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Numerical Investigation of Turbulent Forced Convection Nanofluids Inside an Annulus

Turbulent forced convection heat transfer of Al_2O_3-water nanofluid has been studied numerically under uniform heat flux on the inner and outer walls in an annulus with rough tubes. Solid nanoparticles diameter were considered to be 18, 32, and 67 nm. Two-dimensional elliptic governing equations were used and the second-order upstream difference scheme and finite volume method were used for the discretization of governing equations. SIMPLEC algorithm has been established the relationship between pressure and velocity. The results demonstrated that the surface friction coefficient increases both in the inner and outer wall of the heat exchanger by increasing the diameter of nanoparticles. On the other side, the Nusselt number of nanofluid is greater than the base fluid for a given Reynolds number and nanoparticle volume fraction. Also, the Nusselt number decreases in the inner and outer wall with increasing the diameter of nanoparticles.

Keywords: *Turbulent flow, Forced convection, Annulus, Nanofluid.*

1. Introduction

Heat exchangers are devices that widely used in industry. Different ways exist for increasing heat transfer in heat exchangers which divide into two groups: active and passive. Enhanced heat transfer using nanofluid is a method that presented by Choi [1]. Heat transfer behavior of nanofluid depends on various parameters including thermal conductivity, heat transfer coefficient, viscosity, heat capacity, and volume fraction of nanoparticles. Many studies have been conducted on nanofluids and most scientists believed that the addition of nanoparticles to base fluids such as water and Ethylene Glycol increase heat transfer. Bianco et al. [2] numerically investigated turbulent forced convection heat transfer of water/Al_2O_3 nanofluid in a circular tube. They showed that the heat transfer coefficient of nanofluid is greater than the base fluid and also with increasing volume fraction of nanoparticles, Reynolds number increases. In another study [3], they also analyzed the turbulent convection of the nanofluid inside a circular tube with constant wall temperature. They found that with increasing nanofluid concentration, the Nusselt number increases, furthermore entropy generation and pumping power increase. Williams et al. [4] experimentally investigated heat transfer characteristics of nanofluids in a circular tube in both laminar and turbulent regimes. They found that with increasing volume fraction, the heat transfer increases in both regimes. Minea [5] numerically examined turbulent convective heat transfer in a two-dimensional micro-tube with constant temperature in the wall. The results indicated that convective heat transfer coefficient of nanofluid is higher than base fluid. Increasing the area is one of the ways to increase the heat transfer. Installed Fin on channels is used as common method in the industry, which increases the efficiency of heat transfer systems. Another way to increase system performance is to make rough pipes and heat exchanger with rough tubes in the industry. In addition, this method increases the heat transfer and reduces the fouling. The heat transfer investigated in the enhanced tubes by David et al. [6]. Their study showed that the rough tubes have less fouling and more heat transfer than smooth tubes. Many studies carried out both numerically and experimentally in the heat exchangers. Izadi et al. [7] numerically investigated laminar forced convection heat transfer in an annulus section of a double-tube heat exchanger. They used Single-phase approach for modeling nanofluid and concluded that the heat transfer and surface friction coefficients increase with increaseing volume fraction. Mokhtari et al. [8] numerically studied heat transfer enhancement of a mixed convection laminar Al_2O_3-water nanofluid flow in an annulus. In another study [9], they also examined the effect of nanoparticles mean diameter on the hydrodynamics and thermal characteristic in an annulus by two-phase mixture model. They calculated that Nusselt number decreases with increasing nanoparticle mean diameter while it does not influence significantly the hydrodynamic parameters. Lotfi et al. [10] performed some experiments in the case of increasing heat transfer of multi-walled carbon nanotube (MWNT)/water nanofluids in a shell and tube heat exchanger. The result of their study showed that heat transfer increases in the presence of multi-walled nanotubes in comparison with the base fluid. Zamzamian et al. [11] investigated the effect of CuO/EG, Al_2O_3/EG nanofluids on the forced convective heat transfer coefficient in turbulent flow using a double pipe and plate heat exchangers. They examined the effect of particles concentration and operating temperature on the nanofluids forced convective heat transfer coefficient and found that there is enhancement about 2% to 50% as compared to the base fluid. Eiamsa et al. [12] considered the increase of heat transfer using the TiO_2 nanofluids in a heat exchanger tube equipped with overlapped dual twisted-tapes. They made changes in Reynolds number, overlapped twist ratio, and also volume concentration of nanofluid in order to investigate changes of heat transfer rates, friction factor, and thermal performance. Elias et al. [13] studied the effect of different particle shapes on the overall heat transfer coefficient, heat transfer rate, and entropy generation of shell and tube heat exchanger with different baffle angles and segmental baffle. They found that cylindrical shape shows the best result in the case of heat transfer coefficient and heat transfer rate. Garoosi et al. [14] carried out numerical investigation on the heat exchangers for natural convection of the nanofluid by using the Buongiorno model. They examined the effect of different design parameters on the heat transfer ratio and nanoparticles distribution. They found that there is an optimal volume fraction of the nanoparticles at each Rayleigh number in which the maximum heat transfer rate can be obtained. Wael and Aly [15] studied numerically turbulent heat transfer and pressure drop of nanofluid in coiled tube-in tube heat exchangers. Prasad et al. [16] performed experiments on the U-tube heat exchanger with helical tape inserts. They measured changes of the Nusselt number, thermal conductivity, and friction factor for AL_2O_3 nanofluids with making change in Reynolds number, volume concentrations, and helical tape inserts. As seen in these and similar studies, heat transfer parameters of nanofluids have been less investigated in rough tubes. Therefore, the present study investigates turbulent heat transfer of nanofluids of Al_2O_3/water in a double tube heat exchanger with rough tubes. For the discretization of governing equations, the second-order upstream difference scheme and finite volume method are used. The relationship between pressure and velocity is established by using SIMPLEC algorithm.

