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Computer Simulations of Natural Convection of Single Phase Nanofluids in Simple Enclosures

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Abstract

Recently, employing nanofluids as heat transfer agent becomes an upward trend and a considerable alternative to markedly enhance the heat transfer process. Effective heat transfer during heating and cooling streams is an urgent demand in chemicals, petrochemicals, and pharmaceuticals industries. Obviously, industry relies on computer simulation as a quick and effective tool to test, monitor, analyze, and modify the individual units (e.g. heat exchangers) and the entire process performance as well. In this thesis, the natural convection heat transfer of two nanofluids with different concentrations (0.1, 0.2, 0.5, 1, and 1.5 vol % of TiO_2 and Al_2O_3 nanoparticles in water) in different enclosure is examined using CFD analysis. Computer simulations are performed to find the Nusselt number and the heat transfer coefficient for natural convection of nanofluids in horizontal, tilted square, annulus and triangular enclosure. 3D modeling is done in Pro/Engineer and analysis is done in Ansys.

Keywords: *Nanofluid, Heat transfer coefficient, Single-phase model, Two-phase model.*

1. Introduction

Convective heat transfer is very important for many industrial heating or cooling equipments. The heat convection can be enhanced passively by chang- ing the flow geometry or the boundary conditions or by enhancing the fluid thermo physical properties. An innovative way of improving the thermal conductivities of fluids is to suspend small solid particles in the fluid. Maxwell (1873, 1881) showed the possibility of increasing thermal conductivity of a mixture by adding a volume fraction of solid particles. These fluids containing suspended colloidal nanoparticles have been called nanofluids. Heoretical and experimental investigations have been conducted to estimate the effective thermal conductivity of nanofluids. Some experimental stud- ies (Yoo et al., 2007; Choi et al., 2001) show that the measured thermal conductivity of nanofluids is much larger than the classical theoretical predictions (Hamilton and Crosser, 1962). Other experimental investigations (Putman et al., 2006; Zhang et al., 2007) revealed that the thermal conductivity did not show any anomalous enhancement and, for lower volume fractions, the results agree well with the classi- cal equations (Yamada and Ota, 1980; Hamilton and Crosser, 1962). Many attempts have been made to formulate efficient theoretical models for the prediction of the effective thermal conductivity, but this topic is still seriously incomplete (Chon et al., 2005; Xue, 2003; Xuan et al. 2004; Li et al., 2003). Several investigations revealed that the nanofluid heat transfer coefficient could also be increased by more than 20% in the case of very low nanoparticles concentrations (Kang et al., 2006; Xuan and Li, 2000). Relatively, few theoretical and experimental inves- tigations have been reported on convective heat transfer in confined flows, as also reviewed in the literature (Buongiorno, 2006; Das et al., 2006; Daungthongsuk and Wongwises, 2007; Wang and Mujumdar, 2007). However, only limited experimen- tal studies on the convective heat transfer of nano- fluids as modified heat transfer media have been performed, compared with many results for thermal conductivity enhancement (Chen et al., 2008; Ding et al., 2006; He et al., 2007; Pak and Cho, 1998; Wen and Ding, 2004; Li et al., 2003; Yang et al., 2005). Numerical investigations on nanofluids are car- ried out using two approaches. The first approach assumes that the continuum assumption is still valid for fluids with suspended nanosize particles (Moraveji et al., 2011), while the other approach considers a two-phase model for describing both the fluid and the solid phases (Kalteh et al., 2011). Another approach is to adopt the Boltzmann theory. The single-phase model with physical and thermal properties, all assumed to be constant with tempera- ture, was employed in several studies (Roy et al., 2004; Maiga et al., 2004; Maiga et al., 2006; Maiga et al., 2005). The hydrodynamic and thermal charac- teristics of nanofluids, flowing through a uniformly heated tube, in both laminar and turbulent regimes with adjusted properties, have been investigated (Maiga et al., 2004). The advantages of nanofluids with respect to heat transfer were discussed (Maiga et al., 2005), but it was also found that the inclusion of nanoparticles introduced drastic effects on the wall shear stress. A new correlation was proposed (Maiga et al., 2006) to describe the thermal perform- ance of Al2O3/water nanofluids under the turbulent regime and a numerical study of heat transfer for water/Al2O3 nanofluids in a radial cooling system was carried out by Roy et al. (2004). They found that the addition of nanoparticles in the base fluid increased the heat transfer rates considerably. The single-phase model with physical and thermal properties, all assumed to be constant with tempera- ture, was employed in several studies (Roy et al., 2004; Maiga et al., 2004; Maiga et al., 2006; Maiga et al., 2005). The hydrodynamic and thermal charac- teristics of nanofluids, flowing through a uniformly heated tube, in both laminar and turbulent regimes with adjusted properties, have been investigated (Maiga et al., 2004). The advantages of nanofluids with respect to heat transfer were discussed (Maiga et al., 2005), but it was also found that the inclusion of nanoparticles introduced drastic effects on the wall shear stress. 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Recently, numerous other theoretical investiga- tions by different researchers (Bianco et al., 2009; Ebrahimnia-Bajestan et al., 2011; Farsad et al., 2011; Fazeli et al., 2012; Haghshenas Fard et al., 2010; Kalteh et al., 2012; Kamali and Binesh, 2010; Kamyar et al., 2012; Mahmoodi and Hashemi, 2012; Manca et al., 2012; Rana and Bhargava, 2011; Rostamani et al., 2010; Singh et al., 2012; Tahery et al., 2011; Yang and Lai, 2011; Yu et al., 2011) on nanofluid convective heat transfer were carried out, but all of them did not consider the van der Walls interaction and agglomeration phenomena of the nanoparticles. This study aims to investigate the effect of particle agglomeration and cluster size distribution on the convective heat transfer performance of Al2O3/water nanofluids. Moreover, experiments were performed using Al2O3 nanofluids inside a straight circular tube under uniform heat flux and in the laminar flow regime. The cluster size distribution due to particle agglomeration was analyzed and used for the numerical modeling. The single-phase and the two-phase models for the prediction of nanofluid heat transfer coefficients were developed. The single- phase model with constant and variable physical properties and also the discrete particle two-phase model with particle agglomeration and clustering were considered. A commercial CFD code (Fluent, 2006) was employed to solve the governing equa- tions. The numerical simulation results were also compared with the experimental data and some interesting results were obtained.

