International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:5, No:9, 2011

A CFD Study of Heat Transfer Enhancement in Pipe Flow with Al_2O_3 Nanofluid

P.Kumar

***Abstract*—**Fluids are used for heat transfer in many engineering equipments. Water, ethylene glycol and propylene glycol are some of the common heat transfer fluids. Over the years, in an attempt to reduce the size of the equipment and/or efficiency of the process, various techniques have been employed to improve the heat transfer rate of these fluids. Surface modification, use of inserts and increased fluid velocity are some examples of heat transfer enhancement techniques. Addition of milli or micro sized particles to the heat transfer fluid is another way of improving heat transfer rate. Though this looks simple, this method has practical problems such as high pressure loss, clogging and erosion of the material of construction. These problems can be overcome by using nanofluids, which is a dispersion of nanosized particles in a base fluid. Nanoparticles increase the thermal conductivity of the base fluid manifold which in turn increases the heat transfer rate. In this work, the heat transfer enhancement using aluminium oxide nanofluid has been studied by computational fluid dynamic modeling of the nanofluid flow adopting the single phase approach.

***Keywords*—**Heat transfer intensification, nanofluid, CFD, friction factor

I. Introduction

MANY industrial processes involve heat transfer, which is accomplished using heat transfer fluids such as water, ethylene glycol and engine oil. Thermal properties of these fluids determine the thermal efficiency as well as the size of the equipments. Hence, many different techniques are being employed to improve the thermal properties of these fluids, especially the thermal conductivity. Addition of milli or micro sized solid particles is one of the very old techniques of heat transfer enhancement. Industrially, this technique is not attractive because of the inherent problems such as sedimentation, increased pressure drop, fouling and erosion of the flow channel. These problems can be overcome with nanofluids, which is a dispersion of nanosized particles in a base fluid. The nanosized particles increase the thermal conductivity of the base fluid which in turn increases the heat transfer rate. This property has attracted the attention of researchers in the past decade, though the mechanism is not fully understood yet.

A lot of work has been done recently on the forced convective heat transfer of nanofluids in pipe flow. Wen and Ding [1] studied the convective heat transfer in the entrance region under laminar regime using aluminium oxide nanofluid in a circular tube with constant heat flux. Migration of nanoparticles and the subsequent disturbance of the boundary layer were attributed to the enhancement in heat transfer rate. Zeinali Heris et al [2] compared the heat transfer enhancement by copper and aluminium oxide nanofluids in laminar pipe flow under constant wall temperature conditions and found the aluminium oxide nanofluid better than the copper oxide nanofluid. Hwang et al [3] measured pressure drop and heat transfer coefficient in fully developed laminar pipe flow using constant heat flux conditions. Based on the experimental results they showed that the experimental friction factor was in good agreement with the theoretical predictions using the Darcy equation. Whereas, the Shah equation for heat transfer coefficient prediction under constant heat flux conditions in the laminar regime was found to be inadequate for nanofluids. The enhancement in heat transfer coefficient was found to exceed the enhancement in thermal conductivity by a large margin. The flattened velocity profile caused by the particle migration to the centerline of pipe was proposed to be the possible mechanism for convective heat transfer enhancement.

Kim et al [4] conducted experiments with aluminium oxide and amorphous cabonic nanofluids in the laminar and turbulent regimes and concluded that the mechanism for heat transfer enhancement was different for the two regimes. The delaying and disturbance of the thermal boundary layer was attributed to the heat transfer enhancement in the laminar regime. Whereas, in the turbulent regime, increase in thermal conductivity was responsible for heat transfer enhancement. Rea et al [5] investigated the laminar convective heat transfer and pressure loss for alumina – water and zirconia – water nanofluids in a uniformly heated vertical tube. Heat transfer enhancement was observed to be higher in the entrance region than in the fully developed region. The agreement between the experimental and predicted Nusselt numbers was found to be good. This prompted the authors to conclude that nanofluids behave like homogeneous mixtures and the enhancement in heat transfer was only due to the improved mixture properties with respect to that of water. Ben Mansour et al [6] experimentally investigated the thermally developing laminar mixed convection flow of water and Al_2O_3 mixture inside an inclined tube with a uniform wall heat flux. They observed that a higher particle volume concentration clearly

