International Journal of Energy Applications and Technologies
Available online at
http://dergipark.gov.tr/ijeat/issue/30402
Research Article
Vol. 4
Issue 2
pp. 73-86
Year
2017

EXPERIMENTAL INVESTIGATION OF LAMINAR CONVECTIVE HEAT TRANSFER AND PRESSURE DROP OF AL_2O_3/WATER NANOFLUID IN CIRCULAR TUBE FITTED WITH TWISTED TAPE INSERT

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Abstract

In the present study, convective heat transfer coefficient and pressure drop of Al_2O_3/water nanofluid in laminar flow regime under constant inlet temperature conditions inside a circular finned-tube with a typical twisted tape was experimentally investigated. For this purpose, Al_2O_3 nanoparticles of 20 nm size were synthesized, characterized and dispersed in distilled water to formulate Al_2O_3/water nanofluid containing 3 and 5% volume concentration of nanoparticles. The thermophysical properties like thermal conductivity and viscosity of Al_2O_3 nanofluid is determined through experiments at different volume concentrations and validated. Experiments are conducted in the Reynolds number range of (678–2033) with tape of twist ratio (H/D = 1.85). The effect of different volume concentrations on convective heat transfer coefficient and friction factor was studied. The results emphasize that increasing of particle volume concentration leads to enhance convective heat transfer coefficient. Measurements show the average heat transfer coefficient enhanced about 21-51 % with 5% volume concentration and increased about 19-38 % with 3% volume concentration compared to distilled water. In addition, the average ratio of (f_{nf}/f_{bf}) was about 2.3 for 5% volume concentration. Therefore, there is a significant increase in friction factor for nanofluids. A generalized regression equation is developed for the estimation of Nusselt number and friction factor valid for nanofluid.

Key words: *Typical twisted tape, Heat transfer, Nusselt number.*

Paper submitted: March 04, 2017

Paper accepted: July 13, 2017

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1. Introduction

Traditionally, heat transfer coefficients have been determined both numerically and experimentally over a wide range of Reynolds and Prandtl numbers involving single phase fluids. The performance of thermal equipment operating with fluids such as water, oils and ethylene glycol, has reached the limit. In the last decades, many efforts have been made to produce more efficient heat exchangers in order to conserve energy. These efforts deal with improving the heat transfer rate by means of extended surfaces, mini-channels and microchannels. In addition, many studies have been carried out on thermal properties of suspensions of solid particles in conventional heat transfer fluids to enhance their poor thermal performance.

Heat transfer enhancements can be achieved through using swirl flow devices, which can be classified into two kinds: the first is the continues swirl flow in which the swirling motion persists over the whole length of tube; so the heat transfer coefficient and pressure drop keep constant with the axial distance such as twisted-tape inserting. The other kind of swirl flow is the decaying swirl flow in which the swirl is generated at the entrance of the tube and decays along the flow path leads to decreasing the heat transfer coefficient and pressure drop with the axial distance for example the radial guide vane swirl generator, the tangential flow injection device and the snail swirl generator.

The way to improve heat transfer performance is referred to as heat transfer enhancement or augmentation. Nowadays, a significant number of thermal engineering researchers are seeking for new enhancing heat transfer methods between surfaces and the surrounding fluid. Due to this fact, Bergles, Adrian and Khaled [1, 2, 3] classified the mechanisms of enhancing heat transfer as active or passive methods. Those which require external power to maintain the enhancement mechanism are named active methods. On the other hand, the passive enhancement methods are those which do not require external power to sustain the enhancements' characteristics. In addition, a hybrid technique which includes two or more from each of passive and active technique.

