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Effect of Nano Particle and Aspect Ratio in Natural Convection Heat Transfer in a Rectangular Enclosure: A Numerical Analysis

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ABSTRACT
This numerical study investigates the natural convection heat transfer characteristics of water-based nanofluid in a rectangular enclosure. Nanofluid contains copper as nanoparticle. The effect of volume fraction of particle (φ) and Aspect ratio (A) have been studied in this numerical study. A series of Direct Numerical Simulation (DNS) have been conducted into the range of $0 \leq \varphi \leq 0.1$ and $0.5 \leq A \leq 2.0$ at fixed Rayleigh Number, $Ra = 5 \times 10^6$. A Commercial software ANSYS Fluent v16.1 (student version) has been used for these simulations. These numerical results demonstrate that the heat transfer rate increases almost linearly with respect to the particle volume fraction, however Nusselt number (Nu) decreases. Numerical results also demonstrate that heat transfer rate increases with respect to Aspect ratio up to $A = 1$, after that it starts to decrease. .

Key words: Convection, Aspect Ratio, Nanofluid, Volume fraction, DNS.

1. Introduction

Enhancement of heat transfer in the systems is an essential topic from an energy saving perspective. Over the past decade, nanofluids, have been reported to possess substantially higher thermal conductivity for example, copper has a thermal conductivity 700 time greater than water and 3000 greater than engine oil. This makes them very attractive as heat transfer fluids in many applications. Nanofluids would be useful as coolants in the automobile and electronics industries. However, the reported high thermal conductivity sometimes cannot be reproduced, and the potential mechanisms leading to the enhancement are still under scrutiny. Due to these reasons, nanofluids have been a controversial topic [1]. The past decade has witnessed several studies of convective heat transfer in nanofluids. Khanafer et al. [2] were the first to investigate the problem of buoyancy-driven heat transfer enhancement of nanofluids in a two-dimensional enclosure. Putra et al [3] did the same. Jou and Tzeng [4] numerically investigated the heat transfer performance of nanofluids inside two dimensional rectangular enclosures. Their results show that increasing the volume fraction causes a significant increase in the average heat transfer coefficient. But there is a contradiction in case of Nusselt number's response with volume fraction. Santra et al. [5] have conducted a similar kind of study, up to $\varphi = 10\%$, using the models proposed by Maxwell-Garnett and Bruggeman. Hwang et al. [6] have carried out a theoretical investigation of the thermal characteristics of natural convection of an alumina-based nanofluid in a rectangular cavity heated from below using Jang and Choi's model [7] for predicting the effective thermal conductivity of nanofluids (and various models for predicting the effective viscosity). Oztop and Abu-Nada [8] investigated heat transfer and fluid flow due to buoyancy forces in a partially heated enclosure using nanofluids using various types of nanoparticles.

The natural convection studies corresponding to the parallelepiped enclosures can be classified into two elementary classes: i) heating from a horizontal wall (heating from below); ii) heating from a vertical wall. Calcagni et al. [9] made an experimental and numerical study of free convective heat transfer in a square enclosure characterized by a discrete heater located on the lower wall and cooling from the lateral walls. A steady laminar natural convection in 2D enclosures heated from below and cooled from above for a wide variety of thermal boundary conditions at the sidewalls has been carried numerically by Corcione [10]. A numerical investigation of natural convection of air in a vertical square cavity with localized isothermal heating from below and symmetrical cooling from the side walls was investigated by Aydin and Yang [11]. The same problem by replacing a constant flux heat source instead the localized isothermal heat source at the bottom wall has been analyzed by Sharif and Mohammad [12]. They investigated the effect of aspect ratio and inclination of the cavity on the heat transfer process. The effect of heater and cooler locations on natural convection in square cavities has been reported by Turgoklu and Yucel [13]. Natural convection in a square enclosure heated periodically from part of the bottom wall has been investigated by Lakhal et al. [14]. There are good number of papers which deal with natural convection with non-uniform temperature boundary conditions, for example, natural convection in rectangular enclosure with sinusoidal temperature on the upper wall and adiabatic boundary condition on rest walls Sarris et al. [15] , natural convection in a square cavity with the different boundary conditions: uniform as well as nonuniform heating of bottom as well as side walls Roy et al. [16], and cooling by sinusoidal temperature profiles on equally divided active side wall with other sides are insulated Bilgen and Yedder [17]. Natural convection in air-filled 2D square enclosure heated with a constant source from below and cooled from above is