2. Mathematical modeling

Turbulent forced convection of a nanofluid consisting water and Al_2O_3 is considered in an annulus with uniform heat flux at the wall.

Fig. 1 shows the considered geometrical configuration. Because of symmetry, only one-half of the geometry is considered. Based on the reasons mentioned in [7] the Single-phase approach is used for nanofluid modeling. The nanofluid is a mixture of water and Al_2O_3 particles, with different mean diameters of 18, 32, and 67 nm. The governing equations for steady state are as follows:

Continuity equation:

$$\nabla.(\rho_{eff}\boldsymbol{V}_m) = 0 \tag{1}$$

Conservation of momentum:

$$\nabla.\left(\rho_{eff}\boldsymbol{V}_m\boldsymbol{V}_m\right) = -\nabla p + \nabla.\left[\tau - \tau_t\right] \tag{2}$$

Conservation of energy:

$$\nabla.(\rho_{eff} C_p V_{eff} T) = \nabla.(\lambda_{eff} \nabla T - C_p \rho_m \overline{vt}) \tag{3}$$

In Eq. (2), the shear relation is given by:

$$\tau = \mu_m \nabla V_m \quad , \quad \tau_t = \sum_{k=1}^{n} \Phi_k \rho_k \overline{v_k v_k} \tag{4}$$

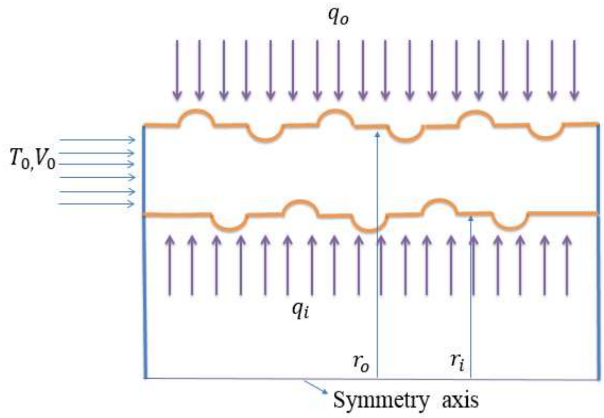


Figure 1. Schematic of the considered heat exchanger.

