

Numerical Investigation on the Heat Transfer Characteristics of Alumina-Water Nanofluid in a Double Pipe Heat Exchanger

Shinu Baby¹, Jijo Johnson²

¹Student, Dept. of Mechanical Engineering, M A College of Engineering, Kerala, India

² Assistant Professor, Dept. of Mechanical Engineering, M A College of Engineering, Kerala, India

Abstract - With increased concern for energy conservation, there has been a steady and substantial increase in research works to improve the design and performance of heat exchangers. The effectiveness and compactness of double pipe heat exchangers can be enhanced either by modifying the geometry or by modifying the heat transfer characteristics of the working fluid. Water, oil and ethylene glycol are the widely used conventional heat transfer fluids that limits the enhanced performance and compactness of these heat exchangers. In the present work, a double pipe heat exchanger using Al_2O_3 nanofluid was numerically investigated for heat transfer enhancement. Effects of nanoparticles concentration and volume flow rate on the average heat transfer coefficient, Nusselt number, pressure drop and friction factor were studied. It was found that the addition of nanoparticles have significant impact on the thermal performance of the heat exchanger. The average heat transfer coefficient and Nusselt number of the base fluid improved by 26% and 12.5% with 4% nanoparticle volume concentration. Addition of nanoparticles to the base fluid enhanced pressure drop by about 10.11%. Approximately 16% enhancement in the heat exchanger effectiveness can be achieved with the addition of 4% alumina particles in base fluid.

Key Words: Heat Exchanger, Nanofluid, Heat Transfer Coefficient, Nusselt Number, Pressure Drop, Friction Factor.

1. INTRODUCTION

Enhancing the heat transfer performance of heat exchangers may save energy, reduce process time, improve thermal performance and lengthen the working life. Due to the demand for high performance thermal systems for heat transfer applications, a number of works are being performed to gain insights into the heat transfer behaviour of these systems for their practical application to heat transfer enhancement. With the advent of high heat flow processes there has been a significant need for new technologies to enhance heat transfer. Using extended surfaces, introducing vibration on the heat transfer surfaces, and use of micro channels are some of the methods to enhance heat transfer rate. For effectiveness, heat exchangers are designed to maximize the heat transfer area of the wall between the two fluids, while minimizing resistance to fluid flow through the exchanger. Performance of heat exchanger can also be affected by the addition of fins or corrugations in one or both directions, which increase

surface area and may channel fluid flow or induce turbulence. Heat transfer rate can also be improved by enhancing the thermal characteristics of the working fluid. Commonly used heat transfer fluids such as water, ethylene glycol, and engine oil have relatively low thermal conductivities.

Nanofluid is a novel heat transfer medium, containing nanoparticles of size varying from 1 to 100 nm which are uniformly and stably distributed in a base fluid. Nanofluids have been considered for applications as advanced heat transfer fluids for almost two decades. However, due to the wide variety and the complexity of the nanofluid systems, no agreement has been achieved on the magnitude of potential benefits of using nanofluids for heat transfer applications.

Experimental investigation of turbulent friction and heat transfer behaviour of metallic oxide dispersed fluids in a circular pipe by Pak and Cho [1] revealed that Nusselt number for the dispersed fluids increased with increasing particle concentration as well as Reynolds number but at constant average velocity, the convective heat transfer coefficient of the dispersed fluid was 12% smaller than that of base fluid water. Heat transfer characteristics of Cu-water nanofluid flowing through a brass tube was experimentally investigated by Xuan and Li [2] and reported 40% enhancement in heat transfer coefficient with 2% nanoparticle concentration in water. Experimental investigations with several nanofluids by Yang et al., [3] in a horizontal tube heat exchanger indicated that the increase in the heat transfer coefficient of the nanofluids is much less than that predicted from a conventional correlation and concluded that near-wall particle depletion in laminar shear flow may be one possible reason for the phenomenon. The flow and heat transfer behaviour of aqueous TiO_2 nanofluids flowing through a straight vertical pipe was experimentally studied by He et al., [4] under both laminar and turbulent flow conditions. Their results showed that addition of nanoparticles enhanced the thermal conduction which enhanced with increasing particle concentration and decreasing particle size. The effect of particle concentration seemed to be more pronounced in the turbulent flow regime for the given flow Reynolds number and particle size. Kulkarni et al., [5] experimentally investigated the thermo-hydraulic characteristics of nanofluids comprised of silicon dioxide (SiO_2) nanoparticles suspended in ethylene glycol and water mixture flowing in a copper tube and revealed that the heat transfer coefficient and pressure loss increased

