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Effects of inclination angle on natural convection in enclosure partially heated and filled with cu-water nanofluid

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Abstract: Effects of inclination angle on natural convection heat transfer and fluid flow in an enclosure with partially heated side walls filled with cu-water nanofluid has been analyzed numerically. A two heat sources maintained at a constant heat flux q'' are embedded in the right and the left wall. The enclosure was cooled from the top and bottom walls. The remaining boundary parts are kept insulated. The angle of inclination is used as a control parameter for improve flow and heat transfer enhancement depicted in our recently work. Using a home developed code a parametric study is conducted and a set of graphical results is presented and discussed to illustrate the effects of inclination angle on the flow and heat transfer characteristics. The inclination angle was varied from 0 to 90° and two mainly case are considered, namely Middle-Middle (MM) and Down-Top (DT) depending of the two vertical sources location along the two side walls. It is found that optimal heat transfer enhancement can be improved. A percentage of heat transfer enhancement using 10% of Cu- nanoparticles is obtained for an inclination angle of 15° at higher Rayleigh number ($Ra=10^5$).

Key words: Nanofluid**,** Natural convection**,** Partially active walls**,** Inclination angle.

1. Introduction

Natural convection heat transfer is an important phenomenon in engineering and industry with widespread applications in diverse fields, such as, geophysics, solar energy, electronic cooling, micro-electromechanical systems, and nuclear energy to mention a few. 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. 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 [2–4]. However, little work has been carried out for the effect of the enclosure inclination angle. Abu-Nada and Oztop [5] studied the effects of inclination angle on natural convection in enclosures filled with Cu-water nanofluids. They found that inclination angle of the enclosure is proposed as a control parameter for fluid flow and heat transfer. They also concluded that lower heat transfer is formed

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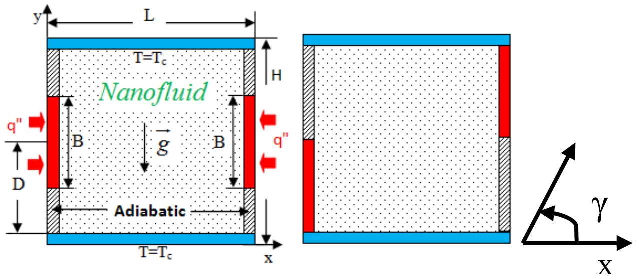
for γ = 90°. Effects of inclination angle on percentage of heat transfer enhancement become insignificant at low Rayleigh number. Recently, Abu-Nada and Chamkha [6] studied the effects of enclosure inclination angle on the Al_2O_3-water nanofluid flow characteristics. They showed that significant heat transfer enhancement can accentuate by inclination of the enclosure at moderate and large Richardson numbers. This present work is a continuation of our last work [7], in which we always seek benefit conditions leading to an optimal thermal transfer in partially heated enclosure filled with nanofluids. The active parts of the vertical left and right side walls are maintained at a constant heat flux q" well the inactive parts are kept insulated; however, the enclosure's top and bottom walls are cooled. Among the nine investigated cases, we keep two interest configurations, namely case DT (Down-Top) for which the rate of transfer is maximum and, in the contrary the case MM (Middle-Middle).

The main aim of this work is to study the effects of inclination angle on flow field and temperature distribution according to the two above cited cases DT and MM. Based on above literature survey and to the author's knowledge, no previous study on effects of inclination angle on natural convection in nanofluid filled enclosure with two partially heated side walls has not been studied yet.

2. Definition of model

A schematic diagram of the considered model is shown in Fig.1 with coordinates. It is a two-dimensional square enclosure of height H, inclination angle (γ) about the horizontal plane and filled with Cu-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 T_C. The remaining boundary parts of the enclosure are adiabatic. The Cu-water nanofluid is assumed to be Newtonian, in thermal equilibrium, and the nanoparticles are kept uniform in shape and size.



