Numerical Investigations of Flow and Heat Transfer Characteristics of Water Based CuO and Al_2O_3 Nanofluids Using Two-Phase Mixture Model

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Abstract: The development of high-performance thermal systems has increased interest in heat transfer enhancement techniques. The application of additives to heat transfer liquids is one of the noticeable effort to enhance heat transfer. The stable suspensions of nanoparticles (typically < 100 nm) in liquids are called nanofluids.

In this study, heat transfer characteristics of two different nanofluids flowing through a circular tube under constant heat flux condition have been investigated numerically. Fluent 6.3 has been used this numerical study. Two-phase mixture model has been implemented two solve the problem. The comparison has been made between calculated and experimental results. The suspended nanosized particles enhance heat transfer and Nusselt numbers by comparing pure water at the same Reynolds numbers. Moreover, pressure drops for the nanofluids is approximately the same as that of pure water. The nanofluids containing CuO has showed bigger heat transfer enhancement than Al_2O_3 in all volume fraction rates.

Keywords: nanofluids, heat transfer, convection, nanoparticles

1. Introduction

Conventional heat transfer fluids such as water, engine oil and ethylene glycol are normally used as heat transfer fluids. Since these conventional fluids have low heat transfer performance the heat transfer enhancement is limited with these conventional fluids. The use of solid particles as an additive suspended into the base fluid is a technique for the heat transfer enhancement. Innovative heat transfer fluids with nanoparticules suspended in them are called “nanofluids”.

Behzadmer et al. (2007) studied turbulent forced convection heat transfer in a circular tube with a nanofluid consisting of water and 1 vol.% Cu numerically. Two phase mixture model has been implemented for the first time to study such a flow field. A single phase model formulation, which has been used frequently in the past for heat transfer with nanofluids, is also used for comparison with the mixture model. Their comparison of calculated results with experimental values shows that the mixture model is more precise than the single phase model. The axial evolution of the flow field and fully developed velocity profiles at different Reynolds numbers are also presented and discussed.

Nguyen et al (2007) have experimentally investigated the behavior and heat transfer enhancement of Al2O3 nanoparticle–water mixture, flowing inside a closed system that is destined for cooling of microprocessors or other electronic components. Experimental data, obtained for turbulent flow regime, have clearly shown that the inclusion of nanoparticles into distilled water has produced a considerable enhancement of the cooling block convective heat transfer coefficient.

Hwang et al. (2008) measured the pressure drop and convective heat transfer coefficient of water-based Al2O3 flowing through a uniformly heated circular tube in the fully developed laminar flow regime. Experimental results show that the convective heat transfer coefficient enhancement exceeds, by a large margin, the thermal conductivity enhancement. They propose that flattening of velocity profile is a possible mechanism for the convective heat transfer coefficient enhancement exceeding the thermal conductivity enhancement.

Heris et al. (2007) investigated laminar flow forced convection heat transfer of Al_2O_3/water inside a circular tube with constant wall temperature experimentally. The Nusselt numbers of nanofluids were obtained for different nanoparticle concentrations as well as various Peclet and Reynolds numbers. Experimental results emphasize the enhancement of heat transfer due to the nanoparticles presence in the fluid. Heat transfer coefficient increases by increasing the concentration of nanoparticles in nanofluid. The increase in heat transfer coefficient due to presence of nanoparticles is much higher than the prediction of single phase heat transfer correlation used with nanofluid properties.

Wen and Ding (2005) studied about formulation of aqueous based nanofluids and its application under natural convective heat transfer conditions. They claimed that very stable titanium dioxide/water nanofluids could be formulated through the mechanical shear mixing and electrostatic stabilization. Both transient and steady heat transfer coefficients were obtained for different concentrations of nanofluids under natural convective conditions. The nanofluids are found to decrease the natural convective heat transfer coefficient; such deterioration increases with nanoparticle concentrations. Possible reasons/mechanisms attributed to such behavior are discussed, including the convection induced by concentration difference, particle–surface and particle–particle interactions, and modifications of the dispersion properties. Further experimental and theoretical works are being carried on to identify the exact causes.

