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CFD simulations on natural convection heat transfer of alumina-water nanofluid with Brownian motion effect in 3-D enclosure

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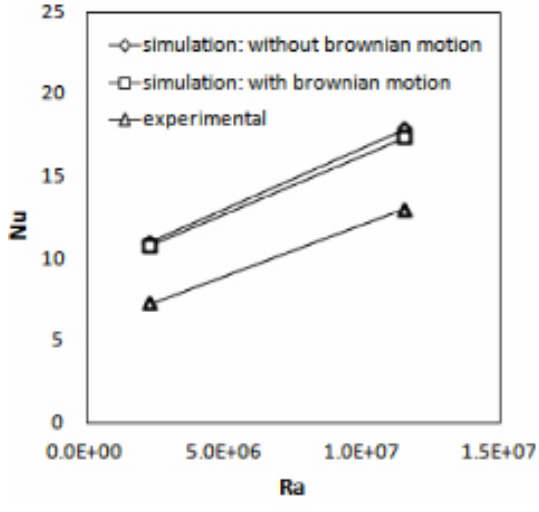
HIGHLIGHTS

- Natural convection in an enclosure, filled with a nanofluid, has been studied numerically.
- The numerical simulation has been carried out using the CFD approach.
- The effect of Brownian motion of nanoparticles on the heat transfer was examined.
- Heat transfer decreases with increase in nanoparticles considering Brownian motion effect.
- Nusselt number increases with Rayleigh number.

GRAPHICAL ABSTRACT



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ABSTRACT

The CFD simulation has been undertaken concerning natural convection heat transfer of a nanofluid in vertical square enclosure, whose dimension, width×height ×length (mm), is 40×40×90, respectively. The nanofluid used in the present study is Al_2O_3-water with various volumetric fractions of the alumina nanoparticles ranging from 0-3%. The Rayleigh number is 10^6, 10^7. Fluent v6.3 is used to simulate nanofluid considering it as a single phase. The effect of Brownian motion on the heat transfer is examined. A comparison between the two studies of with and without the Brownian motion, shows that when the Brownian motion is considered, the solid volume fraction of nanoparticles has dissimilar effects on the heat transfer. The numerical results show a decrease in heat transfer with increase in particle volume fraction considering Brownian motion effects. Moreover, computed result demonstrates an increase of Nusselt number with Rayleigh number as depicted by experimental results.

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1. Introduction

Nanofluids are defined as the new solid-liquid composite materials with nanometer-sized solid particles, typically 1-100 nm, suspended in the base fluid. Nanofluid, a colloid composed of nanoparticles or carbon nanotubes in a base fluid, has been proposed as a highly-effective heat transfer medium in view of its abnormally higher thermal conductivity. The term of nanofluid was first used by Choi [1]. Study various aspects of nanofluids can be examined [2–4]. One of these is natural convection. Natural convection heat transfer is an important phenomenon in engineering systems due to its wide applications in electronic cooling, cooling systems for nuclear reactors, solar energy collectors, heat exchangers, and various thermal systems [5-7]. Enhancement of heat transfer in these systems is an essential topic from an energy saving perspective. The low thermal conductivity of convectional heat transfer fluids such as water and oils is a primary limitation in enhancing the performance and the compactness of such systems. An innovative technique to improve heat transfer is by using nano-scale particles in the base fluid.

Studies on natural convection using nanofluids are very limited and they are related with differentially heated enclosures. The first study concerning natural convection of a nanofluid confined in a differentially heated enclosure seems to be due to Khanafer et al. [8]. They concluded that the heat transfer rate increases with the particle fraction at any given Grashof number. Natural convection heat transfer of nanofluid in a square cavity, heated isothermally from the vertical sides, has been investigated numerically by Ho et al. [9] and santra et al. [10]. Review of pertinent literatures, two different formulas can be found for the effective viscosity and thermal conductivity of the nanofluids which have been considered by [9] while the ostwald-de Waele model for a non-Newtonian shear thinning fluid has been used by [10] to calculate the shear stress. It was found that the uncertainties associated with different models adopted to modelized the nanofluids have a great influence on the natural convection heat transfer characteristics in the enclosure. Putra et al. [11] conducted an experiment on natural convection of alumina-water and Cuo-water nanofluids inside a horizontal cylinder heated from one end and cooled from the other. They found a systematic and definite deterioration in natural convection heat transfer. The deterioration increase with particle fraction and appears more significant for Cuo-water nanofluid. Hwang et al. [12] investigated the buoyancy-driven heat transfer of Al_2O_3-water nanofluid in a rectangular cavity. They showed that the ratio of heat transfer coefficient of nanofluids to that of base fluid is decrease as the size of nanoparticles increases, or the average temperature of nanofluids is decreased. Jang and Choi [13] studied free convection in a rectangular cavity. They reported an increase in heat transfer rate with increase in particle volume fraction. Dinarvand et al. [14] studied mixed convection of a nanofluid over a vertical circular cylinder. Wang et al. [15] conducted a study on natural convection in nanofluid filled vertical and horizontal enclosures. Also, a recent study by Polidori et al. [16] analyzed the heat transfer enhancement in natural convection using nanofluids. References [17-20] present some other interesting researches in this field.

