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Two-phase simulation of nanofluid in a heat exchanger in turbulent flow regime

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Abstract

Turbulent forced convection of different nanofluids consisting of γ-Al_2O_3/water and CuO/water in a concentric double tube heat exchanger has been investigated numerically using two-phase approach. Nanofluids that are used as coolants flowing in the inner tube while hot pure water flows in outer tube. The study was conducted for Reynolds numbers ranging from 20,000 to 50,000 and nanoparticles volume fractions of 2, 3, 4 and 6 percent. The three-dimensional governing equations are discretized using the finite volume approach. Although, two different nanoparticles have almost the same thermal conductivity but using CuO/water nanofluid showing better enhancement in heat transfer that proves thermal conductivity is not the only reason of enhancing heat transfer. Also, CuO/water showing bigger shear stress in comparison of γ-Al_2O_3/water nanofluid. As a result, nanofluids show higher overall heat transfer coefficient in comparison of pure water.

Keywords: Turbulent forced convection, Nanofluid, Two-phase approach, Concentric double tube heat exchanger

Introduction

With progresses of technology heat transfer augmentation is one of the most challenges for developing industrial with high technology. There are many techniques for augmenting heat transfer that can be classified in two groups: (I) Passive techniques that do not need external force and (II) Active techniques that need external power. Passive techniques are included disturbing the boundary layers, improving the thermophysical properties for example, increasing fluid thermal conductivity.

Application of additives to liquids is one way of enhancing heat transfer. Augmenting of fluid thermal conductivity is the main purpose in improvement the heat transfer characteristic of liquids. Whereas in solids thermal conductivity is much bigger than liquids. It is expected adding this solid particles to a liquid enhance thermal conductivity of the base fluid. More than several decades, adding millimetre or micrometre size particles has been known, but, because of lot problems for example: sedimentation, clogging, abrasion and high pressure loss the use of these particles was not efficient (Maxwell, 1881).

Recently progress in material engineering and developing new technologies cause the basis of producing nano-sized particles. By dispersing this nanoparticles to liquid a new class of liquid achieved that first time. Masuda et al. (1993) introduced the liquid suspension of nano-sized particles and then Choi (1995) for the first time proposed the name of nanofluid. Nanofluids change the thermal and hydraulic feature of base fluids and cause enormous heat transfer enhancement. Nanoparticles with comparing of millimetre and micrometre particles have more surface contact that can enhance energy transport between fluids and particles as well as because of low momentum of this particles prevent the erosion and clogging hence nanofluids can be used for small geometries too.

Many researchers investigated the thermophysical properties of nanofluids (Lee, et al, 1999; Li & Peterson, 2006). But Research about the forced convection of nanofluids is important for the practical application of nanofluids in heat transfer devices. For this purpose, different papers focused on the nanofluids convection experimentally and numerically.

Pak and Cho (1998) for the first time investigated experimentally the convective heat transfer inside a circular tube. They investigated the convective heat transfer of γ-Al_2O_3 (13 nm)/water and TiO_2 (27 nm)/water nanofluids in the turbulent flow regime. Constant wall heat flux boundary condition was considered in the analysis. It was indicated that the heat transfer enhancement obtained with γ-Al_2O_3 particles is higher than that obtained with TiO_2 particles. They proposed a new correlation for Nusselt number.

Li and Xuan (2002) presented experimental study to investigate the heat transfer coefficient and friction factor of Cu/water nanofluid in both laminar and turbulent flow regimes. Constant wall heat flux boundary condition was exposed and observed Nusselt enhancements up to 60%. It was seen that the heat transfer coefficient enhancement ratio (heat transfer coefficient of nanofluid divided by the heat transfer coefficient of base fluid) increases with increasing Reynolds number.

Wen and Ding (2004) investigated experimentally the convective heat transfer of Al_2O_3 with the base fluid of water in laminar flow under constant wall heat flux boundary condition. Particle volume fraction varied between 0.6% and 1.6%. The result showed increase of 49% Nusselt number for 1.6% volume fraction. Also for nanofluid the entrance region length was bigger than the base fluid that this length increase with increasing nanofluid particle volume fraction. Particle migration phenomenon that cause dispersing non-uniform thermal conductivity and viscosity and reducing the thermal boundary condition was the reason proposed by them for this anomalous enhancement in heat transfer.