with increasing particle concentration in the turbulent flow regime. The convective heat transfer in turbulent regime inside a copper tube was experimentally investigated by Fotukian and Esfahany [6] using CuO-H₂O nanofluid to reveal that heat transfer coefficients for nanofluids were greater than that of water and increasing the nanoparticle concentration showed a very weak effect on heat transfer coefficient. They reported an increment of 25% for heat transfer coefficient compared to water. Experimental investigation carried out by Fotukian and Esfahany [7] on turbulent convective heat transfer and pressure drop of γ -Al₂O₃-water nanofluid inside a circular tube and their results showed that nanoparticle addition not only enhanced heat transfer coefficient but it also caused pressure drop to increase more than that of the base fluid. The heat transfer coefficient and friction factor of a nanofluid consisting of water and TiO₂ flowing in a horizontal double-tube counter flow heat exchanger under turbulent flow conditions were experimentally studied by Duangthongsuk and Wongwises [8]. They showed that the convective heat transfer coefficient of nanofluid is slightly higher than that of the base liquid by about 6 -11%. They also concluded that the Pak and Cho correlation for predicting the heat transfer coefficient of a nanofluid agreed better with their experimental results than the Xuan and Li correlation. Elsebay et al., [9] made a numerical analysis on the thermal and flow performance of the flat tubes of a radiator using Al₂O₃ and CuO nanofluids for the purpose of resizing of the flat tubes. They reported heat transfer coefficient enhancement of 45% and 38% for Al₂O₃-water and CuO-water nanofluids respectively compared to the base fluid. They also reported an increase in friction coefficient and pressure drop with the use of nanofluids. Al₂O₃-water nanofluid achieved higher length reduction compared to CuO-water nanofluid. Sun et al., [10] experimentally analysed the flow and convective heat transfer characteristics of Fe₂O₃-water nanofluids inside inner-grooved copper and smooth copper tubes. They also analysed the effects of different species and of the mass fractions of dispersants on the stability of nanofluids. The convective heat transfer coefficient enhanced by 33.52% with 0.4% mass fraction of nanoparticles. Pressure drop increased with the increase in the mass fraction of nanoparticles.

Many investigations on heat transfer characteristics of nanofluids had been conducted; however no agreement has been achieved on the magnitude of potential benefits of using nanofluids for heat transfer applications due to wide variety and complexity of nanofluids. The current study aims to numerically investigate the thermal performance of a double pipe heat exchanger using alumina water nanofluid as coolant. This study aims to assess the heat transfer characteristics, pressure drop, friction factor and effectiveness of heat exchanger for different nanoparticle volume concentrations.

2. NUMERICAL MODEL

2.1 Physical Geometry

The geometric model of double pipe heat exchanger is shown in figure 1. Nanofluid passes through the tube side having 6 mm inside diameter and 1m length, while the hot fluid flows through the annulus side. Outer pipe is 16 mm in diameter and the interface is 0.5 mm thick. Three domains were defined for the model, two fluid domains and one solid domain. The cold fluid flows in the inner tube and hot fluid flows in the annulus space in a counter flow fashion.

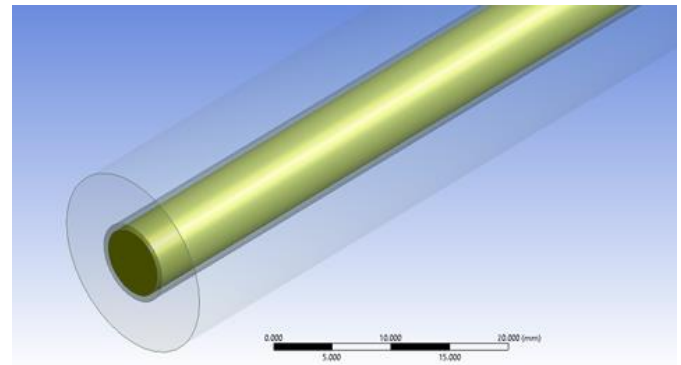


Fig -1: Geometry of the model

2.2 Governing Equations

Using commercial software ANSYS Fluent 15.0 [13], governing equations for flow conditions and heat transfer were numerically solved based on following assumptions: steady-state, incompressible and Newtonian turbulent fluid flows with constant thermo physical properties of the nanofluid. The viscous heating and compression work are both trivial and thus are neglected in the energy equation [11]. The resulting governing equations are:

Continuity equation:

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$

Momentum equation:

$$\frac{\partial u_i u_j}{\partial x_i} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left((\nu + \nu_t) \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right) \quad (2)$$

Energy equation:

$$\frac{\partial u_i T}{\partial x_i} = \rho \frac{\partial}{\partial x_i} \left(\left(\frac{\nu}{Pr} + \frac{\nu_t}{Pr_t} \right) \frac{\partial T}{\partial x_i} \right) \quad (3)$$

where u , ρ , ν and Pr are the average velocity, density, kinematic viscosity and Prandtl number, respectively.