Among the various passive techniques which are effective to improve the thermohydraulic behavior in a single-phase flow, the insert devices like twisted tapes are most frequently used in engineering applications to update an existing heat exchanger. This is mainly due to its low cost, easy installation/removal, reliability and durability. In fact many researchers investigated heat transfer enhancement concept in which swirl and nanofluid were introduced in the flow. In the laminar flow, heat transfer takes place mainly by conduction and molecular diffusion as there is no cross mixing of the fluid. The heat transfer coefficients in laminar flow were generally low. So, for a given heat transfer rate, larger heat transfer whereas will have to be provided as compared with turbulent flow heat transfer situations. Use of twisted tapes for augmentation can be dated back to as early as up to the end of nineteenth century. One of the early researches on heat transfer enhancement by means of twisted tapes was carried out by Whitman [4].

Khwanchit and Smith [5] carried out experimental investigation to evaluate the heat transfer enhancement in circular tube equipped with modified twisted tape with alternate axis (TA). CuO nanoparticle and base fluid water was used in experimental with volume fraction from 0.3% to 0.7% and the range of Reynolds number varied from 830 to 1990. The experimental results found enhance in Nusselt number reached to 12.8% as compared with plain tube. Over the range investigated, the maximum thermal performance factor of 5.53 is found with the simultaneous employment of the CuO/water nanofluid at 0.7% volume and the (TA) at Reynolds number of 1990. In addition, the empirical correlations for heat transfer coefficient, friction factor and thermal performance factor are also developed and reported.

Esmaeilzadeh et al. [6] carried out an experimental study to investigate heat transfer and friction factor characteristics of γ-Al_2O_3/water nanofluid through circular tube with twisted tape inserts with various thicknesses at constant heat flux. In this work, γ-Al_2O_3/water nanofluids with two volume concentrations of 0.5% and 1% were used as the working fluid. The twist ratio of twisted tape remained constant at 3.21, while the thicknesses were changed through three values of 0.5 mm, 1 mm and 2 mm. Results indicated that twisted tape inserts enhanced the average convective heat transfer coefficient, and also more the thickness of twisted tape is more the enhancement of convective heat transfer coefficient is. Also, the highest enhancement was achieved at maximum volume concentration.

Naik et al. [7] investigated experimentally convective heat transfer and friction factor characteristics of water/propylene glycol (70:30% by volume) based CuO nanofluids flowing in a plain tube under constant heat flux boundary condition. Experiments were conducted using CuO nanofluids with 0.025, 0.1 and 0.5% volume concentration and Reynolds numbers range (1000-10000) and considerable heat transfer enhancement in CuO nanofluids is observed. The effect of twisted tape with twist ratios range of (0 to 15) on nanofluids was studied and further heat transfer augmentation was noticed. The experimental results show that the convective heat transfer coefficient increased up to 27.95% in the 0.5% CuO nanofluid in plain tube and with a twisted ratio = 5 it is further increased to 76.06% over the base fluid.

Sundar and Sharma [8] experimentally investigated the fully developed laminar convective heat transfer and friction factor characteristics of different volume concentrations of Al2O3 nanofluid in a plain tube and fitted with different twist ratios of twisted tape inserts. Experiments were conducted with water and nanofluid in the range of $700 < Re < 2200$, particle volume concentration of $0< \varphi < 0.5\%$, and twisted tape twist ratios of $0 < H/D < 15$. The nanofluid heat transfer coefficient is high compared to water and further heat transfer enhancement is observed with twisted tape inserts. The pressure drop increases slightly with the inserts, but is comparatively negligible. A generalized regression equation is developed based on the experimental data for the estimation of the Nusselt number and friction factor for water and nanofluid in a plain tube and with twisted tape inserts.

Suresh et al. [9, 10] conducted a Comparison of thermal performance of helical screw tape inserts with Al_2O_3/water and CuO/water nanofluids through a straight circular duct with constant heat flux boundary condition under a laminar and transition flow. The thermal performance factor of helical screw tape inserts using CuO/water nanofluid was found to be higher when compared with the corresponding value using Al_2O_3/water. The higher enhancement shown by CuO/water nanofluid compared to Al_2O_3/water nanofluid was due to the combined effect of greater surface area to volume ratio and thermal conductivity of CuO nanoparticles compared to Al_2O_3 particles.

