EXPERIMENTAL INVESTIGATION OF ENHANCING INFLUENCE OF Al_2O_3 NANOPARTICLES ON THE CONVECTIVE HEAT TRANSFER IN A TUBE EQUIPPED WITH TWISTED TAPE INSERTS

by

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In this study, the heat transfer rate of Al_2O_3-water nanofluid-flow in a tube with twisted tape inserts is experimentally examined. The twisted tape insert can cause a swirling region in the fluid-flow which will enhance the heat transfer coefficient. On the other hand, the twisted tape inserts have a great influence on the pressure drop. It is known that nanoparticles and twisted tape inserts may both increase the heat transfer rate of the fluid. Thus, in this study, it is tried to demonstrate the combined effect of both phenomena, experimentally. In this way, the Al_2O_3-water nanofluid is used as the carrier liquid in a tube equipped with various twisted tape inserts of different pitches and twist ratios. Moreover, the influence of nanoparticles' volume fractions was studied in the turbulent regime of a tube with the constant temperature boundary condition.

Key words: *heat transfer enhancement, experimental study, nanofluid, convective heat transfer, twisted tape inserts*

Introduction

Carrier fluids (coolant liquids) are used vastly to avoid overheating of the system or simply to increase the heat transfer rate of many different pieces of equipment such as electronic devices, heat exchangers, and in various parts of the transportation industry. The common coolants like water, ethylene glycol or engine oil possess relatively low thermal conductivities regarding conducting solid particles. Thereby, to increase the thermal characteristics of these popular coolants many researchers attempted to apply tiny solid particles to the carrier liquid. In early investigations, particles of the scale of millimeter or micrometer were applied. However, these investigations were subjected to many problems like quick sedimentation and instability difficulties. After the emersion of nanotechnology and particles with the scale of nanometers, a new generation of stable and reliable carrier liquids named as nanofluids were introduced.

In order to increase the heat transfer rate of a system, different active and passive methods are used. The active methods are those which enhance the heat transfer via external sources like the vibration of the surface, oscillation of the fluid-flow, and fluid injection. On the other hand, passive methods enhance the heat transfer rate constantly without any external

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sources. The tube internal facilities like twisted tape inserts, fins, helical tubes and of course the nanoparticles are some examples of the passive methods for heat transfer enhancement.

Some very popular passive methods like fins have their own restrictions, for instance in aerospace industries, in which the main goal is to reduce the system size and weight, fins are not so applicable. Hence, in order to be able to answer the industrial over-growing demand for strong and more efficient heat exchangers, other solutions to this problem must be found. The twisted tape inserts can enhance the heat transfer rate by generating a circulating flow regime in the system. That will result in an increase in both, the heat transfer rate and inevitably the pressure drop of the system. However, the type and twisting methods of the inserts are extremely influential in their efficiency and pressure drop.

Up to this day, many researchers have tried to study the convective heat transfer and pressure drop in a channel with the circular cross-section. Pak and Cho [1] were considered to be among the first authors who tried to investigate the forced convection of nanofluids. They examined two different nanofluids of Al_2O_3-water and TiO_2-water in a tube with the constant heat flux boundary condition on the wall. According to their findings, an addition of 2.78% of Al_2O_3 nanoparticles can boost the heat transfer rate by 75% in the turbulent flow regime. Wen and Ding [2] in an experimental study investigated the laminar convection of nanofluids in a tube with a length of 970 mm and a diameter of 4.5 mm. They stated that the heat transfer coefficient of the nanofluid is greater than the pure liquid, and it increases with an augmentation of Reynolds number or particle volume fraction.

Hwang *et al.* [3], investigated the pressure drop and heat transfer characteristics of Al_2O_3-water nanofluid in a homogeneously heated tube. The flow was considered to be fully developed and laminar. Their results indicated that the nanofluid friction factor may be well calculated from the Darcy equation for the single-phase liquids. Moreover, they claimed that for 0.3% of particle volume fraction, an 8% increment in the heat transfer rate was observed. This enhancement could not be predicted from Shah [4] correlation. Li and Xuan [5] conducted a study to estimate the heat transfer coefficient and the friction factor of Cu-water nanofluid with particle sizes less than 100 mm. their results revealed that in both laminar and turbulent regimes, there is no considerable pressure drop due to the addition of nanoparticles, which means nanofluids with smaller particle sizes will not increase the consumed pumping power, significantly.