2. Experimental Work

Materials and Nanofluid Preparation

In order to investigate the effects of nanoparticles on heat transfer, alumina nanofluids were prepared without any surfactant using deionized water as the base fluid and the two-step method with a stirrer and a sonicator. Before conducting the main heat transfer experiments, stability analysis of 0.5 vol.% alumina nanofluids at different pH were performed to investi- gate the period of the nanoparticles stability in the fluid. Figure 1 show the stability results after 26 days. The results clearly show that the stability period for the same concentration of alumina nanofluids varies with the pH value and the period of stability at low pH is greater than at high pH. Thus, in all the nanofluid experiments pH value was controlled at about 3.

Table 1: TITANIUM OXIDE NANO FLUID PROPERTIES

Volume fraction	Thermal Conductivit y (W/m-k)	Specific Heat (J/kg-k)	Density (kg/m^3)	Viscosity (kg/m-s)
0.1	0.75	3142.33	1282.3 8	0.00125
0.2	1.134	881.02	1566.5 6	0.0015
0.5	3.01	1426.33	2419.1	0.00213 1

Table 2 ALUMINUM OXIDE NANO FLUID PROPERTIES

Volume fraction	Thermal Conductivity (W/m-k)	Specific Heat (J/kg-k)	Density (kg/m^3)	Viscosity (kg/m-s)
0.1	0.78	3130.18	1295.38	0.00125
0.2	1	898.17	1592.56	0.0015
0.5	2.126	1439.5	2484.1	0.002131

3.2 MATHEMATICAL MODELING

Figure shows the geometrical configuration used in this study. The fluid enters with uniform temperature and velocity at the inlet section. The condition of axially and at the outer boundary uniform wall heat flux was considered in this study. The single-phase and two-phase models were also implemented to their predictions

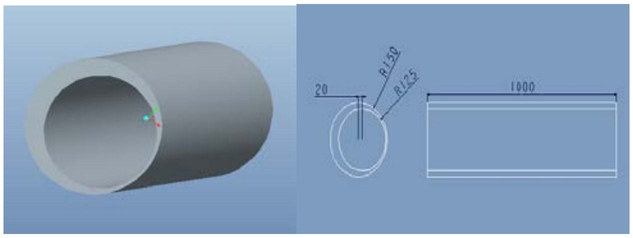


Fig 3.1 *3D & 2D HORIZONTAL SQUARE ENCLOSURE*

3.3 FORMULAS

DENSITY OF NANO FLUID

$$\rho_{nf} = \phi \times \rho_s + [(1-\phi) \times \rho_w]$$

SPECIFIC HEAT OF NANO FLUID

$$C_{p\,nf} = \frac{\phi \times \rho s \times Cps + (1-\phi)(\rho w \times Cpw)}{\phi \times \rho s + (1-\phi) \times \rho w}$$