Realizable k-ε turbulence model is adopted in the current study because it can provide superior performance for flows involving rotation, boundary layers effect under strong adverse gradient of pressure and recirculation [11]. The transport equations in the realizable k-ε model are given below:

Turbulent kinetic energy k equation:

$$\frac{\partial u_i k}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\left(\nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right) + \Gamma - \varepsilon \quad (4)$$

Turbulent energy dissipation ε equation:

$$\frac{\partial u_i \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\left(\nu + \frac{\nu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right) + c_1 \Gamma \varepsilon - c_2 \frac{\varepsilon^2}{k + \sqrt{\nu \varepsilon}} \quad (5)$$

where Γ represents the generation of turbulence kinetic energy k due to the mean velocity gradients and is given by:

$$\Gamma = -u_i u_j \frac{\partial u_i}{\partial x_j} = \nu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_i} \quad (6)$$

The turbulent kinematic viscosity is:

$$\nu_t = c_\mu \frac{k^2}{\varepsilon} \quad (7)$$

The empirical constants for the realizable k-ε turbulence model are:

$$c_1 = \max [0.43, \mu / (\mu_t + 5)]; c_2 = 1.9; \sigma_k = 1.0; \sigma_\varepsilon = 1.2$$

where μ is the dynamic viscosity and σ_k and σ_ε are Prandtl number for k and ε, respectively.

The enhanced wall function is used for the numerical computation near the wall to avoid problems of successive refinements in standard wall function meshes.

2.3 Thermo-Physical Properties of Nanofluids

Homogeneous modelling approach was applied where physical properties of nanofluid is taken as a function of properties of both constituents and their concentrations. Assuming that the nanoparticles were well dispersed in the base fluid the following correlations recommended by [12] were used for evaluating the thermo-physical properties of nanofluids:

Thermal conductivity:

$$k_{nf} = k_{bf} \left(\frac{k_p + 2k_{bf} - 2\phi(k_{bf} - k_p)}{k_p + 2k_{bf} + \phi(k_{bf} - k_p)} \right) \quad (8)$$

Viscosity:

$$\mu_{nf} = (1 + 2.5\phi)\mu_{bf} \quad (9)$$

Density:

$$\rho_{nf} = (1 - \phi)\rho_{bf} + \phi\rho_p \quad (10)$$

Heat Capacity:

$$(\rho C_p)_{nf} = (1 - \phi)(\rho C_p)_{bf} + \phi(\rho C_p)_p \quad (11)$$

2.4 Thermo-Physical Properties of Nanofluids

Numerical investigations were carried out under turbulent regime using alumina nanoparticles and water as base fluid. The momentum boundary condition of no slip and no penetration is set for all the solid walls. Mass flow inlet and pressure outlet boundary were used for both the fluid flows. The outlets were assumed to have a pressure of zero so the inlet pressure is equal to the pressure drop on both the annulus and inner tube sides. The cold fluid flows in the inner tube and hot fluid flows in the annulus space between the two tubes in a counter flow fashion. Table 1 lists the boundary conditions used in the present study.

Table -1: Boundary conditions

Parameter	Value
Volume flow rate of cold fluid (LPM)	0.1, 0.3, 0.5, 0.7, 0.9
Volume flow rate of hot fluid (LPM)	1, 3
Nanoparticles	Al ₂ O ₃
Nanoparticle volume concentration (%)	2, 4
Cold fluid inlet temperature (K)	301
Hot fluid inlet temperature (K)	353

3. CFD SIMULATION

The governing equations were solved by the control volume approach using commercial software ANSYS Fluent 15.0 [13]. The double precision option was adopted for all computations. A second-order upwind scheme was employed to discretize the convection terms, diffusion terms, and other quantities resulting from the governing equations. A staggered grid scheme was used, in which the velocity components are evaluated at the control volume faces, while the rest of the variables governing the flow field are stored at the central node of the control volume. The pressure-velocity coupling was handled with the SIMPLE scheme for pressure linked equations. Fluent solves the linear systems resulting from discretization schemes using a point implicit (Gauss-Seidel) linear equation solver, in conjunction with an algebraic multi-grid method. For all simulations performed in the present study, converged solutions were considered

when the normalized residuals resulting from an iterative process for all governing equations were lower than 10^{-6} .

