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Research Paper

An experimental investigation of heat transfer of Fe_2O_3/Water nanofluid in a double pipe heat exchanger

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ABSTRACT: One way to increase the heat transfer is to use perforated twisted tapes with different hole diameters, which largely improve heat transfer with an increase in the heat transfer area at the constant volume and more mixed flow. In the previous studies, the effect of simultaneous use of nanofluids and perforated twisted tapes is less studied. In this work, the performance of water / iron oxide nanofluid in a double pipe heat exchanger with perforated twisted tapes is investigated under turbulent flow regime. Reynolds number considered is in the range between 2500 to 20500.Iron oxide nanoparticles with diameter of 15 nm are used as nanofluid with the concentration range from 0.12 to 0.2% by volume. The results showed that the addition of nanoparticles increases the heat transfer and the Nusselt number. Also, reducing the twist ratio (H/D=2.5) of perforated twisted tape and using the nanofluid with concentration of 0.2%v/v increase this value by 130%.

Keywords: *Double pipe heat exchanger; Heat transfer; Nanofluids; Nusselt number; Twisted tapes.*

INTRODUCTION

Nanofluid is a new kind of heat transfer medium, containing nanoparticles (1–100 nm) which are uniformly and stably dispersed in a base fluid. These distributed nanoparticles greatly enhance the thermal conductivity of the nanofluid and increase conduction and convection coefficients [1].

Aghayari *et al.* [2] investigated the enhancement of heat transfer coefficient and Nusselt number of a nanofluid containing nanoparticles(γ-AL_2O_3) with a particle size of 20nm and volume fraction of 0.1%–0.3% (V/V). Effects of temperature and concentration of nanoparticles on Nusselt number changes and heat transfer coefficient in a double pipe heat exchanger with counter turbulent flow are investigated. Comparison of experimental results with valid theoretical data based on semi empirical equations shows an acceptable agreement. Experimental results show a considerable increase in heat transfer coefficient and Nusselt number up to19%–24%, respectively. Moreover, it has been observed that the heat transfer coefficient increases with the operating temperature and concentration of nanoparticles.

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Aghayari *et al.* [3] reported experimental results, which illustrated the heat transfer and overall heat transfer coefficient of AL_2O_3 nanoparticles in liquid for turbulent flow in a double pipe heat exchanger. Impacts of the Reynolds number, volume fraction, temperature and nanoparticle source on the overall heat transfer coefficient have been investigated. The experimental results showed that the heat transfer coefficient increases with the Reynolds number and the particle concentration. Aluminum oxide nanofluid with concentrations of 0.2 and 0.3 had high thermal efficiency compared to the base fluid. For example, this amount is 1450000 for water at a constant mass flow rate and a temperature of 50°C. This amount is 1565000 and 1580000 for the nanofluid at the concentrations of

0.2 and 0.3, respectively. Thermal efficiency of water and nanofluid with the concentration of 0.1 is 1103842 and 1123123, respectively (in Reynolds of 23000), which is approximately 1.71% higher than the heat transfer of the base fluid. This increase can be attributed to the immigration of the particles, non-uniform distribution of the thermal conductivity and viscosity of the fluid, which decreases the boundary layer thickness, resulting in the delay in the formation of the thermal boundary layer.

Maddah *et al.* [4] investigated heat transfer and overall heat transfer in a double pipe heat exchanger fitted with twisted-tape elements and titanium dioxide nanofluid experimentally. Titanium dioxide nanoparticles with a diameter of 30 nm and a volume concentration of 0.01% (v/v) were prepared. The effects of temperature, mass flow rate, and concentration of nanoparticles on the overall heat transfer coefficient, heat transfer changes in the turbulent flow regime (Ree" 2300), and counter current flow were investigated. When using twisted tape and nanofluid, heat transfer coefficient was about 10 to 25 percent higher than when they were not used. It was also observed that the heat transfer coefficient increases with operating temperature and mass flow rate. The experimental results also showed that 0.01% TiO_2 water nanofluid with twisted tape has slightly higher friction factor and pressure drop than 0.01% TiO_2/water nanofluid without twisted tape. The empirical correlations proposed for friction factor were in good agreement with the experimental data.