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studied numerically by Nader et al. [18]. A numerical study to investigate the steady laminar natural convection flow in a square cavity with uniformly and non-uniformly heated bottom wall, and adiabatic top wall maintaining constant temperature of cold vertical walls has been performed by Basak et al. [19] with the help of penalty finite element method. In the same geometry, the numerical study deals with natural convection flow in a closed square cavity when the bottom wall is uniformly heated and vertical wall(s) are linearly heated whereas the top wall is well insulated has been reported by Sathiyamoorthy et al. [20].

In this paper effect of volume fraction of nanoparticle in heat transfer in a rectangular enclosure with heating from vertical wall, is observed. Effect of aspect ratio of the enclosure filled with nanofluid also investigated.

2. Physical Model

Fig. 1 shows a schematic diagram of a rectangular enclosure with heating from left vertical wall. The fluid in the enclosure is a water based nanofluid containing Cu as nanoparticles. The problem is solved in transient state. The flow is assumed to be laminar. It is assumed that the base fluid (i.e. water) and the nanoparticles are in thermal equilibrium and no slip occurs between them. The thermo-physical properties of the cu Nano particle and base fluid are given in Table 1. The left wall is hot wall, maintained at a temperature T_h. The right wall is cold wall, maintained at temperature T_c. Temperature of left wall is higher than the right wall. Top and bottom walls are insulated. All four walls remain stationary. Here we assume that fluid velocity at all fluid–solid boundaries is equal to that of the solid boundary i.e. no slip condition and fluid temperature at all fluid–solid boundaries is equal to that of the solid boundary wall temperature i.e. no jump condition. The thermo-physical properties of the nanofluid are assumed to be constant except for the density variation, which is approximated by the Boussinesq model.

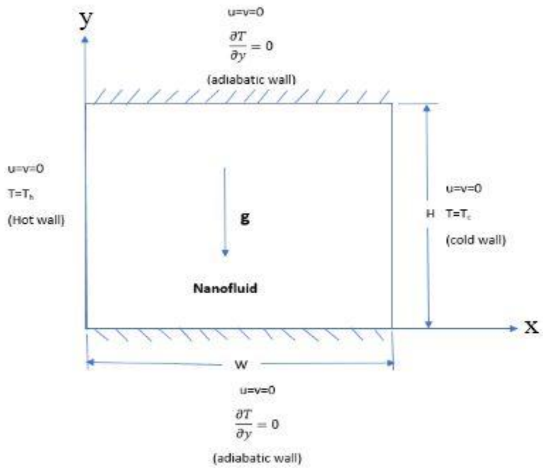


Fig.1 A typical sketch which shows the Problem with necessary condition.

3. Computational Details

The general momentum equation is also called the equation of motion or the Navier-Stoke's equation; in addition, the equation of continuity is frequently used in conjunction with the momentum equation. The equation of continuity is developed simply by applying the law of conservation of mass to a small volume element within a flowing fluid. The governing continuity, momentum and energy equations are-

Continuity: $$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) = 0 \tag{1}$$

x momentum:
$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + \mathrm{v}\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\mu}{\rho}\left(\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y}\right) \tag{2}$$

y momentum:
$$\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + \mathrm{v}\frac{\partial v}{\partial y} = -\frac{1}{\rho_{nf}}\frac{\partial p}{\partial y} + \frac{\mu_{nf}}{\rho_{nf}}\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) - \frac{1}{\rho_{nf}}(\rho\beta)_{nf}\, g(T - T_c) \tag{3}$$

Energy: $$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + \mathrm{v}\frac{\partial T}{\partial y} = \alpha_{nf}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) + \frac{q''}{\rho c_p} \tag{4}$$

Here the term -ρg on the right side of the eq. (3) represents the body force exerted on the fluid element in the negative y direction [21].

