transfer regime in mixed convection. In view of the results, following findings may be summarized:

a) The governing parameter affecting heat transfer is Richardson number Ri = Gr/Re2. For Ri < 1, the flow and heat transfer is dominated by forced convection, for Ri > 1, it is dominated by natural convection and for Ri = 1, it is a mixed regime.

b) For Ri > 1, the average Nusselt number relatively low and has the same order of magnitude for all three cases. For Ri < 1, the forced convection becomes dominant, the natural convection relatively weak, as a result of which Nusselt number is relatively higher.

c) For the two investigated cases, the heat transfer rate and flow strength are enhanced on increasing the Richardson number, subsequently a decrease in the maximum temperature heat source is obtained.

d) The heat transfer rate is also increases on increasing the solid volume fraction of the nanoparticles.

e) It turns out that the case 2 gives the highest heat transfer enhancement. Precisely, the addition of 20% of the SiO2 nanoparticles to the pure water leads to about 41.53% enhancement of heat transfer rate for Ri=0.001.

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