3. Turbulence modeling

The turbulent flow is modeled by Launder and Spalding [17] k–ε model for the mixture. It is expressed by Eqs. (5)- (7):

$$\nabla.(\rho_m V_m k) = \nabla.(\frac{\mu_{t,m}}{\sigma_k} \nabla k) + G_{k,m} - \rho_m \varepsilon \tag{5}$$

$$\nabla.(\rho_m V_m \varepsilon) = \nabla.(\frac{\mu_{t,m}}{\sigma_\varepsilon} \nabla k) + \frac{\varepsilon}{k}(c_1 G_{k,m} - c_2 \rho_m \varepsilon) \tag{6}$$

where

$$\mu_{t,m} = \rho_m c_\mu \frac{k^2}{\varepsilon} \qquad G_{k,m} = \mu_{t,m}(\nabla V_m + (\nabla V_m)^T)$$

$$c_1 = 1.44, c_2 = 1.92, c_\mu = .09, \sigma_k = 1, \sigma_\varepsilon = 1.3 \tag{7}$$

4. Boundary conditions

The boundary conditions are expressed as follows:

•At the inlet of annulus (X = 0):

$$\mathrm{V_{mx} = V_i \quad , V_{mr} = 0 \quad , T = T_i} \tag{8}$$

Turbulent intensity calculated based on the formula [18]:

$$I_0=0.16(Re)^{-1/8} \tag{9}$$

• At the solid –fluid interfaces ($r = r_0$, $r = r_i$):

$$V_{mr} = V_{mx} = 0 \text{, at } r = r_i : -k_{eff}\frac{\partial T}{\partial r} = q_i \text{, at } r = r_0 : -k_{eff}\frac{\partial T}{\partial r} = q_o \tag{10}$$

• At the tube outlet:

The diffusion fluxes are set to zero at the exit for all dependent variables and an overall mass balance correction is obeyed.

5. Nanofluids thermophisical properties

The physical properties are:

Effective density:

The nanofluid density is given by Cho and Pak [19].

$$\rho_m = (1-\phi)\rho_f + \phi\rho_p \tag{11}$$

Where the volumetric concentration is given by Choi et al. [20].

$$\phi = \frac{\rho_f \phi_m}{\rho_f \phi_m + \rho_p (1-\phi_m)} \tag{12}$$

Where ϕ_m is the mass fraction.

An accurate equation is used for calculating the effective heat capacitance [21].

$$(C_p)_{eff} = \left[(1-\phi)(\rho C_p)_f + \phi(\rho C_p)_p\right] / \rho_m \tag{13}$$

The thermal conductivity of the nanofluid is calculated from Chon et al. [22] equation that considers the Brownian motion and mean diameter of the nanoparticles.

$$k_{eff}/k_f = 1 + 64.7 \times \phi^{0.746} \left(d_f / d_p\right)^{0.369} \times \left(k_p / k_f\right)^{0.746} \times pr^{.9955} \times \mathbf{Re}^{1.2321} \tag{14}$$

Where Pr and Re in Eq. (14) are defined as:

$$Re = \frac{\rho_f B_c T}{3\pi\mu^2 l_{bf}}, \quad pr = \frac{\mu_f}{\rho_f \alpha_f}$$

$$\mu = A \times 10^{\frac{B}{T-C}}, C = 140, B = 247, A = 2.414e-5 \tag{15}$$

l_{bf} is the mean free path of water and B_c is Boltzman constant.

Effective viscosity is calculated by the following equation proposed by Masoumi et al. [23] which considers the effects of volume fraction, density, average diameter of nanoparticles, and physical properties of the base fluid:

$$\mu_{eff} = \mu_f + \frac{\rho_P V_B d_P{}^2}{72C\delta}, \quad V_B = \frac{1}{d_p}\sqrt{\frac{18K_bT}{\pi\rho_P d_P}}, \quad \delta = \sqrt[3]{\frac{\pi}{6\phi}}d_P$$

$$C = \mu_f^{-1}\left[\left(c_1 d_p + c_2\right)\phi + \left(c_3 d_p + c_4\right)\right] \tag{16}$$