The Boundary conditions are

At the time t = 0,

$$T = \frac{T_c + T_h}{2}$$

u = v = 0; at all (x,y)

At the time t > 0,

Left wall: u = v =0 $T=T_h$

Right wall u = v =0 $T=T_c$

Top and bottom walls are adiabatic.

Time step size is 0.1 sec, which is shown time dependency test and the total time for the calculation is 120 seconds.

The following dimensionless parameters are defined to show different results later:

$$X = \frac{x}{W} \qquad Y = \frac{y}{H} \qquad \theta = \frac{T - T_c}{T_h - T_c} \tag{5}$$

The effective properties of nanofluid are calculated as follows-

$$\rho_{nf} = (1-\varphi)\rho_f + \varphi\rho_s \tag{6}$$

$$\alpha_{nf} = \frac{k_{eff}}{(\rho c_p)_{nf}} \tag{7}$$

The heat capacitance of the nanofluid is expressed as (Abu-Nada [22]; Khanafer *et al*. [2]):

$$(\rho c_p)_{nf} = (1-\varphi)(\rho c_p)_f + \varphi(\rho c_p)_s \tag{8}$$

The effective thermal conductivity of the nanofluid is approximated by the Maxwell–Garnetts [23] model

$$\frac{k_{nf}}{k_f} = \frac{k_s + 2k_f - 2\varphi(k_f - k_s)}{k_s + 2k_f + \varphi(k_f - k_s)} \quad (9)$$

The use of eq. (9) is restricted to spherical nanoparticles where it does not account for other shapes of nanoparticles. The viscosity of the nanofluid can be approximated as viscosity of a base fluid lf containing dilute suspension of fine spherical particles and is given by Brinkman [24]:

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

Rayleigh no. and Nusselt numbers are defined as-

$$Ra = \frac{g\beta\Delta T H^3}{\upsilon\alpha} \qquad N_u = \frac{hH}{k}$$

Physical property of the Cu-water nanofluid is determined from following Table 1.

Table 1 Thermo-physical properties of base fluid and Nano-particles [25]

Property		Water	Cu
ρ	(kg/m^3)	997.1	8933
C_p	(J/kgK)	4179	385
K	(W/mK)	0.613	400
α*10^7	(m^2/s)	1.47	1163.1
β	(K^{-1})	0.00021	0.000051

4. Results and Discussion

This section includes mesh or grid independence test, time dependency test, model validation with some previously published paper and finally effect of volume fraction and aspect ratios are shown.

4.1 Grid Independence Test

The solutions for different mesh size have been studied in order to determine independence of each solution. Five structured meshes are used and temperature at midplane in y direction are plotted along x direction. Fig. 2 shows variation of temperature with x coordinate. It is clear from graph that there is slight or no change in temperature curve. Finally, 100×100 mesh is selected for square cavity. For other aspect ratios mesh number is selected with reference to this.

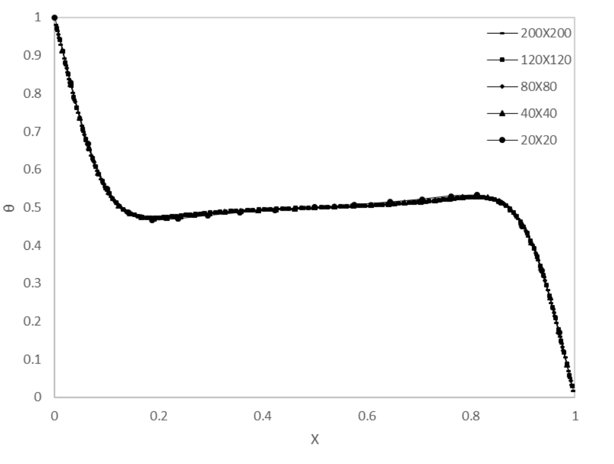


Fig.2 Variation of static temperature along x direction for different mesh (A=1, Ra=5×10^6, φ = 0.1).