With regard to the CFD simulation using FLUENT software, the effect of gravity on sedimentation and clustering of nano ferro-fluids on natural convection heat transfer was studied by Jafari et al. [21]. They used two models in their study: the single phase approach and mixture model. Buoyancy driven heat transfer of nanofluids was studied by Ismail et al. [22] using FLUENT. They studied the effect of volume fraction and Raileigh number in their work using single phase approach.

Motivated by the above-mentioned investigations, the main objective of the present study is to numerically simulate natural convection heat transfer of nanofluid and validate them with the previous experimental results of Ho et al. [23]. In this work, FLUENT v6.3 is employed to study natural convection heat transfer of nanofluid. Simulations have been carried out in a vertical square enclosure, as used in the experimental setup of [23]. The numerical results are presented in terms of non-dimensional parameters such as Nusselt number and Rayleigh number.

2. Thermophysical properties of alumina-water nanofluid

The effective properties of the nanofluid are defined as follows:
Density:

$$\rho_{nf} = (1-\phi)\rho_f + \phi\rho_{np} \tag{1}$$

Heat capacity:

$$C_{nf} = \frac{(1-\phi)(\rho C)_f + \phi(\rho C)_{np}}{\rho_{nf}} \tag{2}$$

The Equations (1) and (2) were introduced by Buongiorno [20].

Thermal expansion coefficient:

$$\beta_{nf} = \frac{(1-\phi)(\rho\beta)_f + \phi(\rho\beta)_{np}}{\rho_{nf}} \quad (3)$$

For the thermal conductivity and viscosity two scenarios are considered: the first scenario neglects Brownian motion and uses $\mu_{eff}=\mu_{static}$ and $k_{eff}=k_{static}$. The second scenario includes Brownian motion and uses $\mu_{eff}=\mu_{static}+\mu_{Brownian}$ and $k_{eff}=k_{static}+k_{Brownian}$.

Thermal conductivity:

$$k_{static} = k_f\left[\frac{\left(k_{np}+k_f\right)-2\phi\left(k_f-k_{np}\right)}{\left(k_{np}+k_f\right)+\phi\left(k_f-k_{np}\right)}\right] \quad (4)$$

This was introduced by Maxwell and Garnett [25]. For the Brownian motion, we have [26]:

$$k_{Brownian} = 5\times10^4\beta\phi\rho_f C_{p,f}\sqrt{\frac{\kappa T}{2\rho_{np}R_{np}}}f(T,\phi) \quad (5)$$

Viscosity:

$$\mu_{static} = \frac{\mu_f}{(1-\phi)^{2.5}} \quad (6)$$

This was introduced by Brinkman [27]. For the Brownian motion, we have [26]:

$$\mu_{Brownian} = 5\times10^4\beta\phi\rho_f\sqrt{\frac{\kappa T}{2\rho_{np}R_{np}}}f(T,\phi) \quad (7)$$

In equations 5 and 7:

$$\kappa = 1.3807\times10^{-23}\ J/K$$
$$\beta = 0.0011(100\phi)^{-0.7272} \quad (8)$$
$$f(T,\phi) = (-6.04\phi+0.4705)T+(1722.3\phi-134.63)$$

In equation 8, κ is Boltzmann constant and β, $f(T,\emptyset)$ are modeling functions. The radius of the Al_2O_3 nanoparticles (R_{np}) is 33 nm was used [23]. Thermophysical properties of the pure water and nanoparticles tabulated in table 1, also thermophysical properties of the nanofluid used in this study are tabulated in table 2.

