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Effect of Al_2O_3/water Nanofluid on Conjugate Free Convection in a Baffle Attached Square Enclosure

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Nomenclature

T_h is temperature of left hot wall, K; T_c is temperature of right cold wall, K; L is length of the enclosure; B_l is length of the baffle; B_p is position of the baffle; B_t is thickness of the baffle; g is acceleration due to gravity; c_p is specific heat, J/(kgK); k is thermal conductivity, W/(m K); k_r is conductivity ratio (k_B/k_{nf}); Ra is Rayleigh number; Pr is Prandtl number; Nu is Nusselt number; X, Y is dimensionless coordinates; ϕ is solid volume fraction; ρ is density, kg/m^3; β is thermal expansion coefficient, 1/K; α is thermal diffusivity, m^2/s; μ is dynamic viscosity, kg/(m s); ν is kinematic viscosity, m^2/s; θ is temperature distribution; η_B os efficiency of the baffle.

Subscripts: *nf* is nanofluid; *f* is base fluid; *s* is solid particle; *av* is average.

1. Introduction

Numerous applications of free convection in differentially heated enclosures can be seen in many industrial applications such as cooling of electronic gadgets, solar collectors, energy storage systems, nuclear reactors, smoke modelling and double glass windows. Many researchers have studied free convection in differentially heated enclosures extensively for different input parameters like aspect ratios, Rayleigh number, fin length, finite fin thickness or finite wall thickness.

The pioneering work on numerical simulation of natural convection inside enclosure is by De Vahl Davis [1] Bilgen [2] examined by inserting a thin fin on the active wall inside enclosure and found considerable effect on both flow field and heat transfer. Abdullatif Ben-Nakhi et al. [3] studied differentially heated cavity with multiple fins attached to both the vertical walls. Decrease in temperature gradient and Nusselt number is observed with increase in number and length of the fins. Abdennacer Belazizzia et al. [4] examined the effect of conduction in the wall on the free convection. Average Nusselt number was observed to increase with either increase in thermal conductivity ratio or Rayleigh number. Costa [5] considered two adiabatic partitions and reported that the different combination of length and position of the partition affect the heat transfer performance. Kalidasan et al. [6] carried out a numerical simulation on free convection inside an open enclosure with partitions. The combined effect of presence of partitions on horizontal insulated walls and overhanging transverse baffle on heat transfer performance of the enclosure was found to be significant. Numerical study of free convection inside a square enclosure with thin baffle attached to the isothermal cold wall was reported by Nagasubramanian et al. [7]. The results show that the reduction in average Nusselt number was observed on the cold wall with the baffle when compared with the hot wall. Ahmed Mahmoudi et al. [8] considered a single horizontal fin attached to the isothermal hot wall of a square enclosure and reported that increase in fin length results in enhancement of fin effectiveness.

All the above studies have been done with the fluid inside the enclosure as air. Hence, later on, various other studies have been done with simple geometry but with a different fluid instead of air like nanoparticle added in a definite proportion with base fluid like water to enhance the heat transfer rate. Eiyad Abu-Nada and Ali J. Chamkha [9] analyzed the effect of CuO-EG-water nanofluid on the enhancement of heat transfer performance of a differentially heated enclosure. Amin Habibzadeh et al. [10] examined the effect of Al_2O_3/water nanofluid on free convective heat transfer of a partitioned square enclosure. The position and height of the partition are found to affect flow field and heat transfer significantly, the effect being more significant at higher Rayleigh numbers. For the range of volume fraction of the nanofluid considered in the study, the average Nusselt number was found to increase with increase in Rayleigh number. Alumina-water and Copper-water nanofluids were found to cause significant enhancement of heat transfer in the case of differentially heated and bottom heated enclosures as reported by Abbasian Arani et al. [11]. In a numerical study of natural convection inside an enclosure filled with nanofluids by Ibtissam El Bouihi and Rachid Sehaqui [12], the heat transfer was found to increase with increase in volume fraction of the nanofluids for a given Rayleigh number. In a numerical study of natural convection in enclosures filled with Au, Al_2O_3, Cu and TiO_2 water based nanofluids by Ternik and Rudolf [13], the average Nusselt number was found to be a function of Rayleigh number and volume fraction of the nanofluids. Cong Qi et al. [14] investigated the effect of Al_2O_3/H_2O nanofluid on free convective heat transfer in a differentially heated square enclosure and found that nanofluid enhances heat transfer. Bhramara Panitapu et al. [15] studied the numerical investigation of differentially heated square cavity with Fe_3O_4 based nanofluid of various volume fractions. The results show that for increasing volume fractions there is certain rate of increase in heat transfer rate. Marta Cianfrini et al [16] analyzed the effect of size of the nanoparticles on heat transfer inside an enclosure filled with Al_2O_3/water nanofluid and reported that higher the size of the nanoparticles better is the heat transfer. Zoubair Boulahia et al. [17] examined the combined effect of copper-water nanofluid and a cold obstacle on heat transfer performance of a square enclosure. It was reported that heat transfer increases with increase in Rayleigh number, volume fraction of nanoparticles and the length of the obstacle. All these studies assumed the baffles / fins to be at a uniform temperature. That is conjugate heat transfer has not been analysed.