4.2 Time Dependency Test

In a similar fashion as mesh dependency test, for different residual values and time step sizes temperature is plotted against x coordinate. Finally, residual value for energy equation is selected as 10^{-9} & with respective others residual as 10^{-6}. And time step size is selected as 0.1 sec.

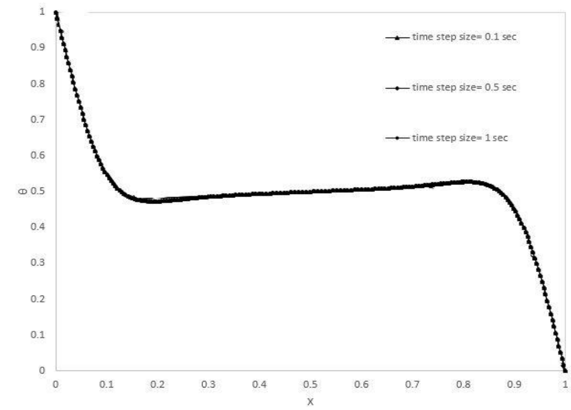


Fig.3 Variation of static temperature along x direction (midplane in y direction) for different time step size (A=1, Ra=5×10^6, φ = 0.1)

4.3 Model Validation

The problem is solved by finite-volume based commercial software package ANSYS Fluent v16.1 (student version). It has been validated against solutions obtained in the literature as shown in Table 2. Natural convection of air inside a square cavity whose two sides are set to differential temperatures while keeping the top and bottom surfaces at adiabatic condition is a classic case for validation. Average Nusselt number for the hot wall calculated from the present model is compared with the data available in the literature and found good agreement for various Rayleigh numbers.

Table 2 Comparison of average Nusselt number for the hot wall obtained by various studies with present study for different Rayleigh numbers.

Ra	Nu						
	Present work	Vahl Davis [22]	Fusegi *et al.*[23]	Comini *et al.* [24]	Khanafer *et al.* [9]	Bilgen [25]	Kobra [26]
10^4	2.240668	2.243	2.302		2.245	2.245	2.2448
10^5	4.526406	4.519	4.646	4.503	4.522	4.521	4.5216
10^6	8.862388	8.799	9.012	8.825	8.826	8.8	8.8262

4.4 Effect of Volume Fraction of Nanoparticle on Heat Transfer

Contours of static temperature are shown in fig.4. Contours are drawn for the time of 120 sec. For same time, with increasing volume fraction there is significant difference in the contours. For lower volume fraction change in temperature distribution is less than that of higher volume fraction. For high volume fraction change in temperature distribution in left upper side and right lower side is more prominent than lower volume fraction. Similarly, in case of stream function (fig.5), maximum value of stream function increases with increase in volume fraction and contour takes a regular shape in higher volume fraction. The change is due to equivalent thermal conductivity.

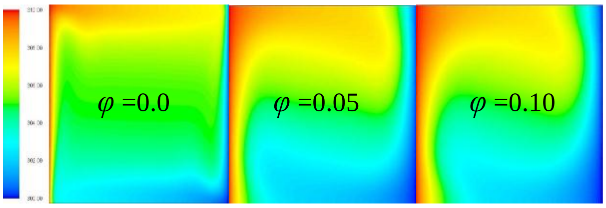


Fig.4 Contours of Static temperature with and without Nano particle (Ra=5×10^6, time=120 sec).