Despite experimental investigation, the numerical simulation is fast progressing. It is seen that there are both single-phase and two-phase models are proposed for the analysis of nanofluids heat transfer.

Khanafer et al. (2003) were the first to simulate nanofluid flow. They analyzed the natural convection flow inside a square enclosure for Cu/Water nanofluid. Their result showed that heat transfer and velocity of nanofluid because of enhancement in thermal conductivity and Brownian motion of nanoparticles is higher than base fluid.

Maiga et al. (2004) numerically studied laminar and turbulent force convection inside a circular tube under constant wall heat flux boundary condition. They used single-phase assumption and took the effect of nanoparticles into account only through the substitution of the thermophysical properties of the nanofluids into the governing equations. They simulated the mixture of Al_2O_3/water and Al_2O_3/ethylene glycol and showed the wall shear stress and heat transfer enhance with increasing volume fraction while the latter nanofluid showed better heat transfer enhancement in identical Reynolds number and volume fraction.

Bianco et al. considered the laminar and turbulent flow of Al_2O_3/Water nanofluid under constant and uniform heat flux at the wall. They analyzed the problem by using both single and two-phase models. The results showed heat transfer enhances with increasing particles volume concentration and Reynolds number and it showed two-phase models for the simulation of nanofluid are satisfactory with comparing of experimental data.

In this study, two-phase model of Volume of Fluid (VOF) has been used for considering flow and heat transfer characteristic of nanofluids. CuO/water and γ-Al_2O_3 /water nanofluids are used in turbulent flow regime in a horizontal concentric double tube heat exchanger. The analyze are conducted for different Reynolds numbers and volume fractions ranging from 20,000 to 50,000 and 2, 3, 4 and 6 percent respectively. For validation of the numerical solution, the results are *compared* with experimental correlation proposed by (Pak and Cho, 1998). The aim of this study is to add more contribution to nanofluids convection.

Mathematical modelling

Geometric configuration

Fig. 1 shows the considered configuration consisting of the double tube counter-flow heat exchanger with a length of 0.65 m and with the inner and the outer diameters of 0.01 m and 0.015 m respectively. Nanofluids that enter the inner tubes composed of CuO/water and γ-Al_2O_3/water with particle diameter of 20 nm. Table 1 shows the thermophysical properties of base fluid and nanoparticles.

Nanofluids properties

Nanofluid thermophysical properties play important role in accuracy of the results. The effect of nanoparticles can be taken into account by using the thermophysical properties of the nanofluid in the governing equations. In analyzing nanofluid as a two-phase flow the interactions between nanoparticles and liquid are modelled too. The following formulas are used to compute the thermophysical properties of the nanofluids under consideration.

Pak and Cho (1998) compared the density values were measured experimentally with hydrometers and those calculated from (1) at various volume concentrations. The maximum deviation between (1) and experimental results was 0.6% at volume concentration of 3.16%. Therefore, this equation is appropriate for using nanofluid density:

$$\rho_{eff} = (1-\phi)\rho_{bf} + \phi\rho_p \tag{1}$$

Also specific heat of nanofluid are achieved by the following equation:

$$c_{peff} = \frac{(1-\varphi)(\rho c_p)_{bf} + \varphi(\rho c_p)_p}{(1-\varphi)\rho_{bf} + \varphi\rho_p} \tag{2}$$