(a) Case MM **(b) Case DT**

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

3. Formulation

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:

$$X=\frac{x}{L}\ ,\ Y=\frac{y}{L}\ ,\ U=\frac{uL}{\alpha_f}\ ,\ V=\frac{vL}{\alpha_f}$$

$$P=\frac{(p+\rho_0 gy)L^2}{\rho_{nf}\alpha_f^2}\ ,\ \theta=\frac{T-T_C}{\Delta T}$$

$$Ra=\frac{g\beta_f \Delta T L^3}{\alpha_f \nu_f}\ ,\ \Delta T=\frac{q''L}{k_f}\quad and\quad Pr=\frac{\nu_f}{\alpha_f}$$

the dimensionless form of the governing equations for unsteady nanofluid flow can be written as [8],

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{1}$$

$$\frac{\partial U}{\partial t}+U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\Pr\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) + Ra\Pr\theta\sin\gamma \tag{2}$$

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

	Pure water	Copper(Cu)
$\rho(kgm^{-3})$	997.1	8933
$\beta(K^{-1})$	21×10^{-5}	1.67×10^{-5}
$k(Wm^{-1}K^{-1})$	0.613	401
$C_p(Jkg^{-1}K^{-1})$	4179	385

$$\frac{\partial V}{\partial t}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ra\Pr\theta\cos\gamma \quad (3)$$

and

$$\frac{\partial \theta}{\partial t}+U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{\alpha_{nf}}{\alpha_f}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \quad (4)$$

where the Rayleigh number Ra, and the Prandtl number Pr are defined as follows:

$$Ra=\frac{g\rho_f\beta_f\Delta T L^3}{\mu_f\alpha_f},\ and\ \Pr=\frac{\nu_f}{\alpha_f}$$

The boundary conditions consist of:

- $U=V=0$ on all four walls.
- $\theta=0\ for\ Y=0,1\ and\ 0\le X\le 1$
- $\frac{\partial\theta}{\partial Y}=0$
  $for\ X=0,1\ and\ 0\le Y\le(D-0.5B)$
  $1\ge Y\ge(D+0.5B)$
- $\frac{\partial\theta}{\partial Y}=-\frac{k_f}{k_{nf}}$
  $for\ X=0,1\ and\ (D-0.5B)\le Y\le(D+0.5B).$

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

$$Nu_s=\frac{hL}{k_f} \quad (5)$$

where h is the convection heat transfer coefficient:

$$h=\frac{q''}{T_s-T_c} \quad (6)$$

Rewrite the local Nusselt number by using the dimensionless parameters:

$$Nu_s(Y)=\frac{1}{\theta_s(Y)} \quad (7)$$

where θ_s is the dimensionless heat source temperature. The average Nusselt number (Nu_m) is determined by integrating Nu_s along the heat source

$$Nu_m=\frac{1}{B}\int_{D-0.5B}^{D+0.5B}Nu_s(Y)\,dY \quad (8)$$

4. Numerical approach

The governing equations (1) - (4) were numerically solved using the classical projection method [8]. 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. [9] 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 [10] whoever, the momentum equations are computed by a red black successive over-relaxation method.

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» [11].

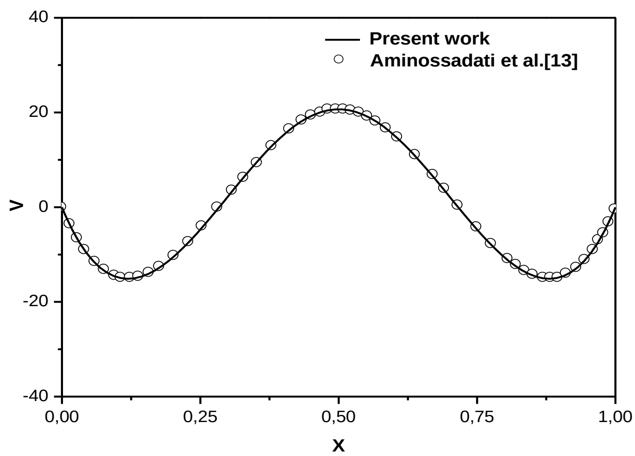


Fig.2: Vertical velocity profiles along the mid-section of the enclosure (Cu–Water, $\phi=0.1$, Ra=10^5, D = 0.5 and B =0.4).

