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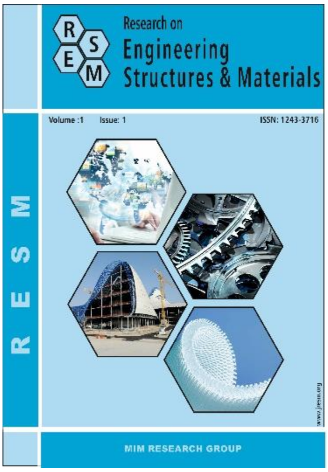

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Research Article

Mixed convection heat transfer of nanofluids in a trapezoidal cavity having an adiabatic square body at its center

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Abstract

In this study, the enhancement introduced by nanoparticles is investigated. Nanosized copper particles are dispersed into water which is confined in a cavity with a trapezoidal cross-section. The bottom and top walls of the cavity are adiabatic while the inclined lateral walls are kept at different constant temperatures. The cavity contains an adiabatic block at its center. The top wall of the cavity slides horizontally with a uniform constant velocity. The effect of adiabatic block size and particle volume fraction on the fluid flow and heat transfer is examined. Grashof number is set to 10^5 whereas Reynolds number is varied to examine the flow and heat transfer for Richardson numbers of 0.1, 1 and 10. The computational results show that heat transfer rate enhances as Richardson number decreases. It is also observed that large size of adiabatic block impedes the fluid circulation which reduces heat transfer rate.

1. Introduction

Mixed convection heat transfer in which both forced convection and natural convection are present is an often encountered process in many engineering applications such as metal casting, solar collectors, heat exchangers and cooling of electronic devices. In order to enhance heat transfer rate and energy management, in recent years, dispersing nano-sized solid particles into base fluids with low thermal conductivities have been employed extensively. Khanafer et al. [1] numerically investigated natural convection in a rectangular cavity filled with water-Cu nanofluids for various Grashof numbers and nanoparticle volume fractions and showed that the nanoparticles alters the structure of fluid flow and increases heat transfer rate significantly. It is also reported that heat transfer increases with the increase in volume fraction of Cu nanoparticles. Waheed [2] numerically studied mixed convection in a rectangular enclosure considering different Richardson numbers, Prandtl numbers and the length-to-height aspect ratio values and setting Reynolds number to 100. It was reported that while heat flux on the heated wall enhances as Richardson number or Prandtl number increases, it decreases as aspect ratio increases. Mahmood and Sebdani [3] investigated natural convection inside a square cavity having adiabatic square bodies at its center and filled with water-Cu nanofluids. They examined the effect of Rayleigh number, adiabatic square body size, nanoparticle volume fraction on the fluid flow and heat transfer. It was reported that as nanoparticle volume fraction increases average Nusselt number increases for all Rayleigh numbers except for $Ra=10^4$ where heat transfer rate is attenuated. Also, it was concluded that as the adiabatic block body size increases, the heat transfer rate decreases for $Ra=10^3$ and $Ra=10^4$ and increases for $Ra=10^5$ and $Ra=10^6$. Recently, a great amount of attention has been paid to investigate heat transfer of nanofluids in trapezoidal enclosures which are

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encountered in various engineering fields. A typical example of such a geometry is the concentrating solar collectors which have reflective side walls, absorbing bottom surface and top cover plate [4]. Saleh et al. [4] studied free convection of a nanofluid-filled trapezoidal enclosure considering different parameters such as Grashof number, nanoparticle material, nanoparticle volume fraction and inclination angle of inclined wall of the enclosure. They concluded that the most effective way to enhance heat transfer rate is attained with the combination of acute sloping wall and Cu nanoparticle with high volume fraction. A similar study is conducted by Nasrin and Parvin [5] where free convection heat transfer in a trapezoidal cavity filled with water–Cu nanofluid is investigated. The effect of various Rayleigh numbers, Prandt numbers, aspect ratios (ratio of base length and height) and nanoparticle volume fractions on the heat transfer was studied. It was reported that the most improvement in heat transfer rate is obtained for Cu nanoparticles with the highest Prandtl number and the lowest aspect ratio. Esfe et al. [6] studied numerically free convection heat transfer inside a trapezoidal enclosure filled with carbon nanotube–EG–water nanofluid for different Rayleigh numbers, nanoparticle volume fractions and aspect ratios. It was shown that as the inclination angle increases the mean Nusselt number decreases at low Rayleigh numbers while it increases until 30^o of inclination angle and then decreases for high Rayleigh numbers. Hasib et al. [7] examined numerically the effect of tilt angle of the cavity on mixed convection of water-Al_2O_3 nanofluid inside two different lid-driven trapezoidal cavities. They reported that as the inclination angle increases, heat transfer rate from heated wall increases for short based trapezoids and decreases for long based trapezoids.