P.Kumar is with the school of Engineering and Science, Curtin University, Sarawak, Malaysia. (phone: 60-85-44 3825; fax: 60-85-44 3838; e-mail: p.kumar@curtin.edu.my).

induces a decrease of the Nusselt number for the horizontal inclination. On the other hand, for the vertical one, the Nusselt number remains nearly constant with an increase of particle volume concentration from 0 to 4%. The apparent contradictory behavior observed between experimental data and analytical/numerical results regarding the heat transfer enhancement of nanofluids prompted them raise serious concerns regarding the applicability of using the single phase and homogeneous fluid model for nanofluids under natural convection effect. Nassan et al [7] compared the performance of Al_2O_3 – water and CuO – water nanofluids in a square duct. They also suggested further theoretical and experimental investigations to understand the heat transfer characteristics of nanofluids in noncircular ducts like triangular ducts, rectangular ducts with different aspect ratios and other possible noncircular ducts with different nanofluids.

Syam Sundar and Sharma [8] conducted forced convective heat transfer experiments in the turbulent regime with pipes employing twisted tape inserts with and without Al_2O_3 – water nanofluid. The increase in pressure drop for nanofluids was found to be negligible, whereas considerable increase in heat transfer coefficient was observed both with and without pipe inserts**.** They also developed generalized correlations for the estimation of Nusselt number and friction factor for pipes with and without inserts. Farajollahi et al [9] compared the heat transfer enhancement of Al_2O_3 – water and TiO_2- water nanofluids in a shell and tube heat exchanger. They observed different optimum volume concentration for both the nanofluids in which the heat transfer characteristics showed maximum enhancement. The nanoparticle with less mean diameter (TiO_2 nanoparticle) had a lower optimum volume concentration. Comparison of the experimental data with that predicted by Xuan and Li [10] correlation was found to have good agreement. Vajjha et al [11] used a mixture of ethylene glycol and water as base fluids to compare the heat transfer enhancement by aluminium oxide, copper oxide and silicon dioxide nanoparticles and developed generalized correlations for the prediction of Nusselt number and friction factor for turbulent pipe flow under constant heat flux conditions.

In annular duct flow, Nasiri et al [12] observed that the heat transfer performance of aluminium oxide and titanium oxide nanofluids to be similar when the concentration was same in the turbulent regime under constant wall temperature conditions. Recently, Sonawane et al [13] observed good heat transfer enhancement with aviation turbine fuel – aluminium oxide nanofluid even at low concentrations.

Earlier numerical investigations on forced convective heat transfer considered nanofluids as a homogeneous fluid and adopted a single phase approach to predict heat transfer enhancement [14, 15]. More recently, the two phase approach has been used by some researchers, but the opinion about these two approaches is varied. Bianco et al [16] observed only a maximum of 11% difference between single and two phase results for the laminar regime. So they opined that single phase approach is good enough to test new nanofluids as it requires information about the particle and the base fluid with no reference to the mixture. Other researchers [17, 18] found the mixture model to be working better than the single phase model in the turbulent regime. Akbari et al [19] for the first time compared three different two phase models and the single phase model in the laminar regime. Single and two phase models were found to be predicting identical hydrodynamic fields but very different thermal ones. A closer look at all the experimental and numerical works reveals that most of the forced convective heat transfer studies in pipe flow have been done with constant wall flux boundary condition either in the laminar or turbulent regime. So in this work, a systematic computational fluid dynamic investigation with constant wall temperature boundary condition covering both laminar and turbulent regimes has been carried out adopting the single phase approach and the results are compared with the experimental results available in the literature.

II. CFD Modelling

A. Geometry Creation and Grid Independence Study

A circular pipe of diameter 0.017 m diameter and length 10 m was used as the geometry. Grid independence study was carried out to find out the optimum grid size without compromising the accuracy of results. Different mesh sizes were tested in order to examine the effect of number of cell volumes on the Nusselt number (Fig.1).

