NUMERICAL ANALYSIS OF NATURAL CONVECTION OF NANO FLUIDS ON SQUARE ENCLOSURES

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Abstract - *Recent decade has witnessed an extensive research in the use of nanoparticle suspended fluids for heat transfer augmentation in electronics cooling, heat exchangers, heat pipes and automotive applications. In this the natural convection inside an enclosure is studied using Al_2O_3 and TiO_2 nano particle dispersed in the water is numerically analyzed. Numerical analysis has been carried out in low concentration of nano particle in the base fluid, concentration used in the study in the range of 0.1% to 0.4% with a liquid side Rayleigh number of the range $2*10^7$ to $8\ 8*10^7$. Effect of nano particle concentration and Rayleigh number on the average heat transfer coefficient and the average nusselt number are presented. The effect on aspect ratio is also examined in the current study. The Al_2O_3 shows enhancement of 33% and TiO_2 shows an enhancement of 22% compared to the base fluid in the heat transfer properties as compared to that of conventional heat transfer fluids. It is also found out that as the Rayleigh number increases the heat transfer of the both nano fluid increases and the opposite trend is found for increasing concentration of the nano fluid*

Keywords *— Natural Convection, Nanoparticle, Al_2O_3 and Tio_2, Rayleigh Number*

1. INTRODUCTION

Natural convection has engrossed a great deal of consideration from researcher since natural convection have intensive application on both natural and engineering field In engineering applications, natural convection heat transfer is used in situation of electrical cooling, double glazed glass, heat exchangers. In nature, convection cells formed from air raising above sunlight-warmed land or H2O are a key feature of all weather systems. Convection is also seen in the increasing plume of hot air from fire, plate tectonics, oceanic currents (thermohaline circulation) and sea-wind formation.

Natural convection in enclosures are also known as internal convection enclosures are finite spaces filled with fluid and bounded by wall. This mode of internal convection can be take place in rooms and furnaces; this is also seen in cooling of electronic cooling systems. Natural convection in enclosures is different from the case of normal natural convection, in this there will be a heated and cooled wall will be in contact with the fluid filled in the enclosure. Natural convection in enclosures with various arrangements has many applications in thermal industry such as glass fabrication, solar collectors, food and drying technologies thermal energy storage tanks Therefore, optimization of such systems could be an aid to save energy. Using a nano fluid instead of conventional working fluids such as water could be a solution to optimize the heat transfer processes in enclosures. Therefore, optimization of such systems could be an aid to save energy. Using a nano fluid instead of conventional working fluids such as water could be a solution to optimize the heat transfer processes in enclosures.

Simon Ostrach (1972) discusses the multifaceted nature of the natural convection occurrences in enclosures. It discusses the natural convection inside rectangular cavity shapes configuration. In rectangular cavities, the two-dimensional is selected for the convective motion produced by the buoyancy force on the fluid in a rectangle and to the related heat transfer. The two long sides are vertical boundaries held at dissimilar temperatures and the short sides can be adiabatic. To formulate the boundary value problem that defines this occurrences it is expected that the motion is two-dimensional and steady, the fluid is incompressible and frictional heating is insignificant, and the variance between the hot wall and cold wall temperatures is insignificant comparative to the absolute temperatures of the cold wall.

Choi and Eastman (1995) theoretically studied the impact of suspended nano sized solid particle in convectional heat transfer liquid and suggested an new novel class of heat transfer liquid can be made by suspending nano size particles of metal oxides in the traditional hat transfer liquids. The newly produced liquid shows high thermal conductivity than those are now used normal heat transfer liquid, they are the next level heat transfer liquid with best hope for heat transfer augmentation. In their studies the outcomes on the thermal conductivity of nano fluid made up of nano particle of copper were shown, the chief benefits of the nano particle mixed solution is studied.

Omid Mahian.et.al (2016), investigated natural convection inside square and triangle cavities with silica nano fluid using theoretical correlations. Their main aim is to find out the heat transfer properties like heat transfer coefficient and average Nusselt number. The heat transfer properties of the enclosure are calculated using the correlation that are the function of thermo physical properties of the nano fluid. Mainly the studies which are used to predict the thermo physical properties of the nano fluids like thermal conductivity, density, viscosity etc. uses the classic correlation models. Sometime these might lead to substantial error. This kind of errors can be e avoided by taking the results from the experimental procedure. So to find the thermo physical properties of silica nanofluids, silica nanoparticle dispersed water is made with the diameter of silica nanoparticle equal to 7nm. Next At a temperature range of 25 to 60 °C thermal conductivity viscosity and density of the nanofluids with volume fraction .5%, 1%, 2% are measured. Measured data are compared with the outcomes of theoretical models.