The main aim of the present study is to investigate fluid flow and heat transfer of nanofluids filled in a trapezoidal cavity having a square adiabatic block at its center. The effect of nanoparticle volume fraction together with Richardson number and size of adiabatic block on the fluid flow and heat transfer is examined.

2. Mathematical Formulation and Numerical Model

In this study, the mixed convection in a 2D trapezium enclosure of side length L and height H, and containing an adiabatic square block (length of side *l*) at its center is considered. The enclosure is filled with Cu-water nanofluid. The trapezoid has a unity aspect ratio (ratio of height to length). Horizontal walls of the enclosure are assumed to be adiabatic and isothermal inclined lateral walls are maintained at different constant temperatures (T_c=295 K and T_h=305 K). The top wall of cavity moves from left to right with a constant velocity of U. The geometry and boundary conditions are depicted schematically in Fig. 1.

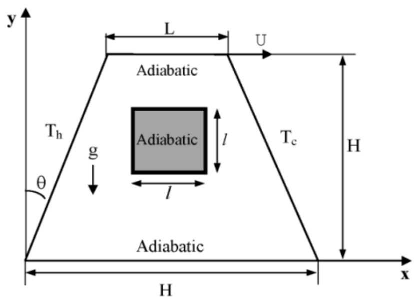

Fig. 1. Schematic representation of the problem and boundary conditions

2.1. Thermophysical properties

It is assumed that the flow is steady, two-dimensional, laminar and incompressible, and viscous dissipation is negligible. Single phase approach which assumes both the fluid phase and nano-sized solid particles are in thermal equilibrium state and flow at the same velocity is employed. Thermophysical properties of the materials are given in Table 1. Thermophysical properties except density are assumed to be constant. Boussinesq approximation is employed to account for the variations of density thus buoyancy terms in the momentum equation.

Table 1 Thermophysical properties of materials

Material	Density (ρ) (kg/m^3)	Heat Capacity (C_p) (J/kgK)	Thermal Conductivity (k) (W/mK)	Thermal Expansion (β) (1/K)	Dynamic Viscosity (μ) (Pa.s)
Cu	8954	383	400	16.7×10^{-6}	-
Water	997.1	4179	0.6	210.0×10^{-6}	0.00091

2.2. Governing equations

The continuity, momentum and energy equations based on the assumptions given above can be expressed in non-dimensional form as:

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

$$u^*\frac{\partial u^*}{\partial x^*} + v^*\frac{\partial u^*}{\partial y^*} = -\frac{\rho_{bf}}{\rho_{nf}}\frac{\partial p^*}{\partial x^*} + \frac{v_{nf}}{v_{bf}}\frac{1}{Re}\left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}}\right) \tag{2}$$

$$u^*\frac{\partial v^*}{\partial x^*} + v^*\frac{\partial v^*}{\partial y^*} = -\frac{\rho_{bf}}{\rho_{nf}}\frac{\partial p^*}{\partial y^*} + \frac{v_{nf}}{v_{bf}}\frac{1}{Re}\left(\frac{\partial^2 v^*}{\partial x^{*2}} + \frac{\partial^2 v^*}{\partial y^{*2}}\right) + \frac{(1-\phi)\rho_{bf}\beta_{bf} + \phi\rho_p\beta_p}{\rho_{nf}\beta_{bf}} Ri.\theta \tag{3}$$

$$u^*\frac{\partial \theta}{\partial x^*} + v^*\frac{\partial \theta}{\partial y^*} = \frac{\alpha_{nf}}{\alpha_{bf}}\frac{1}{Re\ \ Pr}\left(\frac{\partial^2 \theta}{\partial x^{*2}} + \frac{\partial^2 \theta}{\partial y^{*2}}\right) \tag{4}$$