Where subscripts of eff, bf and p indicate the effective properties, base fluid (water in this case) and particles respectively. Chon et al. (2005) proposed a correlation for thermal conductivity. This correlation except the volume fraction and particles diameter, considers the temperature and Brownian motion which is defined as:

$$\frac{k_{eff}}{k_f} = 1 + 64.7 \times \phi^{0.746} \times \left(\frac{d_{bf}}{d_p}\right)^{0.369} \times \left(\frac{k_p}{k_f}\right)^{0.746} \times Pr^{0.9955} \times Re^{1.2321} \tag{3}$$

where Prandtl and Brownian Reynolds numbers expressed as:

$$Pr = \frac{\mu}{\rho_f \alpha_f}, Re = \frac{\rho_f k_b T}{3\pi\mu^2 L_f} \tag{4}$$

where L_f is the base fluid mean free path (0.17 nm for water) and μ is temperature-dependent viscosity of the base fluid which is defined as:

$$\mu = A \times 10^{\frac{B}{T-c}} \tag{5}$$

the constants A, B, C for water are equal to $2.414\exp(-5)$, 247 and 140 respectively.

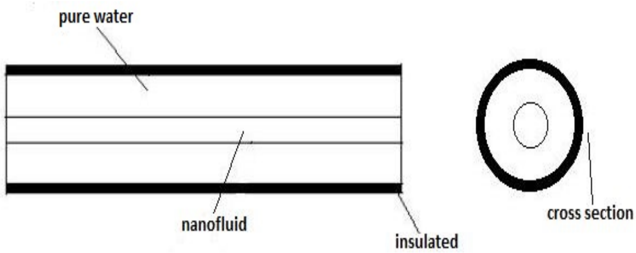

Figure 1. Schematic of considering configuration

Table 1. Thermophysical properties of materials under consideration.

Material	Density (Kg/m^3)	Thermal conductivity (W/m k)	Specific heat (J/Kg K)
Water	1000	0.6	4179
γ-Al2O3	3880	36	773
CuO	6310	32.9	550.5

One equation for dynamic viscosity of nanofluid is defined as:

$$\frac{\mu_{nf}}{\mu_{bf}} = 123\phi^2 + 7.3\phi + 1 \qquad (6)$$

Equation (6) was obtained by Maiga et al (2007) with curve fitting based on experimental data of Wang et al. (1999). And, this equation was successfully used by Maiga et al (2007) for simulating of single-phase nanofluid consist of γ-Al_2O_3/water nanofluid. In the present study (6) is used for simulating two-phase of γ-Al_2O_3/water nanofluid.

Another equation for effective dynamic viscosity of nanofluid which consider the effects of density, diameter of particles and volume fraction was proposed by Masoumi et al. (2009) as follows:

$$\mu_{nf} = \mu_{bf} + \frac{\rho_p V_B d_p^2}{72c\delta}, \delta = \sqrt[3]{\frac{\pi}{6\phi}} d_p \qquad (7)$$

where $V_B = \frac{1}{d_p}\sqrt{\frac{18k_bT}{\pi\rho_p d_p}}$, is the Brownian velocity of nanoparticles. In (7) the $c = \mu_f^{-1}\left[\left(c_1 d_p + c_2\right)\phi + \left(c_3 d_p + c_4\right)\right]$, are obtained from experimental data and are expressed as:

$$c_1 = -1.133\times10^{-6}, \quad c_2 = -2.771\times10^{-6}$$

$$c_3 = 9\times10^{-8}, \quad c_4 = -3.93\times10^{-7}$$

In the present study, (7) is used for calculating of CuO/water nanofluid effective dynamic viscosity.

Governing equations

In VOF model the two different phases do not interpenetrate. For each additional phase the volume fraction of that phase is added in the computational cell. If volume fraction of nanoparticles phase is indicated as ϕ, then the following three conditions are possible:

- $\phi = 0$ (The cell just contains base fluid)
- $\phi = 1$ (The cell just contains nanoparticles)
- $0 < \phi < 1$ (The cell contains the interface between the nanoparticles and base fluid phase respectively).