Fotukian and Naser [6] also studied the pressure drop and heat transfer rate of Al_2O_3-water nanofluid in a circular tube. Their experiments were carried out in a tube with a constant wall temperature and for the turbulent regime and particle volume fractions less than 0.3%. They observed that the pressure drop in the system will not necessarily increase due to the increase of particle volume fraction and as a matter of fact the highest pressure drop was 20% which was observed at a particle volume fraction of 0.03% for the nanofluid.

According to more recent researches, metal oxides can provide higher thermal conductivities for the nanofluid and hence these nanoparticles gain large popularity among researchers. Zeinali Heris [7] used two different nanofluids of Al_2O_3-water and Cu-water in a circular tube with constant temperature on the walls. According to their findings for both nanofluids, the heat transfer rate would augment with the increment of Peclet number and particle concentration. In this study, they tried to compare the influence of various metal oxide nanoparticles on the system. One year later, Zeinal Heris *et al.* [8] in another experimental investigation tried to investigate the forced convection of Al_2O_3-water nanofluid in the same circular tube with a constant-temperature wall. The result of this study also indicated that in the laminar regime, the heat transfer coefficient will increase with the nanoparticle concentration.

Djordjević *et al.* [9] presented a numerical study on the heat transfer of a spiral coil tube in the laminar, transitional, and turbulent flow regimes. Their results revealed that the secondary flows are extremely significant in the flow and heat transfer rate of the spiral tube. Al-Sammarraie and Vafai [10] examined forced convective heat transfer and fluid-flow characteristics through a convergent pipe. They found a remarkable heat transfer augmentation. Maghrebi *et al.* [11] investigated the influence of nanoparticle volume fraction on the heat transfer rate of a 2-D incompressible plane forced jet flow. They used Al_2O_3 and CuO nanoparticles in water and according to their results the temperature amplitude, velocity-time histories, and the turbulent intensities are decreased by the increment of particle volume fraction. The transport of Al_2O_3-water nanofluid through concave/convex pipes has been recently studied by Al-Sammarraie *et al.* [12]. They reported a maximum heat transfer enhancement of 41%.

In another numerical study, Soltanipour *et al.* [13] investigated the heat transfer enhancement of γ-AL_2O_3-water nanofluid in a curved duct. They reported a remarkable heat transfer enhancement for the nanofluid with respect to the smooth tube and pure water. Hejazi *et al.* [14] used twisted tape inserts in the condensation process of R-134a vapor and propose an experimental correlation for calculating the pressure drop. Moreover, Akhavan-Behabadi *et al.* [15] tried to investigate the heat transfer increment due to twisted tape inserts in R-134a condensers. Lim *et al.* [16] studied the heat transfer augmentation of laminar flow in heat exchangers equipped with twisted tape inserts and they found that the induced swirling flow will increase the heat transfer rate of the system. In their work, several various operational factors are taken into account. Salimpour and Yarmohammadi [17] studied the impact of inserting twisted tapes on the pressure drop of the R-400A condensation process. They further studied the heat transfer enhancement induced by twisted tapes in those condensers [18]. Yarmohammadi *et al.* [19] performed a multi-objective optimization of R-404A vapor condensation in swirling flow using genetic algorithms.

Man *et al.* [20] also introduced a new type of twisted tape insert and observed its influence on the heat transfer rate and pressure drop of the system. Garg *et al.* [21] presented a review paper on novel applications of twisted tape inserts and provided a fine literature review for future investigations. Zheng *et al.* [22] in a numerical study investigated the influence of dimpled twisted tape in a circular tube and presented the heat transfer characteristics of the Al_2O_3-water nanofluid for different nanoparticle diameters and concentrations. Bhuiya *et al.* [23] examined the heat transfer increment in a circular tube with a perforated double counter twisted tape insert in the turbulent regime, experimentally. Also in another study, Nanan *et al.* [24] investigated the flow and heat transfer mechanisms in a heat exchanger. The device is equipped with twisted cross-baffle tabulators. They stated that these baffles can enhance the heat transfer rate by up to 70%.