Maddah *et al.* [5] studied thermal and physical behavior of the Al_2O_3 Nano-ûuid in a horizontal double pipe counter-ûow heat exchanger ûtted with modified twisted tapes under turbulent ûow conditions. The experiments with and without typical twisted tapes and nanofluid were performed under similar operation condition and validated with existing well-established correlations to verify experimental setup. Al_2O_3 nanoparticles with diameter of 21 nm dispersed in water and the concentration was varied from 0.2 to 0.9% by volume. The nanoûuid considered as non-Newtonian fluid due to the shear-thinning rheological behavior. The mathematical concept of geometrical progression was applied to prepare modified twisted tapes. Pitch length of the proposed twisted tapes and consequently the twist ratios changed along the twists with respect to the Geometrical Progression Ratio (GPR) whether reducer (RGPR<1) or increaser (IGPR>1). The experiments were performed using modified tapes with seven different Geometrical Progression Ratios (RGPR = 0.6, 0.75and 0.85, typical twist with GPR=1, IGPR = 1.2, 1.5 and 2) over a Reynolds number range of 5000 to 21000. Regarding the experimental data, utilization of RGPR twists together with nanofluids tend to increase heat transfer and friction factor by 12% to 52% and 5% to 28% as compared with the tube with the typical twisted tapes (GPR=1)and nanofluid. Contrarily, performances were weakened by using for IGPR twists 0.6 to 0.92 and 0.75 to 0.95 times of those in the typical twisted tapes and nanofluid. Over the range investigated, heat transfer rates were enhanced in terms of Nusselt numbers by the RGPR twisted tape with nanofluid up to 4 and with IGPR twists up to 1.84 times of that in the plain tube. Sheikholeslami *et al.* [6] investigated natural convection in a concentric annulus between a cold outer square and heated inner circular cylinders in presence of static radial magnetic field numerically using the lattice Boltzmann method. The numerical investigation was performed for different parameters such as the Hartmann number, nanoparticles volume fraction and Rayleigh number. Domiri Ganji *et al.* [7] studied nanofluid flow and heat transfer between parallel plates considering Brownian motion using differential transformation method. Soleimani *et al.* [8] investigated natural convection heat transfer in a semi-annulus enclosure filled with nanofluid. In their study, they used the Control Volume based Finite Element Method (CVFEM). Domiri Ganji *et al.* [9] investigated Ferrofluid flow and heat transfer in a semi annulus enclosure in the presence of magnetic source considering thermal radiation. In their study, Sheikholeslami *et al.* [10] examined Lattice Boltzmann Method for simulating of the effect of magnetic field on hydrothermal behavior of nanofluid in a cubic cavity. They used Koo–Kleinstreuer–Li correlation in calculating the effective viscosity and thermal conductivity of nanofluid. The impacts of active parameters such as Hartmann number, nanoparticle volume fraction and Rayleigh number on flow and heat transfer were also investigated. Sheikholeslami *et al.* [11] investigated effects of spatially variable magnetic field on ferro fluid flow and heat transfer using control volume based finite element method(CVFEM). Moreover, the combined effects of ferrohydrodynamic and magnetohydrodynamic were considered.

EXPERIMENTAL

Nanofluid Preparation

The nanofluid used in the experiment was 99.0+% pure iron oxide pre-dispersed in water, with an average particle size of 15 nm. The nanofluid was mixed with deionized water. To prepare experimental concentrations, nanofluids with less than 3% nanoparticles were found to be stable and the stability lasted over a week; no intermediate mixing was considered necessary. Fig. 1 represents the morphology of Fe_2O_3 nano-particles by using TEM.

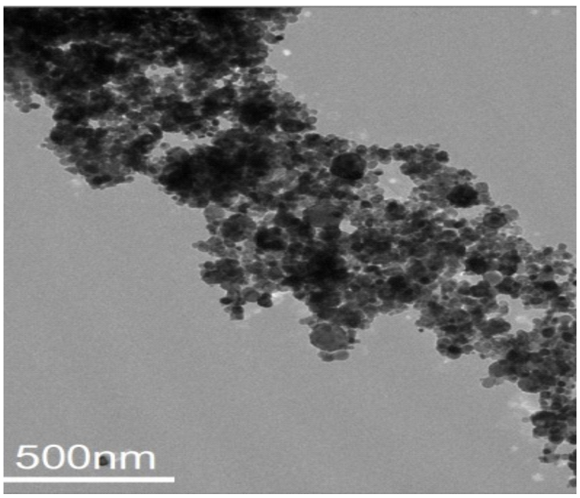

Fig. 1: TEM photograph of Fe_2O_3 particles.