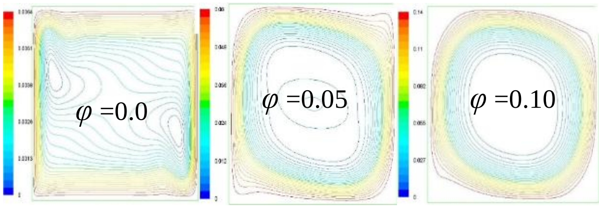


Fig.5 Contours of stream function(streamlines) with and without Nano particles (Ra=5×10^6, time=120 sec).

Fig. 6 shows variation of local heat transfer coefficient and Nusselt number at hot wall and fig.7 shows average heat transfer coefficient and Nusselt number at hot wall for different volume fraction. Fig. 6 shows that local heat transfer coefficient increases with increase in volume fraction whereas local Nusselt number decreases with increases in volume fraction. Considering single graph in fig.6 it is seen that heat transfer coefficient first increases to a maximum value, then starts to decrease and goes to a minimum value. Fig. 7 shows that average heat transfer coefficient increases with increase in volume fraction and Nusselt number decreases with increase in volume fraction. Which justify the previous studies [2-3]**.** The reason behind this is, with increase in convective heat transfer coefficient thermal conductivity also increases and increase in conductive heat transfer is more prominent than convection.

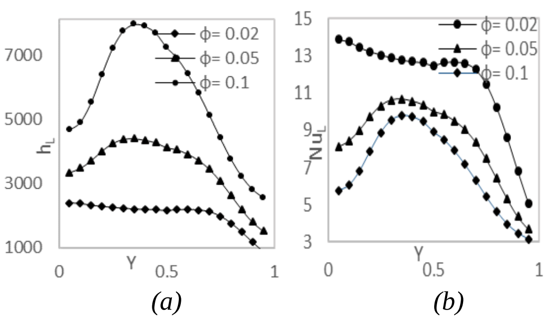


Fig.6 Variation of local (a) convective heat transfer coefficient and (b) Nusselt number along hot wall for different volume faction (Ra=5×10^6, time=120 sec)

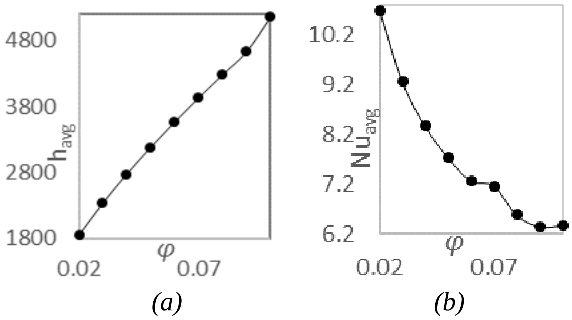


Fig.7 Variation of average (a) convective heat transfer coefficient and (b) Nusselt number with volume fraction along hot wall (Ra=5×10^6, time=120 sec).

4.5 Effect of Aspect Ratio of Enclosure on Heat Transfer

Contours of static temperature are shown in fig. 8. Contours are drawn for the time of 120 sec. For same time, with increasing aspect ratio there is significant difference in the contours. Analyzing, contours it is prominent that temperature distribution is maximum when aspect ratio is unity. But maximum magnitude of stream function (fig. 9) increases up to aspect ratio 1.5 then start to decrease.

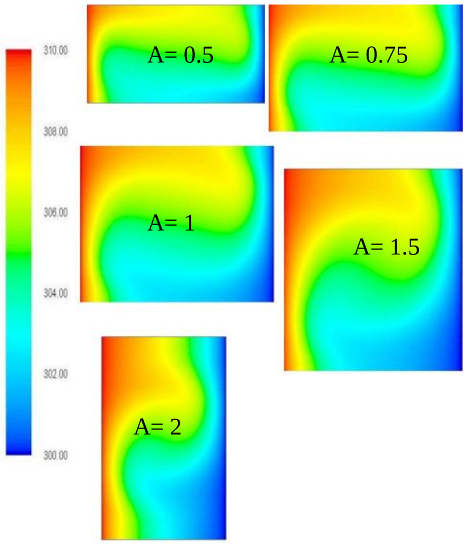


Fig. 8 Contours of Static temperature for different Aspect Ratio (Φ = 0.10, Ra=5×10^6, time=120 sec).