$$C_1 = -0.000001133, \quad C_2 = -0.000002721$$

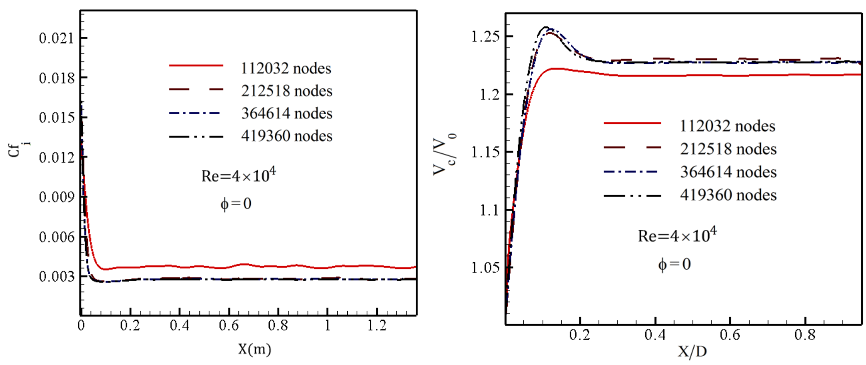
$$C_3 = -0.00000009, \quad C_4 = -0.000000393$$

6. Numerical method and validation

This set of coupled nonlinear differential equations discretizes with the control volume technique. The second-order upwind method is used for the convective and diffusive terms while the SIMPLEC procedure is introduced for the velocity–pressure coupling. A structured non-uniform grid distribution uses to discretize the computation domain. It is finer near the tube entrance and the wall where the velocity and temperature gradients are large. Several different grid distributions test to ensure that the calculated results are grid independent. As shown in Fig. 2, increasing the grid numbers do not significantly change the thermal and hydrodynamic characteristics of the nanofluid.

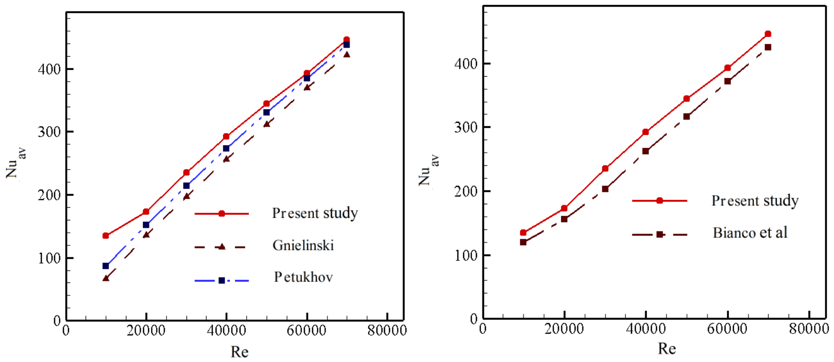
In order to demonstrate the validity, precision of the model, and numerical procedure, comparison is done with the available experimental and numerical simulation. As it is shown in Fig. 3 (a, b), good agreements are observed between the results. Fig.3.a. shows the comparison of the calculated results with the results obtained by Gnielinski [24] and Petukhov [25].

Another comparison is also done with the numerical results obtained by Bianco et al. [9]. (See Fig.3.b.).



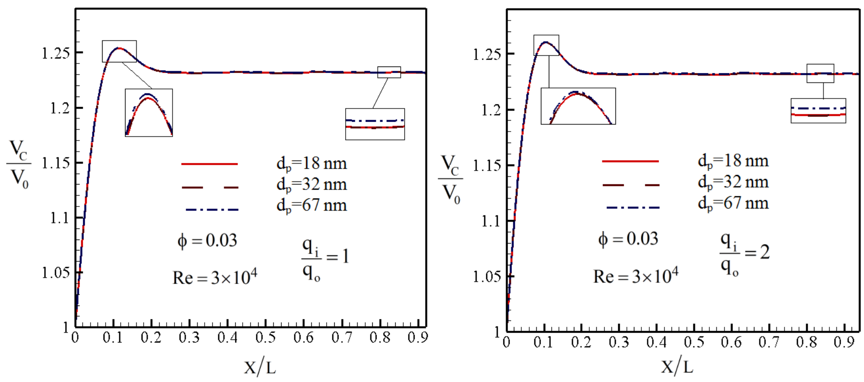
(a) **(b)**

Figure 2. (a) Grid independence test surface friction coefficient in the inner wall **(b)** Grid independence test: centerline axial velocity.