The present numerical code was validated against the results of other natural convection studies in enclosures filled with nanofluid developed by Aminossadati et al. [12] in terms of computational data (see Table 3 in Reference [12]) and v- velocity component profile as shown in Fig.2 (Cu–Water, $\phi = 0.1$, Ra=10^5, D = 0.5 and B =0.4). Our results are in excellent agreement with those of [12].

5. Results and discussion

The streamlines and the isotherms for the Cu-water nanofluid ($\phi = 0.1$) for the two investigated case MM and DT with different inclination angles at Ra =$10^5$are shown in Fig.3and Fig.4 respectively. For the sake of comparisons, the streamlines and the isotherms for pure water ($\phi = 0$) are also shown in this figures. As can be seen from Figure 3, the structure of flow and the shape of the two counter rotating cells are sensitive to the inclination angle and addition of nanoparticles. In fact, isotherms indicate that the addition of nanoparticles becomes more effective for γ=90° due to the increasing of the buoyancy driven forces.

Thermal stratification in the core region is reduced with increasing inclination angle, resulting in a stronger flow field.

As showing in Fig.3 and Fig.4, the results prove the existence of two counter-rotating circulating cells for all different inclination angles according to the two investigated cases.

The variation of the local Nusselt number along the left heat source at different inclination angles is presented in Fig.5. As seen from this figure, for γ=0°, local Nusselt number decreases along the higher half of the heat source. Minimum of Nusselt number moves about the middle of the heat source by increasing the angle of inclination from γ=0° to γ=90°.

Fig. 5-(a) shows that the localization of minimum Nusselt number decreases from Y=0.648 to Y=0.5 while γ is varied from 0° to 90°.

Similarly, Fig. 5-(b) shows that the localization of minimum Nusselt number moves from Y=0.429 to Y=0.335 when γ is augmented from 0° to 90°. Fig. 6 presents the vertical velocity profiles along the mid-section of the enclosure. For the MM case, Fig.6-a

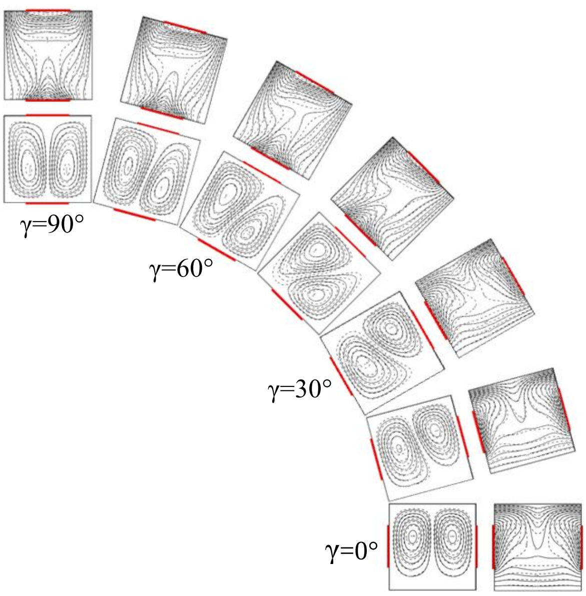


Fig.3: Streamlines (on the left) and isotherms (on the right) for different inclination angles of the cavity (from 0 to90°) at Ra=10^5, (----) for $\phi = 0.1$, (- - -) for $\phi = 0$, case MM.