Salman et al. [11] presented a comparison study on thermal performance conic cut twist tape inserts in laminar flow of nanofluids through a constant heat fluxed tube. Three tape configurations , namely, quadrant cut twisted tape (QCT), parabolic half cut twisted tape (PCT), and triangular cut twisted (VCT) of twist ratio y = 2.93 and cut depth de = 0.5 cm were used with 1% and 2% volume concentration of SiO_2/ water and TiO_2/water nanofluids. Typical twist tape with twist ratio of y = 2.93 was used for comparison. The results show that the heat transfer was enhanced by increasing of Reynolds number and nanoparticles concentration of nanofluid. The results have also revealed that the use of twist tape enhanced the heat transfer coefficient significantly and maximum heat transfer enhancement was achieved by the presence of triangular cut twist tape insert with 2% volume concentration of SiO_2 nanofluid.

In the present paper, convective heat transfer coefficient and pressure drop of Al_2O_3/water nanofluid in laminar flow regime under constant inlet temperature conditions inside a circular tube were experimentally investigated in order to present a deep insight into the current issue.

2. Nanofluid Preparation

In the present study, γ-Al_2O_3 water based nanofluid contains Al_2O_3 nanoparticles with 99% purity procured from US Research Nanomaterials, Inc. is used in the experiments after appropriate dilution. Al_2O_3 nanoparticles of size 20 nm were mixed with distilled water and stabilizers and then sonicated continuously by ultrasonic vibrator generating ultrasonic pulses of 720W at 40 kHz for 10 h to break down agglomeration of the nanoparticles, prior to being used as the working fluid. The nanofluid was mixed with deionized water to prepare experimental nanofluid with concentrations 3 and 5%. Nanoparticles were found to be stable and the stability lasted over a two day.

In the calculations, the thermal conductivity, density and specific heat of Al_2O_3 nanoparticle are taken as 40 W/m.ºC, 3970 kg/m^3 and 765 J/kg.ºC respectively as given by [12]. About 3 L of the nanofluid is prepared to the desired concentration by dilution for the conduct of heat transfer experiments. The volume of distilled water ΔV to be added for attaining a desired concentration φ_2 is evaluated with Eq. (2) with the values of V_1 and φ_1 known a priory.

$$\varphi\,\% = \frac{(m_p/\rho_p)}{(m_p/\rho_p)+(m_{bf}/\rho_{bf})} \tag{1}$$

$$\Delta V = V_2 - V_1 = V_1\left(\frac{\varphi_1}{\varphi_2}-1\right) \tag{2}$$

The thermo physical properties of nanofluids such as viscosity, specific heat and density at each concentration are determined respectively with Brookfield digital viscometer model DV-E, Specific heat apparatus (ESD – 201) and digital balance. The measured data are presented in **Table 1**. The description of Brookfield Rheometer is presented by Namburu et al. [13]. The viscometer was calibrated using ethylene glycol and water (60:40 % by weight). The obtained readings were compared with data from the American Society of Heating, Refrigerating and Air – conditioning Engineers (ASHRAE) handbook [14]. It can be seen that the ASHRAE data and the experimental values match nicely (maximum difference of ± 2%) with temperature ranging from 20 °C to 55 °C, as shown below in **Fig. 1**. The thermal conductivity of nanofluid is estimated based on mixture relation given by Buongiorno [15]:

Table 1. The measured value of properties at 25°C.