Although the influence of nanoparticles on the flow regime is vastly studied in the open literature [25-27], but to the best of the authors' knowledge, the influence of twisted tape inserts on heat transfer rate of Al_2O_3-water nanofluid in a circular tube was not experimentally investigated so far, thus here we try to study this issue for various Reynolds numbers and different types of twisted tape inserts.

The test apparatus

In order to study the convective heat transfer characteristics of the nanofluid in a constant temperature tube with a twisted tape insert, an experimental device was built and installed according to fig. 1 schematic design. In this device, to maintain the temperature of the tube wall, a vapor bath was applied.

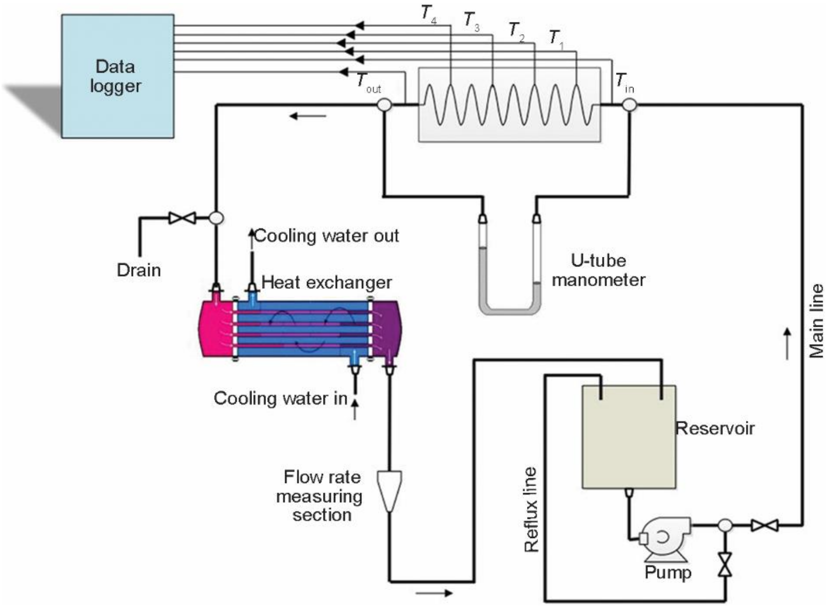

Figure 1. The schematic diagram of the test apparatus

As can be seen, the test apparatus has a main circulating line for the nanofluid which consists of various subdivisions such as reservoir, pump, test section, cooling heat exchanger, flow rate measuring system and some thermocouples to record the temperature data in the test section. The twisted tape insert is 1 meter long and 14 mm wide. The width of the inserts is considered to be 1 mm less than the tube inner diameter to simplify the placement of the inserts. The tapes thickness is 0.7 mm and they are made of steel. In this study, different twisted tape inserts with pitches of 4, 7, 10, and 14 cm were applied. To calculate the twist ratio, y, twisted tape pitch, H, is divided by the twisted tape width, D_i; that is $y = H/D_i$. The tapes and their effective parameters are demonstrated in fig. 2.