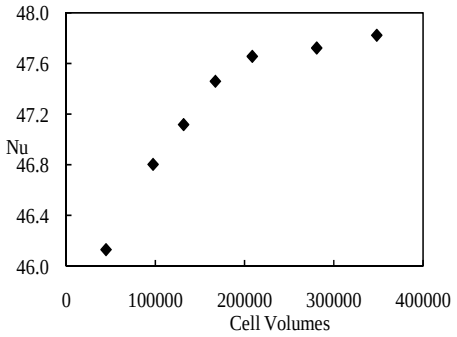

Fig.1 Grid Independence study

It can be observed that the Nusselt number for water increases linearly till an optimum number of cell volumes is reached. Beyond this, any further increase in the number of cell volumes only increases the computational time, without any significant improvement in the Nusselt number. Similar trend was also observed with the nanofluids. So this “optimum” mesh size was selected for further study with both water and the nanofluids. The thermo physical properties of the nanofluid were calculated using the correlations suggested by Syam Sundar and Sharma [8].

B. Governing Equations

Steady state simulations were carried out by solving mass, momentum and energy conservation equations, which are expressed as:

$$\frac{\partial \rho}{\partial t} + \nabla.\left(\rho \vec{u}\right) = 0 \quad (1)$$

$$\frac{\partial \left(\rho \vec{u}\right)}{\partial t} + \nabla.\left(\rho \vec{u}\vec{u}\right) = \rho g - \nabla P + \nabla.\left(\bar{\tau}\right) \quad (2)$$

$$\frac{\partial \left(\rho e\right)}{\partial t} + \nabla.\left(\vec{u}\left(\rho e + P\right)\right) = \nabla.\left(K_{eff} \nabla T + \left(\bar{\tau}_{eff}.\vec{u}\right)\right) \quad (3)$$

where,

$$\bar{\tau}_{eff} = \mu\left(\left(\nabla \vec{u} + \nabla \vec{u}^{T}\right) - \frac{2}{3}\nabla.\vec{u}I\right) \quad (4)$$

Turbulent flow was modeled using the RNG version of k-ε viscous model with thermal enhancement near the wall as swirling flow of the fluid is involved inside the pipe. The k and ε equations are as given below.

$$\frac{\partial}{\partial t}\left(\rho k\right) + \frac{\partial}{\partial x_i}\left(\rho k u_i\right) = \frac{\partial}{\partial x_j}\left(a_k \mu_{eff} \frac{\partial k}{\partial x_j}\right) + G_k + G_b - \rho\varepsilon - Y_M + S_k \quad (5)$$

$$\frac{\partial}{\partial t}\left(\rho \varepsilon\right) + \frac{\partial}{\partial x_i}\left(\rho \varepsilon u_i\right) = \frac{\partial}{\partial x_j}\left(a_\varepsilon \mu_{eff} \frac{\partial \varepsilon}{\partial x_j}\right) + C_{1\varepsilon}\frac{\varepsilon}{k}\left(G_k + C_{3\varepsilon}G_b\right) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon \quad (6)$$

C. Boundary conditions

Water at a temperature (315 *K*) was used as the working fluid. The numerical studies were carried out with uniform velocity profile at the inlet of the horizontal pipeline. The direction of the flow was defined normal to the boundary. Turbulent intensity, *I* and the hydraulic diameter, D_h were specified for an initial guess of turbulent quantities (k and ε). The turbulent intensity was estimated for each case based on the formula $I = 0.16(Re)^{-1/8}$ and was set at 5% from calculations. Outflow boundary condition was used at the outlet boundary. The wall of the pipe was assumed to be perfectly smooth with zero roughness height. A constant wall temperature of 289 K was used at the wall boundary.

D. Numerical Solution Strategy

The commercial CFD solver FLUENT 6.3.26 was used to perform the simulations, based on finite volume approach to solve the governing equations with a segregated solver. The second-order upwind scheme was used for discretization of convection terms, volume fraction, energy, turbulent kinetic and turbulent dissipation energy. This scheme ensures, in general *especially* for tri or tetrahedral mesh flow domain, satisfactory accuracy, stability and convergence. The SIMPLE algorithm was used to resolve the coupling between velocity and pressure fields. The convergence criterion is based on the residual value of calculated variables such as mass, velocity components, turbulent kinetic (k), turbulent dissipation energies (ε), energy and volume fraction. In the present calculations, the initial residual values were set to 10^{-4} for all variables, except for energy for which 10^{-6} is used. The under-relaxation factors used for the stability of the converged solutions are set at their default values. The numerical simulation was decided as converged when the sum of normalized residuals for each conservation equation and variables was less than the set residual values. However, the residual for the continuity equation reached a minimum plateau before the value of 10^{-4}, thus additionally, the mass balance is monitored on the flux report and was used as a secondary indicator of convergence when the net imbalance is less than 1% of the inlet flux through the domain boundary