The tracking of the interface between the phases is accomplished by the solution of continuity equation for secondary phase (8) and then finding the volume fraction of primary phase conducts by the (9):

$$\nabla.(\phi_p \rho_p \vec{V}_p) = 0 \tag{8}$$

$$\sum \phi_k = 1 \tag{9}$$

The conservation of momentum and energy equations are given as (10) and (11) respectively:

$$\rho \vec{V} \cdot \nabla \vec{V} = -\nabla P + \nabla.(\mu(\nabla \vec{V} + \nabla \vec{V}^T)) \tag{10}$$

$$\nabla.(\vec{V}(\rho E + P)) = \nabla.(k \nabla T) \tag{11}$$

Turbulence modelling

In order to close the governing equations the standard k-ε two-equation eddy-viscosity model is used. This model proposed by Launder and Spalding (1972) and it is based on the solution of equations for turbulent kinetic energy k and the turbulent dissipation rate ε. Their equations can be expressed as follows:

$$\nabla.(\rho_{mix} V_{mix} k) = \nabla.(\frac{\mu_{t,mix}}{\sigma_k} \nabla k) + G_{k,mix} - \rho_{mix} \varepsilon \tag{12}$$

$$\nabla.(\rho_{mix} V_{mix} \varepsilon) = \nabla.(\frac{\mu_{t,mix}}{\sigma_\varepsilon} \nabla \varepsilon) + \frac{\varepsilon}{k}(c_1 G_{k,mix} - c_2 \rho_{mix}) \tag{13}$$

where

$$\mu_{t,mix} = \rho_{mix} c_\mu \frac{k^2}{\varepsilon} \tag{14}$$

$$G_{k,mix} = \mu_{t,mix}(\nabla V_{mix} + (\nabla V_{mix})^T) \tag{15}$$

With constant values of

$$c_1 = 1.44, c_2 = 1.92, c_\mu = 0.09 \text{ and } \sigma_\varepsilon = 1.3, \sigma_k = 1$$

Boundary conditions

Above equations are solved for the following boundary equations. At the inner tube inlet, uniform temperature profile with $T_{in} = 298\,K$ are assumed. Pure water enters in the annulus with uniform and constant velocity and temperature $V_{in,an} = 2.407 \frac{m}{s}$ and $T_{in,an} = 360\,K$ respectively. The inner tube is without thickness and outer tube is thermally insulated. At tubes outlet fully developed conditions and on the walls, the non-slip conditions are considered. Moreover, a constant turbulent intensity equal to 1% is imposed for both sides.

Computational procedure and validation

In the numerical solution, finite volume method is utilized for solving the above equations. PRESTO and QUICK Scheme is used for pressure correction and volume fraction respectively. For other equations second order upwind is adopted for numerical solution.

The SIMPLE algorithm is used for pressure-velocity coupling. Different non-uniform grids for heat exchanger inner and outer tubes are tested to insure independency of solution Fig. 2. 315000 and 537600 cells for the inner and the outer tube is sufficient for the present study respectively. Finer mesh is used near the wall because of higher velocity and temperature gradient. Mean Nusselt number is calculated as follows:

$$Nu = \frac{h_{ave} d}{k_{eff}} \tag{16}$$

Also overall heat transfer coefficient based on the inner tube heat transfer coefficient (h_{in}) and the outer tube heat transfer coefficient (h_{out}) can be expressed as follows :

$$U = \frac{1}{\frac{1}{h_{in}} + \frac{1}{h_{out}}} \tag{17}$$

The outer heat transfer coefficient (h_{out}) is constant and equal to 10530.82 for boundary conditions as mentioned before. In Fig. 3 validation takes place with a correlation proposed by Dittus and Boelter(1930) for pure fluid in turbulent pipe flow. The average and maximum error for the inner tube is 2% and 3.6% respectively. Also comparison with respect to overall heat transfer coefficient is conducted by the correlation proposed by Pak and Cho (1998) for nanofluid heat transfer Fig. 4.

Results and discussion

Results of numerical solution of convective heat transfer of different nanofluids in concentric heat exchanger with two-phase model (VOF) and different volume concentration (Maxwell, 1881; Masuda et al, 1993; Choi, 1995; Pak and Cho, 1998) at turbulent flow regime are presented. The mean diameter of CuO and γ-Al_2O_3 nanoparticles are assumed to be 20 nm. Fully developed (hydrodynamically and thermally) turbulent flow is assumed for the outer and the inner tube for $(\frac{L}{D_h}) = 130$ and $(\frac{L}{D_h}) = 65$ respectively (Incropera & Bergman, 2006). Constant uniform velocity inlet and turbulent intensity for pure water equal to 2.407 m/s and 1% are assumed in the annulus for all runs. Therefore, heat transfer coefficient of outer tube (h_{out}) is constant and equal to 10530.82. Comparison with experimental correlation proposed by Pak and Cho (1998) for nanofluid in term of U_{ave} also has taken place.

