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The Effect of Reynolds Number on the Thermal and Hydrodynamic Characteristics of Turbulence Flow of the Nanofluid in the Heat Exchanger

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Abstract. Reynolds number is one of the most important parameters in investigation of heat transfer in double tube heat exchangers. In this paper, the effect of this parameter has been investigated on the convective heat transfer coefficient and surface friction coefficient of the wall. Turbulent forced convection heat transfer of nanofluid flow of Al_2O_3 /water in a double tube heat exchanger with rough tubes in the annular portion was numerically studied. The finite volume method and the second-order upstream difference scheme were used for the discretization of the governing equations. The volume fraction and mean diameter of nanoparticles were assumed to be 4% and 32 nm, respectively. After reviewing the results, it was observed that the convective heat transfer coefficient in the outer and inner walls of heat exchanger increases with the increase of the Reynolds number. The wall surface friction coefficient, which decreases by increasing Reynolds number, is another examined parameter.

Keywords: Reynolds number, heat exchanger, convective heat transfer coefficient, surface friction coefficient

1. INTRODUCTION

Heat exchangers are devices that transfer heat from the hot fluid to the cold fluid. Enhanced heat transfer in heat exchangers is one of the topics of interest and many studies have been done in this regard. [1, 2, 3] One of the ways to increase the heat transfer in heat exchangers which was first suggested by Maxwell is adding metallic solid particle to base fluid such as water [4]. Choi [5] also suggested adding metallic nanoparticles and nonmetallic particles to the base fluid. With the advent of nanofluid subject, extensive experimental and numerical studies have been conducted by researchers all over the world. They believe that the heat transfer would be improved by adding the solid nanoparticles (diameter of particles less than 100 nm) [6, 7, 8]. Behzadmehr et al., [9] investigated turbulent forced convection heat transfer of Cu/water nanofluid with 1% of volume fraction of copper in a circular tube by using two-phase mixture method. They used two-phase mixture model for the first time. They observed that Nusselt number and the heat transfer coefficient increase by increasing Reynolds number.

Namburu et al. [10] investigated the numerical study of the turbulent flow and heat transfer characteristics of nanofluid in a circular tube under constant heat flux in the wall. Three different nanoparticles including SiO_2, CuO, and Al_2O_3 were used in this study. They reported that the nanofluids containing nanoparticles with smaller diameter have higher viscosity and Nusselt number. Furthermore, at the same Reynolds number for CuO/water nanofluid with 6% volume fraction, Nusselt number increases to 35% relative to the base fluid.

Torii [11] examined the heat transfer of turbulent flow for different nanofluids in a circular tube under constant heat flux in the wall. In this research, three nanoparticles including

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diamond, CuO, and Al_2O_3 were used. In this study, the effect of nanoparticles on increasing heat transfer, thermal conductivity, viscosity, and pressure drop in turbulent flow was studied. He concluded that the heat transfer, viscosity, and pressure drop were increased by enhancing the volume fraction of the particles.

Fotukian and Nasr Esfahany [12] experimentally investigated the forced convective heat transfer of the turbulent flow of γ-Al_2O_3/water nanofluid with the volume fraction less than 0.2% in a circular tube. They showed that addition of small amounts of nanoparticles to the base fluid augmented heat transfer, remarkably. Their results showed a 48% increase in heat transfer in nanofluids compared with pure water for 2% volume fraction. In another study [13], they used CuO/water nanofluid with a volume fraction less than 0.24 percent. In this case, the heat transfer considerably enhances by adding of a small amount of nanoparticles to base fluid.