E. Data Reduction

The area weighted average temperature and static pressure were noted at the inlet and outlet surfaces of the pipe. The friction factor and average heat transfer coefficients were calculated as follows.

$$f = \Delta P / 2\left((L/D)\,\rho v^2\right) \quad (7)$$

$$Q = m\,C_p\,(T_o - T_i) \quad (8)$$

$$h = Q / A\,(T_w - T_b) \quad (9)$$

Where

A - Heat transfer area

Tb - $(T_o + T_i)/2$ (10)

T_w - Wall Temperature

Nu - hD/k (11)

III. Results And Discussion

A. Validation of Numerical Results

Numerical results were made credible by comparing them with data from correlations available in the literature. Nusselt number for the base fluid (water) in the turbulent regime was compared with that of Gnielinski [20] correlation (Fig. 2). Similarly, Blasius formula from White [21] was used for friction factor comparison in the turbulent regime (Fig. 3). It can be seen from Fig. 2 that Numerical Nu are in very good agreement with that of the correlation values. Friction factor comparison is in the acceptable limit, though not as good as the Nu comparison.

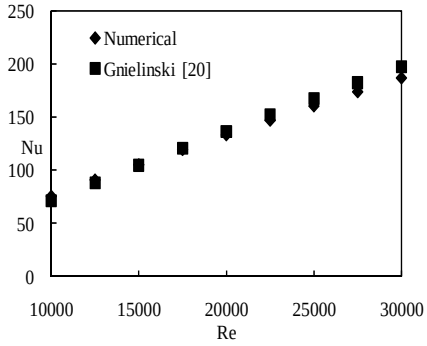

Fig. 2 Comparison of Numerical Nu for water in the turbulent regime with Gnielinski Correlation

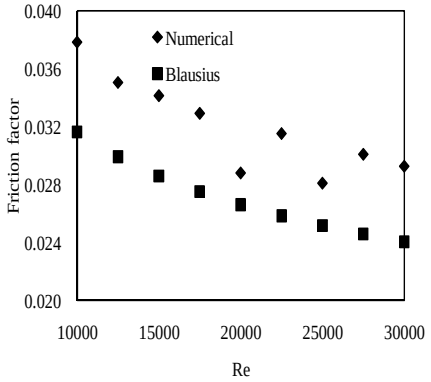

Fig. 3 Comparison of Numerical friction factor for water with Blasius Correlation in the turbulent regime

B. Turbulent Regime

The effect of nanofluid volume fraction on heat transfer enhancement is shown in Fig.4. It can be observed that at the lowest volume fraction (0.02%) there is practically no enhancement in heat transfer. But as the volume fraction increases, Nusselt number increases significantly over that of the base fluid. This can be attributed to the enhancement in the thermal conductivity of the base fluid.

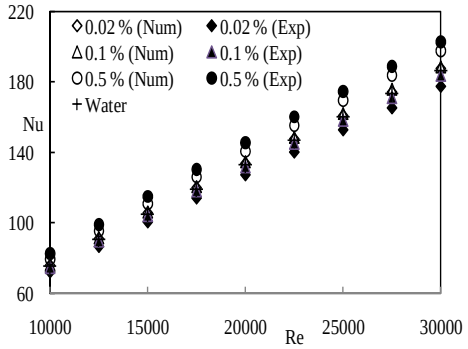

Fig. 4 Effect of volume fraction of Al_2O_3 nanofluid on heat transfer enhancement in the turbulent regime

The increase in Nusselt number becomes more accentuated as the Reynolds number increases. A threefold increase in Nusselt number is observed without increase in the pressure drop (Fig. 5). This observation suggests that the use of nanofluid is a very effective method of heat transfer enhancement. Though at higher volume fractions increase in pressure drop will become significant and it is important to find the optimum volume fraction for each application.