As far as conjugate heat transfer is considered, Ishak Hashim and Ammar Alsabery [18] examined conjugate free convection in a square enclosure filled with nanofluid and heated from below. With decrease in thermal conductivity ratio, the average Nusselt number is found to increase. Ahmed Elatar et al. [19] analyzed the fin efficiency and effectiveness by considering a single horizontal fin attached to the isothermal hot wall of a square enclosure. When compared to studies on enclosures with isothermal thin fins / baffles, studies on enclosures with thick fins / baffles (conjugate heat transfer) is scarce.

To the best knowledge of the authors, no work has been reported in the literature to address the combined effect of thick fins / baffles (conjugate heat transfer) and nanofluids on free convective heat transfer in a square enclosure. This is the motivation for the present work. The present work aims to investigate numerically conjugate free convection heat transfer of Al_2O_3/water nanofluid inside a square enclosure with a thick baffle attached to its hot wall.

2. Mathematical model

Fig. 1 shows the 2-D square enclosure investigated the present work. The left vertical wall is isothermally hot. The right vertical wall is isothermally cold, and the top and bottom walls are insulated. A baffle of length B_l and thickness B_t is attached to the hot wall at a position B_p from the bottom adiabatic wall. Conjugate heat transfer takes place between the baffle and Al_2O_3/water nanofluid, which is assumed as an incompressible and Newtonian fluid. Fluid phase and nanoparticles are assumed to be in thermal equilibrium with each other. The Boussinesq approximation is assumed to be valid, which considers density variation only in the buoyancy term.

2.1. Governing equations

Numerical investigation of free convective heat transfer is analyzed inside a square enclosure using water-based Al_2O_3 nanofluid as the working medium. Isothermal and no slip boundary conditions were imposed on both hot and cold wall, other walls are assumed as adiabatic.

Heat transfer is assumed to be laminar and steady state, with constant fluid properties except for buoyancy. The governing equations for the previous assumptions are solved computationally using finite volume based FLUENT software.

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

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

$$\rho_{nf}\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu_{nf}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+\rho_{nf}\, g\beta\left(T-T_c\right), \tag{3}$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha_{nf}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right). \tag{4}$$

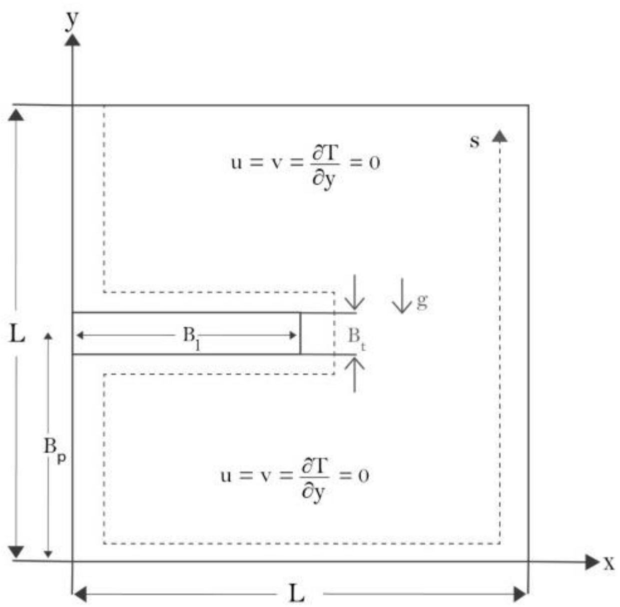


Fig. 1 Geometrical configuration

2.2. Boundary conditions

The governing equations for the present problem are elliptic in nature and thus would need the specification of the boundary conditions on all the four boundaries. In this study, all the four sides and the baffle are solid boundaries:

- left wall is maintained at high temperature T_h;
- right wall is maintained at low temperature T_c;
- top and bottom walls are adiabatic;
- a baffle is attached to the left hot wall with the length B_l and thickness B_t;
- no slip on all four walls and baffle.