Experimental Setup

Experimental apparatus used in this study is depicted in Fig. 2. The apparatus consists of a test section (heat exchanger), two tanks, two magnetic gear pumps, and a pump for transporting nanofluid as the hot fluid and the other for the cold water. The test section is a counter current double pipe heat exchanger with the length of 1.20 m. In this exchanger, the nanofluid flows into the pipe and cold water in the annular space of the pipe. The inside pipe is made of a soft steel tube with the inner diameter of 0.006 m, outer diameter of 0.008 m, and thickness of 0.002 m while the outside pipe is of steel tube with the inner diameter of 0.014 m, outer diameter of 0.016 m, and thickness of 0.002 m. To reduce the heat loss along the axis, the top and bottom of the test section are insulated with the plastic tubes. To measure the inlet and outlet temperatures of the nanofluid and cold water at the inlet and outlet of the test section, 4 RTD thermometers are used. It is necessary to measure the temperature at six stations altogether at the outer surface of the test section for finding out the average Nusselt number. All six evaluated temperature probes are connected to the data logger sets. The pressure drop across the test section is measured by using inclined U-tube manometers. The 15-liter tanks made of stainless steel are used for the storage of nanofluid and cold water. To maintain the temperature of the fluid, a cooling tank and a thermostat are used. An electric heater and a thermostat installed on it are used to maintain the temperature of the nanofluid. Measured Nusselt number error depends on the measurement of the temperature and the flows of the cold water and nanofluid. During the test, the wall temperature of the test section, the mass flow rate, and the inlet and outlet temperatures of the nanofluid and cold water are measured.

Fig. 2: Experimental setup.

The twisted tapes were made from copper sheet with tape thickness (δ) of 0.001 m and length of 1.2 m. The tape thickness of 0.001 m was chosen to avoid an additional friction in the system that might be occurred by the thicker tape. To produce the modified twisted tape, the typical twists changed by changing twist ratio (H/D) to 5.1, 3.2 and 2.5 (Fig. 3).

The experimental data were used to calculate Nusselt number of nanofluids with various particle volume concentrations. For fluid flows in a concentric tube heat exchanger, the heat transfer rate of the hot fluid (nanofluid) in the inner tube can be expressed as[12]:

$$Q_{(nf)} = \mathring{m}_{nf} C_{p\,nf} (T_{out} - T_{in}) \tag{1}$$

Where $\overset{\circ}{m}_{nf}$ is the mass flow rate of the nanofluid (hot fluid), T_{out} and T_{in} are the outlet and inlet temperatures of the nanofluid (hot fluid), respectively.

While the heat transfer of the cold fluid (water) for the outer tube is [12]:

$$Q_{bf} = \overset{\circ}{m}_{bf} C_{pbf}(T_{in} - T_{out}) \tag{2}$$

Where $\overset{\circ}{m}$ is the mass flow rate of the water (cold fluid), T_{out} and T_{in} are the inlet and outlet temperatures of the water (cold fluid), respectively. The average heat transfer rate between nanofluid and cooling water is calculated as follows [13]:

$$\frac{Q_{nf} + Q_{bf}}{2} \tag{3}$$

The density and specific heat of the nanofluids are calculated by use of the Pak and Cho[14] correlations, which are defined as follows:

$$\rho_{nf} = (1 - \varphi_V)\rho_f + \varphi_v \rho_p \tag{4}$$

Subscripts f, p and nf refer to the base fluid, the nanoparticles, and the nanofluid, respectively. φ_V is the nanoparticle volume concentration.The heat transfer coefficient of the test fluid, h_i ,can be calculated by the following equation [15]:

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{D_i \mathrm{Ln}\frac{D_o}{D_i}}{2k_w} + \frac{D_i}{D_o} + \frac{1}{h_o} \tag{5}$$