VISCOSITY OF NANO FLUID

$$\mu_{nf} = \mu_w (1+2.5\phi)$$

THERMAL CONDUCTIVITY OF NANO FLUID

$$K_{nf} = \frac{Ks + 2Kw + 2(Ks - Kw)(1+\beta)^3 \times \phi}{Ks + 2Kw - (Ks - Kw)(1+\beta)^3 \times \phi} \times k_w$$

3.3. CFD analysis of annulus enclosure with Titanium Oxide Volume Fraction 0.1

Fig 3.2.1 Pressure

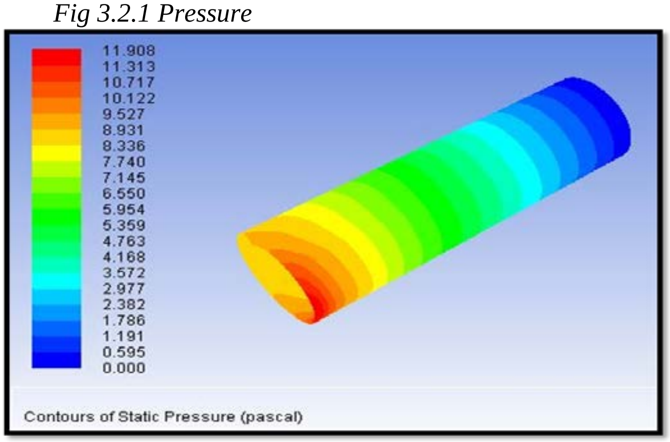
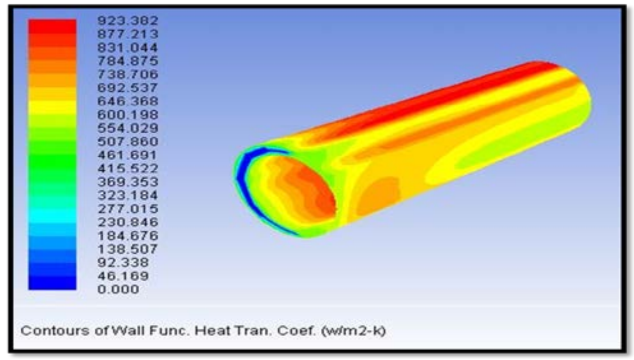
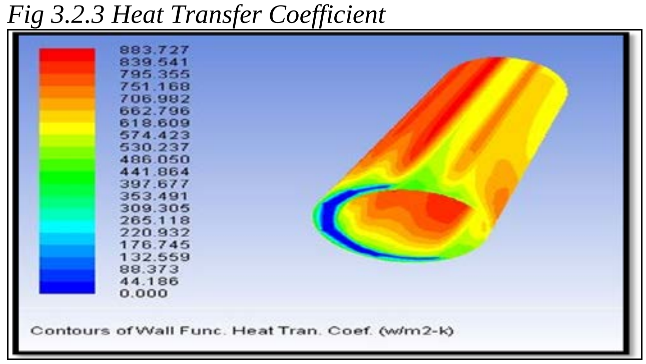