3.1 Grid Independence Study

Four different meshes were considered for establishing the grid independence of the model under the present study. In each case, the cold fluid outlet temperature was recorded and tabulated as shown in table 2 to achieve the grid independency. From the tabulated results, it was concluded that grid independency was achieved for mesh-3 and this mesh was used for further numerical computations. The computational domains are meshed with face meshing and it contain a mix of hexahedral and wedge elements. Figure 2 shows the meshed view of the generated model.

Table -2: Grid independence study

Grid	Temperature (K)
Mesh-1 (553621)	326.524
Mesh-2 (862356)	326.352
Mesh-3 (1233729)	326.233
Mesh-4 (1570157)	326.233

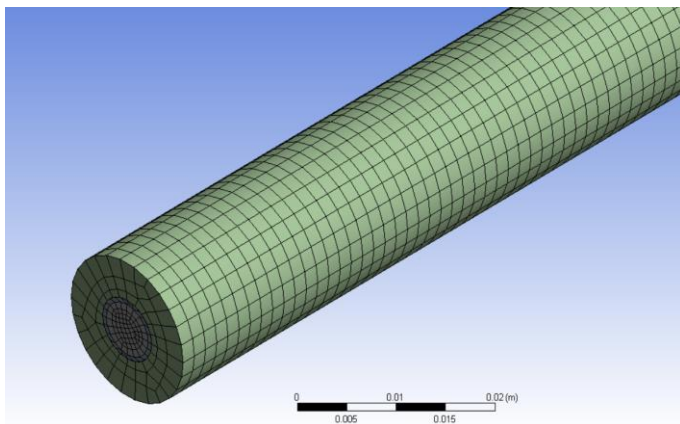


Fig -2: Meshed view of the generated model

3.2 Model Validation

The numerical results obtained by present study were compared with the experimental results of Shriram et al. [14] and Sahin et al. [15] to validate the numerical model. Shriram et al. [14] carried out experimental study to investigate the heat transfer characteristics of Al_2O_3 -water nanofluids as a coolant used in concentric tube heat exchanger. Figure 3 shows the comparison of heat transfer coefficient for 2% nanoparticle volume concentration in base fluid for current numerical study and experimental investigation of Shriram et al. [14]. The numerical results showed good agreement with the experimental results. The maximum deviation of the numerical results from the experimental study in the literature were less than 14% for hot fluid volume flow rate of 1 LPM with 2% nanoparticle volume concentration in the base fluid.

The friction factor values obtained from the current numerical study were validated by comparing them with the experimental results of Sahin et al. [15] in the literature. Figure 4 shows the comparison of friction factor for the current numerical study and the experimental study done by Sahin et al. [15] for base fluid water. The maximum deviation for numerical results was found to be less than 9% from the experimental study. Therefore, the present results are in good agreement with the results reported Shriram et al. [14] and Sahin et al. [15], thus, validating the numerical model used in the present study.

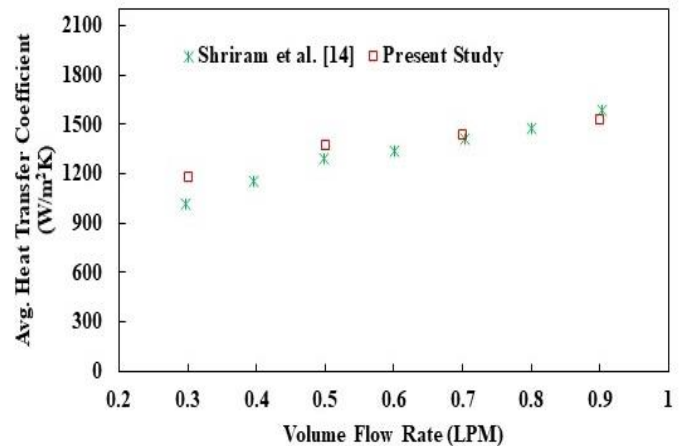


Fig -3: Variation of average heat transfer coefficient for hot fluid volume flow rate of 1 LPM

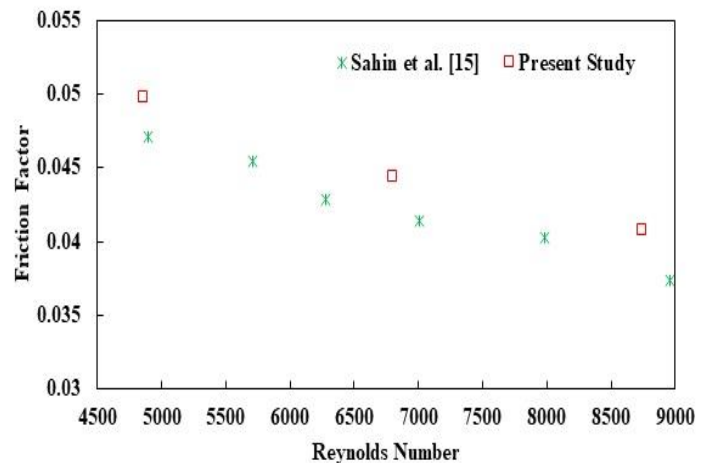


Fig -4: Variation of friction factor for base fluid water

4. RESULTS AND DISCUSSIONS

Heat transfer performance and flow dynamics in a concentric tube heat exchanger with alumina-water nanofluids was evaluated in terms of heat transfer coefficient, Nusselt number, pressure drop and friction factor. It not only highlights the influence of flow velocity but also the effect of volume concentration of alumina