Figure 4 depict overall heat transfer enhancement by increasing volume fractions of nanoparticles for different Reynolds numbers. As shown in Figs. 4, overall heat transfer enhances by augmenting of nanoparticle concentration and also Reynolds number. Effect of Reynolds number is more pronounced on heat transfer enhancement when compared to nanoparticles volume concentration. For example for CuO/water nanofluid for a fixed volume concentration of $\phi = 2\,\%$, for Re=20,000 to 50,000 overall heat transfer enhanced from 5406.32 to 7221.13. On the other hand, for Re=20,000 and volume concentration varying from 0 to 6% the enhancement is 4620.75 to 5590.66. But with combination of these two parameters simultaneously overall heat transfer as high as 7375.05 can be achieved in comparison of the value 4620.53 for Re=20,000 and $\phi = 0$. The maximum and average deviation for Re=50,000 in comparison of experimental data are 3.8% and 3.1% for γ-Al_2O_3 /water nanofluid and 4.5% and 3.5% for CuO/water nanofluid respectively.

It is obvious that numerical prediction of γ-Al_2O_3/water nanofluid is closer to experimental data. One of this reason is dynamic viscosity of γ-Al_2O_3/water nanofluid which is purely experimental. Also Pak and Cho (1998)'s experimental study was carried out for γ-Al_2O_3/water and TiO_2/water nanofluids. It should be mentioned that this experimental correlation is valid for volume concentration ranges between 0 to 3% and other nanoparticles volume concentration is just showing the trends. Hence, using enhanced models for thermal conductivity and viscosity which considering more heat transfer mechanisms of nanofluids can promote the numerical solution.

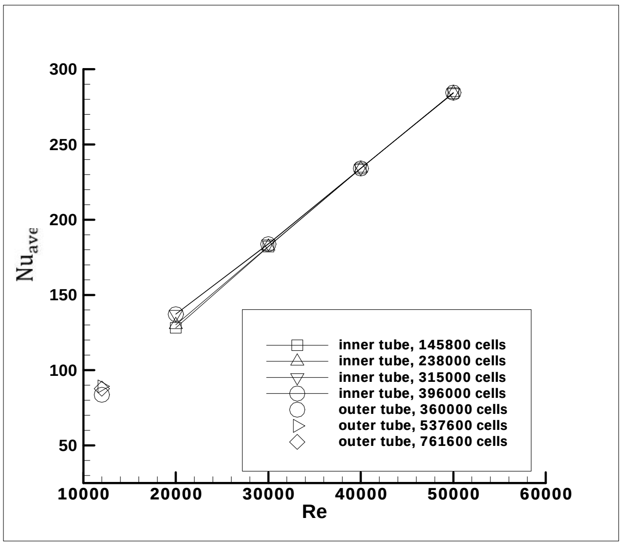

Figure 2. Different grids for independency of solution

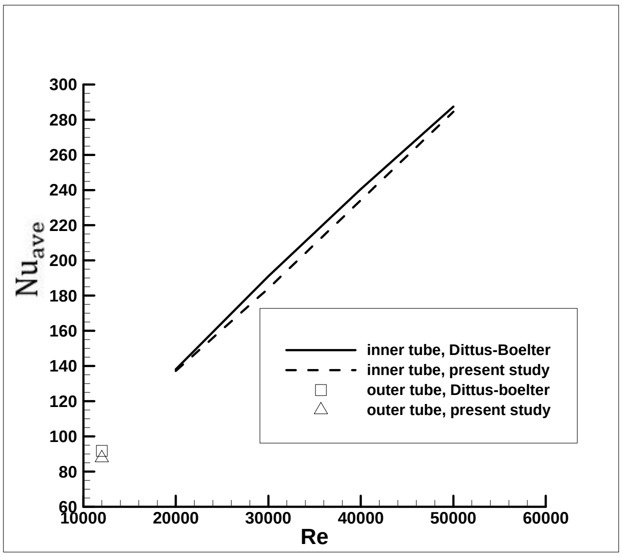

Figure 3. Grid validation for inner tube and annulus by correlation proposed by Dittus and Boelter (1930)