Fig 3.2.2 Heat Transfer Coefficient



Volume Fraction 0.2

Fig 3.2.3 Heat Transfer Coefficient



Volume Fraction 0.5

Fig 3.2.4 Pressure

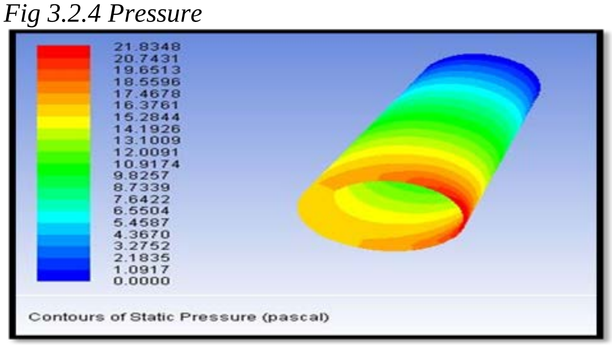
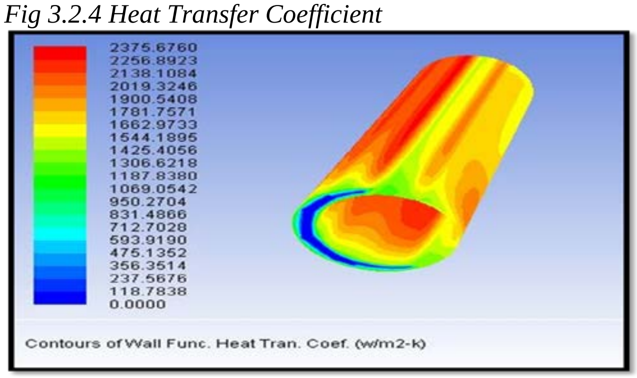


Fig 3.2.4 Heat Transfer Coefficient



CFD analysis of annulus enclosure with Aluminum Oxide Volume Fraction 0.1

Fig 3.2.7 Pressure

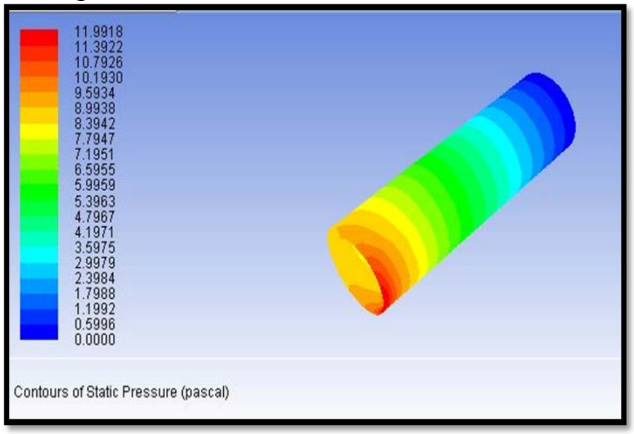
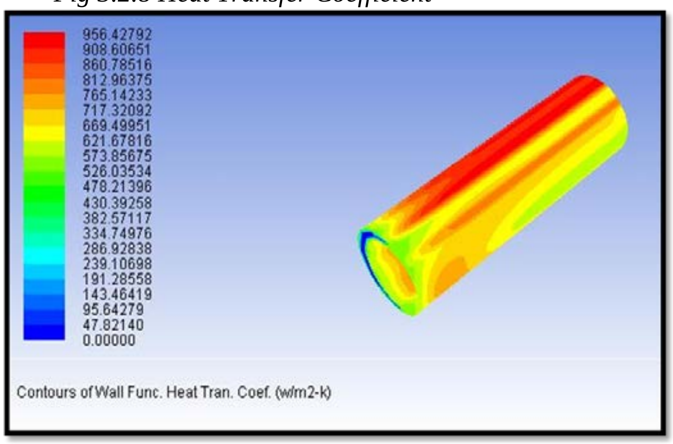


Fig 3.2.8 Heat Transfer Coefficient



Volume Fraction 0.2

Fig 3.2.9 Pressure

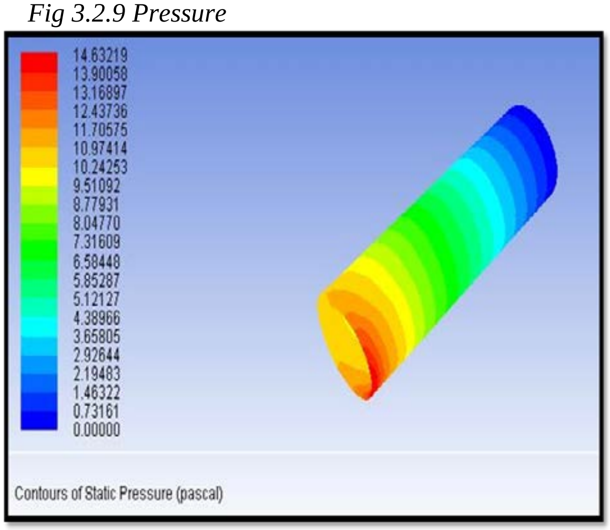
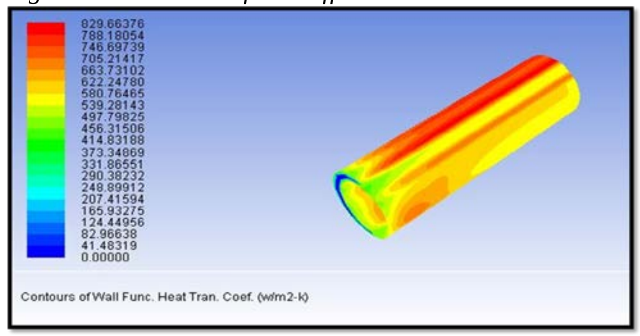
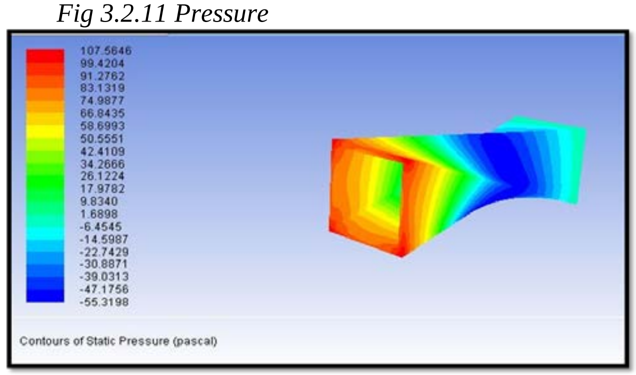