nanoparticles on the heat transfer performance of a heat exchanger.

4.1 Variation of Heat Transfer Coefficient

Figure 5 shows the effect of particle concentration on the heat transfer coefficient at different volume flow rates. Results revealed that the average heat transfer coefficient increases with increase in volume flow rate and concentration of the nanofluid. At all concentrations and flow rates the heat transfer coefficient of nanofluid is higher than the base fluid water. Maximum enhancement of 26% and 18% was obtained with 4% alumina. For a given volume flow rate the increase in average heat transfer coefficient might be due to the higher thermal conductivity of the nanofluid which is several orders of magnitude higher than water and also due to a large energy exchange process resulting from the chaotic movement of nanoparticles. The effective thermal conductivity is reinforced with any increase in the concentration of nanoparticles as it increases the turbulence of the fluid and these particles are near the surface of the internal wall of the heat exchanger which promotes fast heat exchange.

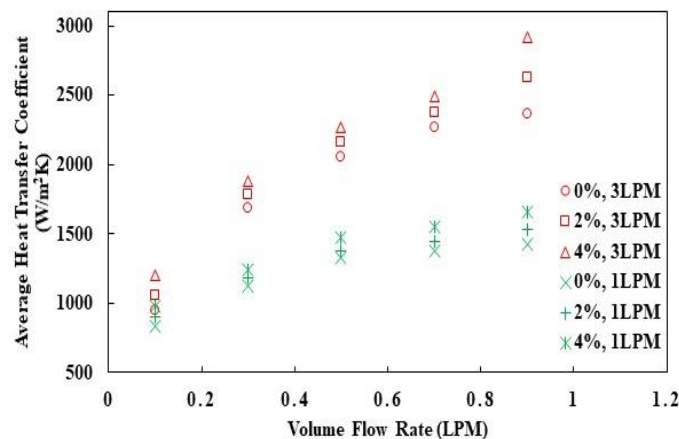


Fig -5: Variation of heat transfer coefficient of cold fluid for hot fluid flow rates of 1 and 3LPM

Increasing the hot and cold water flow rate leads to an enhancement in the heat transfer rate and results in an increase in the heat transfer coefficient of the nanofluid. Increasing the flow rate induces eddies in the flow, which aids increased heat transfer. At low flow rates, an agglomeration of nanoparticles might happen which will reduce the heat transfer enhancement. However, at high flow rates, the agglomeration effect is reduced significantly and the dispersion of nanoparticles intensifies the mixing, which results in a significant increase in heat transfer coefficient.

4.2 Variation of Nusselt Number

Effect of volume concentration of nanofluid on the Nusselt number at different volume flow rate of the cold

fluid is shown in figure 6. Results showed that the Nusselt number increased with nanofluid volume concentration and volume flow rate. An enhancement of 12.5% and 5.38% was achieved with 4% nanoparticle volume concentration. But the effective increase is not as pronounced as in the case of average heat transfer coefficient. This might be due to the fact that Nusselt number is influenced by both thermal conductivity as well as convective heat transfer coefficient and increase in thermal conductivity might be more pronounced than the corresponding increase in heat transfer coefficient. It should be noted that the enhancement in heat transfer coefficient is attributed not only by thermal conductivity but also by other factors such as interaction of particle-particle, particle-liquid, micro convection in nanofluids, Brownian motion and particle migration. Also enhancement of Nusselt number is more pronounced at higher volume flow rates, which might be due to the enhanced turbulence at higher Reynolds number which increases the heat transfer properties of the nanofluid.