In this study two phase mixture model wa applied to study turbulent heat transfer forced convection flow of nanofluids in a uniformly heated tube.

2. Mathematical formulation

2.1. Mixture model

The mixture model is based on a single fluid two phase approach. Each phase has its own velocity and own volume fraction, primary phase and the secondary phase. The dimensional equations are independent from the time. Hydraulic diameter and turbulent intensity have been specified for each Reynolds number. Nanofluid consists of water-Al_2O_3 and water-CuO.

The simulation is a two-dimensional (axisymmetric) steady and forced turbulent convection flow of nanofluid(water-CuO and water-Al_2O_3). The horizontal circular tube has diameter of 0.0115 m and a length of 0.84 m. The fluid and particles insert the circular tube with uniform axial velocity and temperature.

Results in this study illustrate the effect of the Reynolds number on the turbulent forced convection flow characteristics of a nanofluid consisting of water and %0.5, %1, %2, %3, %4 volume fraction CuO with 33 nm and Al_2O_3 with 50 nm mean diameter.

The governing equations for the fluid flow are :

Countinuty equation for the mixture

$$\nabla.\left(\rho_m V_m\right) = 0 \tag{1}$$

Momentum

$$\nabla.\left(\rho_m V_m V_m\right) = -\nabla p_m + \nabla.[\tau - \tau_l] + \rho_m g + \nabla.\left(\sum_{k=1}^{n} \phi_k \rho_k V_{dr,k} V_{dr,k}\right) \tag{2}$$

Energy

$$\nabla.\left(\phi_k V_k \left(\rho_k h_k + p\right)\right) = \nabla.(\lambda_{eff} \nabla T - C_p \rho_m \overline{vt}) \tag{3}$$

Volume fraction

$$\nabla.\left(\phi_p \rho_p V_m\right) = -\nabla.(\phi_p \rho_p V_{dr,p}) \tag{4}$$

Mixture density

$$\rho_m = \sum_{k=1}^{n} \phi_k \rho_k \tag{5}$$

Mixture viscosity

$$\mu_m = \sum_{k=1}^{n} \phi_k \mu_k \tag{6}$$

Solid viscosity model was determined from experimental work of Miller and Gidaspow(1992).

$$\mu_s = -0.188 + 537.42\phi \tag{7}$$

Where ϕ is solid volume fraction and μ_s is in unit of centipose.

Drift velocity(V_k is secondary phase velocity)

$$V_{dr,k} = V_k - V_m \tag{8}$$
$$\tau = \mu_m \nabla V_m \tag{9}$$
$$\tau_t = -\sum_{k=1}^{n} \phi_k \rho_k \overline{v_k v_k} \tag{10}$$

Slip velocity(V_p is secondary phase velocity)

$$V_{pf} = V_p - V_f \tag{11}$$

Drift velocity is

$$V_{dr,p} = V_{pf} - \sum_{k=1}^{n} \frac{\phi_k \rho_k}{\rho_k} V_{fk} \tag{12}$$

the relative velocity is given by Manninen et al. (1996),

$$V_{pf} = \frac{\rho_p d_p^2}{18\mu_f f_{drag}} \frac{(\rho_p - \rho_m)}{\rho_p} a \tag{13}$$

The drag function is given by Schiller and Naumann (1935)

$$f_{drag} = \begin{cases} 1 + 0.15\,\mathrm{Re}_p^{0.687} & \mathrm{Re}_p \le 1000 \\ 0.0183\,\mathrm{Re}_p & \mathrm{Re}_p > 1000 \end{cases} \tag{14}$$

d is the diameter of the particles of secondary phase , a is the acceleration of secondary-phase.

$$a = g - (V.\nabla)V_m \tag{15}$$

2.2. Turbulence model

In this study $\kappa - \varepsilon$ turbulence model has been used (Launder 1972). $\kappa - \varepsilon$ turbulence model contains two additional equations. These are turbulence kinetic energy (κ) and dissipation (ε) rate.