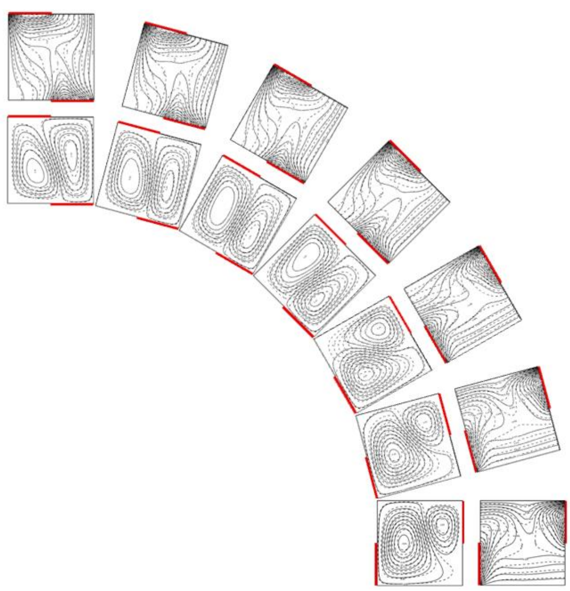


Fig.4: Streamlines (on the left) and isotherms (on the right) for different inclination angles of the cavity (from 0 to 90°) at Ra=10^5, (----) for $\phi = 0.1$, (- - -) for $\phi = 0$, case DT.

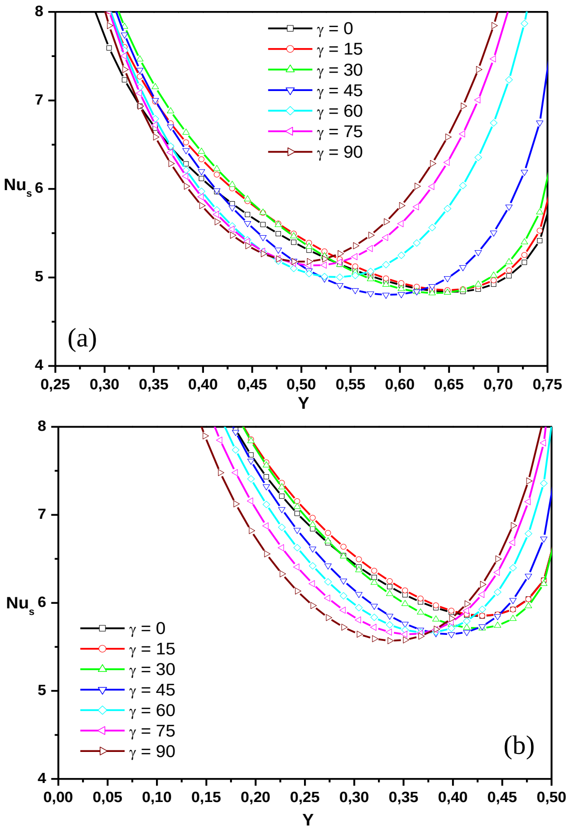


Fig.5: Variation of Nusselt number versus y coordinate along the left heat source at different inclination angles, for the MM case (a), and the DT case (b).