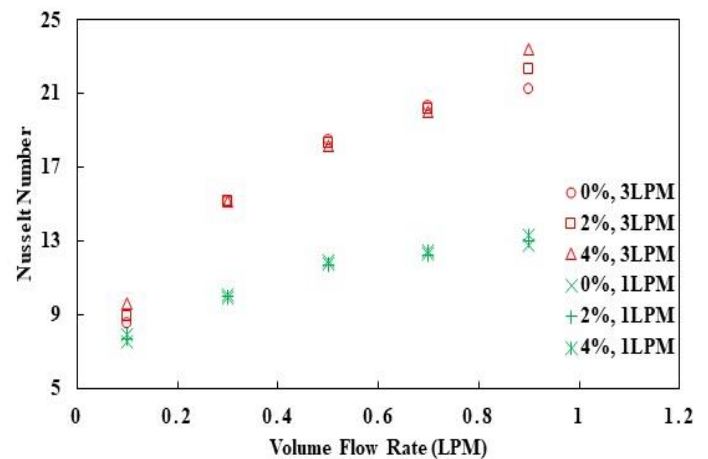


Fig -6: Variation of Nusselt number of cold fluid for hot fluid flow rates of 1 and 3LPM

4.3 Variation of Pressure Drop

Figure 7 shows the variation of pressure drop with increasing particle concentration and volume flow rate of alumina nanofluid for hot fluid volume flow rates of 1 and 3LPM. It is a significant parameter for better performance of the heat exchanger as an increase in pressure drop may lead to higher pumping power, so it is important to manage its variation during the heat transfer process. At low flow rates the pressure drop of the base fluid and the nanofluid are negligible, but it increases with flow rate as well as volume concentration of nanofluid. This variation is more pronounced at higher velocities due to the increased turbulence.

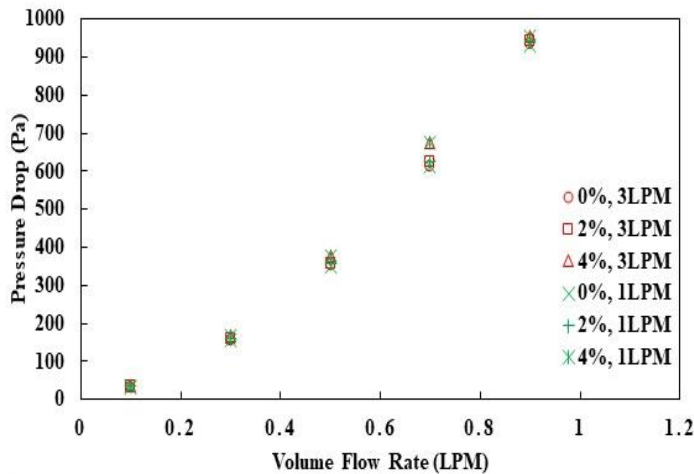


Fig -7: Variation of pressure drop of cold fluid for hot fluid flow rates of 1 and 3LPM

The pressure drop increased by about 10% with 4% volume concentration of nanofluid. One of the most important factors influencing the pressure drop of a fluid is its dynamic viscosity. Dispersing the nanoparticles in the fluid increases its viscosity which lead to higher pressure drop at higher particle concentrations. Pressure drop of the nanofluid is more than water as its viscosity is higher than the base fluid but it is negligible. Therefore it is possible to have higher heat transfer rate with marginal increase in pumping power with the use of nanofluids as heat transfer medium. This enhances the performance of the heat transfer system.

4.4 Variation of Friction Factor

The effect addition of alumina nanoparticles on the friction factor of water at different volume flow rates is depicted in figure 8. For all volume concentrations of the nanofluid the friction factor decreased with increase in volume flow rate of the cold fluid. Fluid flow is initially laminar and the friction factor is higher, however, as the flow rate increases the flow becomes fully developed turbulent flow and the friction factor inside the tube becomes negligible and decreases with increasing volume flow rate. The friction factor of the nanofluid at 4% volume concentration reduced by about 5% compare to the base fluid as it's pressure drop is negligible and the density is more than the base fluid.

Figure 9 compares the numerically obtained values of friction factor for hot fluid flow rate of 3LPM with those calculated using Blasius equation which is valid for both laminar and turbulent flow. The maximum deviation between the numerically and empirically calculated values occur at low Reynolds number, but as the flow becomes more turbulent the maximum deviation reduces considerably and the two values are comparable and shows good agreement with each other. At higher Reynolds number the maximum deviation reduced to about 12.6%.