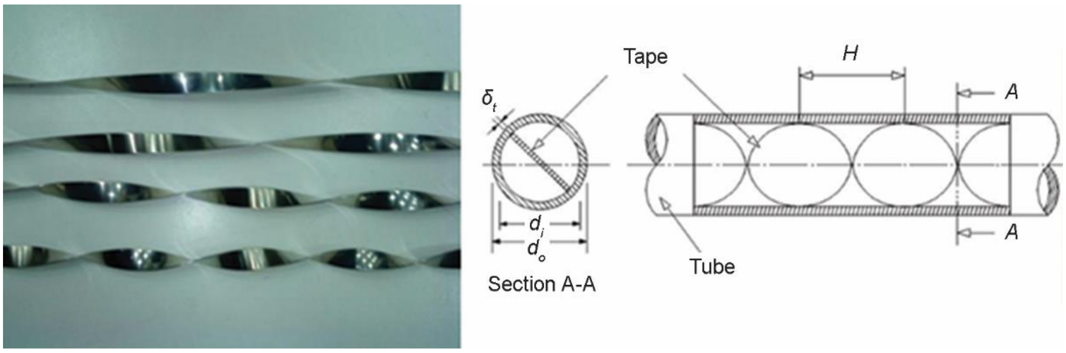

Figure 2. The applied twisted tape inserts

Experimental procedure

At first, half of the vapor bath is filled with water and the intake reservoir is filled with nanofluid. Then the thermal elements are switched on, and we will wait until the vapor around the test section provides a steady constant temperature on the tube wall. After the initial heating process and when the water is boiling, in order to achieve a steady-state condition, we can switch off all of the thermal elements but two. Then the nanofluid will be pumped into the test section and the fluid-flow can be regulated via two valves, which are devised one at the outlet and one on the mainline before the test section. Finally, the coolant water which is provided from the city water will flow in the heat exchanger and all of the preliminary preparations are done for the experiment.

Afterward, the system is left to work on its own until the steady-state condition is achieved. The steady-state condition is achieved when the variation of flow and thermal characteristics of the system is negligible. After several tests, it was observed that the steady-state condition is valid after 20 minutes of the flow rate regulation in the system. The experiments were carried out for pure water and two different nanofluids with particle concentrations of 0.5% and 0.1%. Moreover, the flow rates of the system, as well as the twisted tape insert types, were varied to observe their influence on the heat transfer rate. After finishing the measurements of each cycle of the experiment, the nanofluid is exerted from the test section and to assure that no nanoparticle is left in the section the tube is evacuated by water. Finally, the tube would be dried off by the high-pressure wind.

Before starting the experiments, a sonicator was used to prepare stable nanofluid. Due to this process, no visible sedimentation was observed. Also, according to the nature of the swirling flow, the chance of sedimentation was low.

Calculation process and validation

To demonstrate the equation derivation and ensure the reliability of the results, at first, the data reduction equations and then a comparison of test apparatus results with previous studies on a circular straight tube filled with pure water will be presented, here. Afterward, the results regarded to the heat transfer rate of the tubes with twisted tape inserts would be displayed and discussed. Moreover, a comparative study on the tubes' heat transfer for various nanofluids' concentrations is carried out.

Data reduction equations

In order to estimate the heat transfer coefficient and the Nusselt number of the tube, the recorded data including the mean temperatures of the inlet and outlet, the surface temperature of various locations of the tube, volumetric flow rate, and thermophysical characteristics of the fluid are used.

As can be seen, the tubes are long enough to assume that the whole test section was in a fully-developed regime. The convective heat transfer coefficient of the tube wall may be computed from:

$$\dot{m}c_p(T_{\text{out}} - T_{\text{in}}) = hA_{\text{p}}\Delta T_{\text{LMTD}} \tag{1}$$

where $\dot{m}$ is the mass-flow rate, A – the peripheral area of the tube (the heat transfer area), and T_{in}, T_{out} – the fluid bulk temperatures at the inlet and outlet of the test section, respectively. Also, ΔT_{LMTD} is the logarithmic mean temperature difference, which is calculated from:

$$\Delta T_{\text{LMTD}} = \frac{\Delta T_{\text{in}} - \Delta T_{\text{out}}}{\ln \dfrac{\Delta T_{\text{in}}}{\Delta T_{\text{out}}}} \tag{2}$$

where $\Delta T_{LMTD} = T_s - T_{out}$, with T_s being the tube surface temperature. Since the tubes are very thin, and the conductive heat transfer resistance of the tube walls can be neglected, the following correlation is valid for calculating the heat transfer coefficient, h:

$$h = \frac{\rho c_p Q(T_{out} - T_{in})}{\pi d_i l_h \Delta T_{LMTD}} \tag{3}$$

Thus, the average Nusselt number of the tube wall is obtained from:

$$\mathrm{Nu} = \frac{hd}{k} \tag{4}$$

where d is the inner diameter of the tube and for non-circular tubes, it can be replaced by the hydraulic diameter and k – the conductive heat transfer coefficient of the fluid.