$$\nabla(\rho_m V_m \kappa) = \nabla.\left(\frac{\mu_{t,m}}{\sigma_k}\nabla\kappa\right) + G_{k,m} - \rho_m\varepsilon \tag{16}$$

$$\nabla.(\rho_m V_m \varepsilon) = \nabla.\left(\frac{\mu_{t,m}}{\sigma_\varepsilon}\nabla\varepsilon\right) + \frac{\varepsilon}{\kappa}(C_1 G_{k,m} - C_2\rho_m\varepsilon) \tag{17}$$

$$\mu_{t,m} = \left(\rho_m C_\mu \frac{\kappa^2}{\varepsilon}\right) \tag{18}$$

$$G_{k,m} = \mu_{t,m}(\nabla V_m + (\nabla V_m)^T) \tag{19}$$

$C_\mu = 0.09$, $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $\sigma_\kappa = 1.0$, $\sigma_\varepsilon = 1.3$

$G_{k,m}$ is turbulence kinetic energy generation due to average velocity gradient. σ_κ, σ_ε are Prandtl numbers for turbulence kinetic energy and dissipation rate, $C_{1\varepsilon}$ ve $C_{2\varepsilon}$ are constants. $\mu_{t,m}$ is eddy viscosity[Fluent 2006].

2.3. Boundry conditions

Uniform axial velocity, temperature have been specified at the tube inlet, turbulent intensity and hydraulic diameter [Fluent 2006] have been also specified. At the tube outlet section, the flow and temperature fields are assumed fully developed (*(x/D)* > 10). Pressure-outlet boundary condition has been implemented for the outlet section. Only half of the tube was modeled due to the symmetry. On the upper wall of the tube, the no-slip boundary condition was imposed. The wall is subjected to a uniform heat flux. On the lower wall of the modeled domain, the axis boundary condition was applied. In the present analysis, the near wall treatment was based on enhanced wall functions [Fluent 2006].

2.4. Numerical procedure

The CFD code Fluent was used for solving this problem. Second order upwind scheme was employed to discretize equations. Pressure and velocity were coupled using Semi Implicit Method for Pressure Linked Equations [SIMPLE] (Patankar 1980).

2.5. Grid optimization

100 x 150, 115x160, 150x200, 200x250 in r-direction and in *x*-direction grids were tested. All gave similar values of velocity and temperature at the outlet. Therefore, 100×150 was accepted as the ideal grid size.

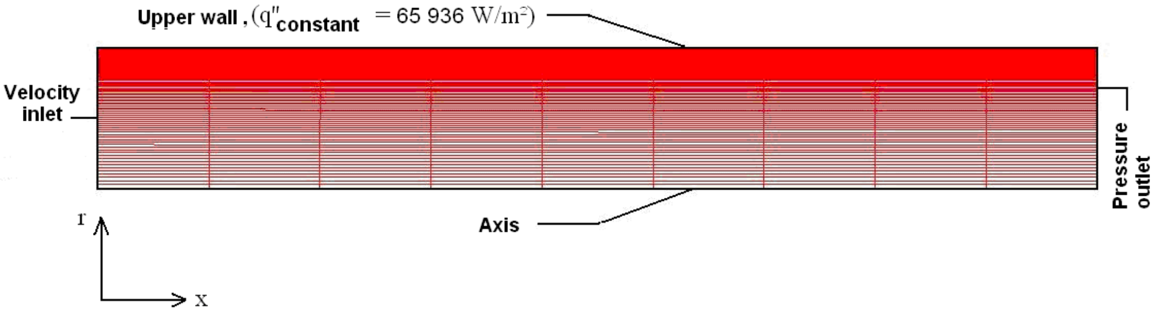

Fig. 1. Grid figure used in the present simulation, axisymmetric from X-axis.