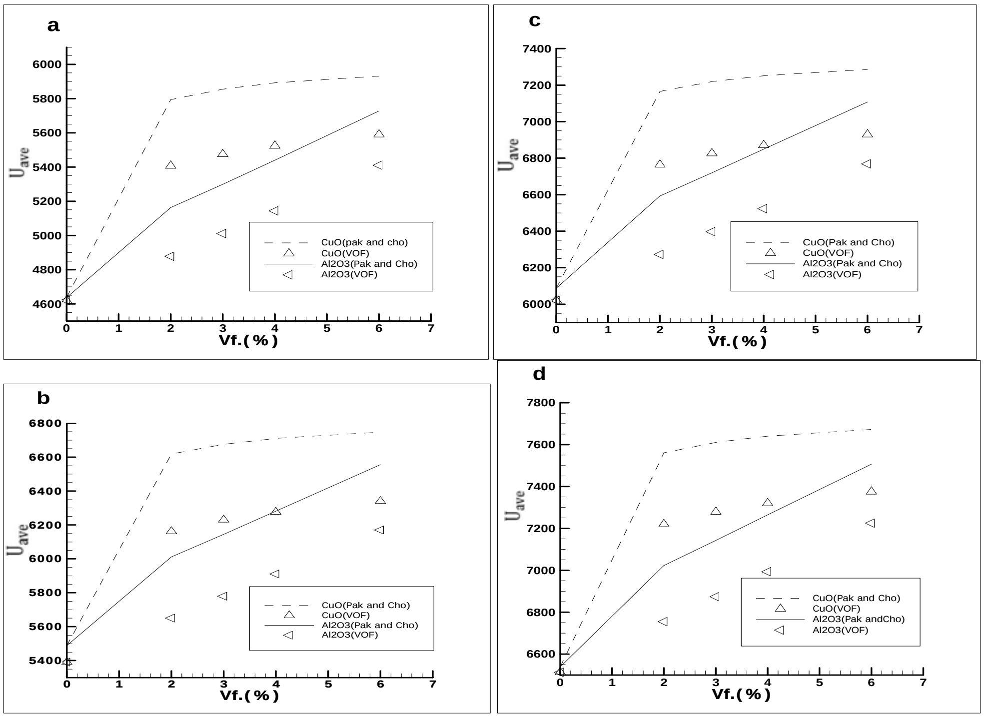

Figure 4. Overall heat transfer coefficient enhancement for (a) Re=20,000 (b) Re=30,000 (c) Re=40,000 and (d) Re=50,000

As a result of Fig. 4, heat transfer enhancement using CuO/water is more than γ-Al_2O_3/water nanofluid, however the thermal conductivity of γ-Al_2O_3 nanoparticle is almost bigger than CuO one. Consequently, thermal conductivity of nanoparticles is not the dominant parameter that specifies the convection heat transfer enhancement of the nanofluid. Average wall shear stress of different nanofluids by augmenting volume concentration of nanoparticles with different Reynolds numbers are shown in Fig. 5. As a result, wall shear stress increase with increasing of Reynolds number and volume concentration of nanoparticles. Although CuO/water nanofluids predicts more enhancement in heat convection than γ-Al_2O_3/water, but wall shear stress of former is much bigger than the latter as depict in Fig. 5. For example for volume fraction of 2% and Reynolds number of 30,000 the average wall shear stress of γ-Al_2O_3/water and CuO/water nanofluids are 36.7 and 100.98 respectively.