3. Numerical simulation

3.1 Geometry creation and grid arrangement

The geometry and the grid were generated using GAMBIT the preprocessing module of the FLUENT v6.3. GAMBIT is an integrated preprocessor for CFD analysis. The geometry is shown in Fig. 1. The geometry is a vertical square enclosure. This enclosure was differentially heated across two vertical walls; while the remaining side walls of

Table 1. Thermophysical properties of pure water and nanoparticles [23]

Fluid	Density (kg/m³)	Specific heat (J/kg K)	Thermal conductivity (W/mK)	Viscosity (Pa s)	Thermal expansion (1/K)
Pure water					
At Tm=296 K	997	4179	0.51	0.000930	0.000210
At Tm=299 K	995	4179	0.50	0.000875	0.000242
Al_2O_3	3600	765	40	-	

Table 2. Thermophysical properties of nanofluid

Fluid	Density (kg/m³)	Specific heat (J/kg K)	Thermal conductivity (W/mK)	Viscosity (Pa s)	Thermal expansion (1/K)
Nanofluid properties					
Nanofluid 1%					
At Tm=296 K	1023.03	4058.86	0.5248	0.0009536	0.0002029
At Tm=299 K	1021.05	4058.63	0.5146	0.0008972	0.0002337
Nanofluid 2%					
At Tm=296 K	1049.06	3944.68	0.54	0.0009781	0.0001961
At Tm=299 K	1047.10	3944.24	0.5294	0.0009203	0.0002259
Nanofluid 3%					
At Tm=296 K	1075.09	3836.04	0.5555	0.001003	0.0001897
At Tm=299 K	1073.15	3835.42	0.5446	0.0009442	0.0002185

enclosure were thermally insulated. The physical boundary conditions for the geometry are defined as hot wall (left), cold wall (right) and adiabatic walls.

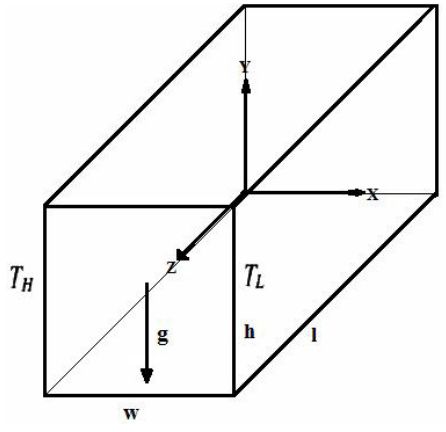

Fig. 1. schematic of 3-D enclosure

4. Computational model

4.1. Assumption

The nanofluid is incompressible and the flow is laminar. Also it is assumed that the liquid and solid are in thermal equilibrium with each other and thus no slip occurs between them. The nanoparticles in the base fluid may be easily fluidized and consequently the effective mixture behaves like a single phase fluid. The resultant mixture may be considered as a conventional single phase fluid. The effective thermophysical properties of nanofluid are dependent upon the temperature and volume fraction. The problem investigated is a three dimensional steady state laminar flow in vertical square enclosure and are governed by the following model equations.

4.2. Governing equations

4.2.1. Continuity equation

The principle of conversation of mass given by:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho U) = 0 \tag{9}$$

4.2.2. Momentum equation

The principle of conversation of momentum is in fact an application of Newtonian's second law of motion to an element of fluid:

$$\frac{\partial}{\partial t}(\rho U) + \nabla.(\rho UU) = -\nabla P + \nabla \tau + B \tag{10}$$

4.2.3. Energy equation

The principle of conversation of energy given by:

$$\frac{\partial}{\partial t}(\rho h) + \nabla.(\rho UC_p T) = \nabla.(k\nabla T) \tag{11}$$

5.2. Numerical procedure

The governing equations were numerically solved using segregated solver. Laminar model was used to simulate the natural convection flow using SIMPLE scheme for pressure-velocity coupling and PRESTO was used for pressure. Steady state solver was used for all simulations. Second order upwind discretization scheme were employed for all simulations. Iteration is terminated when the maximum of the residues a value less than 10^{-6}. All simulations carried out in FLUENT software. FLUENT solver uses control volume approach to solve heat transfer problem.