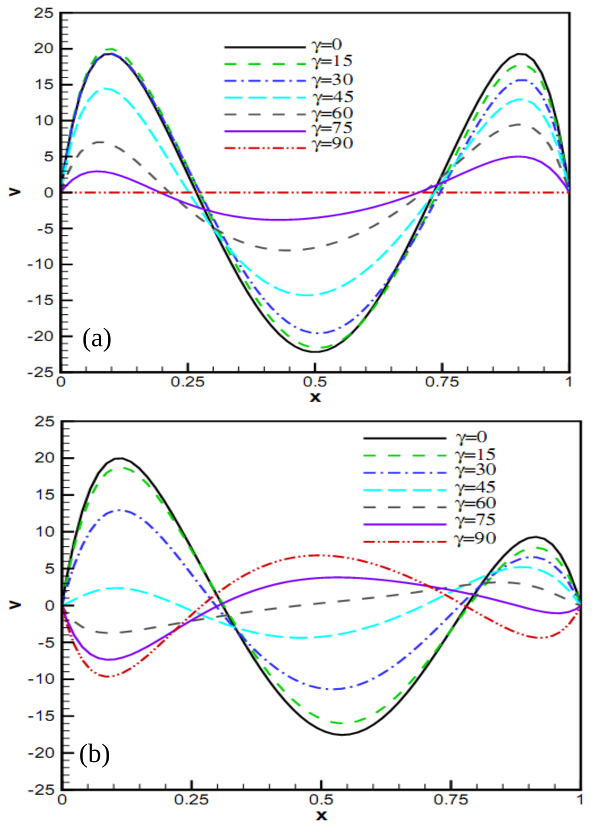


Fig.6: Vertical velocity profiles along the mid-section of the enclosure for different inclination angles for the MM case (a), and the DT case (b).

exhibits a symmetric behaviour about the cavity mid plane due to the symmetry of the heat sources about the mid plane. In addition, this subfigure shows that the maximum velocity decreases by increasing tilt angle.

Consider the case DT, Fig.6-b shows an asymmetric behaviour for all angles of inclination. Indeed, by increasing the inclination angles from 0° to 45° the maximum velocity at the left is greater than that on the right. This can be explained by the distance that the fluid needs to travel in the circulating cell to exchange heat between the heat sources and the top or bottom cold wall. For the range of angle inclination varied from 60° to 90°, the direction of movement is reversed and the maximum velocity increases again.

The enhancement of heat transfer rate can be controlled by the inclination angle as shown in Fig. 7 in terms of histograms.

When the MM case is concerned (Fig. 7a), enhancement rate increases by increasing tilt angle and a maximum of heat transfer enhancement of about 14% is achieved at an angle of inclination equal to 90 °. However, according to the DT case (Fig. 7b), a maximum of heat transfer enhancement of about 1.5% is reached at an angle of inclination equal to 15 °.

6. Conclusions

The influence of inclination angle on natural convection heat transfer and fluid flow in an enclosure with partially heated side walls filled with cu-water nanofluid has been analyzed. Results have clearly indicated that inclination angle of the enclosure is proposed as a control parameter for fluid flow and heat transfer enhancement. It is found that the inclination angle effects in terms of percentage of heat transfer enhancement at the left source become more pronounced at the MM case than that at the DT case. In fact, the present result shows that the DT case is enhanced, by comparison to our recently work, at an angle inclination of 15°.

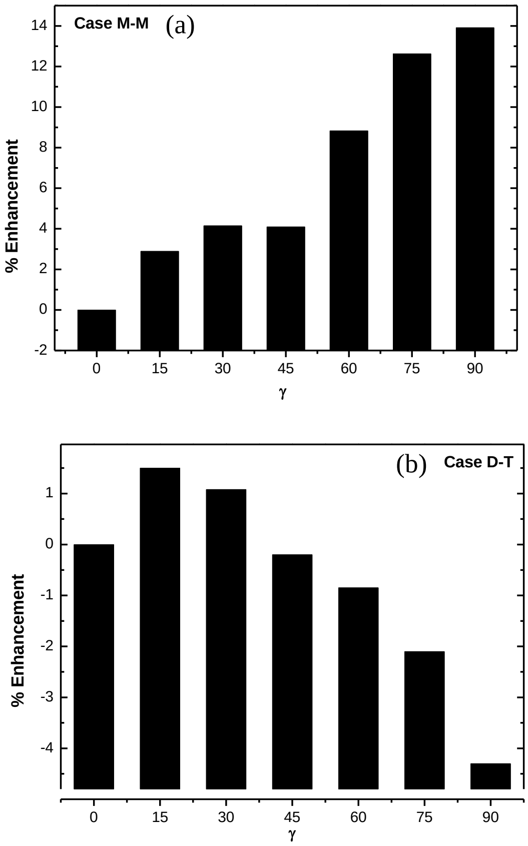


Fig.7: Histograms of enhancement of heat transfer due to variation of angle inclination for the MM case (a), and the DT case (b) at the left source.

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