In order to compare the experimental results of the straight tube with the Sieder and Tate [28] and Gnielinsky [29] correlations, the mean Nusselt number of the pure liquid is calculated in the laminar and turbulent regimes. Moreover, the Dittus-Boelter [30] equation for the Nusselt number of turbulent flow is compared to the recorded data.

– The Seider-Tate correlation for laminar flow:

$$\mathrm{Nu} = 1.86\left(\mathrm{Re}\,\mathrm{Pr}\frac{d}{l}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} \tag{1}$$

– The Gnielinsky correlation for turbulent flow:

$$\mathrm{Nu} = \frac{\frac{f}{8}(\mathrm{Re} - 1000)\mathrm{Pr}}{1 + 12.7\left(\frac{f}{8}\right)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \tag{2}$$

– The Dittus-Boelter correlation for turbulent flow:

$$\mathrm{Nu} = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{3}$$

Equation 5 is used for the range of $100 < \mathrm{Re} < 2100$, while eq. (6) is applicable to the flow with $3000 \leq \mathrm{Re}\ 5\times10^6$ and $0.2 < \mathrm{Pr} < 5$ condition. Equation (7) is also valid for $\mathrm{Re} > 10000$ and $0.16 < \mathrm{Pr} < 6$. All of these correlations are proposed for the fully-developed regime. In eq. (6), f is the friction factor. Also, the s and b indices in eq. (5) refer to the fluid average temperature and the tube wall temperature, respectively. It is worth mentioning that eq. (6) may be used for both the constant temperature and constant heat flux walls.

Test apparatus validation

Before carrying out the main experiments on the nanofluid and tubes with twisted tape inserts, in order to examine the experimental device precision and reliability, the Nusselt number of the plain tube (tube without any insert) with distilled water was tested against the three existing correlations. Figure 3 presents the validation results in which, the experimental results of the present study were found to be in excellent agreement with those of Seider-Tate, Dittus-Boelter, and Gnilinski.

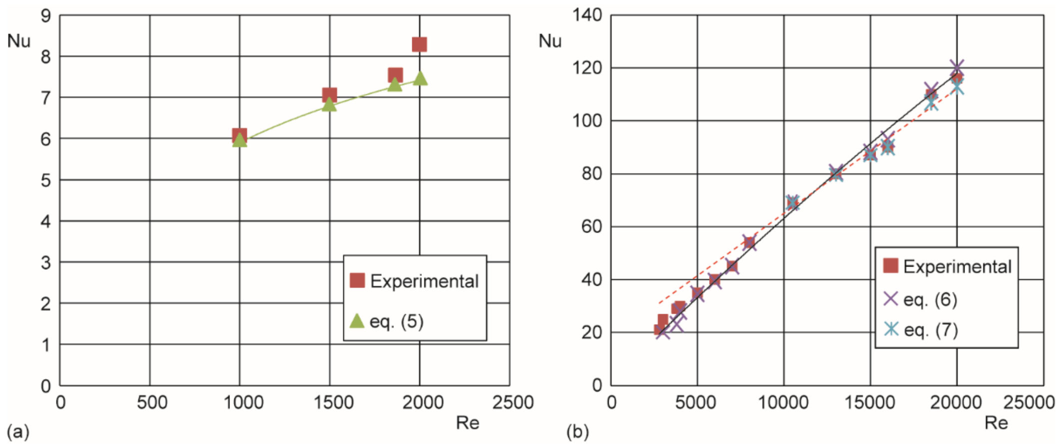

Figure 3. The Nusselt number variation of the plain tube *vs.* Reynolds number for pure water flow, (a) laminar flow regime, (b) turbulent flow regime.