The total surface heat flux (q") was computed from hot wall in each case using surface integrals. With the q", the surface-averaged heat transfer coefficient, $\overline{h}_{nf}$ was then calculated based on the measured temperature difference between the hot and cold walls, as

$$\overline{h}_{nf} = \frac{\overline{q''}}{T_h - T_c} \tag{12}$$

Further, the heat transfer coefficient was presented as the average Nusselt number, which may be defined based on the measured thermal conductivity of the nanofluid as

$$\overline{Nu}_{nf} = \frac{\overline{h}_{nf} w}{k_{nf}} \quad \text{or} \quad \overline{Nu}_{nf} = \frac{\overline{q''} w}{k_{nf}(T_h - T_C)} \tag{13}$$

Moreover, based on the temperature difference across and the characteristics length, w, of the enclosure, the Rayleigh number was determined as

$$Ra_{nf} = \frac{g\beta_{nf} w^3 (T_h - T_c)}{\alpha_{nf} \upsilon_{nf}} \tag{14}$$

Thermophysical properties of the nanofluid were evaluated based on the mean temperature between the hot and cold walls,

$$T_m = \frac{T_h + T_c}{2} \tag{15}$$

5. Grid independence study

The grid independence study has been conducted

in enclosure for pure water with three different grid sizes such as 41^3, 61^3 and 81^3. The hot and cold walls are maintained isothermally at temperature of 297 K and 295 K, respectively. Velocity magnitude profiles are plotted at the mid-section of the enclosure as shown in Fig. 2. From Fig. 2, it is very clear that grid size 61^3 and 81^3 gave same results for velocity magnitude. Grid size of 61^3 is shown in Fig. 3. Thus, to save time and speed up the convergence of computing, the grid size of 61^3 was used in all simulations.

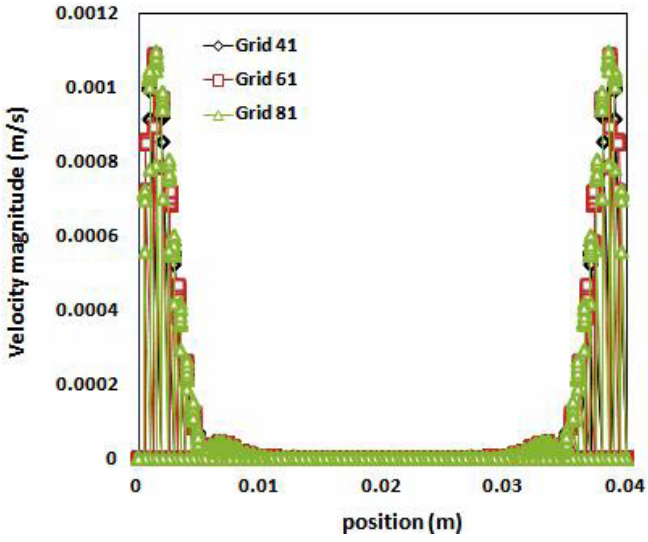

Fig. 2. Velocity magnitude at middle enclosure for showing grid independent

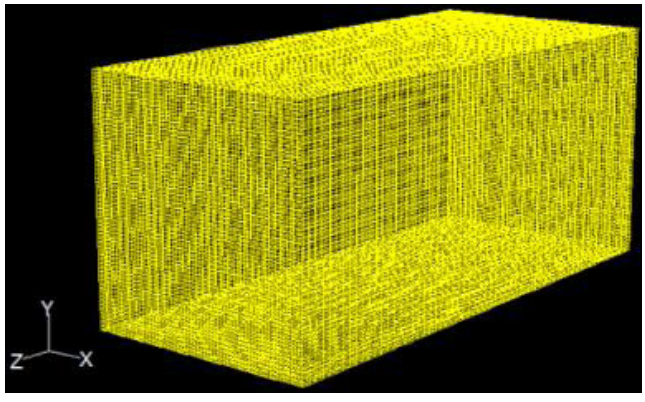

Fig. 3. Grid for vertical square enclosure

6. Results and discussion

In this section, we will discuss about the results obtained from numerical simulations. Nusselt number versus Rayleigh number is displayed.