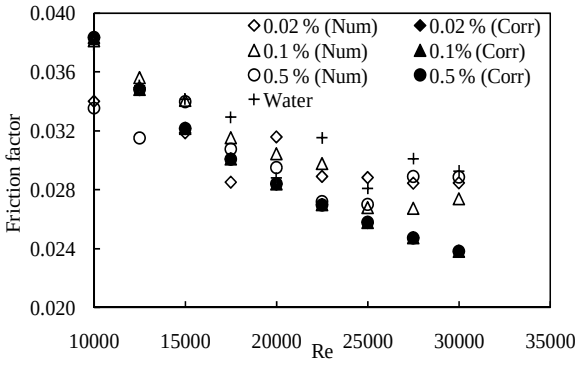

Fig. 5 Effect of volume fraction of Al_2O_3 nanofluid on pressure drop in the turbulent regime

The numerical Nusselt number and the friction factor were compared with the experimental data of Syam Sundar and Sharma [8]. It can be seen that the Nusselt number agrees very well with the experimental values, whereas the friction factor is slightly over predicted.

C. Laminar regime

Fig. 6 shows the effect of volume fraction of nanofluid on heat transfer enhancement in the laminar regime. The numerical result has been compared with the Seider – Tate [22] equation. It can be easily observed that enhancement in the laminar regime is not as significant as in the turbulent regime for all volume fractions as calculated by [22]. A similar trend is predicted by the numerical results for Re less than 600 and the comparison with Seider – Tate values is found to be good. Beyond this Re, the trend predicted by the numerical results and the correlation is very different.

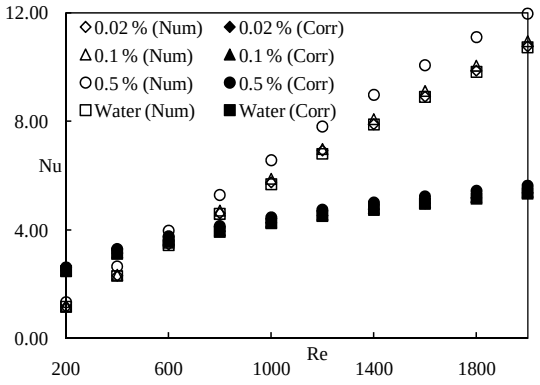

Fig. 6 Effect of volume fraction of Al_2O_3 nanofluid on heat transfer enhancement in the laminar regime

Contradictarily, Zeinali Heris et al [2] observed very good agreement between their experimental results and the correlation values. So it can be said that the single phase approach fails to predict the heat transfer in the laminar regime. Similar observation has also been made by Akbari et al [19]. They reported the single phase model to under predict heat transfer for laminar flow mixed convection with constant heat flux boundary condition. Interestingly, Bianco et al [16] for the same type of flow and boundary condition, found single phase model to predict heat transfer coefficient within 11% difference. This is considered to be good result according to them as this will be useful to test new nanofluids.

IV. Conclusions

Heat transfer enhancement in pipe flow by Al_2O_3 nanofluid has been investigated numerically using the single phase approach for constant wall temperature boundary condition. Both the experimental values and the numerical predictions show that heat transfer enhancement in the laminar regime is not as significant as in the turbulent regime. Model predictions in the turbulent regime agree very well with experimental values of Syam Sundar and Sharma [8]. As pointed out by Akbari et al [19], single phase approach does not predict heat transfer coefficient as accurately as in the turbulent regime. More research needs to be done to arrive at a definitive conclusion on the efficacy of single phase approach for laminar flow heat transfer prediction.