2. Problem Definition and Mathematical Formulations

The flow configuration which is chosen for the analysis is square enclosure whose bottom wall and top walls are isothermally kept at a temperature of Th and the side walls are kept at a temperature of Tc and the remaining walls is kept at adiabatically insulated. Th is greater than Tc. The enclosure is filled with nano fluid is modelled as a dilute solid – liquid mixture with a uniform volumetric fraction of nanoparticles dispersed within a base fluid (water) represented in figure 1.

The Boussinesq approximation is assumed to be valid for the buoyancy-driven flow in the enclosure. In addition, effects of the compression work and viscous dissipation are assumed negligible. The enclosure wall thickness is negligible and the nanofluid in the enclosure is consideration a single phase fluid the physical properties of the nanofluid is taken as the function of the both constituent and their concentration

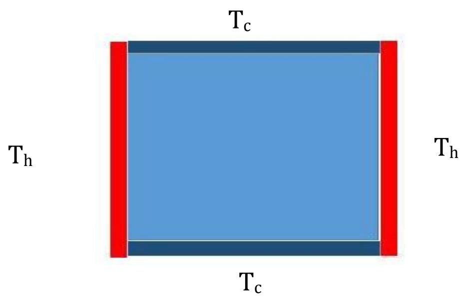

Fig 1 Flow configuration with horizontal walls are kept at T_c and vertical walls are kept at T_h

The temperature and flow field in this problem is investigated domain can be predicted using the following the following equation for a three dimensional steady flow in cartesian coordinates

Continuity equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad 1$$

Momentum equation in x direction

$$u\frac{\partial u}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial w}{\partial z}= \qquad 2$$

$$\frac{1}{\rho_{nf}}\left(-\frac{\partial p}{\partial x}+\mu_{nf}\nabla^2 u+(g\beta)_{nf}\,g_x\left(T_h-T_h\right)\right)$$

Momentum equation in y direction

$$u\frac{\partial u}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial w}{\partial z} \qquad 3$$

$$=\frac{1}{\rho_{nf}}\left(-\frac{\partial p}{\partial y}+\mu_{nf}\nabla^2 v+(g\beta)_{nf}\,g_y\left(T_h-T_h\right)\right)$$

Momentum equation in z direction

$$u\frac{\partial u}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial w}{\partial z}= \qquad 4$$

$$\frac{1}{\rho_{nf}}\left(-\frac{\partial p}{\partial x}+\mu_{nf}\nabla^2 u+(g\beta)_{nf}\,g_x\left(T_h-T_h\right)\right)$$

Energy equation

$$\frac{\partial T}{\partial x}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\frac{1}{\rho_{nf}C_{p.nf}}\left(\frac{\partial}{\partial x}\left(k_{eff}\frac{\partial T}{\partial x}\right)\right)+ \qquad 5$$

$$\left(\frac{\partial}{\partial y}\left(k_{eff}\frac{\partial T}{\partial y}\right)\right)+\left(\frac{\partial}{\partial x}\left(k_{eff}\frac{\partial T}{\partial x}\right)\right)$$

One of the first model thermal conductivity of nanofluids is proposed by Maxwell in 1881 for storing liquid mixture with relative large a particle. The model was based on dissolution of heat conduction equation true stationary random suspension of sphere. Duty unpredictability future of the nanofluid still researchers is doing wide range of experimental and theoretical studies. In this problem the effective thermal conductivity of the nanofluid is calculated using following equation

$$k_{nf}=k_{bf}\left(\frac{k_p+k_{bf}+2\varphi\left(k_{nf}-k_p\right)}{k_p+k_{bf}+2\varphi\left(k_{nf}-k_p\right)}\right) \qquad 6$$

Density and specific heat capacity of inner fluid is defined as a function of the particle volume concentration an individual properties it is derived from derived from a classic formula for a two phase mixture. The following equations are used for finding the density and specific heat capacity respectively.

$$\rho_{nf}=(1-2.5\varphi)\rho_{bf} \qquad 7$$

$$\left(\rho C_p\right)_{nf}=\left(1-\varphi\right)\left(\rho C_p\right)_{bf}+\varphi\left(\rho C_p\right)_p \qquad 8$$

In this problem the numerical investigation are carried out under laminar flow condition using two different nanofluids with different concentration different railway number. In this problem the volume of concentration of the nanofluids is varied from 0 % to .4 % Rayleigh number was vary from 2*10^7 to 8*10^7. The temperature assign to the walls are based on Rayleigh number choose.