2.3. Physical properties of nanofluid

The density of nanofluid has been estimated using the following formula developed for conventional solid-liquid mixtures:

$$\rho_{nf}=\left(1-\varphi\right)\rho_f+\varphi\rho_s, \tag{5}$$

where: φ refers to the volume fraction of nanoparticles and the subscripts s, f, nf denote the solid particle, base fluid and nanofluid respectively. The specific heat is obtained as:

$$\left(\rho c_p\right)_{nf} = (1-\varphi)\left(\rho c_p\right)_f + \varphi\left(\rho c_p\right)_s. \tag{6}$$

The thermal expansion coefficient of a nanofluid can be determined by:

$$\left(\rho\beta\right)_{nf} = (1-\varphi)\left(\rho\beta\right)_f + \varphi\left(\rho\beta\right)_s. \tag{7}$$

Thermal diffusivity of the nanofluid is:

$$\alpha_{nf} = \frac{k_{nf}}{\left(\rho c_p\right)_{nf}}. \tag{8}$$

The effective thermal conductivity of nanofluid is:

$$\frac{k_{nf}}{k_f} = \frac{k_s + 2k_f - 2\varphi\left(k_f - k_s\right)}{k_s + 2k_f + \varphi\left(k_f - k_s\right)}. \tag{9}$$

The dynamic viscosity of the nanofluid can be given as:

$$\mu_{nf} = \frac{\mu_f}{(1-\varphi)^{2.5}}. \tag{10}$$

The Rayleigh and Prandtl numbers are define by:

$$Ra = \frac{g\beta_{nf}\left(T_h - T_c\right)L^3}{\nu_{nf}\alpha_{nf}}, \tag{11}$$

$$Pr = \frac{\nu_{nf}}{\alpha_{nf}}. \tag{12}$$

The local Nusselt number is defined by:

$$Nu(y) = -\left(\frac{k_{nf}}{k_f}\right)\frac{\partial T(y)}{\partial x}\frac{L}{T_{x=0} - T_c}. \tag{13}$$

And the average Nusselt number is:

$$Nu_{av} = \int_0^L Nu(y)\,dy. \tag{14}$$

3. Solution procedure

Governing equations of steady, 2-D laminar, Boussinesq approximation models for free convective heat transfer analysis has been solved using finite volume method based FLUENT software. Simple algorithm is employed in carrying out the analysis with default values for relaxation factor. Convergence criterion was set as 10^{-6} for continuity, x-momentum, y-momentum and energy. The uniform fine mesh was created inside the square enclosure. Properties for pure water and Al_2O_3 water-based nanofluids are considered. The thermo-physical properties of water and nanoparticles considered in the present work are shown in Table 1. Fig. 2 presents the convergence and stability of the solution. 80000 iterations were found to be enough to reach the stable solution.

In order to test the grid independency of the numerical simulation carried out, a differentially heated square cavity filled with air (Pr = 0.707) at $Ra = 10^6$ is considered and the numerical results for different uniform grid sizes, namely, 81 x 81, 121 x 121 and 151 x 151, are obtained. These results are presented in Table 2. A grid system of 151 x 151 was used for all the calculations.

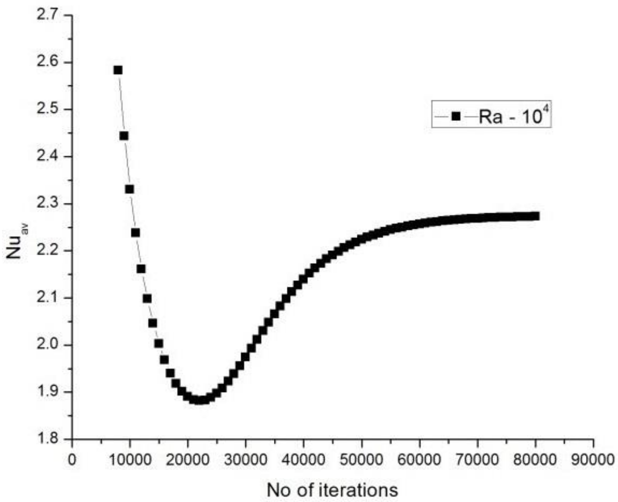


Fig. 2 Effect of no of iterations

4. Validation and comparison with previous research

As a part of validation, the present work is compared with published works for average Nusselt number in the case of a square enclosure without baffle as shown in Table 3. In addition, comparison is made for average Nusselt number for the case of square enclosure with a thick baffle with the results presented by Nag et al. [24] and Elatar et al. [19].