(a) **(b)**

Figure 3. (a) Comparison of the axial evolution of average Nusselt number in an annulus with the results obtained by Gnielinski [24] and Petukhov and **(b)** with the numerical results obtained by Bianco et al [2].



(a) **(b)**

Figure 4. (a) Effect of nanoparticles mean diameter on the dimensionless axial velocity at $\frac{q_i}{q_o} = 1$ **(b)** at $\frac{q_i}{q_o} = 2$.

7. Results and discussion

In a numerical fashion, this paper investigated hydrodynamics and thermal behavior of turbulent forced convection heat transfer of nanofluid flow (AL_2O_3 and water) of nanoparticles different sizes with mean diameter (d_p= 18, 32, and 67 nm) in a two -dimensional annulus. Investigations are carried out in the Reynolds number Re=3×10^4 and volume fraction of 3% of variation of the inner wall heat flux to the outer wall heat flux. The axial velocity along with the annulus centerline at different nanoparticle size are shown in Fig. 4. As it is clearly seen, after the annulus inlet, the boundary layer growth pushes the fluid toward the centerline region, and causes an increase of the centerline velocity. As the nanoparticles size increases, the velocity maximum point moves further upstream, since increase of axial momentum transports the generated turbulence in the flow direction. After the maximum point, the velocity decreases at the centerline in order to respect the continuity equation. It is interesting to note that the maximum of the dimensionless centerline velocity increases as nanoparticle mean diameter increases. This effect is due to the fact that the corresponding velocity profiles become more non-uniform as d_p increases. On the other hand increasing the inner wall heat flux to the outer wall heat flux do not change appreciably the maximum velocity.

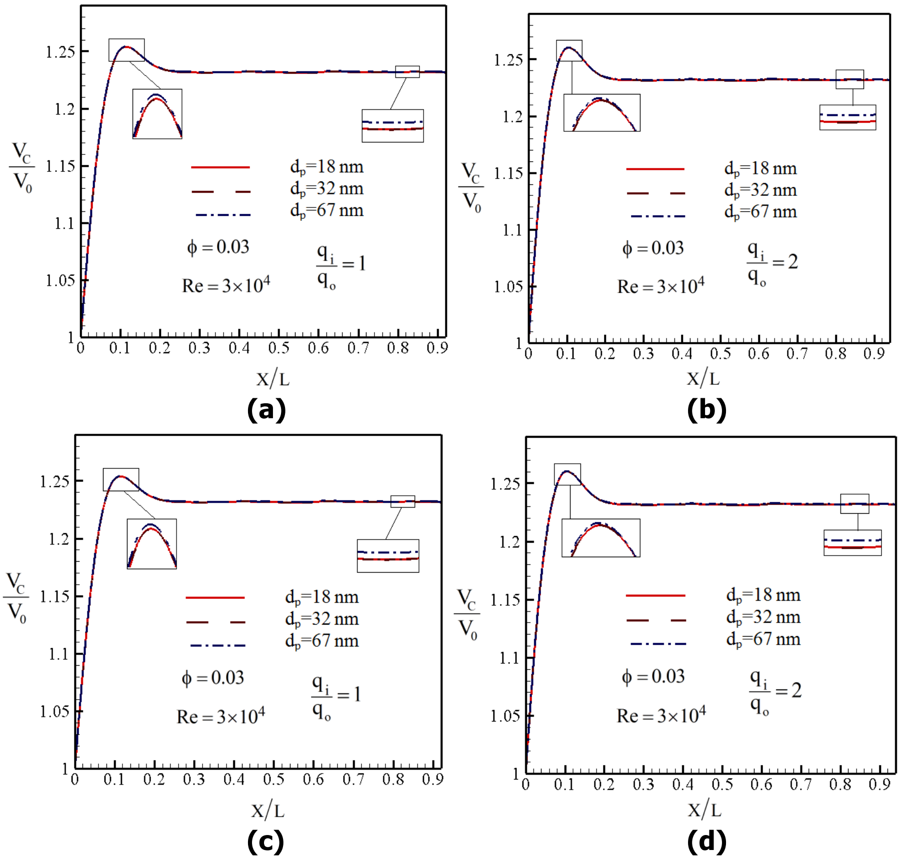
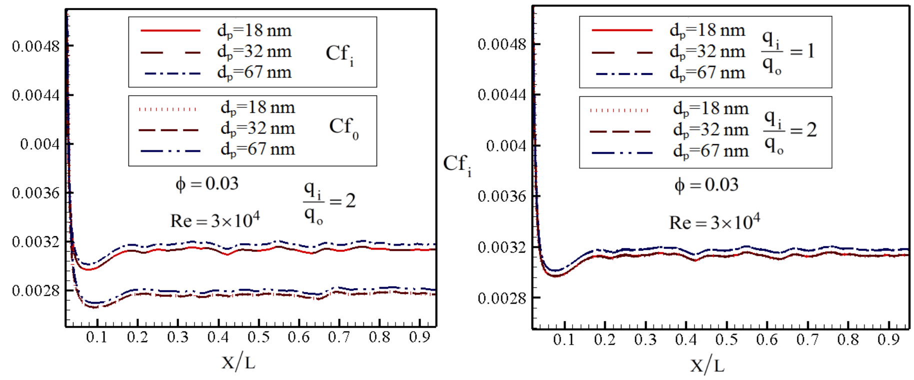