6.1. Pure water (Ø=0%)

Numerical simulation conducted for pure water (0% volume fraction). Nusselt number calculated for two different Rayleigh numbers corresponding to two different temperatures of 296 and 303 K at hot wall and 295 K at cold wall. Fig. 4 shows the numerical results validated against experimental results for pure water. As expected, both the numerical and experimental results show increase in Nusselt number with increase in Rayleigh number. The computed results are seen to be in good agreement with the experimental results. Also, computed results show that the grid size of 61^3 is suitable to the simulation of problem.

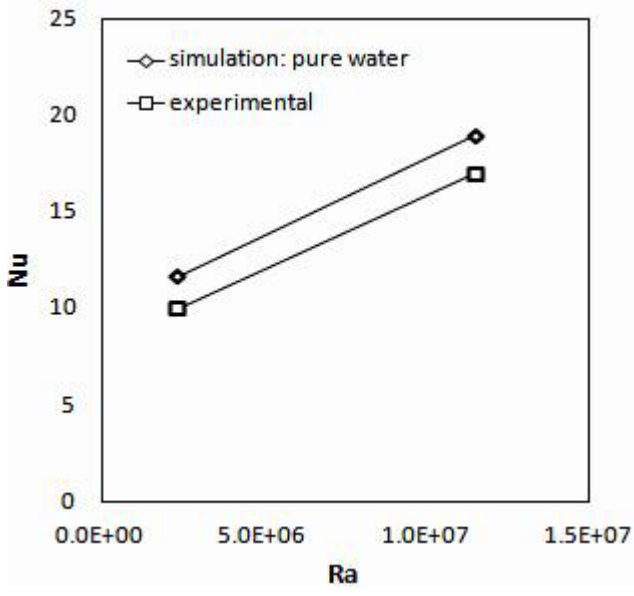

Fig. 4. Variation of Nusselt number with Rayleigh number for pure water

6.2. Nanofluid (Ø=1-3%)

Fig. 5 shows the variation of the Nusselt number versus the Rayleigh number for 1% volume fraction of nanoparticles. When the nanoparticles are added to pure water, the Nusselt number decreases compared to that of pure water. Also, Figs. 6 and 7 represent the variation of the Nusselt number with the Rayleigh number for 2 and 3% volume fraction of nanoparticles, respectively. In these cases, again, Nusselt number decreases with increase in volume fraction of nanoparticles. For all cases, the numerical results predict the similar trend as that of the experimental results. This phenomenon is paradoxical which is pointed by Jafari et al [21]. Typically, when the nanoparticles are added to the base fluid, the thermal