The obtained numerical results of the present study are found to be in good agreement with the published results as seen in Table 4. Furthermore, the average Nusselt number results presented by Ternik and Rudolf for natural convection flow of Al_2O_3 water-based nanofluid in a square enclosure without baffle. The present Nusselt numbers are found to be in good agreement with the published results as shown in Table 5.

Table 1

Thermo-physical properties of water and the nanoparticles (Abbasian Arani et al. [11])

Physical properties	Water	Al_2O_3
c_p, J / kg K	4179	765
P, kg / m^3	997.1	3970
k, W / m K	0.613	40
β, K^{-1}	21 x 10^{-5}	0.85 x 10^{-5}
μ, kg / m s	10.03 x 10^{-4}	---

Table 2

Grid independence study for $Ra = 10^6$, Pr = 0.707

Grid size	Nu_{av}	% difference
81 x 81	8.997	---
121 x 121	8.898	1.106
151 x 151	8.868	0.338

Table 3

Comparison of average Nusselt number on cold wall

Ra	10^4	10^5	10^6
Nag et al. [20]	2.24	4.51	8.82
Shi and Khodadadi [21]	2.247	4.532	8.893
Ben-Nakhi and Chamkha [22]	2.244	4.524	8.856
Tasnim and Collins [23]	2.244	4.5236	8.8554
Ahmed Elatar et al. [19]	2.234	4.517	8.948
Present work	2.245	4.524	8.868

Table 4

Comparison of average Nusselt number on the cold wall for k_r = 7750, B_l = 0.2 and $Ra = 10^6$

B_t	0.02	0.04	0.1
Nag et al. [24]	8.861	8.888	9.033
Elatar et al. [19]	8.672	8.71	8.947
Present work	8.715	8.816	9.014

Table 5

Comparison of average Nusselt number on right or left wall for Al_2O_3 water-based nanofluid for $Ra = 10^5$, φ = 0 and 0.06

φ	0	0.06
Ternik. P and Rudolf. R [14]	4.7	5.6
Present work	4.7034	5.5766

5. Results and discussions

A detailed parametric study has been carried out to analyze the combined effect of conjugate heat transfer and nanofluid inside a square enclosure with the baffle position on the isothermal hot vertical wall with Rayleigh number, baffle length, baffle position, baffle thickness, baffle conductivity ratio and volume fraction of Al_2O_3/water nanofluid, as shown in Table 6. Results of numerical simulation are reported in the form of flow fields, isotherms, efficiency of baffle and Nusselt number.

Figs. 3 and 4 represents the flow fields and isotherms inside enclosure with three different baffle lengths (B_l = 0.25, 0.5, 0.75), three different baffle positions (B_p = =0.25, 0.5, 0.75) for Rayleigh number 10^4, baffle conductivity ratio (k_r = 100) and baffle thickness (B_t = 0.04). It is evident from Fig. 3 that the vortex is blocked by the baffle below and above the surface of the baffle for B_p = 0.25 and 0.75 respectively.

Table 6

Parameters and its ranges

Parameters	Range
Rayleigh number *Ra*	10^4, 10^5 and 10^6
Baffle length B_l	0.125 to 0.875
Baffle thickness B_t	0.02 to 0.1
Thermal conductivity ratio k_r	10, 100 and 1000
Volume fraction φ	0 to 0.2

Further, it is understood that longer the baffle higher is the weakening of the vortex due to blockage effect irrespective of baffle position. Increase in Rayleigh number strengthens the vortex (figures not shown). At the highest Rayleigh number ($Ra = 10^6$) studied, vortex is found to be so strong as to overcome the blockage effect due to baffle.

It can be inferred from the isotherms, shown in Fig. 4, that the region above the baffle for the position B_p = =0.75 has low concentration of isotherms indicating low heat transfer. On the other hand, one can observe dense isotherms in the upper half along the right cold wall, which shows higher heat transfer in this portion.