3. Results and Discussion

3.1. Validation of the present simulation

The tube has a diameter of 0.0115 m and a length of 0.84 m. The fluid enters the tube with a constant inlet temperature T_{in} of 293 K and with uniform axial velocity V_{in}. The Reynolds number was varied from 10000 to 80000. In order to validate the computational model, the numerical results were compared with the theoretical data available for the conventional fluids. The Nusselt number computed with simulation for developed turbulent flow were compared with the Eq. 20 given by Petukhov, (Incropera 2000),

$$Nu_D = \frac{(f/8)\, Re_D\, Pr}{1.07 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \quad (20)$$

Fig. 2 shows the comparison of Nusselt numbers from Petukhov equation and computed values from present study for water. The maximum deviation and average deviation of computed Nusselt number from equation given by Petukhov is 9.9 and 6.4%, respectively.

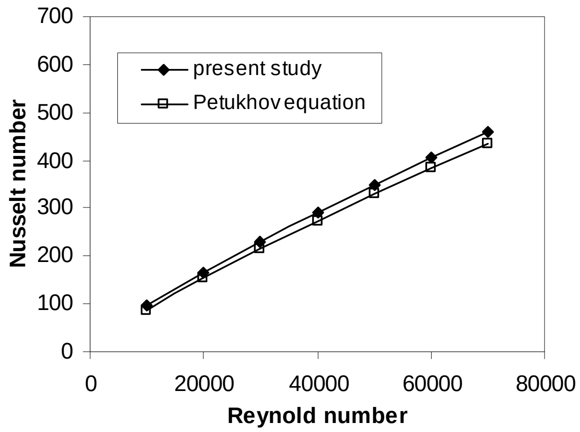

Fig. 2. Comparison between computed values of Nusselt numbers and Petukhov equation

The friction factor values were compared with the Darcy friction factor given by Blasius [White] is presented as Eq. (21)

$$f = 4C_f = 4(0.0791 Re^{-1/4}) \quad (21)$$

Fig. 3 displays the comparison of Darcy friction factor from Blasius equation and computed values from this numerical study. An excellent agreement is observed and maximum deviation of computed values from Blasius equation is 2.8 % for friction factor over the range of Reynolds numbers studied.

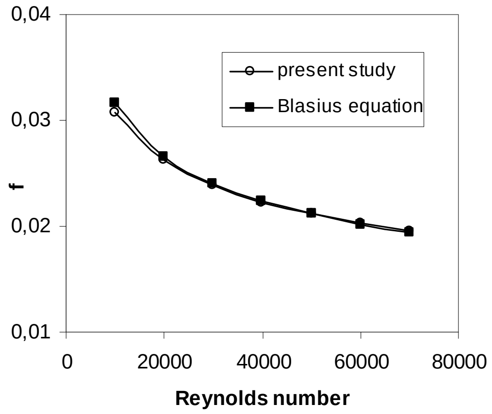

Fig.3. Comparison of Darcy friction factor by Blasius equation and computed values

3.2 Effect of nanoparticule volume concentration on the Nusselt number

Fig. 4 shows heat transfer coefficient as a function of the Reynolds number for the different Al2O3 nanoparticle volume concentrations. It is seen from Fig 4. that heat transfer coefficient increases with increasing volume fraction.

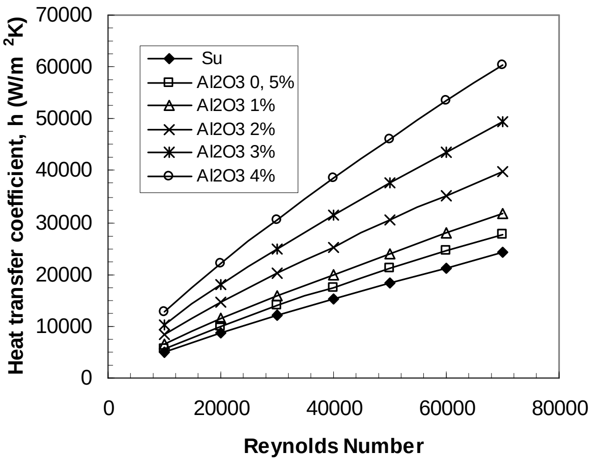

Fig. 4. Effect of volume fraction ration for Al_2O_3 nanofluids on heat transfer coefficient