3. Numerical Model

Numerical simulations were carried out using commercially available CFD software ANSYS Fluent-16.2. A first-order upwind scheme was employed to discretize the convection terms, diffusion terms, and other quantities resulting from the governing equations. The pressure-velocity coupling was carried out using Semi Implicit Method for Pressure Linked Equations (SIMPLE) scheme and second order scheme as pressure interpolation scheme. For the momentum and energy interpolation scheme Quadratic Upstream Interpolation for Convective Kinematics (QUICK) model is used in this problem. Throughout the iterative process, the residuals were continually monitored until convergence.

3.1 Grid Independence

By considering four different mesh element for discretizing the interior domain of the square enclosure the grid independence was established. In each case the average heat transfer coefficient at the heated wall was obtained and tabulated in table 1. From the result obtained it is found that the average heat transfer coefficient is almost the same for all the cases, as shown only a small percentage difference. From the result obtained it is concluded that the solution is independent of the mesh

Table 1 Grid Independence

Number of Element	Average Heat Transfer Coefficient (W/(m^{2}K))
100569	276.2
195112	276.03
280965	276.47
360256	276.39

3.2 Validation

The result obtained from the numerical simulation of this problem were compared with the results obtained from the Prediction of Heat Transfer Coefficient, Nu, as function of Rayleigh number in Rayleigh-Bénard natural convection by Dinh et al. in 1997 carried out numerical study to investigate the heat transfer prediction of the base fluid that is the water in the enclosure. Figure 3.4 shows the comparison of the average heat transfer coefficient of the base fluid by the present numerical simulation and with Prediction of Heat Transfer Coefficient, Nu, as function of Rayleigh number in Rayleigh-Bénard natural convection (Dinh et at, 1997). The maximum deviation of the result of the numerical result and the result from literature were less than ±10

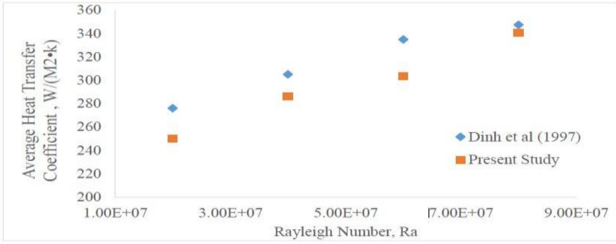

Fig 2 Variation of heat transfer coefficient with Rayeligh number

Figure 3 shows the comparison of the average Nusselt number of the base fluid by the present numerical simulation and with Prediction of Heat Transfer Coefficient, Nu, as function of Rayleigh number in Rayleigh-Bénard natural convection (Dinh et at, 1997). The maximum deviation of the result of the numerical result and the result from literature were less than ±18% This comparison of data shows that the deviation of results obtained from the numerical results and that are available in the literature are the in the acceptable limits

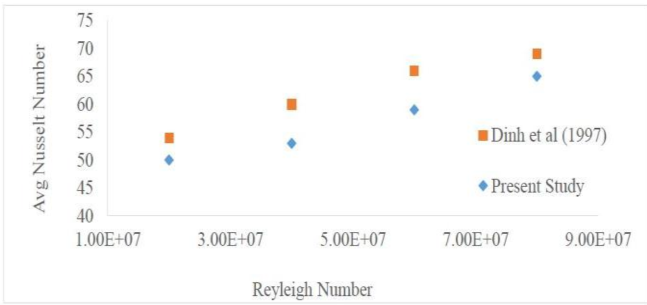

Fig 3. Variation of heat transfer coefficient with Rayeligh number

4.Results and Discussions

4.1 Effect of Particle Concentration on heat transfer coefficient

In the flow configuration the effect of Al2O3 and TiO2 nano particles on average heat transfer coefficient on the heated plate is analyzed at different Rayleigh number are shown in figure 4 and 5. The figure 4 and 5 shows the results for effect of particle concentration and Rayleigh number on the heat transfer coefficient that obtained from the numerical simulations for TiO_2 and Al_2O_3 nano particles dispersed in water.

In this case the opposite walls are heated and the other wall opposite to it is cooled. By doing this natural convection current is doubled up since both heating and cooling of the liquid is done in the adjacent walls. This back to back heating and cooling of the fluid in the enclosure speeds up the natural convective current inside the enclosure. This fast convective current increase the heat transfer of the fluid. The fluid get less dense in the heated wall then goes to the cooled wall which makes the fluid get dense and the process is repeated once again in the cycle this process is the prime reason for the fluid motion inside the enclosure. The heat transfer of the fluid gets augmented on this speed motion of the fluid. The heat transfer coefficient of the fluid get increases as the temperature difference increase.