Investigation of the heat transfer with nanofluid in a variety of heat exchangers has been developed. Rabienataj Darzi et al., [14] experimentally investigated characteristics of the heat transfer and flow of Al_2O_3-water nanofluid in a double tube heat exchanger. They measured effective viscosity of nanofluid at different temperatures ranging from 27 to 55 ° C. In this experiment, they considered the Reynolds number ranging from 5000 to 20000 and in volume fraction of upper than 1%. Their results showed that there is a good potential to improve the thermal performance of the heat exchangers by adding nanoparticles in the study area. Finally, they found an equation on the basis of Reynolds number and the volume fraction of nanoparticles to compute Nusselt number. The numerical and experimental study of nanofluid performance in a compact minichannel plate heat exchanger was performed by Ray et al. [15]. For this purpose, they used three nanofluid including aluminum oxide; copper oxide and silicon dioxide in ethylene glycol and water mixture base fluid to study their performance in the heat exchanger. Regarding to the volume fraction of dilute particles of 1%, they observed performance improvement for each of the three nanofluid compared to the base fluid. On the other hand, the convective heat transfer coefficient increases for all three nanofluid, but the volumetric flow rate and the pumping power requirement increase. Halelfadl et al. [16] investigated the heat transfer properties of aqueous carbon nanotubes in a coaxial heat exchanger under laminar regime. They used the nanofluid in the inner portion and the distilled water in the annular portion of the exchanger to stabilize the temperature of the wall and assumed the Reynolds number ranging from 500 to 2500. They concluded that in a volume fraction of 0.026 of the Carbon nanotubes the heat transfer increases compared with the base fluid averagely about 12%. They also found out that increasing the aspect ratio of the nanotubes and the base fluid with a lower thermal conductive ability helps to improve and increase the heat transfer of nanofluid.

Elias et al. [17] investigated the effect of different nan oparticles forms in a shell and tube heat exchanger by using different baffles angle and operative nanofluid numerically. They used four types of nanoparticles with diverse shapes of cylinder, brick, blade and platelet. They observed that cylindrical nanoparticle has better performance in the overall coefficient of heat transfer and heat transfer rate in comparison to other forms at different angles of the baffle.

Mohammad et al. [18] numerically investigated the effect of various types of nanofluids and Reynolds numbers on the characteristics of heat transfer and fluid flow in a square-shaped micro-channel heat exchanger. For this purpose, they used four nanoparticles including aluminum oxide, silicon dioxide, silver, and titanium dioxide, with three volume fractions of 2%, 5%, and 10% and water as the base fluid. They observed that nanofluid increases thermal characteristic and performance of nanofluid whereas the pressure drop also increases slightly. They also found with increasing Reynolds number, the pumping power enhances and the efficiency is reduced.

Zamzamian et al. [19] investigated the effect of the nanoparticles volume fraction and operating temperature on the convective heat transfer coefficient in plate and double pipe heat exchangers under turbulent flow. They used Al_2O_3 and CuO nanoparticles in water and Ethylene Glycol base fluids with the volume fraction ranging from 0.1% to 1%, respectively. They reported that the convective heat transfer coefficient increases with increasing the volume fraction of nanoparticles and the operating temperature.

In the light of previous studies, it is observed that most of the researchers used smooth tubes for studying heat transfer in heat exchanger. In this paper, the effect of Reynolds number is investigated numerically on thermal and hydrodynamic characteristics of nanofluid turbulent flow of Al_2O_3/water in the annular portion of the double tube heat exchanger with rough tubes under constant heat flux in the wall. The finite volume method and the second-order upstream difference scheme are used for the discretization of the governing equations.

2. MATHEMATICAL MODELING AND SOLUTION

Studied geometry is shown in Fig. 1. It is a double tube exchanger with circular cross section made of copper, length =1.40 m, interval radius r=0.01m, and external radius r=0.02m. Nanofluid is assumed as mixture of water and Al_2O_3 nanoparticles with a diameter of 32 nm.

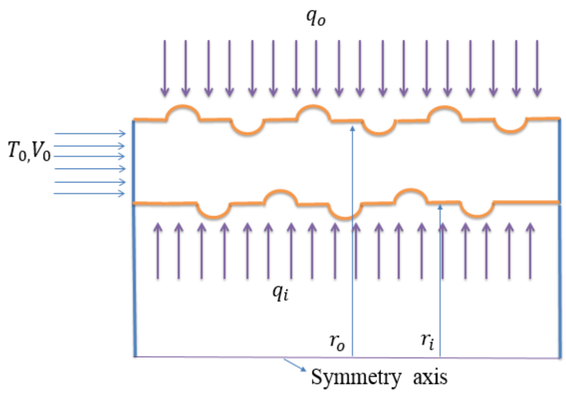

Figure 1. Studied geometry (schematic of a double tube heat exchanger)

2.1. Governing equations

The applied governing equations in the solution is expressed as follows:

Continuity equation:

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

Where ρ_{eff} represents the effective density.