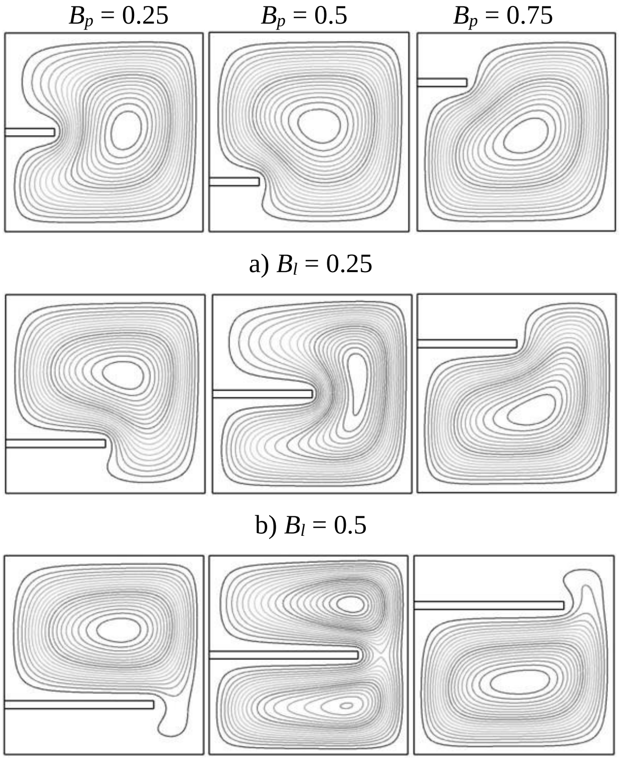


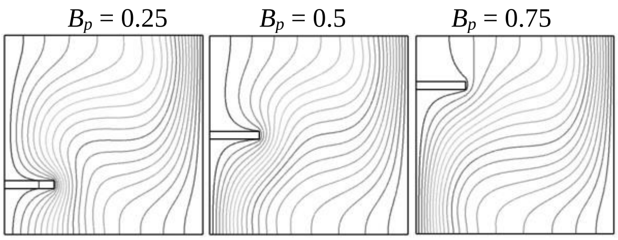
Fig. 3 Streamline at $Ra = 10^4$

This effect is strongly felt for the baffle length B_l = =0.75. It is observed (figures not shown) that increase in Rayleigh number causes denser isotherms along the left hot wall and right cold wall, indicating thinner boundary layer and higher heat transfer rate. The effect of thermal conductivity ratio on temperature distribution along the baffle in the presence of nanofluid is illustrated in Fig. 5 for the case of B_p = 0.5 and $Ra = 10^6$, B_t = 0.04 and volume fraction of the nanofluid φ = 0.1. The temperature of the baffle at any location is found to increase with increase in conductivity ratio k_r. This is because higher the value of thermal conductivity ratio k_r smaller is the conductive resistance of the baffle, leading to higher baffle temperature at any location along the length of the baffle. Also it is seeing that the baffle temperature decreases along the baffle length irrespective of thermal conductivity ratio k_r considered. When compared to thermal conductivity ratio k_r = 1000, the rate at which baffle temperature decreases is higher for thermal conductivity ratios k_r = 10 and 100, as evident by the dimensionless baffle tip temperatures of 0.57, 0.74 and 0.95 for thermal conductivity ratios k_r = 10, 100 and 1000 respectively.

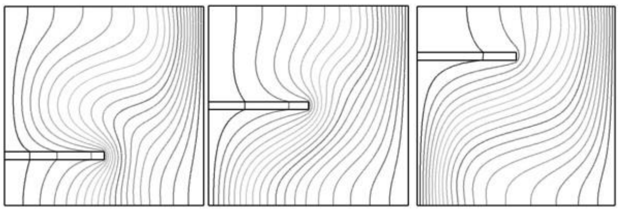
The efficiency of the baffle is defined as the ratio of heat transferred from the baffle in the actual case to the heat transferred from the baffle assuming uniform temperature on along the length of the baffle (θ_B = 1), as defined below.

$$\eta_B = \frac{Q_{real}}{Q_{isothermal}}. \tag{15}$$

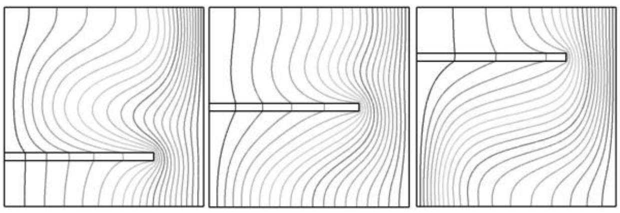
Fig. 6 presents the efficiency of the baffle as a function of baffle length for (Ra = 10^6, B_p = 0.5, B_t = 0.04) for different thermal conductivity ratios of baffle. With increase in baffle length, baffle efficiency is found to decrease due to the decrease in baffle temperature. This decrease in baffle efficiency is more pronounced for lower values of thermal conductivity ratios (k_r = 10 and 100).