	Distilled water	γ-Al_2O_3 (20 nm)/Distilled water	
φ (vol. %)	0 %	3 %	5 %
μ ($N.s/m^2$)	0.998×10^{-3}	1.26×10^{-3}	1.63×10^{-3}
Cp (kJ/kg.K)	4.183	3.808	3.601
ρ (kg/m^3)	995.7	1084.9	1142.01
k (W/m.K)	0.600	0.631	0.652

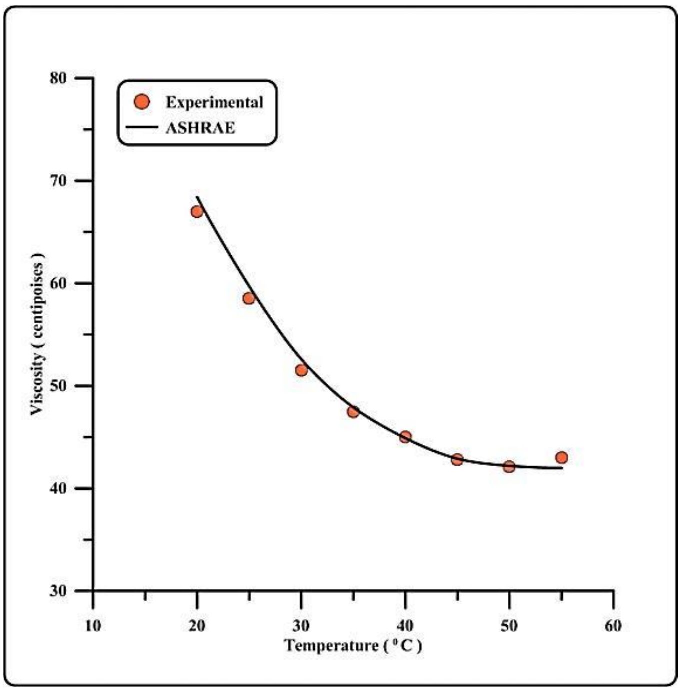

Fig. 1. Comparison of ASHRAE viscosity values and experimental data.

3. Experimental Apparatus and Procedure

The experimental setup for measuring the convective heat transfer and pressure drop characteristics is shown schematically in **Fig. 2**. It mainly consists of a test section, receiving tank in which working fluids are stored, heating system, thermometer, flow meter, pressure measurement system, and data logger system. The working fluids were circulated through the loop by using constant speed circulating pump of suitable capacity. The type of the pump [Marquis - 0.37 kW], China industry and the pump works with a constant voltage (220 V) and (2.5A) and (H_{max} = 30 m). The pump is able to operate with a fluid temperature up to 70 °C. The pump could deliver a maximum flow rate of 30 liters per minute. Nanofluids were driven by the pump from the reservoir to flow through the test loop.

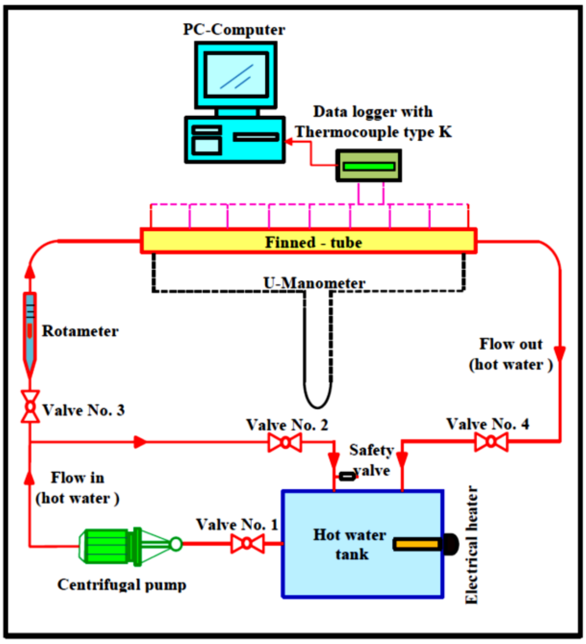

Fig. 2. Schematic diagram of experimental test rig.