Where D_i and D_o are the inner and outer diameters of tubes, respectively. U_i is the overall heat transfer coefficient based on the inside tube area, h_i and h_o are the individual convective heat transfer coefficients of the fluids inside and outside the tubes, respectively and k_w is the thermal conductivity of the tube wall. is given by:

$$Q = U_i A_i \Delta T_{lm} \tag{6}$$

Where ΔT_{lm} is the logarithmic mean temperature difference. The convection heat transfer from the test section can be written as [16]:

$$Q_{(convection)} = h_i A_i ((T_{\tilde{W}} - T_b) \tag{7}$$

$$T_b = \frac{T_{out(nano\ fluid(hot\ fluid))} + T_{in(nano\ fluid(hot\ fluid))}}{2} \tag{8}$$

$$\left(T_{\tilde{W}} = \sum \frac{T_W}{6}\right) \tag{9}$$

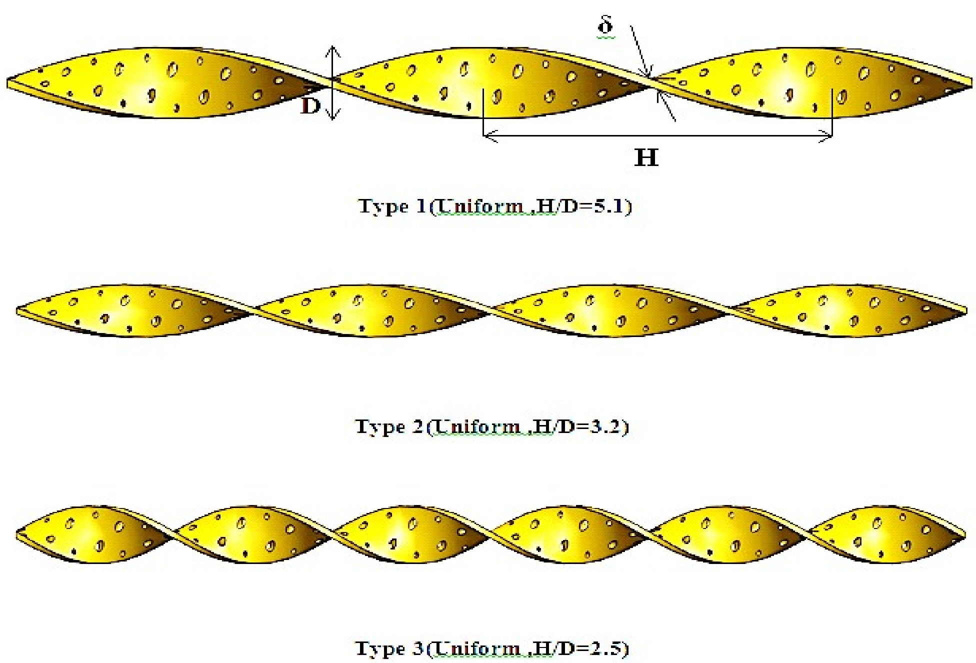

Fig. 3: Schematic of tested twisted-tape inserts.

T_w is the local surface temperatureat of the outer wall of the inner tube. The average surface temperature T_w is calculated from 6 points of T_w lined between the inlet and the exit of the test tube. The heat transfer coefficient, h_i and the Nusselt number, Nu are estimated as follow [17]:

$$h_i = \overset{\circ}{m}_{(\text{nano fluid})} C_{p(\text{nano fluid})} (T_{out} - T_{in}) / A_i ((\tilde{T_W} - T_b)$$

$$\overset{\circ}{m}_{(\text{nano fluid(hot fluid)})} C_{p(\text{nano fluid(hot fluid)})}$$

(10)

$$Nu_{nf} = \frac{h_i d_i}{k_{nf}} \tag{11}$$

Where the effective thermal conductivity (k_{nf}) of the nanofluids can be evaluated by Timofeeva correlations model that is given as follows [18]:

$$k_{nf} = (1 + 3\varphi) k_w \tag{12}$$

The thermal conductivity of the nanofluids is calculated from the Wasp [19] model, which is defined as follows:

$$k_{nf} = \left[\frac{k_p + 2k_{m-} 2\varphi(k_w - k_p)}{k_p + 2k_{w+} \varphi(k_w - k_p)} \right] k_w \tag{13}$$