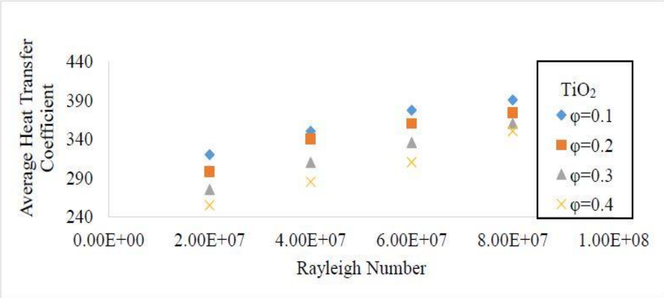

Fig 3 Variation of average heat transfer coefficient with particle concentration of TiO_2

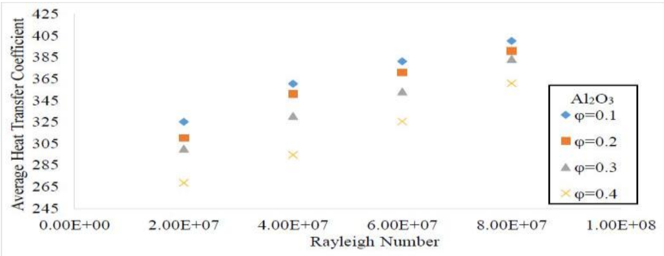

Fig. 4 Variation of average heat transfer coefficient with particle concentration of Al_2O_3

The adjacent cooling and heating of the fluid which makes the fluid more efficient in the heat transfer. The density of the fluid is of the fluid get altered at the walls so that the viscosity of the fluid also decrease so that the movement of the fluid is more therefore the heat transfer occurs at a faster rate.

4.2 Effect of Particle Concentration on Nusselt Number

In the flow configuration the effect of Al_2O_3 and TiO_2 nano particles on heat transfer coefficient is analyzed at different Rayleigh number are shown in figure 5 and 6. The figure 5 and 6 shows the results for effect of particle concentration and Rayleigh number on the nusselt number that obtained from the numerical simulations for TiO_2 and Al_2O_3 nano particles dispersed in water.

This flow configuration is a unique one which adjacent wall are kept at dissimilar temperature, so in this, the convective heat current developed is high which makes the fluid inside the enclosure keep moving inside the enclosure. The movement of fluid in the enclosure is a good way of identifying the augmentation of the heat transfer. This increased heat transfer will shows the increased nusselt number at the heated wall.

The average nusselt number is showing an increasing trend on increase in the Rayleigh number for a given particle concentration for both the nano particle. The increased heat transfer properties of the nanofluid inside the enclosure are the reason for the increase in the average nusselt number on the heated plate.

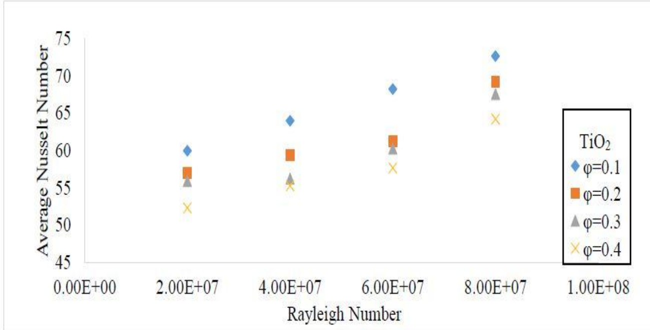

Fig 5 Variation of average Nusselt number with particle concentration of TiO_2

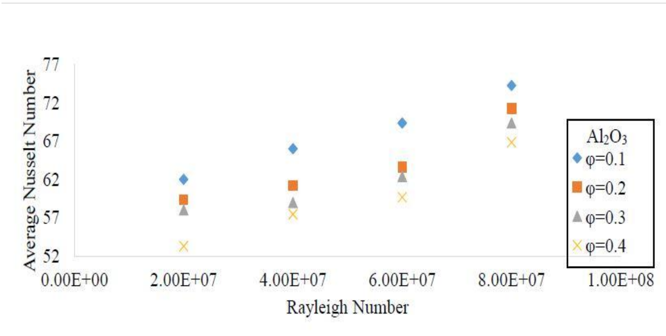

Fig 6 Variation of average Nusselt number with particle concentration of Al_2O_3

4.3 Effect of Aspect Ratios

For the analysis of aspect ratio, Al2O3 nano particles dispersed in the base fluid that is the water is selected because the maximum enhancement of heat transfer properties are seen. In this the height and length of the enclosure is changed in order to find the optimum heat transfer for that we are choosing 3 aspect ratios, AR=1/4 and 2. Figure 7 ,8 shows the effect of aspect ratio on the nusselt number in particle concentration and Rayleigh number