The parameters employed for the non-dimensionalization of governing equations are given in Eq.(5) and Eq.(6).

$$x^* = \frac{x}{H}, y^* = \frac{y}{H}, u^* = \frac{u}{U}, v^* = \frac{v}{U}, p^* = \frac{p}{\rho_{bf}U^2}, \theta = \frac{T - T_c}{T_h - T_c} \tag{5}$$

Reynolds (Re), Prandtl (Pr), Grashof (Gr) and Richardson (Ri) numbers are defined as:

$$Re = \frac{UL\rho_{bf}}{\mu_{bf}},\ \ Pr = \frac{v_{bf}}{\alpha_{bf}},\ \ Gr = \frac{g\beta_{bf}H^3(T_h - T_c)}{{v_{bf}}^2},\ \ Ri = \frac{Gr}{Re^2} \tag{6}$$

In these equations, x^* and y^* are non-dimensional space coordinates, u^* and v^* are non-dimensional form of the velocity components, p^* is non-dimensional pressure, θ is non-dimensional temperature, ρ is fluid density at T_c temperature, α is the thermal diffusivity of fluid, ν is the kinematic viscosity of the fluid, β is thermal expansion coefficient and g is acceleration of the gravity which is varied to obtain the desired Gr number. Here, ϕ is the solid particle volume fraction and n_f, b_f and s subscripts represents nanofluid, base fluid and solid particles, respectively. The density, heat capacity (C_p), thermal expansion coefficient, thermal diffusivity, dynamic viscosity (μ) and thermal conductivity (k) of the nanofluid are calculated by the following equations, respectively [7];

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_p \tag{7}$$

$$\left(\rho C_p\right)_{nf} = (1-\phi)\rho_{bf}c_{p_{bf}} + \phi\rho_p c_{p_p} \tag{8}$$

$$(\rho\beta)_{nf} = (1-\phi)\rho_{bf}\beta_{bf} + \phi\rho_p\beta_p \tag{9}$$

$$\alpha_{nf} = \frac{k_{eff}}{\left(\rho c_p\right)_{nf}} \tag{10}$$

$$\mu_{nf} = \frac{\mu_{bf}}{(1-\phi)^{2.5}} \tag{11}$$

$$\frac{k_{nf}}{k_{bf}} = \frac{k_s + 2k_{bf} - 2\phi\left(k_{bf} - k_s\right)}{k_s + 2k_{bf} + \phi\left(k_{bf} - k_s\right)} \tag{12}$$

The local Nusselt number (Nu_l) for the heated wall can be described as;

$$Nu_l = -\frac{k_{nf}}{k_{bf}}\vec{\nabla}\theta.\vec{n}\bigg|_{hot\ wall} \tag{13}$$

where $\boldsymbol{n}$ is surface normal vector. Mean Nusselt number is obtained from integration of local Nusselt number along the heated wall of the cavity:

$$Nu = \frac{1}{A_{hot\ wall}}\int_{A_{hot}} Nu_l\, dA \tag{14}$$

Second order upwind scheme is used to discretize the convective terms in momentum and energy equations. The pressure-velocity equation is coupled by SIMPLE algorithm. The governing equations together with the boundary conditions are solved by a finite volume based program, ANSYS Fluent 15.0.

3. Grid independency and Model Validation

A grid sensitivity study was conducted to assure a grid independent solution by observing mean Nusselt number. Finally, a grid which yields insignificant change in Nusselt number with 8665 cells was chosen.

The validation of numerical model was done in two steps. At first, mixed convection in a lid-driven cavity with insulated vertical walls and isothermal horizontal walls (the upper horizontal wall is at higher temperature than the lower one) is solved for various

Reynolds numbers and compared with previously published results in Table 2. As seen from the table, the mean Nusselt number calculated in the present study agree pretty well with the previous works.

The numerical model is further validated in Fig.2 with the results of Nasrin and Salma [5] where natural convection of water–Cu nanofluid in a trapezoidal enclosure is studied. The top and bottom walls of the enclosure are insulated while the inclined lateral walls are maintained at different temperatures. As seen in the figure, results for the mean Nusselt numbers show good overall agreement.

Table 2 Comparison of mean Nusselt number of the present study with those of previously published works.