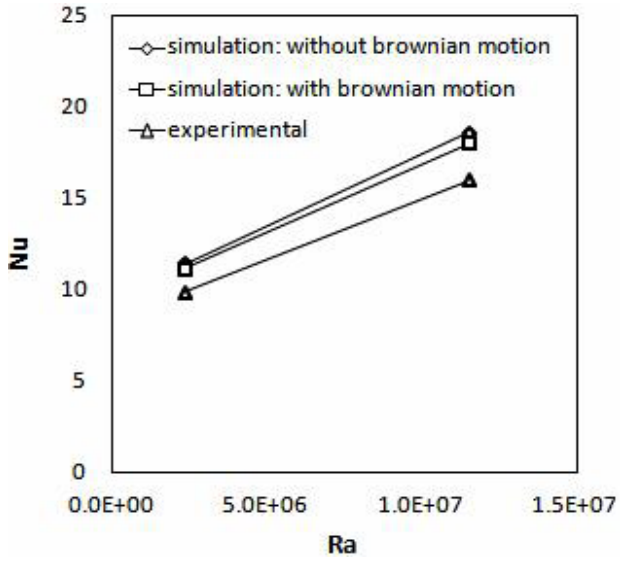

Fig. 5. Variation of Nusselt number with Rayleigh number for $\emptyset$=1%

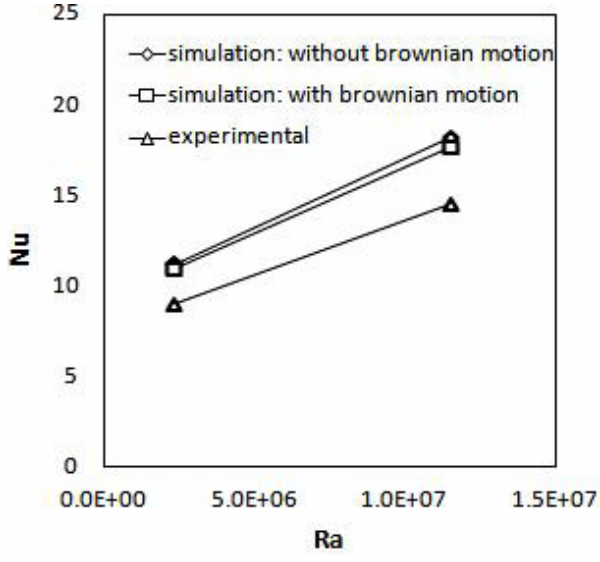

Fig. 6. Variation of Nusselt number with Rayleigh number for $\emptyset$=2%

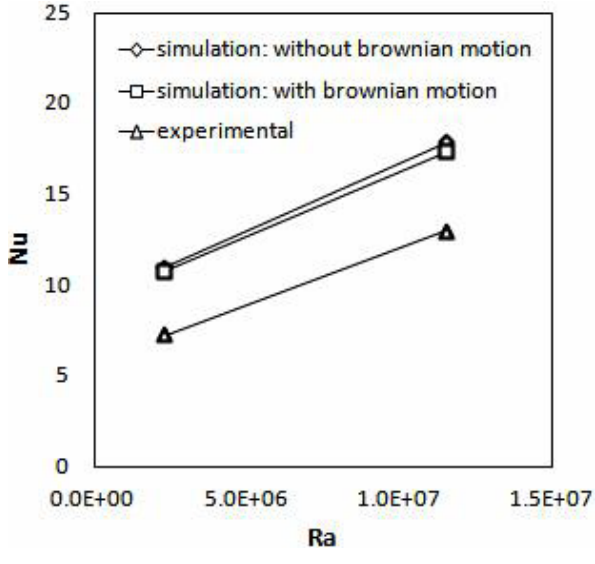

Fig. 7. Variation of Nusselt number with Rayleigh number for $\emptyset$=3%

Comparison of numerical results and experimental results shows that, with increase in particle volume fraction, a considerable difference in experimental and numerical results are observed. There are several reasons for this problem. This may be due to the selection of single phase approach which takes into account the effective mixture properties but not the particles into consideration. Thermal conductivity models have also played a decisive role in the numerical results. Theoretical models for predicting the thermal conductivity of nanofluids are different experimental measurements, thus leading to different numerical result and the experimental. It should be noted that the Rayleigh's number of the nanofluids used in computations are different from experimental value due to difference in the values of material properties.

6.3. Brownian motion effect

According to Figures 5–7, it can be seen that the values of the Nusselt number are generally lower when the Brownian motion is considered. Also, these values are closer to the experimental results.

6.4. Temperature contours

Figures 8–11 show isotherms for pure water and the nanofluid with 1–3% volume fraction of nanoparticlrs, respectively. It is observed that with