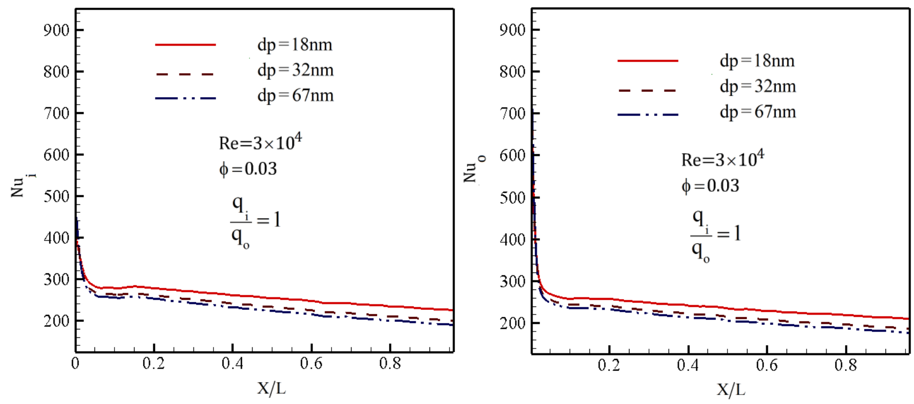
Figure 5. Effect of nanoparticles mean diameter on inner and outer wall surface friction coefficients for $\frac{q_i}{q_o} = 1$ and $\frac{q_i}{q_o} = 2$.



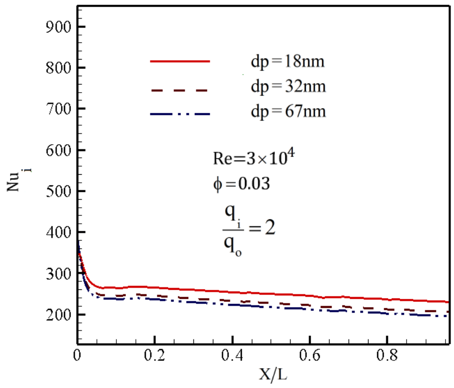
(a) **(b)**

Figure 6. Effect of nanoparticles mean diameter on the surface friction coefficient.

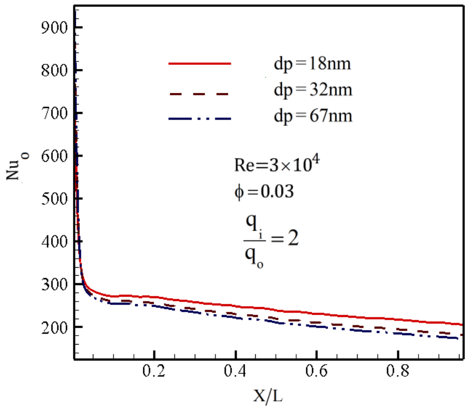
Fig.5. shows the surface friction factor changes along the annulus centerline dimensionless. As can be seen in front of the annulus due to large surface friction coefficient to the velocity gradient will tend to very high and by moving the tube length with a sharp slope down to and reaches a minimum and then remains constant. As shown in Fig. 5, the surface friction coefficient increases both in the inner and outer wall of the double tube heat exchanger with increasing nanoparticles mean diameter. In Fig.6.a., comparison of the surface friction coefficient of the inner wall with outer wall in annulus is done at $q_i/q_o=2$, given Reynolds number ($Re=3\times10^4$) and volume fraction of 3% . As shown in this figure, the surface friction coefficient in the inner wall is more than the outer wall. On the other side, as shown in Fig.6.b., with increasing ratio of the inner wall heat flux to the outer wall ($q_i/q_o=1$ to $q_i/q_o=2$), significant effect on the surface friction coefficient is not found.