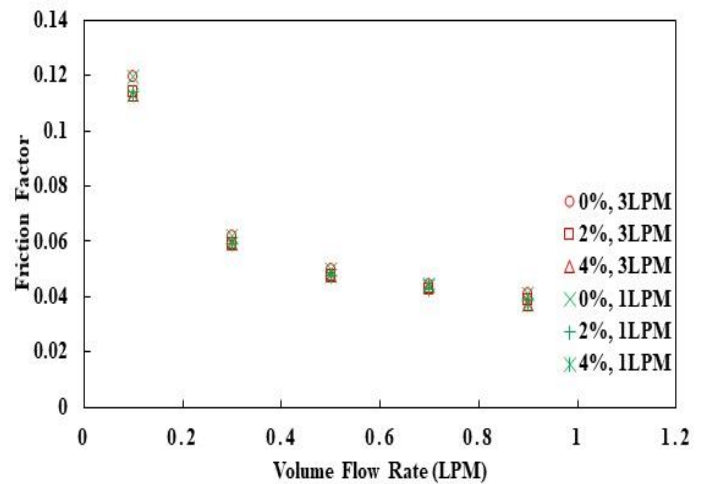


Fig -8: Variation of friction factor of cold fluid for hot fluid flow rates of 1 and 3LPM

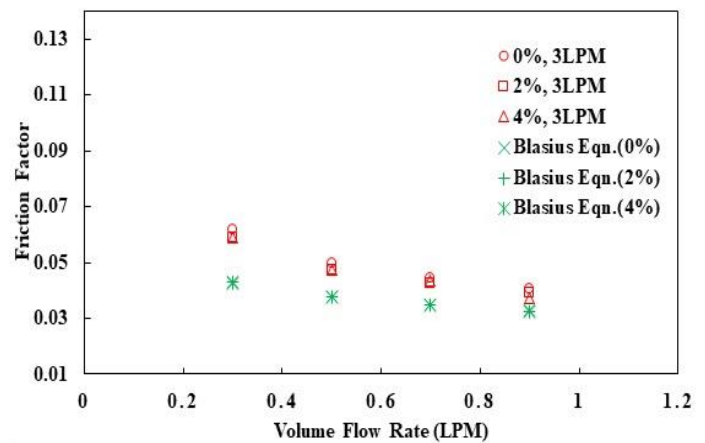


Fig -9: Comparison of numerically and empirically calculated values of friction factor

4.5 Effectiveness of Heat Exchanger

The variation of effectiveness of heat exchanger with different nanoparticle concentrations in the base fluid is depicted in figure 10. At all particle concentrations the effectiveness of the heat exchanger was higher than the base fluid and an enhancement of 16% was obtained with 4% nanoparticle concentration. This increased performance of heat exchanger might be due to the enhanced heat transfer characteristics of the nanofluid. Results also showed that enhanced performance of heat exchanger was more pronounced at higher volume flow rates, which might be due to the increased turbulence which induces eddies in the flow.

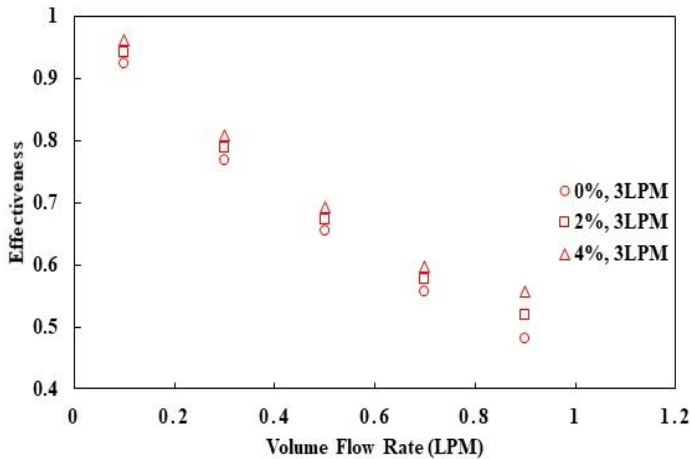


Fig -10: Effectiveness of heat exchanger at different concentrations of nanofluid

5. CONCLUSIONS

In the present study, a numerical model was developed to compute and analyse the heat transfer characteristics of a double pipe heat exchanger using nanofluids on the cold side. The influences of the nanoparticle volume concentration and volume flow rate were examined. The results showed that, the heat transfer characteristics of the heat exchanger enhanced with the use of nanofluids compared to the conventional base fluid. The thermal conductivity of base fluid enhanced with addition of nanoparticles and the degree of thermal conductivity enhancement depended on the amount of nanoparticles added to the base fluid.

The numerical results showed that the average heat transfer coefficient and pressure drop across the pipe were dependent on the volume concentration and the flow rate of the nanofluid. These parameters increased with volume concentration of the nanofluid. For a hot fluid flow rate of 3 LPM, about 26% and 11% enhancement in heat transfer coefficient was achieved with addition of 4% and 2% alumina particles respectively. The pressure drop of base fluid increased by about 10% with 4% concentration of alumina nanoparticle. Nusselt number of the base fluid increased by about 12.5% and 5.38% at 4% alumina particle concentration for hot fluid volume flow rate of 3 LPM and 1 LPM, respectively. Although friction factor decreased with the increase in flow rate, there was no significant variation with nanoparticle concentration. About 5% reduction in friction factor was observed at higher volume flow rates with the addition of 4% alumina nanoparticles. The effectiveness of heat exchanger enhanced by 16% and 12.5% with 4% volume concentration of the nanofluid for a hot fluid flow rate of 3 LPM and 1 LPM respectively. An enhancement of 8% and 6.25% was obtained with 2% concentration of nanofluid. Therefore Al_2O_3 -water nanofluid can be considered as next generation heat transfer fluid in heat exchanger applications.