Results and discussion

In what follows, the influence of twisted tape inserts on the heat transfer rate of the tubes is investigated and the impacts of effective parameters like Reynolds number, Prandtl number, twist ratio and nanoparticles volume fraction are discussed.

Figure 4 illustrates the mean Nusselt number of the tubes with twisted tape inserts for distilled water. It is clear that with the increment of Reynolds number and decrement of the twist ratio of the inserts, the mean Nusselt number of the tube is increased. Reynolds augmentation means an increment in the system velocity and stream function which provides a higher heat transfer coefficient and thus offers a better heat transfer rate. On the other hand, the twist ratio plays a significant role in the heat transfer of the tube and the tube which is not equipped by the inserts possesses the lowest Nusselt number. As the inserts' twist ratios are increased, the twisted tape shape approaches to a straight blade which induces less turbulence into the fluid-flow, and as a result, the Nusselt number decreases. This difference between the heat transfer coefficients of a twisted tape inserted tube and the corresponding values of a plain tube are more remarkable at higher values of Reynolds number. It is interesting that the percentage of heat transfer increments at all the Reynolds number ranges is approximately the same. However, in higher Reynolds numbers, the absolute values of the changes are more distinguishable for higher fluid velocities. For instance, at Re = 20000, the maximum Nusselt number is achieved at the tube possessing the tape with a twist ratio of $y = 4$, where Nusselt number is enhanced from 140.4 (for the plain tube) to 200.3 (for the tube with a twisted tape of $y = 4$).

Figure 5 demonstrates the results of the average Nusselt number of tubes with twisted tape inserts for the Al_2O_3-water nanofluid-flow with 0.1% of nanoparticle concentration. In this figure, also it is obvious that the Nusselt number augments with Reynolds number and nanoparticle concentration increment. The maximum Nusselt number is observed at Re = 20000. Moreover, the twist ratio of 4 which is the lowest one, presents the highest heat transfer coefficients. Also, it goes without saying that the nanoparticles' concentration augmentation boosts the heat transfer coefficients.

Figure 6 presents the Nusselt number results of the tube with twisted tape inserts for nanofluid with 0.5% of Al_2O_3 volume fraction. It is obvious from the graph that the Nusselt number increases with Reynolds number. In this case, also the highest Nusselt number is

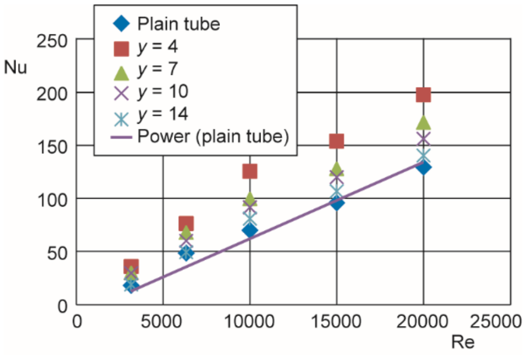

Figure 4. Variation of Nusselt number of water against Reynolds number for different twist ratios

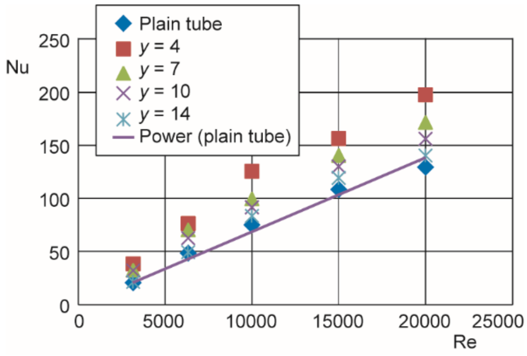

Figure 5. The Nusselt number variation of 0.1% nanofluid *vs.* Reynolds number for various twist ratios of the insert

observed at Re = 20000 and y = 4. Moreover, the particle concentration increase augments the heat transfer rate. The highest Nusselt number which is achieved in this study is 220.