A three liter capacity galvanized material vessel equipped by drain valve is used as fluid reservoir. The vessel is insulated with one layer of asbestos rope tape wound and over the asbestos tape winding approximately 40 mm thickness of glass wool is lined to minimize the heat loss to the atmosphere. The water is heated using single electric heater with 4kW capacity and the desired temperature of the hot water is controlled by the temperature controller. A safety valve is installed at tap on the tank to release pressure in the event of increased pressure inside the tank caused by the heater.

A straight aluminum finned- tube with 1.5m length, 22 mm inner diameter and 32 mm outer diameter was used as the test section. The test is provided with nine (K–types, TP – 01) thermocouples, seven are fixed to the surface with a space distance of (250 mm) between each thermocouple and two located to measure the working fluid inlet and outlet temperatures. All these thermocouples have 0.1 °C resolution and are calibrated before fixing them at the specified locations.

The twisted tape was made from copper sheet with tape thickness (δ) of 1 mm, width (W) of 17mm, and length of 1.5m. The tape thickness of 1mm was chosen to avoid an additional friction in the system that might occur by the thicker tape as shown in **Fig. 3** and dimensions of twisted tape insert are given in **Table 2**. The two ends of a strip were held on a lathe, one at the headstock end and the other at the tail stock end by special devices made in the factory. The strip is then subjected to twist by turning the chuck manually. One insert with twist ratio of (1.85) is made and test performed with it. The heat conduction in the copper tape along its length is assumed to be negligible.

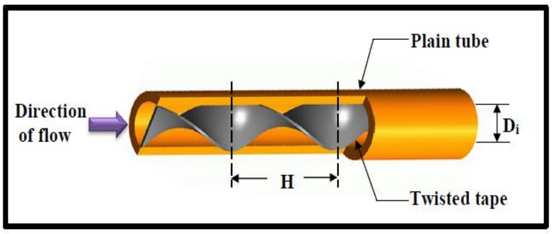

Fig. 3. Diagram of a twisted tape insert inside a tube.

Table 2. Dimension of twisted tape insert.

Twisted Tape Type	Twist Ratio (y)	Pitch (H) (mm)	Width (W) (mm)	Thickness (δ) (mm)
Typical	1.85	40.8	17.15	1

The pressure drop is measured with U-tube manometer by connecting its two ends to 2 mm holes located on either side of the test section. Carbon tetrachloride is used as the manometric liquid for determining the pressure drop at different flow rates. A flow meter capable of measuring in the range of 10-100 LPH is connected to the test section.

A constant value of 4kW is supplied by the heater to the test section. The sensor is adjusted to attain a liquid average temperature of 60 ºC in the inlet test section with a maximum variation of ±1 ºC. The outer surface temperature of the test section is monitored and observed to vary between 49 and 58 ºC. A data logger is connected to record the surface temperature of the aluminum finned-tube (test section) and the inlet and outlet temperatures of liquid every five second to determine the state of the experiment. At steady state, the temperatures, the flow rate and the power input to the heater are recorded.

After the experimental set up is assembled, the storage tank is filled with the working fluid. Experiments are conducted with water and nanofluids to determine friction factor and heat transfer coefficients for flow in a tube. Distilled water was tested prior to nanofluid after completion of construction and calibration of the flow loop, testing of the loop's functionality for measuring Nusselt number and viscous pressure loss. The initial test was done with distilled water, whose performance and properties are taken practically. Five initial tests were run with different flow rates. Then the carefully prepared nanofluids at different volume concentrations of 3% and 5% are used and the procedure repeated.

The working fluid flow rate in the test section is calculated from the flow meter readings and validated with manual measurements. The experimental heat transfer coefficient is estimated with the Newton's law of cooling. The properties of the fluid are considered at the mean temperature. Required data to estimate heat transfer coefficient and friction factor are recorded at different flow Reynolds number range of (678–2033) with flow of water and nanofluid with twisted tape insert in a circular finned- tube. To ensure the steady state condition for each run, the period of around 15–20 minutes depending on Reynolds number value was taken prior to the data record.