Momentum equation:

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

Energy equation:

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

Where in Eq. (2), τ represents the shear stress and is given by:

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

C_p represents the specific heat and λ is indicative of thermal conductivity.

The turbulent flow is modeled by Launder and Spalding [20] k–ε model. The k–ε model is considered by two equations, one for the turbulent kinetic energy and the other for the dissipation rate.

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

$$\nabla.(\rho_mV_m\varepsilon)=\nabla.(\frac{\mu_{t,m}}{\sigma_\varepsilon}\nabla k)+\frac{\varepsilon}{k}(c_1G_{k,m}-c_2\rho_m\varepsilon) \tag{6}$$

$$\mu_{t,m}=\rho_mc_\mu\frac{k^2}{\varepsilon}\,,\; G_{k,m}=\mu_{t,m}(\nabla V_m+(\nabla V_m)^T) \tag{7}$$

$$c_1=1.44, c_2=1.92, c_\mu=.09, \sigma_k=1, \sigma_\varepsilon=1.3 \tag{8}$$

2-2. Nanofluids thermophysical properties

Effective density expresses according to equation (9) is given by [21]

$$\rho_m=\left(1-\phi\right)\rho_f+\phi\rho_p \tag{9}$$

Where ϕ represents the volume fraction of nanoparticles and is given by [22].

$$\phi=\frac{\rho_f\phi_m}{\rho_f\phi_m+\rho_p\left(1-\phi_m\right)} \tag{10}$$

Where ϕ_m is the mass fraction

The effective specific heat capacity of nanofluid expresses according to equation (11) is given by [23].

$$\left(C_p\right)_{eff}=\left[\left(1-\phi\right)\left(\rho C_p\right)_f+\phi\left(\rho C_p\right)_p\right]\Big/\rho_m \tag{11}$$

Thermal conductivity coefficient of the nanofluid of Chon et al [24] correlation, is obtained as follows:

$$k_{eff}/k_f = 1 + 64.7 \times \phi^{0.746} \left(d_f/d_p\right)^{0.369} \left(k_p/k_f\right)^{0.746} \times pr^{.9955} \times \mathbf{Re}^{1.2321} \tag{12}$$

Where Pr and Re in Eq. (12) are defined as

$$pr = \frac{\mu_f}{\rho_f \alpha_f}, \quad Re = \frac{\rho_f B_c T}{3\pi\mu^2 l_{bf}}, \quad \mu = A \times 10^{\frac{B}{T-C}}, C = 140, B = 247, A = 2.414e-5 \tag{13}$$

l_{bf} is the mean free path of water and B_c is Boltzman constant($B_c = 1.3807 \times 10^{-23}$ J/K).

Effective viscosity is calculated by the following equation proposed by Masoumi et al. [25] which considers the effects of volume fraction, density, and average diameter of nanoparticles and physical properties of the base fluid:

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

$$C = \mu_f^{-1}\left[\left(c_1 d_p + c_2\right)\phi + \left(c_3 d_p + c_4\right)\right] \tag{14}$$

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

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

2.3. Boundary conditions

The uniform velocity V_0 and entrance temperature T_0= 293K at the inlet of the annular portion and the static pressure at the outlet are assumed to be known. Also, the heat flux q_i and q_0 is considered on the exchanger walls. The finite volume method and the second-order upstream difference scheme are used for the discretization of displacement parameter of governing equations. The SIMPLEC algorithm is used for the relationship between pressure and velocity parameters.

2.4. Numerical solution and validation

To assess the independence of mesh solution, the results of Re=4×10^4 parameters are compared. To determine the optimal node, different nodes are compared according to various parameters that surface friction coefficient of the external wall is given in Fig.2. Comparison of the present study and classical equation [26] is performed in Fig.3. As seen in this figure, the present study has small error compared with the classical equation and therefore, it is valid.