It is shown in Fig. 5 that the influence of Al_2O_3 nanoparticle volume concentration on the Nusselt number. Nusselt number increases with increasing volume fraction ratio. The increase in the Nusselt number is about 2.4 times with 4% volume fraction ratio over the water at Reynolds number of 70000. The particle volume fraction is one of the main factors affecting the Nusselt numbers of the nanofluids.

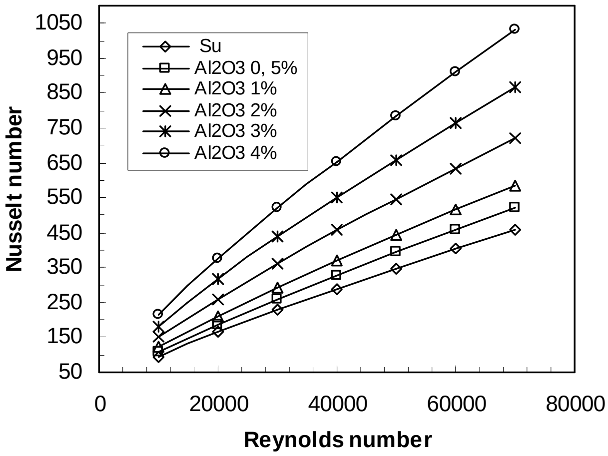

Fig. 5. Effect of volume fraction ration for Al_2O_3 nanofluids on Nusselt number

Fig. 6 shows heat transfer coefficient as a function of the Reynolds number for the different CuO nanoparticle volume concentrations. It is seen from Fig 6. that heat transfer coefficient increases with increasing volume fraction.

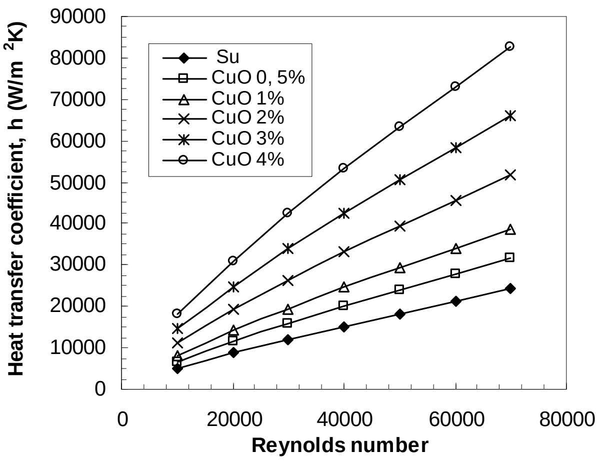

Fig.6. Effect of volume fraction ration for CuO nanofluids on heat transfer

It is shown in Fig. 7 that the influence of CuO nanoparticle volume concentration on the Nusselt number. Nusselt number increases with increasing volume fraction ratio. The increase in the Nusselt number is about 3 times with 4% volume fraction ratio over the water at Reynolds number of 70000. The particle volume fraction is one of the main factors affecting the Nusselt numbers of the nanofluids. The heat transfer enhancement is achieved with CuO nanofluids more than Al_2O_3. It is because conductivity of CuO and heat transfer area for the same volume fraction ratio higher than Al_2O_3 nanoparticles.