Experimental uncertainty was calculated using Kline and McClintock method [16]. The uncertainties associated with experimental data are calculated on the basis of 96 % confidence level. The calculations indicated that the uncertainties involved in the measurements is around ± 3% and ± 4.1% and ± 5% for Reynolds number, Nusselt number and friction factor respectively.

4. Data Reduction

4.1. Thermophysical Properties of Nanofluid

The physical and thermal properties such as density and specific heat of the nanofluid are calculated using different formulae presented in the literature as outlined below with equations (3) to (6). All properties are calculated using bulk temperatures between inlet and outlet.

$$\rho_{nf} = \left(\frac{m}{V}\right)_{nf} = \frac{\rho_{bf} V_{bf} + \rho_p V_p}{V_{bf} + V_p} = (1-\varphi)\rho_{bf} + \varphi\rho_p \tag{3}$$

$$Cp_{nf} = \frac{(1-\varphi)\rho_{bf} Cp_{bf} + \varphi\rho_p Cp_p}{\rho_{nf}} \tag{4}$$

$$\mu_{nf} = (1 + 7.3\varphi + 123\varphi^2)\mu_{bf} \tag{5}$$

$$k_{nf} = k_{bf}(1 + 7.74\varphi) \tag{6}$$

Where φ is the volume concentration and μ is the dynamic viscosity. The index nf, bf and p refers to nanofluid, base fluid and particle properties respectively. In fact there is no reliable data base for viscosity and thermal conductivity of nanofluids and their values can vary significantly depending on the relation used to calculate them.

4.2. Heat Transfer and Hydrodynamics Calculation

The heat flux supplied from the heater system is given by:

$$Q = I \,.\, V \tag{7}$$

Where: I , V are current and voltage.
The amount of the heat transferred from the heater coil to the nanofluid is given by:

$$Q_{fluid} = \dot{m}_{nf}\, Cp_{nf}\, (T_i - T_o) \tag{8}$$

Where:
$\dot{m}_{nf}$: Mass flow rate of the nanofluid.
C_{nf} : Specific heat of the nanofluid.
T_i , T_o: Inlet and outlet temperature of the nanofluid at the test section respectively, as shown in **Fig. 4**.

It is necessary to know the heat transfer coefficient of the nanofluid before any calculations. The local heat transfer coefficient is given by:

$$h(z) = \frac{\dot{q}_w}{(\Delta T)_z} \tag{9}$$

Where: $(\Delta T)_z$ is the difference between the inner wall temperature of the tube $T_{si}(z)$ and the temperature of the nanofluid T(z) at distance z from the entrance of the tube.
To calculate T_{si} (z), the conduction equation in the cylinder is used:

$$\dot{q}_w = \frac{Q_{fluid}}{\pi D_i L} = \frac{2k[T_{si}(z) - T_{so}(z)]}{D_i \times \ln\left(\frac{r_o}{r_i}\right)} \tag{10}$$

Where:
Di : The inner diameter of the pipe.
L : The length of the pipe.

Also, from the energy balance in the tube, the mean temperature of nanofluid can be expressed by [17]:

$$T_o(z + \Delta z) = T_{in}(z) - \frac{\dot{q}_w\, \pi D}{\dot{m}_{nf}\, Cp_{nf}} z \tag{11}$$

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Where: T_{in} (z), $\dot{m}_{nf}$ and Cp_{nf} are the nanofluid temperature at the inlet of test section, the mass flow rate and the heat capacity of the nanofluid respectively.

Thus, the local heat transfer coefficient becomes:

$$h(z) = \frac{\dot{q}_w}{T_{si}(z) - T(z)} \tag{12}$$

Where: T(z) = [Tin(z) +To (z+Δz)] / 2 is the nanofluid bulk temperature.
[[

The local Nusselt Number has been calculated from the following equation:

$$Nu(z) = \frac{h(z)\,D}{k_{nf}} \tag{13}$$

The average value of Nusselt number in the thermal region can be expressed by the integral:

$$\overline{Nu} = \frac{1}{L}\int_0^L Nu(z)\,dz \tag{14}$$

Simpson rule 1/3 is used to calculate the above integration.