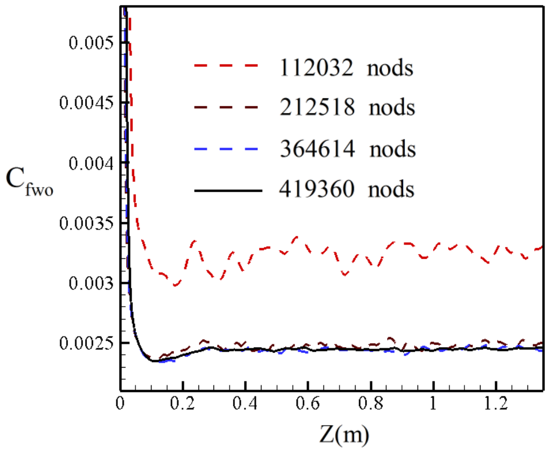

Figure 2. Surface friction coefficient of the outer wall for selecting optimal node

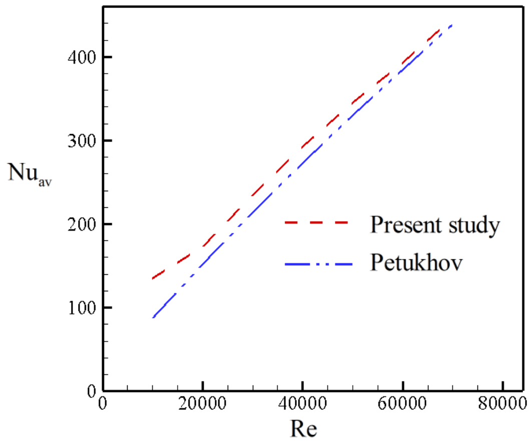

Figure 3. Validation of the present study with previous studies

3. RESULTS AND DISCUSSION

The surface friction coefficient on the wall is one of the most important parameters in a double tube heat exchanger. Fig.4. indicates variations in the surface friction coefficient on the inner wall of the heat exchanger with rough tubes and solid-liquid volume fraction of 3%, at different Reynolds numbers. As seen, surface friction coefficient decreases with increasing Reynolds number. This coefficient is a function of shear stress on the wall, the density of nanofluid, and flow velocity. With increasing Reynolds number, shear stress increases. On the other hand, by increasing Reynolds number, the flow velocity will be too high. Bringing these parameters results in reduced surface friction coefficient with Reynolds number increasing.

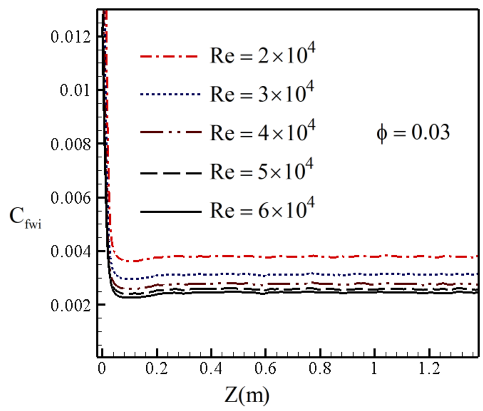

Figure 4. Changes in surface friction coefficient of the inner wall of the exchanger for different Reynolds numbers

Fig.5. shows changes in surface friction coefficient on the outer wall of the heat exchanger with solid-liquid volume ratio of 3%, at different Reynolds numbers. Also in this case, the surface friction coefficient decreases with increasing Reynolds number. By comparing Fig.4 and Fig.5, it is observed that the surface friction coefficient of the outer wall of the exchanger is smaller than the inner wall.

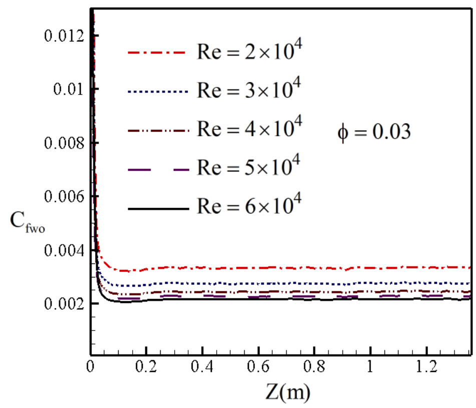

Figure 5. Changes in surface friction coefficient of the outer wall of the exchanger for different Reynolds numbers.