Fig 3.2.10 Heat Transfer Coefficient



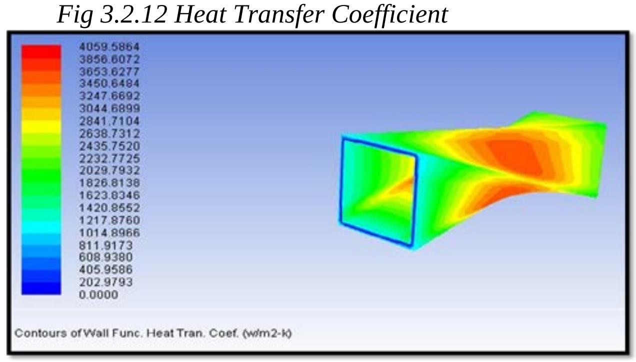
TWISTED SQUARE TITANIUM OXIDE Volume Fraction 0.5

Fig 3.2.11 Pressure



TWISTED SQUARE ALUMINUM OXIDE

Fig 3.2.12 Heat Transfer Coefficient



3.3 RESULT TABLES

Table 3.3.1 ANNULUS ENCLOSURE

Material	Volume fraction	Pressure (Pa)	Heat transfer coefficient (W/m^2-k)
TiO_2	0.1	11.908	923.382
	0.2	14.467	883.727
	0.5	21.8348	2375.6760
Al_2O_3	0.1	11.99178	956.42792
	0.2	14.63219	829.66
	0.5	22.24	1942.812

Table 3.3.2 ANNULUS ENCLOSURE

Material	Nusselt number	Heat transfer rate(W)	Mass flow rate(kg/s)
TiO_2	1271.176	10.410156	0.0006895
	779.3007	3.5878	0.00082159
	789.2611	9.69921	0.00136566
Al_2O_3	1226.189	10.87109	0.0006999
	829.66	3.65429	0.00083123
	913.8344	9.9921	0.0014095

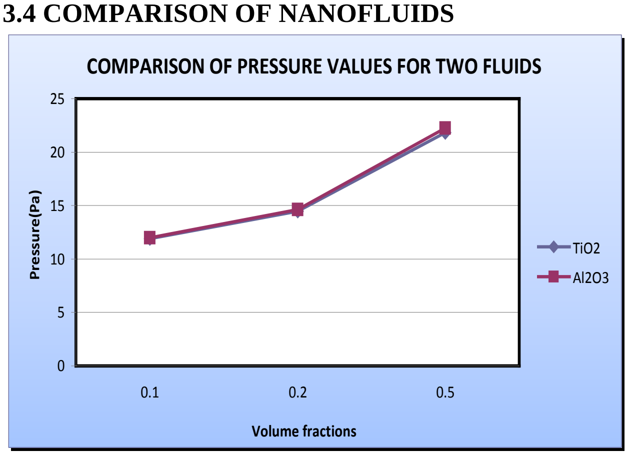
Table 3.3.3 TWISTED SQUARE ANNULUS ENCLOSURE