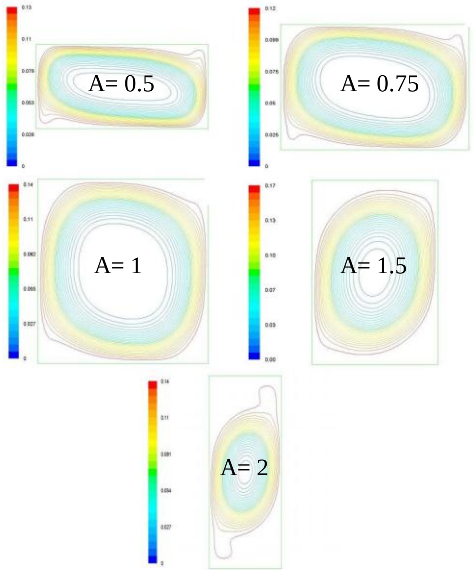
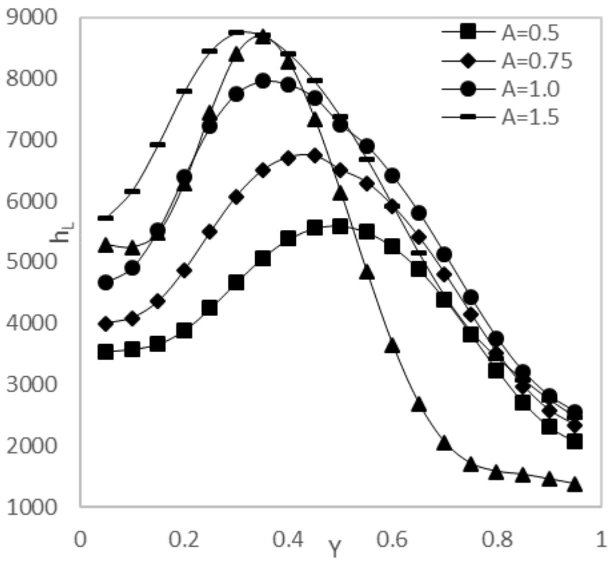
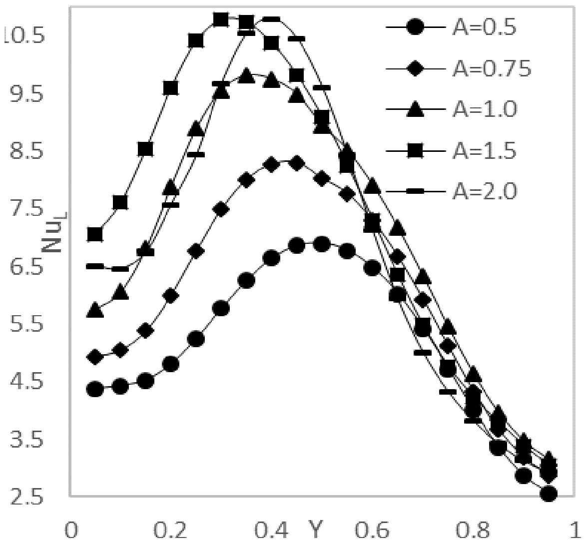


Fig. 9 Contours of Stream function (Streamlines) for different Aspect Ratio ($\Phi = 0.10$, Ra=5×10^6, time=120 sec).