References

[1] D.Wen, Y.Ding, "Experimental investigation into convective heat transfer of nanofluids at the entrance region under laminar flow conditions", Int.J.Heat and Mass Transfer, 47, 5181- 5188, 2004.
[2] S.Zeinali Heris, S.Gh.Etemad, M.Nasr Esfahany, "Experimental investigation of oxide nanofluids laminar flow convective heat transfer", Int.Comm.in Heat and Mass Transfer, 33, 529-535, 2006.
[3] K.S.Hwang, S.K.Jang, S.U.S.Chio, "Flow and convective heat transfer characteristics of water-based Al2O3 nanofluids in fully developed laminar flow regime", Int.J.Heat and Mass Transfer, 52, 193-199, 2009.
[4] D.Kim, Y.Kwon, Y.Cho, C.Li, S.Cheong, Y.Hwang, J.Lee, D.Hong, and S.Moon, "Convective heat transfer characteristics of nanofluids under laminar and turbulent flow conditions", Current Applied Physics, 9, e119-e123, 2009.
[5] U.Rea, T.McKrell, L.W.Hu, J.Buongiorno,"Laminar convective heat transfer and viscous pressure loss of alumina–water and zirconia–water nanofluids", Int.J.Heat and Mass Transfer, 52, 2042-2048, 2009.
[6] R.Ben Mansour, N.Galanis, C.T.Nguyen, "Experimental study of mixed convection with watereAl2O3 nanofluid in inclined tube with uniform wall heat flux", Int.J.Thermal Sciences, 1-8, 2010.
[7] T.H.Nassan, S.Zeinali Heris, S.H.Noie, "A comparison of experimental heat transfer characteristics for Al2O3/water and CuO/water nanofluids in square cross-section duct", Int.Comm.in Heat and Mass Transfer ,Article in Press, 2010.
[8] L.Syam Sundar and K.V.Sharma, "Turbulent heat transfer and friction factor of Al2O3 Nanofluid in circular tube with twisted tape inserts", 53, 1409 - 1416, 2010.
[9] B.Farajollahi, S.Gh.Etemad, M.Hojjat, "Heat transfer of nanofluids in a shell and tube heat exchanger", 53, 12 – 17, 2010.
[10] Y.Xuan, Q.Li, "Investigation on convective heat transfer and flow features of nanofluids, J. Heat Transfer 125, 151–155, 2003.
[11] R.S.Vajjha, D.K.Das, D.P.Kulkarni," Development of new correlations or convective heat transfer and friction factor in turbulent regime for nanofluids", Int.J.Heat and Mass Transfer, Article in Press, 2010.
[12] M.Nasiri, S.Gh.Etemad, R.Bagheri," Experimental heat transfer of nanofluid through an annular duct", 38, 958-963, 2011.
[13] S.Sonawane, K.Patankar, A.Fogla, B.Puranik, U.Bhandarkar, S.sunil Kumar, "An experimental investigation of thermo-physical properties and heat transfer performance of Al2O3-Aviation Turbine Fuel nanofluids, Applied Thermal Engineering, Accepted Manuscript, 2011.
[14] S.E.B.Maiga, S.J.Palm, C.T.Nguyen, G.Roy, N.Galanis, "Heat transfer enhancement by using nanofluids in forced convection flows", Int.J.Heat and Fluid flow, 26, 530 – 546, 2005.
[15] M.Akbari, A.Behzadmehr, F.Shahraki, "Fully developed mixed convection in horizontal and inclined tubes with uniform heat flux using nanofluid", Int.J.Heat and Fluid flow, 29, 545 – 556, 2008.
[16] V. Bianco, F. Chiacchio, O. Manca, S. Nardini, "Numerical investigation of nanofluids forced convection in circular tubes", 29, 3632 – 3642, 2009.
[17] R. Lotfi, Y. Saboohi, A.M. Rashidi, "Numerical study of forced convective heat transfer of Nanofluids: Comparison of different approaches", Int.Comm.in Heat and Mass Transfer, 37, 74 – 78, 2010.
[18] V. Bianco, O. Manca, S. Nardini, "Numerical investigation on nanofluids turbulent convection heat transfer inside a circular tube", Article in Press, 2010.
[19] M. Akbari, N. Galanis, A. Behzadmehr, "Comparative analysis of single and two-phase models for CFD studies of nanofluid heat transfer", 50, 1343 – 1354, 2011.
[20] V. Gnielinski, New equations for heat and mass transfer in turbulent pipe and channel flow, Int. Chem. Eng. 16, 359–368, 1976.
[21] F.M. White, Viscous Fluid Flow, McGraw Hill, New York, 1991.
[22] E.N. Seider, G.E. Tate, "Heat transfer and pressure drop of liquid in tubes", Ind. Eng. Chem. 28 (12), 1429–1435, 1936.