As shown in Fig. 5(a) to Fig. 5(d) the effect of Reynolds number on increasing the wall shear stress is more than volume fraction of nanoparticles. For Re=20,000 and $\phi = 0$ to $\phi = 6\,\%$ for example for CuO/water nanofluid the wall shear stress increases from 15.6 to 72.8. On the other hand, for a fixed value of $\phi = 2\,\%$ and Re=20,000 to Re=50,000 the wall shear stress varying from 57.9 to 239.93 that the effect of increasing Reynolds number on average wall shear stress is more pronounced.

Conclusion

In the present paper, turbulent forced convection of γ-Al_2O_3/water and CuO/water nanofluids inside a double tube concentric heat exchanger was numerically investigated using two-phase approach (VOF).

Two-phase approach underestimates the overall heat transfer coefficient but a comparison with experimental correlation proposed by Pak and Cho (1998) showed that the numerical results are at good agreement of this correlation.

Results showed for a fixed Reynolds number of 20,000, for CuO/water and just $\phi = 6\,\%$ overall heat transfer enhancement about 17% achieved in comparison of pure water ($\phi = 0$). Also using simultaneously Reynolds number and ϕ, for Re=50,000 and $\phi = 6\,\%$ overall heat transfer enhancement about 37% was achieved in comparison of Re=20,000 and $\phi = 0$. Then nanofluid can be good substitution for pure water in heat transfer devices for optimizing the energy.

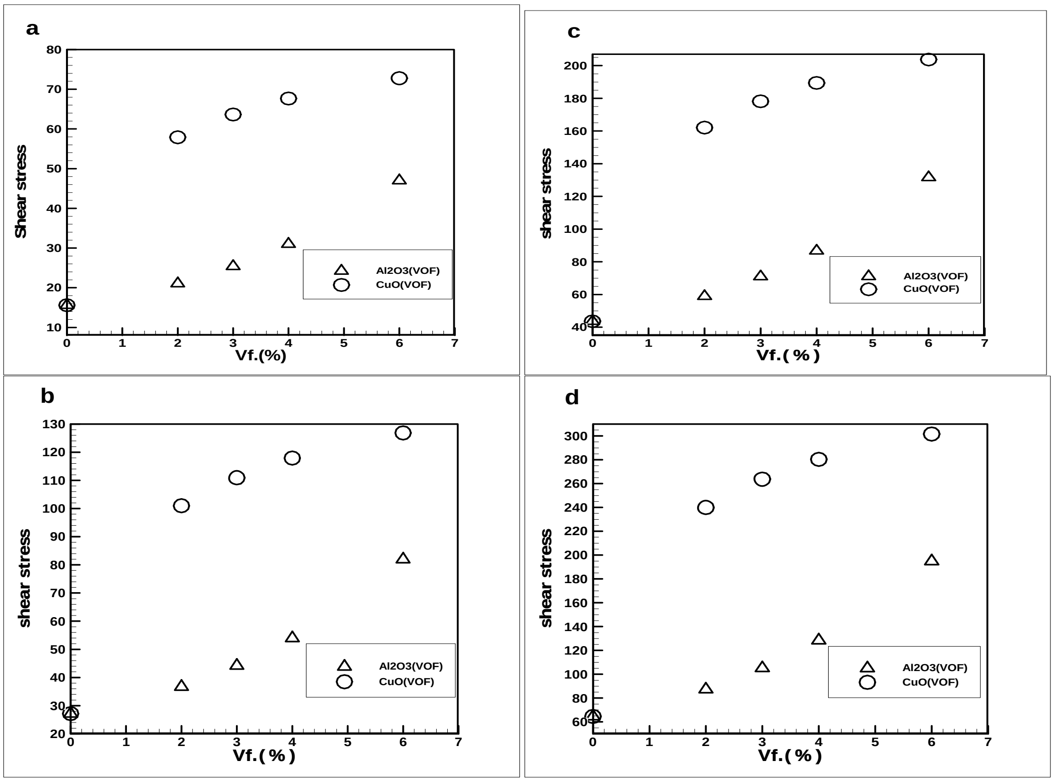

Figure 5. Average wall shear stress prediction for (a) Re=20,000 (b) Re=30,000 (c) Re=40,000 and (d) Re=50,000

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