Re	Present study	Alinia et al.[8]	Waheed[2]	Khanafer et al.[9]	Iwatsu et al.[10]
100	2.00	2.02	2.03	2.02	1.94
400	4.01	4.03	4.02	4.01	3.84
1000	6.37	6.48	6.55	6.42	6.33

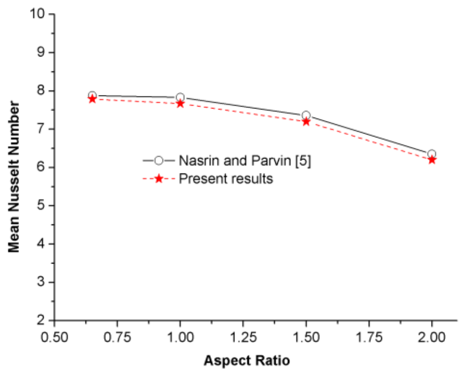

Fig. 2. Comparison of the mean Nu numbers of the present study with those of Nasrin and Salma [5] ($Ra=10^5$)

4. Results and Discussions

Computations are carried out for various Richardson numbers (0.1, 1 and 10), adiabatic block sizes (l/H=0.30, 0.35 and 0.40) and nanoparticle volume fractions (ϕ=0%, 5% and 10%). The results are presented in terms of streamline and isotherm patterns. The effect of adiabatic square block size on the flow and non-dimensional temperature (θ) fields is shown in Fig. 3 and Fig. 4, respectively for different Richardson numbers. As seen in these figures, the heated fluid rises up along the hot wall then moves horizontally and falls down along the cooling wall which creates a primary clockwise cell in the enclosure for all the cases considered in the study. Besides, a secondary cell is developed between the adiabatic block and top wall for all the cases. As Richardson number increases, the size of secondary cell decreases. It is also observed that both hydrodynamic and thermal boundary layer become thicker by increasing Richardson number. Increasing the size of adiabatic block impedes the flow to entrain the lower part of the cavity.

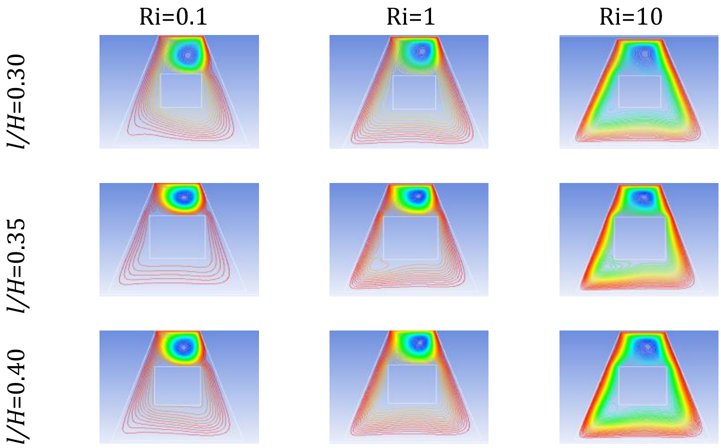

Fig. 3. Streamlines for different Ri numbers and adiabatic block sizes

(Gr=10^5, ϕ=5%)