Fig 7 shows the results that obtained from the numerical study for an aspect ratio AR=1/4, it is clear from the graph that the average nusselt number is showing an increasing trend with increase in the Rayleigh number on a given concentration of nano particle dispersed in water. This is due to the convection dominance in the enclosure It is also found that the as the concentration of the nano fluid increase the average nusselt number is keep going down

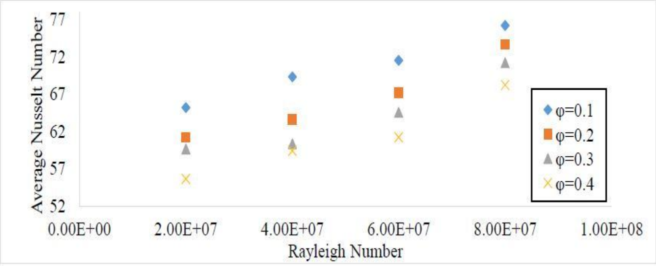

Fig 7 Variation of average nusselt number on particle concentration for AR=1/4

In figure 98 the graph is plotted for average nusselt number at the heated plate and the Rayleigh ratio equals to 2. The result obtain is that of the same trend as the pervious result, but the main finding found in the study is that the heat transfer by convection has less effect on the enclosure having aspect ratio greater than 1. In this the average nusselt number obtain is less than that of AR=1, this is because as height increase the heat transfer is dominated by the conduction mode, convection has less effect in the heat transfer process. From the results obtained it is found that the as the aspect ratio is increasing the nusselt number is showing a decreasing trend on Rayleigh number in a given volume concentration. Enclosure having high aspect ratios shows high values of heat transfer coefficient, but due to the conductive dominance of the heat transfer performance, high aspect ratios of the enclosure shows lower values of the nusselt number. The convective heat transfer performance is better for the enclosure having low aspect ratio due to multicellular flow patterns formation in the enclosure

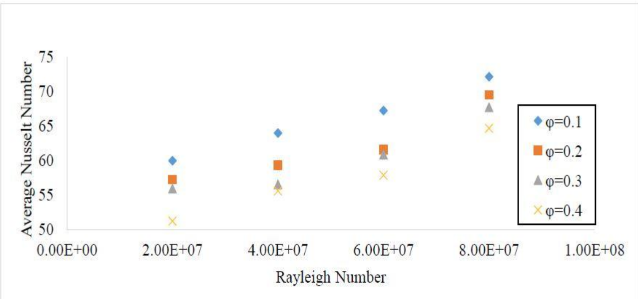

Fig 8 Variation of average nusselt number on particle concentration for AR=2 From the results obtained it is found that the as the aspect ratio is increasing the nusselt number is showing a decreasing trend on Rayleigh number in a given volume concentration. Enclosure having high aspect ratios shows high values of heat transfer coefficient, but due to the conductive dominance of the heat transfer performance, high aspect ratios of the enclosure shows lower values of the nusselt number. The convective heat transfer performance is better for the enclosure having low aspect ratio due to multicellular flow patterns formation in the enclosure.

As the length of the enclosure increases the surface for the conduction heat transfer increases which reduces the convective heat transfer, so that the convective heat transfer coefficient reduces and the thermal conductivity increases. This decreasing convective heat transfer coefficient and the increase in the thermal conductivity are responsible for the decreasing in the average nusselt number of the heated plate.

5. Conclusions

In the present study the numerical analyses of natural convection is studied on an enclosure filled with Al2O3 and TiO2 nano particle dispersed in the water was investigated using numerical methods. From the study it was clearly found out that as the Rayleigh number increases the heat transfer coefficient of the heated plate is enhanced when nano fluid is used compared to the conventional heat transfer fluid. It is found that the flow configuration which is analyzed showed a maximum enhancement for TiO2 and Al2O3 are 20% and 29% respectively in the concentration range of 0.1-0.4%

From the results obtained it is found that the as the aspect ratio is increasing the nusselt number is showing a decreasing trend on Rayleigh number in a given volume concentration. Enclosure having high aspect ratios shows high values of heat transfer coefficient, but due to the conductive dominance of the heat transfer performance, high aspect ratios of the enclosure shows lower values of the nusselt number. The convective heat transfer performance is better for the enclosure having low aspect ratio due to multicellular flow patterns formation in the enclosure.

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