The Prandtl and Reynolds number are:

$$Pr_{nf} = \frac{\mu_{nf}\,Cp_{nf}}{k_{nf}} \tag{15}$$

$$Re_{nf} = \frac{\rho_{nf}\,u\,D_i}{\mu_{nf}} \tag{16}$$

Based on the practically measured pressure drop, Darcy friction factor can be calculated using the expression:

$$f = \frac{\Delta P}{2(L/D)\,\rho\,V^2} \tag{17}$$

Where Δp is the pressure drop across the test section measured by the manometers.

$$\Delta P = \gamma_{nf}\left(\frac{\rho_{ccl_4}}{\rho_{nf}} - 1\right) H \tag{18}$$

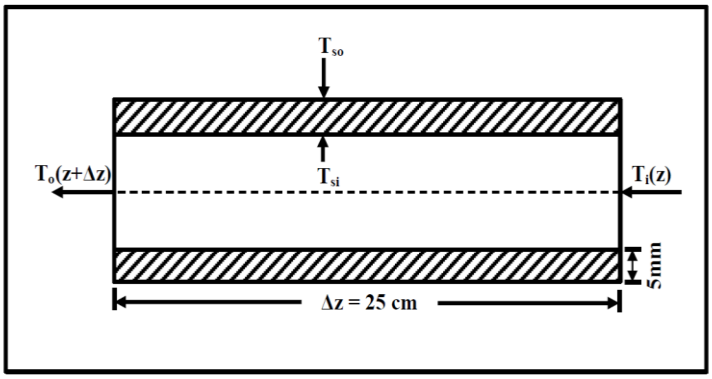

Fig. 4. Section of the pipe of length Δz and at distance (z).

5. Results and Discussion

5.1. Validation of the Experimental System

The Nusselt number results of the tube induced with a plain twisted tape insert were validated with their experimental results as well as with a correlation derived by Hong and Bergles [18], as shown in the equation (19).

$$\mathrm{Nu} = 5.17\left[1+0.005484\times \mathrm{Pr}^{0.7}\left(\frac{\mathrm{Re}}{y}\right)^{1.25}\right]^{0.5} \tag{19}$$

Fig. 5 shows the validation results of the average Nusselt number between the experimental result of the typical twisted tape insert (TT) and Hong & Bergles correlation. The experimental result is in good agreement with the experimental data of Hong and Bergles correlation and falls within a deviation of (16%).

In order to examine the degree to which the experimental results for the friction factor match experimental results and correlations, validation was carried out on the experimental results of the friction factor. The friction factor of the finned-tube induced with plain twisted tape insert were validated with the correlation of Donevski and Kulesza [19], which has been mentioned in the equation (20).

$$f = \left(\frac{15.625}{\mathrm{Re}}\right)\left[1+\left(\frac{0.85}{y^2}\right)\right]^{0.5}\left[\frac{\pi+2-\frac{2\delta}{\mathrm{D}}}{\pi-\frac{4\delta}{\mathrm{D}}}\right]^{2}\left[\frac{\pi}{\pi-\frac{4\delta}{\mathrm{D}}}\right] \tag{20}$$

As depicted in **Fig. 6**, the friction factor for the experimental result has a maximum discrepancy of (18 %) with the Donevski and Kulesza correlation. The above observations indicate that the experimental results are in good agreement with the correlation results. The deviations shown in the comparison are possibly as a result of different conditions under which each result was obtained.