Afterward, the addition of Al_2O_3 nanoparticles to pure water (as a base fluid) was investigated and the enhancing influence of these nanoparticles in the heat transfer rate of the tubes with twisted tape inserts was studied, more closely. Hence, the carrier liquid with three different Al_2O_3 concentrations of 0, 0.1, and 0.5% was applied into the system and the results were presented for various Reynolds numbers and twist ratios.

Results in fig. 7 are the variations of Nusselt *vs.* Reynolds numbers for pure water as well as nanofluids with concentrations of 0.1 and 0.5% for twist ratio of 4. As is evident from this figure, adding nanoparticles enhances heat transfer at all Reynolds numbers. However, the heat transfer enhancement at low Reynolds number is less remarkable due to the small absolute values of heat transfer rate at the low Reynolds number regimes.

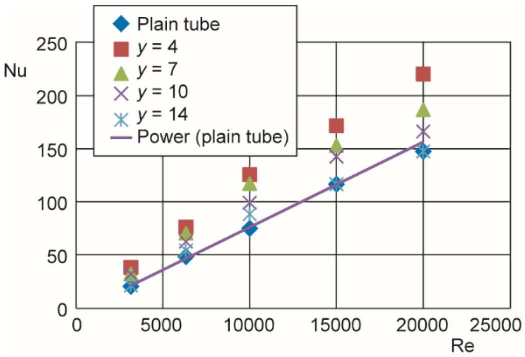

Figure 6. The Nusselt number variation of 0.5% nanofluid *vs.* Reynolds number for various twist ratios of the insert

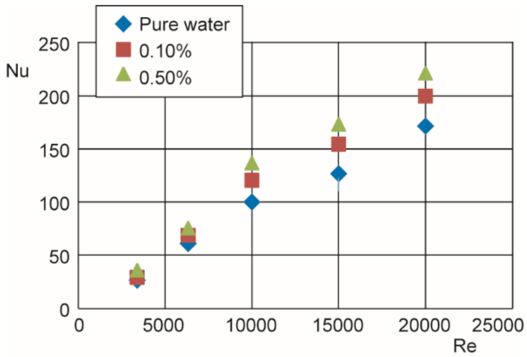

Figure 7. The Nusselt number number variation *vs.* Reynolds number for twist ratio of 4

Figure 8 represents the variation of Nusselt number with Reynolds number for the tube containing the twisted tape insert with y = 7. Comparing this figure with fig. 7, it is seen that at lower twist ratios, the heat transfer coefficients are more vigorous, expectedly. The trend of variations of both figures are similar; however, the superiority of higher volume ratios of nanoparticles is more distinguishable at higher twist ratios.

Figure 9 illustrates similar presentations for tubes with a twist ratio of 10. Here again, the heat transfer coefficient increase is more visible at higher Reynolds numbers. It is considered to be due to higher values of heat transfer coefficients at higher Reynolds numbers. Meanwhile, the relative increase in Nusselt numbers is more pronounced at lower Reynolds numbers. For example, for nanoparticle concentration of 0.5%, the heat transfer coefficient increment is 65% and 23%, at Reynolds numbers of 3000 and 20000, but the Nusselt number absolute increment is 15.0 and 32.1, respectively.

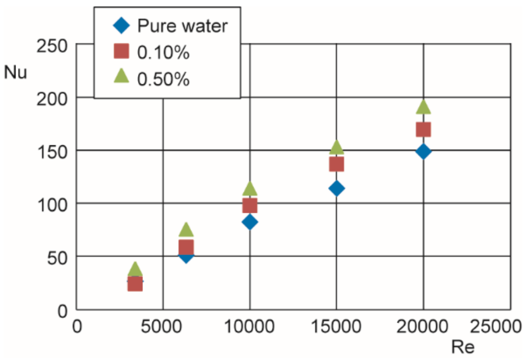

Figure 8. The Nusselt number variations *vs.* Reynolds number for twist ratio of 7