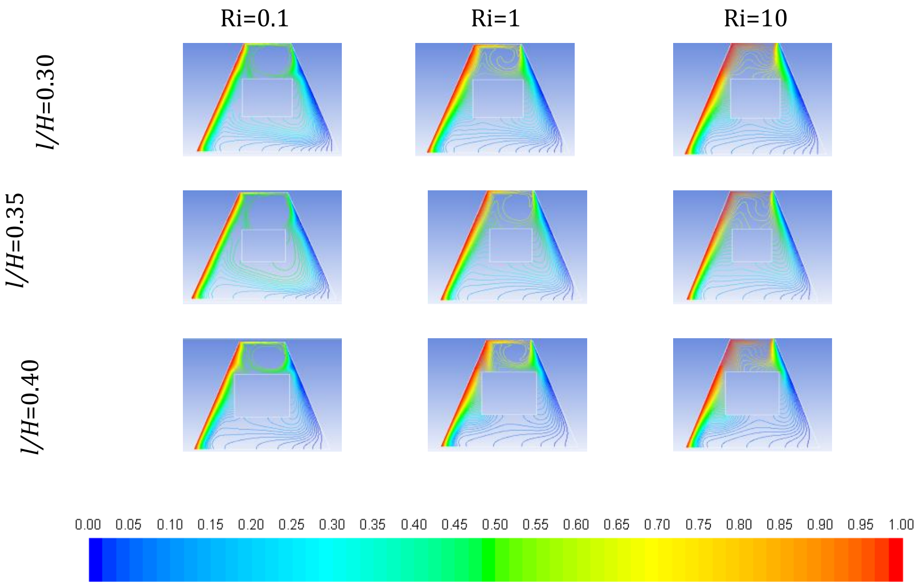

Fig. 4 Non-dimensional temperature distribution for different Ri numbers and adiabatic block sizes (Gr=10^5, ϕ=5%)

The computed mean Nusselt number on the hot wall for the governing parameters considered in this study is presented in Fig. 5. As seen in the figure, as Richardson number increases the mean Nusselt number decreases since the effect of forced convection diminishes and flow and thermal fields are dominated mostly by the natural convection. Also, Nusselt number decreases with the increase of adiabatic block size since the adiabatic block impedes the entrainment of the induced flow to lower part of the cavity. With addition of Cu nanoparticles to water, Nusselt number increases for all the cases studied. The improvement is almost at the same level regardless of Richardson number. It is interesting to note that Nusselt number decreases slightly with the increase of adiabatic block size (increasing from l/H=0.30 to l/H=0.35). However with the further increase in the adiabatic block size, Nusselt number decreases considerably.

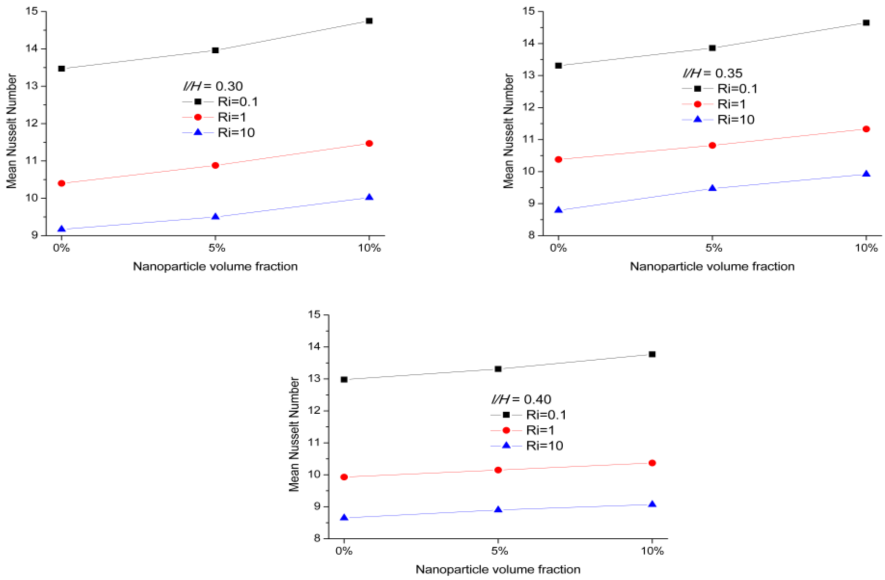

Fig. 5 The computed mean Nusselt numbers for different nanoparticle volume fractions, Richardson numbers and size of adiabatic blocks

5. Conclusions

In this work, mixed convection of water-Cu nanofluid in a two dimensional trapezium enclosure is investigated. The top wall of the enclosure which is moving on its plane at a constant speed and the bottom wall are insulated while the lateral inclined sides are maintained at constant different temperatures. An adiabatic square block is located at the center of the cavity. A parametric study is performed for various Richardson numbers (Ri=0.1, 1 and 10), adiabatic block sizes (l/H=0.30, 0.35 and 0.40) and Cu nanoparticle volume fractions (ϕ=0%, 5% and 10%). In the light of the results obtained in this study, the following conclusions can be drawn:

- Heat transfer is enhanced considerably with the addition of Cu nanoparticles to water.
- Heat transfer enhances by the decrease in Richardson number.
- For the large adiabatic block sizes, heat transfer reduces significantly regardless of Richardson number and solid particle volume fraction.

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