Variation of local heat transfer coefficient and Nusselt number are shown in fig. 10. Fig. 10(a) shows that local heat transfer coefficient increases with increase in aspect ratios. If aspect ratio further increased, maximum value of heat transfer coefficient also increases, but after a peak value it decreases sharply than that for other aspect ratios. Similar phenomena occur for local Nusselt number, as shown in fig.10(b). Considering figure 11, it is clear that average heat transfer coefficient as well as average Nusselt number increases with increase in aspect ratios up to one, after that, start to decrease. Which justify the previous investigations [30].



(a)



(b)

Fig. 10 Variation of local (a) convective heat transfer coefficient and (b) Nusselt number along hot wall for different aspect ratio ($\Phi = 0.10$, Ra=5×10^6, time=120 sec).

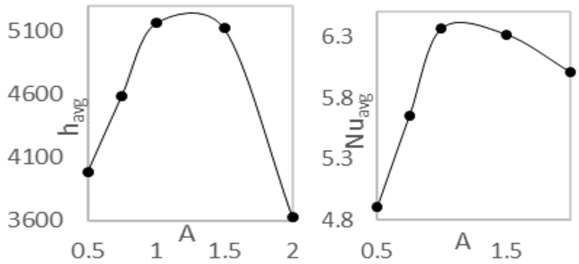


Fig.11 Variation of average (a) convective heat transfer coefficient and (b) Nusselt number with aspect ratio along hot wall ($\Phi = 0.10$, Ra=5×10^6, time=120 sec).

5. Conclusion

A comprehensive investigation on natural convection in a rectangular enclosure filled with nanofluid is presented. The investigation is done to show the enhancement in heat transfer due to use of nanofluid instead of using pure fluids. The parameters investigated are, the solid volume fraction and aspect ratio. The results clearly show that-

- The amount of heat transfer is increased remarkably with increase in volume fraction of nanoparticles, but the Nusselt number decreases with increase in the same.

- Amount of heat transfer as well as Nusselt number increased with increase in aspect ratio up to one, beyond this value heat transfer and Nusselt number decreased. It is found that maximum heat transfer (and Nusselt number) is obtained at aspect ratio One.

REFERENCES

[1]http://web.mit.edu/nanoengineering/research/nanofluids.shtml

[2] Khanafer, K., Vafai, K., Lightstone, M., 2003. Buoyancy-driven heat transfer enhancement in a two-dimensional enclosure utilizing nanofluids 46, 3639–3653

[3] Nandy Putra, Wilfried Roetzel, Sarit K Das. Natural convection of nano-fluids. *Int journal of Heat and Mass Transfer* 39(2003) 775-784

[4] R.Y. Jou, S.C. Tzeng, Numerical research of nature convective heat transfer enhancement filled with nanofluids in rectangular enclosures, *Int. Commun. Heat Mass Transfer* 33 (2006) 727–736

[5] A .K. Santra, S. Sen, N. Chakraborty, Study of heat transfer augmentation in a differentially heated square cavity using copper–water nanofluid, *Int. J.Thermal Sci*. 47 (2008) 1113–1122

[6] K.S. Hwang, J.H. Lee, S.P. Jang, Buoyancy-driven heat transfer of water-based Al_2O_3 nanofluids in a rectangular cavity, *Int. J. Heat Mass Transfer* 50 (2007) 4003–4010

[7] S.P. Jang, S.U.S. Choi, The role of Brownian motion in the enhanced thermal conductivity of nanofluids, *Appl. Phys. Lett.* 84 (2004) 4316–4318.

[8] Oztop Hakan F, Abu-Nada Eiyad. Numerical study of natural convection in partially heated rectangular enclosures filled with nanofluids. *Int Journal of Heat and Fluid Flow* 29 (2008) 1326–1336

[9] B. Calgagni, F. Marsili, M. Paroncini, Natural convective heat transfer in square enclosures heated from below, *Appl. Therm. Eng*. 25 (20 05) 2522–2531.

[10] Corcione M. Effects of the thermal boundary conditions at the sidewalls upon natural convection in rectangular enclosures heated from bellow and cooled from above. *Int J Therm Sci* 2003;42:199–208.