a) B_l = 0.25



b) B_l = 0.5



c) B_l = 0.75

Fig. 4 Isothermal lines at Ra = 10^4

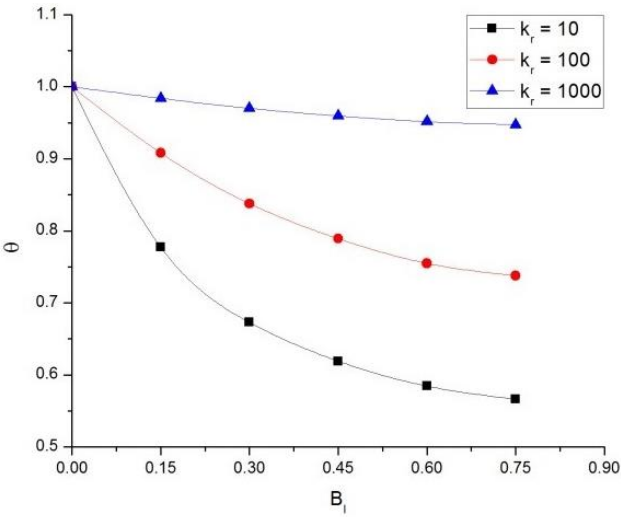


Fig. 5 Baffle temperature distribution for (Ra = 10^6, B_p = =0.5, B_t = 0.04, φ = 0.1)

Fig. 7 is a plot of local Nusselt number versus height of the hot wall for different fluids (air, water and Al_2O_3/water nanofluid). It is observed that the local Nusselt number is high near the bottom of the hot wall (Y = 0), as there exists high temperature difference between wall and the fluid. Among three fluids studied, Al_2O_3/water nanofluid exhibits highest Nusselt number while air exhibits the lowest Nusselt number. Fig. 8 is a plot of the average Nusselt number against the Rayleigh number for different fluids (air, water and Al_2O_3/water nanofluid). Results indicate, as expected, that mean Nusselt number increases with the increase of Ra and the highest values found for Al_2O_3/water nanofluid.

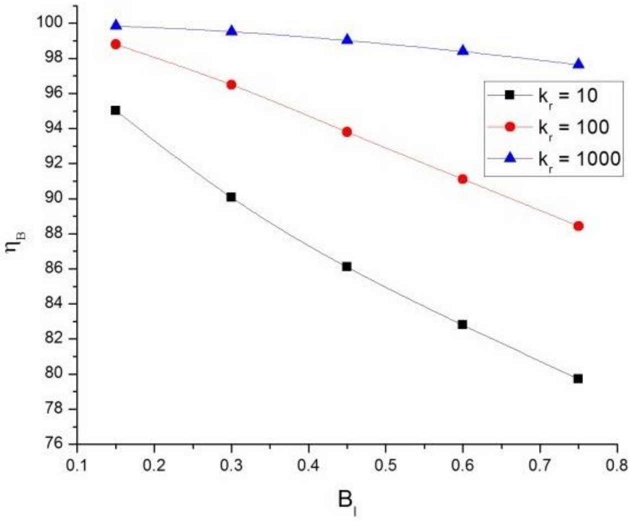


Fig. 6 Baffle efficiency as a funtion of baffle length for (Ra = 10^6, B_p = 0.5, B_t = 0.04, φ = 0.1)

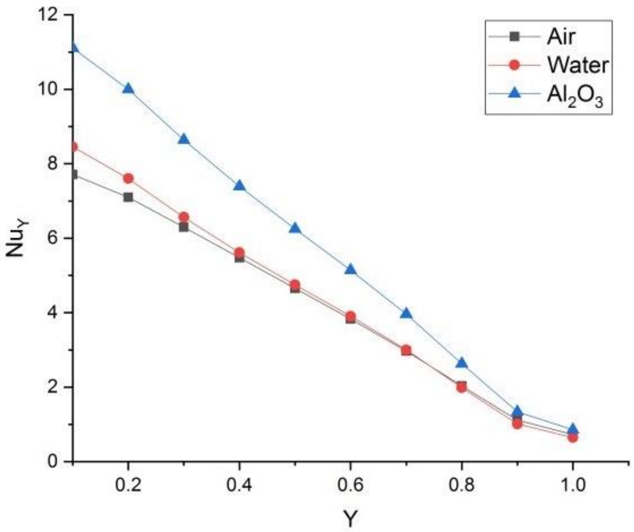


Fig. 7 Variation of local Nusselt number along hot wall for (Ra = 10^5, k_r = 100)

The conjugate heat transfer between the baffle and fluid inside the enclosure gets strengthened due to lesser baffle conductive resistance with higher baffle conductivity ratio. Accordingly, heat transfer is enhanced as the temperature drop along the length of the baffle decreases. Higher average Nusselt number values are observed for Al_2O_3/water nanofluid, with highest value for Al_2O_3/water nanofluid of volume fraction 0.2. However, the effect of baffle thickness on average Nusselt number is observed to be insignificant (figure not shown) for the entire range of parameter studied.