(a) **(b)**



(c)



(d)

Figure 7. Effect of nanoparticles mean diameter on the Nusselt number.

Fig.7. shows the effect of nanoparticles mean diameter on the local Nusselt number along the annulus length for a given Reynolds number Re=3×10^4 at two different wall heat flux ratio (q_i/q_o=1 to q_i/q_o=2). In all cases, the Nusselt number decreases with increase of nanoparticles mean diameter.

The Nusselt number becomes high in the entrance and then goes monotonically to its asymptotic value further downstream. At the beginning of the annulus, after the annulus input rate is too high, this is due to the proximity of the wall temperature and bulk fluid temperature. At constant Reynolds number Re=3×10^4 and the volume fraction of 3%, with increasing nanoparticles size, both Nusselt number at the inner and outer wall decreases. As can be seen in the Fig 8.a., the Nusselt number in the inner wall heat flux is generally higher than the outer wall heat flux. On the other hand, as shown in Fig.8.b., with increasing ratio of the inner wall heat flux to the outer wall (q_i/q_o=1 to q_i/q_o=2), the Nusselt number increases.

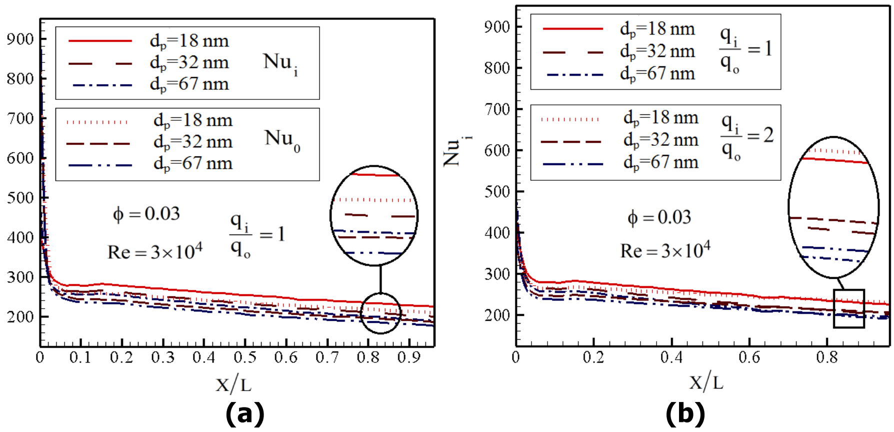


Figure 8. Effect of nanoparticles mean diameter on the Nusselt number.

8. Conclusions

In this paper, the effect of nanoparticles diameter on hydrodynamic and thermal parameters of forced convective heat transfer of Aluminum Oxide/water nanofluids turbulent flow is numerically studied in a circular segment of a two-tube heat exchanger with rough tubes. To solve the governing equations, single-phase model is used and SIMPLEC algorithm is also used for their discretization. The uniform heat flux in the inner and outer wall of the exchanger is also assumed. The results indicated that surface friction coefficient increases both in the inner and outer wall of the heat exchanger by increasing the diameter of nanoparticles. The surface friction coefficient in the inner wall is more than the outer wall. However, with increasing ratio of the inner wall heat flux to the outer wall, significant effect on the surface friction coefficient is not found. Nusselt number is another studied parameter in this paper which decreases with increasing the diameter of nanoparticles in the inner and outer wall. Nusselt number in the inner wall is also more than the outer wall. However, Nusselt number increases with increasing heat flux ratio of the inner wall to the outer wall.

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