REFERENCES

- [1] B.C. Pak, Y.I. Cho (1998), "Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles", *Exp. Heat Transf.*, Vol. 11, pp. 151-170.
- [2] Y. Xuan, Q. Li (2003), "Investigation on convective heat transfer and flow features of nanofluids", *J. Heat Transf.*, Vol. 125, pp. 151-155.
- [3] Y. Yang, Z.G. Zhang (2005), "E.A. Grulke, W.B. Anderson, G. Wu, Heat transfer properties of nanoparticle-in-fluid dispersions (nanofluids) in laminar flow", *Int. J. Heat Mass Transf.*, Vol. 48, pp. 1107-1116.
- [4] Y. He, Y. Jin, H. Chen, Y. Ding, D. Cang, H. Lu (2007), "Heat transfer and flow behavior of aqueous suspensions of TiO_2 nanoparticles (nanofluids) flowing upward through a vertical pipe", *Int. J. Heat Mass Transf.*, Vol. 50, pp. 2272-2281.
- [5] D.P. Kulkarni, P.K. Namburu, H.E. Bargar, D.K. Das (2008), "Convective heat transfer and fluid dynamic characteristics of SiO_2 -ethylene glycol/water nanofluid", *Heat Transf. Eng.*, Vol. 29 (12), pp. 1027-1035.
- [6] S.M. Fotukian, M. Nasr Esfahany (2010a), "Experimental study of turbulent convective heat transfer and pressure drop of dilute CuO /water nanofluid inside a circular tube", *Int. Commun. Heat Mass Transf.*, Vol. 37, pp. 214-219.
- [7] S.M. Fotukian, M. Nasr Esfahany (2010b), "Experimental investigation of turbulent convective heat transfer of dilute $\gamma-Al_2O_3$ /water nanofluid inside a circular tube", *Int. J. Heat Fluid Flow*, Vol. 31, pp. 606-612.
- [8] W. Duangthongsuk, S. Wongwises (2009), "Heat transfer enhancement and pressure drop characteristics of TiO_2 -water nanofluid in a double-tube counter flow heat exchanger", *Int. J. Heat Mass Transf.*, Vol. 52, pp. 2059-2067.
- [9] Elsebay, M., Elbadawy, I., Shedid, M., and Fatouh, M. (2016), "Numerical resizing study of Al_2O_3 and CuO nanofluids in the flat tubes of a radiator", *Applied Mathematical Modelling*, Vol. 40, pp. 6437-6450.
- [10] Lei, B. S. W., and Yang, D. (2015), "Flow and convective heat transfer characteristics of Fe_2O_3 -water nanofluids inside copper tubes", *International Communications in Heat and Mass Transfer*, Vol. 64, pp. 21-28.
- [11] Anas El M., A. Laknizi, Said S., A. B. Abdellah, Mohamed M., Mustapha El M. (2017), "Numerical design and investigation of heat transfer enhancement and performance for an annulus with continuous helical

baffles in a double-pipe heat exchanger”, *Energy Conservation and Management*, Vol. 133, pp. 76-86.

- [12] Peyghambarzadeh, S., Hashemabadi, S., Naraki, M., and Vermahmoudi, Y. (2013), “Experimental study of overall heat transfer coefficient in the application of dilute nanofluids in the car radiator”, *Applied Thermal Engineering*, Vol. 52, pp. 8-16.
- [13] ANSYS FLUENT 15.0 User Guide, November 2013, ANSYS Inc., Southpointe, Canonsburg.
- [14] Shriram S. Sonawane, Rohit S. Khedkar, Kailas L. Wasewar (2013), “Study on concentric tube heat exchanger heat transfer performance using Al_2O_3 -water based nanofluids”, *International Communications in Heat and Mass Transfer*, Vol. 49, pp. 60-68.
- [15] Bayram Sahin, Gül Gedik Gültekin, Eyuphan Manay, Sendogan Karagoz (2013), “Experimental investigation of heat transfer and pressure drop characteristics of Al_2O_3 -water nanofluid”, *Experimental Thermal and Fluid Science*, Vol. 50, pp. 21-28.