The influence of baffle conductivity ratio (k_r) on average Nusselt number is plotted in Fig. 9 for three different fluids air, water, Al_2O_3/water nanofluids of volume fraction 0.1 and 0.2. The average Nusselt number is observed to increase with increase in thermal conductivity ratio irrespective of the fluid consider.

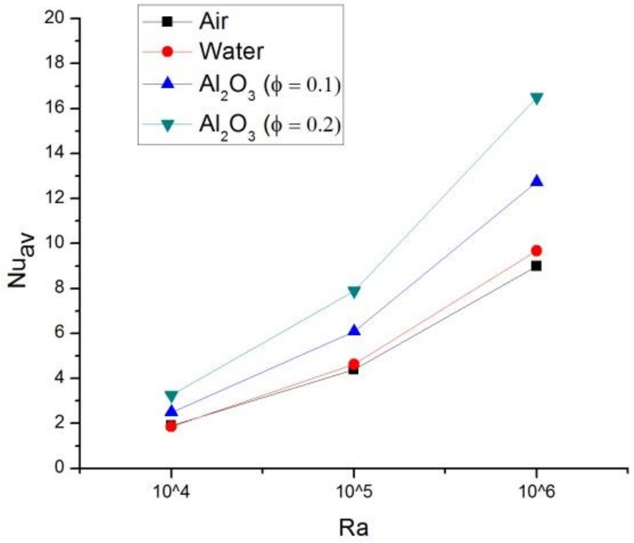


Fig. 8 Nu_{av} as a function of Ra for (B_p = 0.5, B_l = 0.5, B_t = =0.04, k_r = 100)

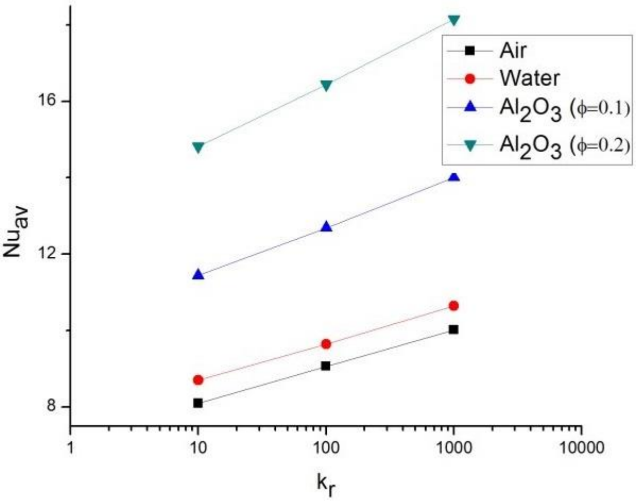


Fig. 9 Variation of average Nusselt number with Conductivity ratio for (Ra = 10^6, B_p = 0.5, B_l = 0.75, B_t = 0.04)

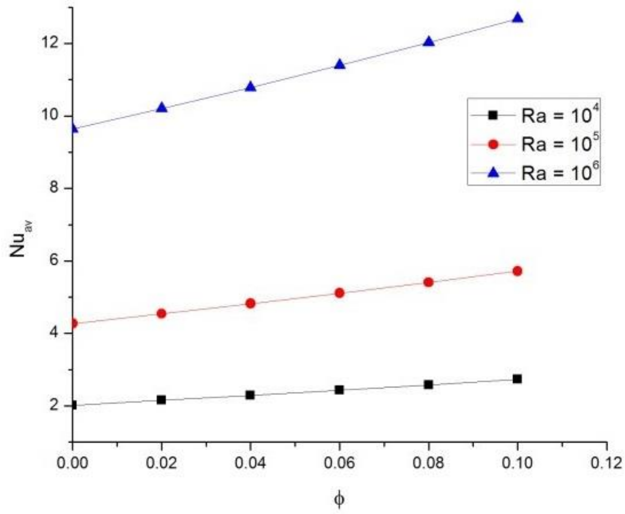


Fig. 10 Average Nusselt number as a funtion of for (B_l = 0.5, B_p = 0.5, B_t = 0.04)