[11] Aydin O, Yang WJ. Natural convection in enclosures with localized heating from below and symmetrical cooling from sides. *Int J Numer Methods Heat Fluid Flow* 2000;10:5: 519–529.

[12] Sharif MAR, Mohammad TR. Natural convection in cavities with constant flux heating at the bottom wall and isothermal cooling from the sidewalls. *Int J Therm Sci* 2005;44:865–78.

[13] Turgoklu H, Yucel N. Effect of heater and cooler locations on natural convection in square cavities. *Numer Heat Transfer* 1995;A 27:351–8.

[14] Lakhal EK, Hasnaoui M, Vasseur P, Bilgen E. Natural convection in a square enclosure heated periodically from part of the bottom wall. *Numer Heat Transfer* 1995;A 27:319–33.

[15] Sarris IE, Lekakis I, Vlachos NS. Natural convection in a 2D enclosure with sinusoidal upper wall temperature. *Numer Heat Transfer* 2002;A 36:513–20.

[16] Roy S, Basak T. Finite element analysis of natural convection flows in a square cavity with non-uniformly heated wall(s). *Int J Eng Sci* 2005;43:668–80.

[17] Bilgen E, Yedder RB. Natural convection in enclosure with heating and cooling by sinusoidal temperature pronles on one side. *Int J Heat Mass Transfer* 2007;50:139–50.

[18] Nader BC, Brahim BB, Taieb L. Influence of thermal boundary conditions on natural convection in a square enclosure partially heated from below. *Int Commun Heat Mass Transfer* 2007;34:369–79.

[19] Basak T, Roy S, Balakrishnan AR. Effects of thermal boundary conditions on natural convection flows within a square cavity. *Int J Heat Mass Transfer* 2006;49:4525–35.

[20] Sathiyamoorthy M, Basak T, Roy S, Pop I. Steady natural convection flows in a square cavity with linearly heated side wall(s). *Int J Heat Mass Transfer* 2007;50:766–75.

[21] Ozisik M. Necati(1985). Heat Transfer(Int. Edition).Singapore: McGraw-Hill. ISBN 0-07-047982-8

[22] Abu-Nada, E., 2008. Application of nanofluids for heat transfer enhancement of separated flows encountered in a backward facing step. *Int. J. Heat Fluid Flow* 29, 242–249.

[23] J.C. Maxwell-Garnett, Colours in metal glasses and in metallic films, *Philos. Trans. Roy. Soc*. A 203 (1904) 385–420

[24] Brinkman, H.C., 1952. The viscosity of concentrated suspensions and solutions. *J. Chem. Phys*. 20, 571–581.

[25] Ogut E.B. Natural convection of water-based nanofluids in an inclined enclosure with a heat source. Int Journal of Thermal Sciences 4 8 (2009) 20 63–2073

[26] D.V. Davis, Natural convection of air in a square cavity a bench mark numerical solution. *Int J Numer Method Fluids*, 3 (1983) 249–64.

[27] T. Fusegi, J.M. Hyun, K. Kuwahara, and B. Farouk, A numerical study of three-dimensional natural convection in a differentially heated cubical enclosure, *Int. J. Heat Mass Transfer,* 34 (1991) 1543–1557.

[28] G. Comini, G. Cortella, and M. Manzan, A stream function-vorticity-based finite element formulation for laminar convection, *Num Heat Transfer*, Part B, 28 (1995) 1-22.

[29] Kobra F, Quddus N, Alim Abdul. Heat transfer enhancement of Cu-water nanofluid filled in a square cavity with a circular disk under a magnetic field. 10th *International Conference on Mechanical Engineering*, ICME 2013.

[30] Alom parvez, Kumar Ashok, Kapoor S, Ansari S.R. Numerical investigation of natural convection in a rectangular enclosure due to partial heating and cooling at vertical walls .*Commun Nonlinear Sci Numer Simulat* 17 (2012) 2403–2414