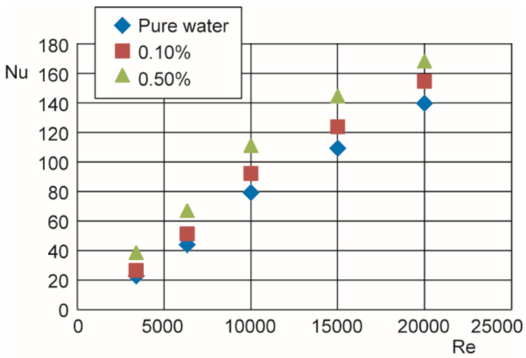

Figure 9. The Nusselt number variations *vs.* Reynolds number for twist ratio of 10

Figure 10 contains the graphs for Nusselt number variations at different Reynolds numbers for distilled water and two concentrations of nanofluid. Comparing this figure to figs. 7-9, a similar trend is observed. However, it is clear that lower twist ratios contribute less to heat transfer augmentation.

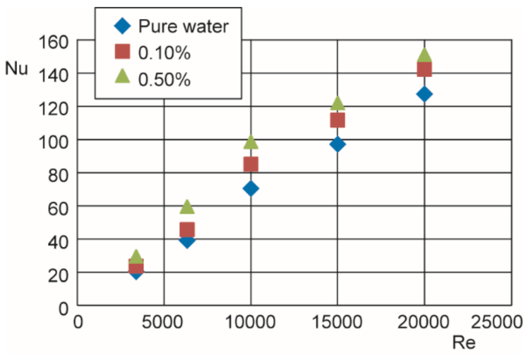

Figure 10. The Nusselt number variations *vs.* Reynolds number for twist ratio of 14

The increment of heat transfer coefficient in nanofluids can be related to the boost of thermal conductivity of nanofluid with respect to the pure water. Moreover, the Brownian motion of nanoparticles and the reduction of the boundary-layer thickness may increase the heat transfer rate as well.

Finally, based on the acquired data and invoking the least square regression analysis, the following correlation, eq. (8), has been developed to predict the Nusselt number of twisted tape inserted tubes.

$$\mathrm{Nu} = 0.028\mathrm{Re}^{0.873}\mathrm{Pr}^{038}y^{-0.4}(1 + \varphi)^{0.4} \tag{8}$$

It is interesting to see that the influence of the Reynold number on the mean Nusselt number of the tubes is much more pronounced than other parameters. While the twist ratio of the inserts provides the same order of significance than the Prandtl number of the flow.

Conclusion

An experimental study was performed to investigate the effect of introducing twisted tape inserts on the heat transfer coefficients of pure water and Al_2O_3-water nanofluids flow. The main findings of these experiments are as follows.

- Nusselt number increases with the increment of Reynolds number.
- At the same Reynolds numbers, higher Nusselt numbers are observed at lower twist ratios of the inserts.
- Nanoparticles increase heat transfer coefficients. This increment is due to the enhanced thermal conductivity of the nanofluid and the Brownian motion of nanoparticles.
- The enhancing effect of nanoparticles is more remarkable at lower Reynolds numbers. While its magnitude is much greater in higher Reynolds numbers.
- The highest Nusselt number achieved in this study was 220 for the twist ratio of 4 and the nanoparticles volume fraction of 0.5%.

Nomenclature

A_p – peripheral area of the tube, [m^2]
c_p – specific heat, [Wkg^{-1}K^{-1}]
d – diameter, [m]
f – friction factor
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
l_h – length, [m]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
Nu – Nusselt number
Pe – Peclet Number (= Re Pr)
Pr – Prandtl number
Q – volumetric flow rate, [m^{3}s^{-1}]
Re – Reynolds number
T – temperature, [K]
T_s – the tube surface temperature, [K]
ΔT_{LMTD} – logarithmic mean temperature difference, [K]
y – twist ratio

Greek symbols

μ – viscosity, [Pa·s]
ρ – density, [kgm^{-3}]
φ – nanoparticle concentration

Subscripts

LMTD – logarithmic mean temperature difference
i – inner
in – inlet condition
o – outer
out – outlet condition

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