Moreover, k_{nf} is the thermal conductivity of the nanofluid, k_p is the thermal conductivity of the nanoparticles, and k_m is the thermal conductivity of the base fluid. Viscosity of the nanofluid is calculated using Einstein [20] correlations as below:

$$\mu_{nf} = (1 + 2.5\varphi) \mu_w \tag{14}$$

Brinkman [21] suggested the equation for calculating the viscosity of the suspension, which is defined as follows:

$$\mu_{nf} = \frac{1}{(1-\varphi)^{2.5}} \mu_w \tag{15}$$

The Reynolds number and Prandtl's number of nanofluid are calculated using following relations [23]:

$$Re_{nf} = \frac{\rho_{nf} V_{nf} d_i}{\mu_{nf}} \tag{16}$$

$$Pe_{nf} = \frac{\mu_{nf} Cp_{nf}}{K_{nf}} \tag{17}$$

Where, V_{nf} is the average velocity of nanofluid flowingthrough the copper tube. Nusselt number of the nanofluid flow is computed from the following equation:

$$Nu_{nf} = \frac{h_{nf} d_i}{K_{nf}} \tag{18}$$

Gnielinski correlation for turbulent flow through a tube [23]:

$$Nu = 0.012(Re^{0.87} - 280) Pr^{0.4} \tag{19}$$

Similarly to the heat transfer coefficient, the friction factor of the nanofluid is calculated from[22]:

$$f_{nf} = \frac{2D\Delta P_{nf} l}{u_m^2 l \rho_{nf}} \tag{20}$$

Where f_{nf} is the friction factor of the nanofluid, ΔP_{nf} is the measured pressure drop of the nanofluid, l is the length of the tube, D is the diameter of the tube, ρ_{nf} is the density of the nanofluid, and u_m is the mean velocity of the nanofluid.

Uncertainty analysis

A systematic error analysis was used to estimate the errors in the experimental analysis. The uncertainties calculated with the maximum possible error for the parameters and various instruments are given in Tables 1 and 2, respectively.

RESULTS AND DISCUSSION

$$Re = \frac{4m^{\cdot}}{\pi D \mu'}, \quad \frac{U_{Re}}{\text{Re}} = \left(\sqrt{\left(\frac{U_{\dot{m}}}{\dot{m}}\right)^2} + \sqrt{\left(\frac{U_\mu}{\mu}\right)^2} \right) \sqrt{\left(\frac{U_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{U_\mu}{\mu}\right)^2} =$$

$$\sqrt{(0.0001^2) + (0.01^2)} = 0.01\%$$

(b) Heat transfer coefficient

$$h = \frac{q}{T_w - T_b}, \frac{U_h}{\text{h}} = \sqrt{\left(\frac{U_q}{q}\right)^2 + \left(\frac{U_{T_w} - T_b}{T_w - T_b}\right)^2} =$$

$$\sqrt{(0.2130)^2 + (0.0687)^2} = 0.2238\%$$

(c) Nusselt number

$$Nu = \frac{hD}{K} \quad , \quad \frac{U_{Nu}}{Nu} = \sqrt{\left(\frac{U_h}{h}\right)^2 + \left(\frac{U_K}{K}\right)^2} =$$

$$\sqrt{(0.1876)^2 + (0.08)^2} = 0.2039\%$$

Table 1: Uncertainties of parameters.

S. no.	Variable name	Uncertainty error,%
1	Reynolds number, Re	0.01
2	Heat transfer coefficient, h	0.2238
3	Nusselt number, Nu	0.2039

Table 2: Uncertainties of experimental instruments.

Name of instrument	Range of instrument	Variable measured	Least division in measuring instrument	Min. and max. values measured in experiments	Uncertainty (%)
Thermocouple	-200 to 1372?	Balk temperature	0.1?	20-50?	0.126
Thermocouple	-30 to 550?	Wall temperature	0.1?	40 -50?	0.264
Flow meter	0-0.16 gr/s	Volumetric flow rate	0.001 gr/s	0.03 - 0.14 gr/s	0.678

These experiments are performed to obtain heat transfer coefficient and mean Nusselt number of nanofluid in a double pipe heat exchanger with perforated twisted tapes of different diameters. The heat transfer experiments were performed in the heat exchanger under the different conditions and then continued with the perforated twisted tape inserts. As shown, Reynolds number range was from 2500 to 21000. Experiments were performed without perforated twisted tapes under turbulent flow. Within full length of the tube, the flow was expanded thermally and hydrodynamically. To evaluate the experimental setup thermally, the obtained results for Nusselt number of the turbulent flow were compared and analyzed based on Gnielinski theory. Then, the perforated twisted tapes were fitted into the same tube and the experiments were continued. Fig. 4 shows the experimental Nusselt number for water versus Reynolds number. As seen from this figure, the experimental data are in good agreement with the theoretical model.