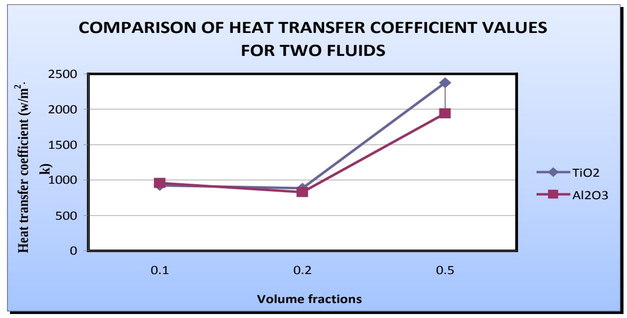
Material	Volume fraction	Pressure (Pa)	Heat transfer coefficient (W/m^2-k)
TiO_2	0.1	59.58	1882.54
	0.2	72.48	1857.93
	0.5	107.5646	5017.0420
Al_2O_3	0.1	60.0395	1951.1440
	0.2	73.394	1736.7971
	0.5	110.0846	4059.5864

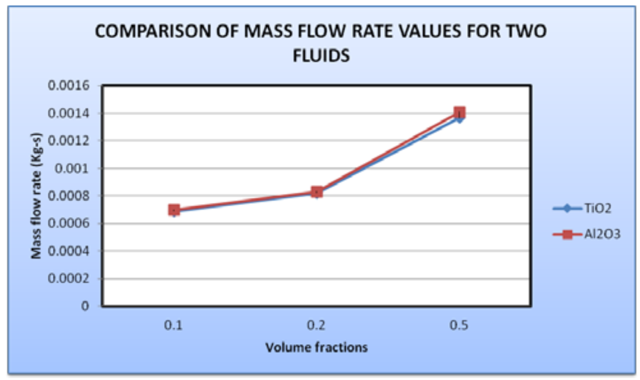
Table 3.3.4 TWISTED SQUARE ANNULUS ENCLOSURE

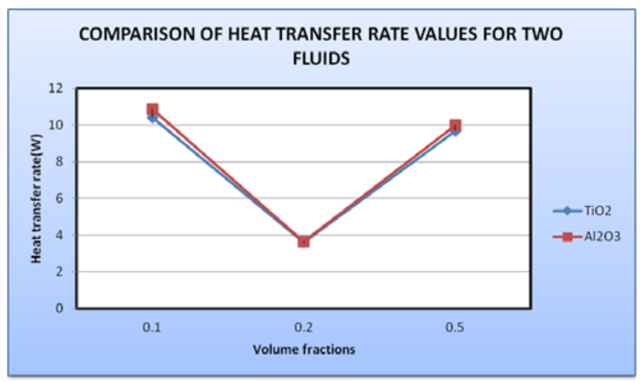
Material	Nusselt number	Heat transfer rate(W)	Mass flow rate(kg/s)
TiO_2	2510.0533	65.25	0.0043005
	1638.3862	23.2988	0.005444
	1666.791	96.281	0.013877
Al_2O_3	2501.466	67.398	0.004435
	1736.7971	24.888	0.005720
	1909.495	78.2734	0.01115

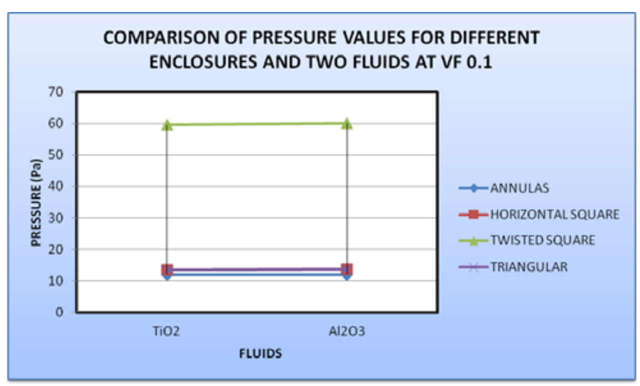
3.4 COMPARISON OF NANOFLUIDS











4. Conclusions

I conclude that computer simulations are performed to find the Nusselt number and the heat transfer coefficient for natural convection of nanofluids in horizontal, tilted square, annulus and triangular enclosure. The output values considered for comparison are pressure, heat transfer coefficient, heat transfer rate, nusselt number and mass flow rate.

1. By comparing the results between different enclosures, the heat transfer coefficient, Nusselt number and heat transfer rate are more when twisted square enclosure is taken.
2. By comparing the results between fluids, the values are better for Al_2O_3 than TiO_2.

So it can be concluded that using twisted square enclosure and fluid TiO_2 is better.

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