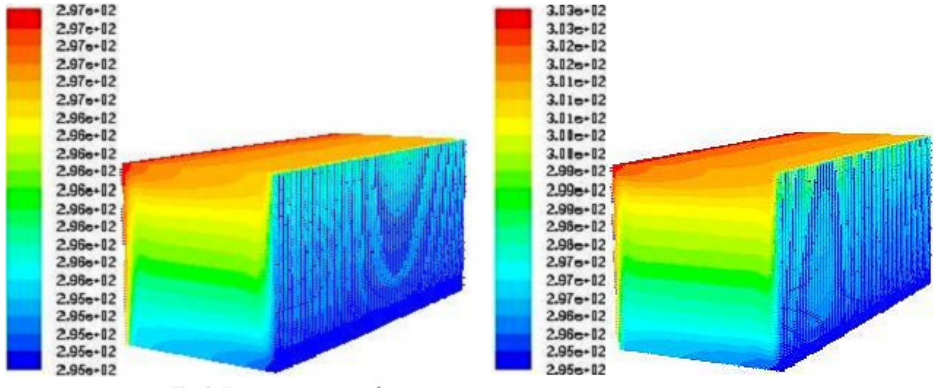

Fig. 8. Temperature contour for pure water

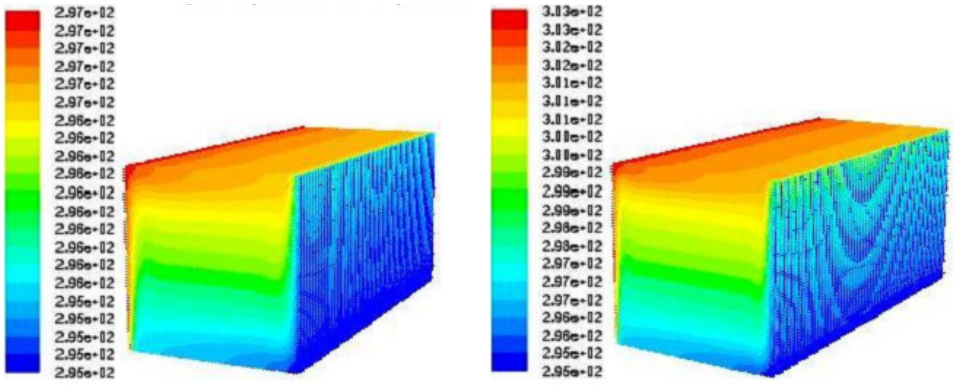

Fig. 9. Temperature contour for $\emptyset$=1%

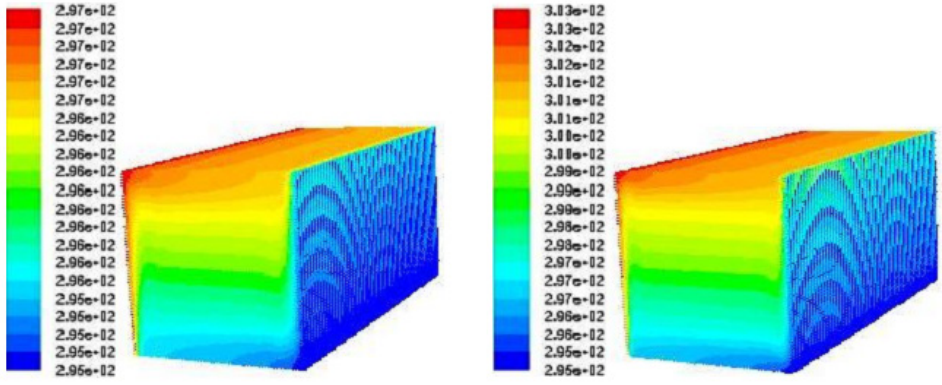

Fig. 10. Temperature contour for $\emptyset$=2%

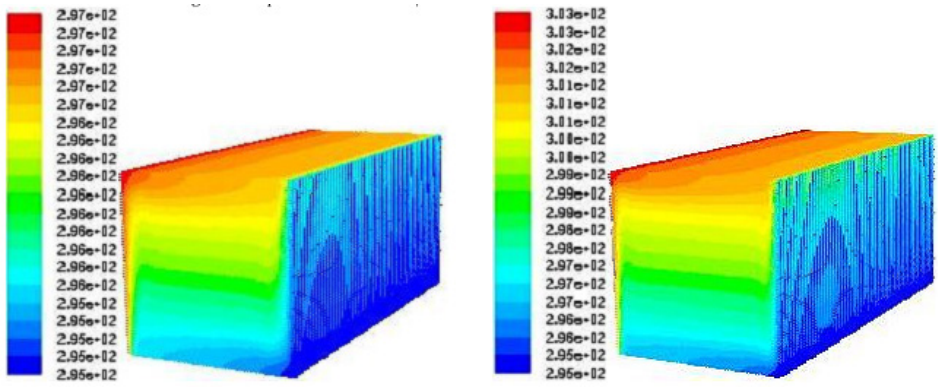

Fig. 11. Temperature contour for $\emptyset$=3%

increase in hot wall temperature, the temperature drops very gradually from top to bottom.

7. Conclusions

Natural convection heat transfer in a vertical square enclosure, filled with an Al_2O_3-water nanofluid has been studied numerically. The numerical simulations were carried out using the computational fluid dynamic (CFD) approach. It is assumed that the nanofluid is a single phase fluid. The effects of parameters such as the Rayleigh number, volume fraction of nanoparticles and Brownian motion on the heat transfer are examined. Comparisons with previously published work were performed and the results were found to be in good agreement. Thus, CFD can be effectively implemented for simulations of the nanofluid with further improvement over theoretical models that can account for temperature effects. A comparison between the two studies of with and without the Brownian motion, shows that when the Brownian motion is considered, the solid volume fraction of nanoparticles has dissimilar effects on the heat transfer at different Rayleigh number.

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