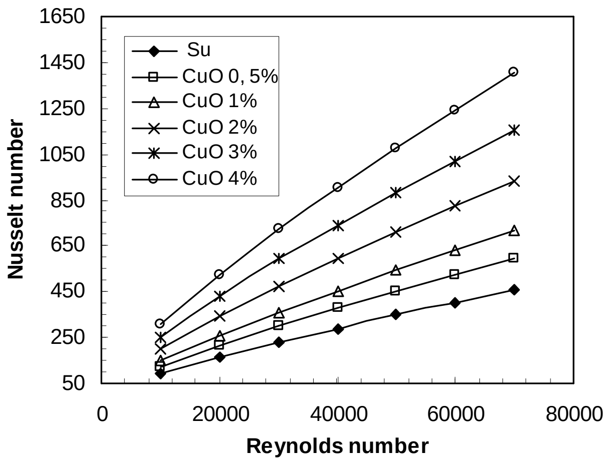

Fig. 7. Effect of volume fraction ration for CuO nanofluids on Nusselt number

3.3 Effect of nanoparticule volume concentration on the Friction factor

The axial evolution of th local frictional coefficient is shown in Fig.8 and 9 for Al_2O_3 and CuO, respectively. As expected, the frictional coefficient decreases as the Reynolds number increases. It is shown from figures that the nanoparticles do not have a significant effect on its value. This observation is also confirmed by the result of Xuan and Li (2003).

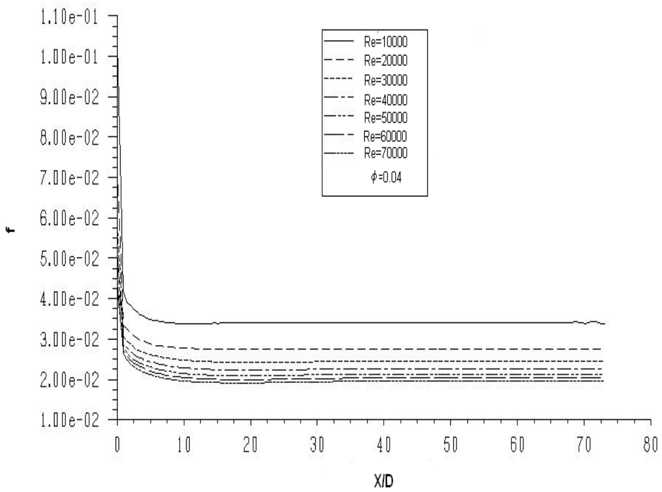

Fig. 8. Effect of Reynolds number on axial evolution of the local frictional coefficient for Al_2O_3 nanofluids

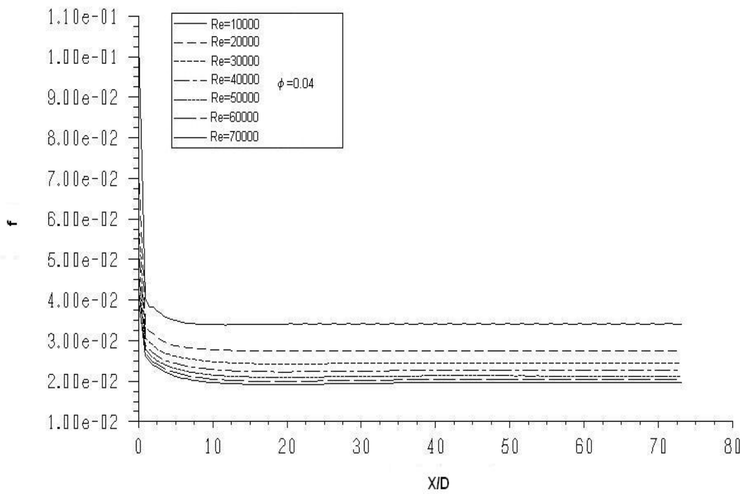

Fig. 9. Effect of Reynolds number on axial evolution of the local frictional coefficient for CuO nanofluids

Conclusion

Turbulent heat transfer and friction factor characteristics in a circular tube with nanofluids consisting of Al_2O_3 and CuO of water were investigated numerically, by using two phase mixture model. Adding 4% nanaoparticules of Al_2O_3 increases the Nusselt number more than 2 times and adding 4% nanaoparticules of CuO increases the Nusselt number more than 3 times. It does not have ant significant effect on the pressure drop penalty.

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