Although it can be said that Reynolds number range for Gnielinski theory divides the range considered into two parts, a good agreement between described correlations even outside their range is clear from Fig. 4. This is more apparent when the maximum error for Gnielinski equation is 5.68%. The obtained results for Nusselt number in different concentrations of nanofluid and different geometries of perforated twisted tapes are represented in Figs. 5-7. The difference between Nusselt number for different concentrations of nanofluid and different geometries of perforated twisted tapes increases with Reynolds number and decreases with decreasing it. This increase is maximum for the concentrations of 0.2 and 0.12% v/v. One reason for this difference in heat transfer at high Reynolds numbers is the high viscosity of nanofluid. In general, the fluid containing rod-shaped particles, due to severe reactions, has high viscosity and high density in shear flow. Particle concentration and movement of particles in the flow are two factors affecting the heat transfer.

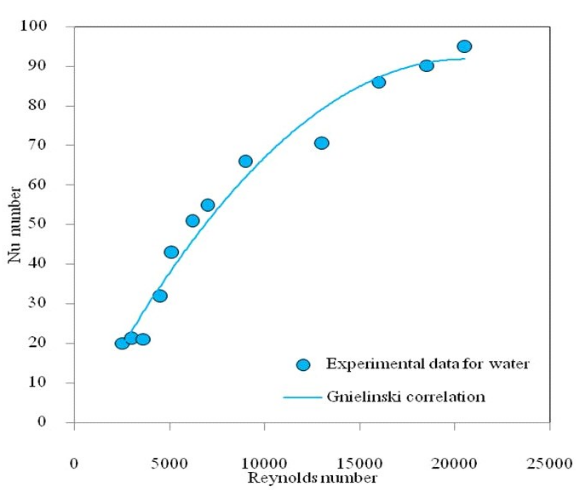

Fig. 4: Comparison between the measured Nusselt number and gnielinski correlation for distilled water.

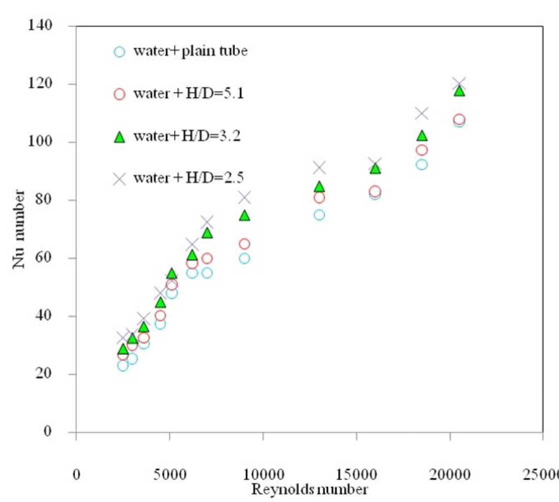

Fig. 5: Nusselt number for different geometries of twisted tapes for punching the flow of water in the heat exchanger.

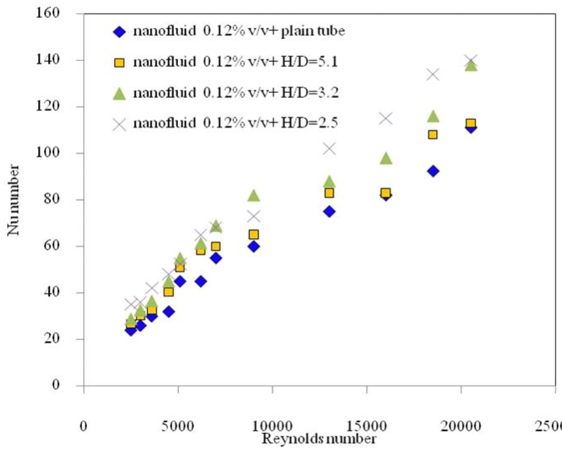

Fig. 6: Nusselt number for different geometries of the twisted tapes, Nanofluids flow heat exchanger with volume concentration of 0.12%.