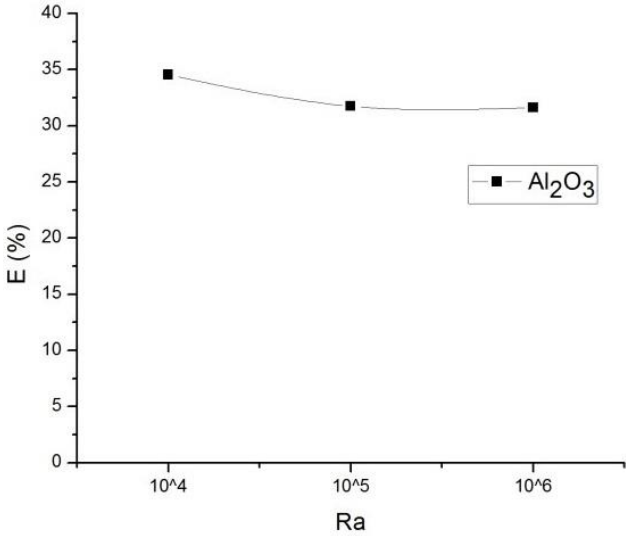


Fig. 11 Heat transfer enhancement as a function of Ra, for (B_p = 0.5, B_l = 0.5, B_t = 0.04, φ = 0.1)

The effect of volume fraction of the nanofluid on heat transfer performance of the enclosure is illustrated in Fig. 10, which is a graph of average Nusselt number with volume fraction of nanofluid. Average Nusselt number is found to increase monotonically with volume fraction of the nanofluid. The rate at which average Nusselt number increases with volume fraction is higher at higher Rayleigh number due to strengthened convective heat transfer.

Fig. 11 shows the enhancement of heat transfer due to Al_2O_3/water nanofluid. It is clearly evident that the minimum enhancement of heat transfer (E) is 30% for the entire range of Rayleigh number considered in the present study, which indicates that it is very much beneficial to use nanofluid for enhanced heat transfer inside enclosure.

In order to quantify the effectiveness of nanofluid on conjugate free convective heat transfer inside the enclosure the parameter E is defined as:

$$E(\%) = \frac{Nu_{av,nf} - Nu_{av,f}}{Nu_{av,f}} X100. \tag{16}$$

6. Conclusions

Conjugate free convective heat transfer inside a square enclosure, with a horizontal baffle fixed to the left hot wall, filled with Al_2O_3/water nanofluid has been numerically investigated. The heat transfer performance of the enclosure is analyzed for different parameters like volume fraction of nanofluid, thermal conductivity ratio, length and position of baffle, and Rayleigh number. The results are summarized as below.

- Flow field and heat transfer are affected by the presence of baffle significantly.
- Conjugate heat transfer is significantly enhanced by higher value of thermal conductivity ratio for all the fluids considered for analysis in the present work.
- Heat transfer enhancement by the nanofluid is found to be considerable, the minimum enhancement being 30% for nanofluid of volume fraction 0.1 and compared to that of water. Hence use of nanofluid is certainly beneficial.
- Average Nusselt number increases with increase in volume fraction of the nanofluid.

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EFFECT OF Al_2O_3/WATER NANOFLUID ON CONJUGATE FREE CONVECTION IN A BAFFLE ATTACHED SQUARE ENCLOSURE

S u m m a r y

A numerical study of conjugate free convection heat transfers of Al_2O_3/water nanofluid inside a differentially heated square enclosure with a baffle attached to its hot wall has been carried out. A detailed parametric study has been carried out to analyze the effect of Rayleigh number ($10^4 \leq Ra \leq 10^6$), length, thickness and position of baffle, conductivity ratio and volume fraction of the nanoparticle ($0 \leq \varphi \leq 0.2$) on heat transfer. The thermal conductivity ratio of the baffle plays a major role on the conjugate heat transfer inside the enclosure. Higher the baffle length better is the effectiveness of the baffle. The average Nusselt number is found to be an increasing function of both thermal conductivity ratio and volume fraction of the nanofluid. The minimum enhancement of conjugate heat transfer is 30% when Al_2O_3/water nanofluid of 0.1 volume fraction is used for the entire range of Rayleigh number considered.

Keywords: conjugate free convection, square enclosure, baffle, nanofluid.

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