According to conducted studies with solid-liquid volume ration of 3% and uniform heat flux on the exchanger's walls, it become clear that convective heat transfer coefficient increases with Reynolds number increasing. This is shown in Fig.6. It shows the effect of Reynolds number on the convective heat transfer coefficient for the inner wall of the exchanger. With increasing Reynolds number, the flow velocity, flow turbulence, and turbulence kinetic energy increase, thereby the convective heat transfer coefficient will be increased by increasing these parameters. Changes in the convective heat transfer coefficient of the exchanger's external tube is plotted in Fig.7. Due to the reasons mentioned above, in this case with increasing Reynolds number, the convective heat transfer coefficient increases. By comparing Fig.6 and Fig.7, it is observed that the convective heat transfer coefficient in the outer wall of the exchanger is smaller than the inner wall.

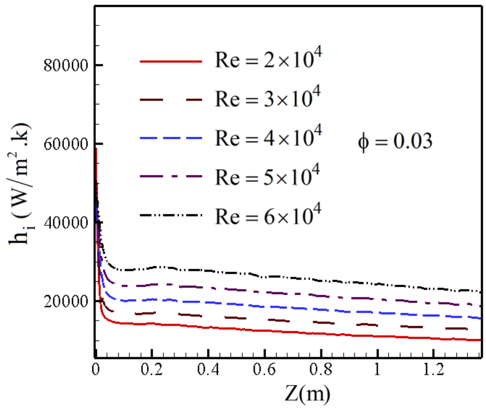

Figure 6. Changes in convective heat transfer coefficient of exchanger's inner tube for different Reynolds numbers.

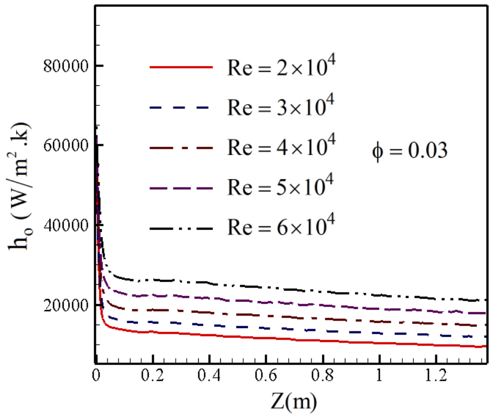

Figure 7. Changes in convective heat transfer coefficient of exchanger's outer tube for different Reynolds numbers.

4. CONCLUSION

In the present study, forced convection heat transfer of the turbulent flow of AL_2O_3/ water nanofluid was studied numerically in annular portion of double tube heat exchanger with rough tubes by the finite volume method and the second-order upstream difference scheme. Results show that with increasing Reynolds number, surface friction coefficient of both inner and outer walls of the exchanger will decrease in a constant volume fraction and specified diameter of nanoparticles. On the other hand, with the same constant parameters mentioned above, it is observed that with increasing Reynolds number, the forced convective heat transfer coefficient of the nanofluid turbulent flow will be increased in the inner and outer walls of the double heat exchanger.

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Nomenclature

Cf_i	Inner wall surface friction coefficient, $Cf_i = \tau_i / (\frac{1}{2}\rho_m V_{in}^2)$
Cf_o	Outer wall surface friction coefficient, $Cf_o = \tau_o / (\frac{1}{2}\rho_m V_{in}^2)$
C_p	specific heat of the fluid, J/kg K
d_f	molecular diameter of base fluid (nm)
d_p	nanoparticle diameter (nm)
D_h	hydraulic diameter, m
h_i	Local convective heat transfer coefficient in the inner wall $h_i = q_i / (T_{wi} - T_m)$
h_o	Local convective heat transfer coefficient in the outer wall $h_o = q_o / (T_{wo} - T_m)$
L	channel length, m
P	pressure (Pa)
q_i	Inner wall heat flux, W/m^2
q_o	outer wall heat flux, W/m^2
Re	Reynolds number, $Re = (\rho_m V_{in} D_h)/\mu_m$

T,t	time averaged and fluctuating temperature
V,v	time averaged and fluctuating velocity
X	axial coordinate, m

Greek letters

α	Thermal diffusivity
ε	dissipation of turbulent kinetic energy, m^2/s
ϕ	volume fraction of nanoparticle
λ	thermal conductivity of the fluid, $W/m\,K$
μ	fluid dynamic viscosity, $Kg/m.s$
ρ	fluid density, Kg/m^3
τ	shear stress, Pa