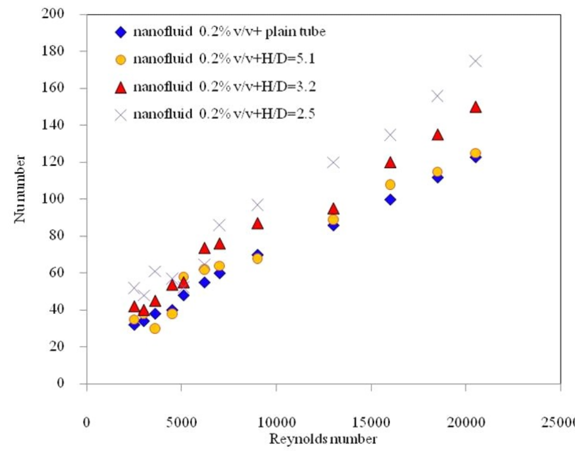

Fig. 7: Nusselt number for different geometries for the twisted tapes, Nanofluids flow heat exchanger with volume concentration of 0.2%.

The transitional move is assumed the main mechanism for increasing the thermal conductivity of nanoparticles. The migration of finer particles increases thermal conductivity coffecient of nanofluids more than the coarse particles.

Twisted-tape inserts increase the heat transfer coefficients with relatively small increase in the pressure drop. They are known to be one of the earliest swirl flow devices employed in the single phase heat transfer processes. Because of the design and convenience of application, they have been widely used over decades to generate the swirl flow in the fluid. Size of the new heat exchanger can be reduced significantly by using twisted tapes in the new heat exchanger for a specified heat load. Thus, it provides an economic advantage over the fixed cost of the equipment. Twisted tapes can be also used for retrofitting purpose. They can increase the heat duties of the existing double tube heat exchangers. Twisted tapes with multitude bundles are easy to fit and remove and thus enable tube side cleaning in fouling situations. Inserts such as twisted tape, wire coils, ribs, and dimples mainly obstruct the flow and separate the primary flow from the secondary flows. This causes the enhancement of the heat transfer in the tube flow. Inserts reduce the effective flow area, thereby increasing the flow velocity.

In the double pipe heat exchangers, due to the high thermal efficiency of nanofluid compared to water, nanofluids can be used in different heating and cooling processes. Therefore, water flow rate can be decreased by increasing the heat transfer. Probably, this solves the used water and produced wastewater problems in the big industries such as oil and petroleum. In addition, the dimensions of the heat exchangers can be reduced by increasing the heat transfer. Addition of the nanoparticles into the fluids increases the heat transfer coefficient and thereby decreases the operational and production costs. Efficiency increases with the heat transfer. So the power needed for pumping and heat transfer area decreases. This, in turn, reduces the fixed costs. Also, increasing efficiency controls the transferred heat better and decreases the destructive impacts of energy on the environment.

CONCLUSION

In the present study, Fe_2O_3 nanofluid was used to investigate the heat transfer. After all, it can be said that:

Heat transfer (regardless of water or nanofluid, with and without the twisted tape) greatly increases with Reynolds number. Addition of the nanoparticles to the base fluid improves the heat transfer partly. This is evident when the average variation of Nusselt number for the heat exchanger without twisted tape inserts and at the concentrations of 0.12 and 0.2%v/v is 3.56, 5.3 and 8.10% respectively.

Simultaneous application of nanoparticles and perforated twisted tape inserts enhances the heat transfer due to the mixture of the fluid along with the using nanoparticles and twisted tape.

Heat transfer increases with a decrease in the twist ratio. The mean variation of Nusselt number in for twist ratios of 5.1, 3.2 and 2.5 is 5.9, 22.1 and 38.9% respectively.

Simultaneous application of nanofluid and twisted tape inserts increases the heat transfer.

Maximum increase in Nusselt number is achieved for the 0.2%v/v Fe_2O_3 nanofluid and twist ratio of 2.5. This increase occurs in Reynolds number of 2500, which is 132.2% compared to the experiment performed with water and without the twisted tape inserts.

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