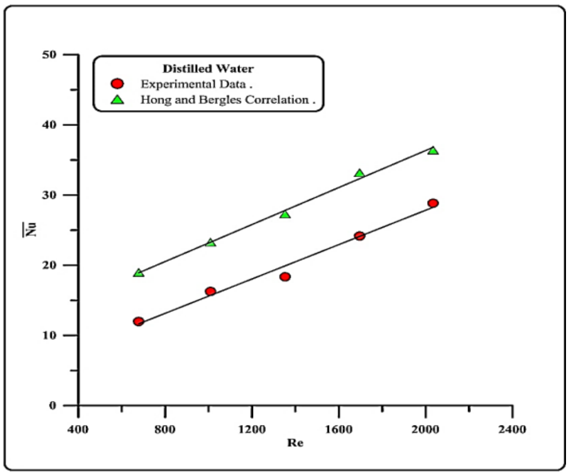

Fig. 5. Validation of ($\overline{\mathbf{Nu}}$) with experimental works for laminar flow.

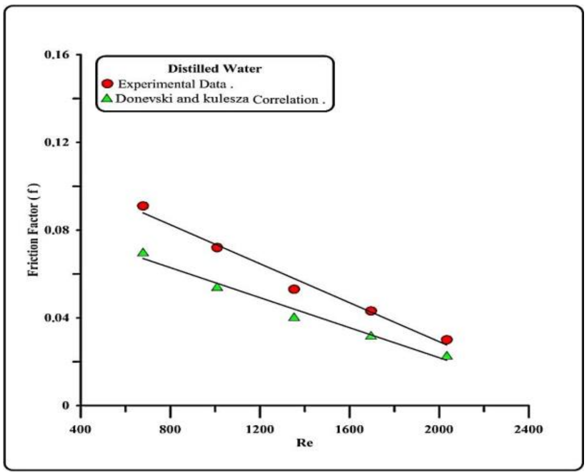

Fig. 6. Validation of (f) with experimental works for laminar flow.

5.2. Tape Effect on the Thermal and Hydrodynamic Characteristics of the Flow

5.2.1. Thermal effect of twisted tape

The recorded data from the experiments were converted into local values of heat transfer coefficient which in turn converted into local values of Nusselt number along the tube, as shown in F**ig.7**. From this figure, it is noticed that the values of Nu are high at the thermal entrance region and then becomes less when the flow advances towards the end of the tube due to the increasing effect of boundary layer. The presence of twisted tape causes an increase in the values of Nusselt number because of the interruption in the fluid flow and decreasing the thermal boundary layer thickness caused by the twisted tape.

The variation in average Nusselt number with Reynolds number for the finned tube with typical twisted tape is shown in **Fig. 8**. In general, the average Nusselt number increases as the Reynolds number increases for all the ranges of Reynolds number. For the tube with twisted tape, the x and y velocities are not zero, therefore, a rotational or secondary flow is created with a desired effect which promote mixing in the plane normal to the bulk flow direction. This mixing works to maintain a high temperature gradient close to the tube wall and in turn increases the enhancement in heat transfer.

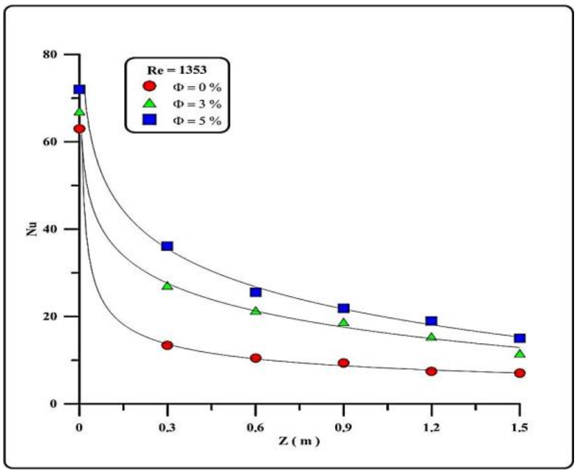

Fig. 7. The variation of local (Nu) along the tube for laminar flow.

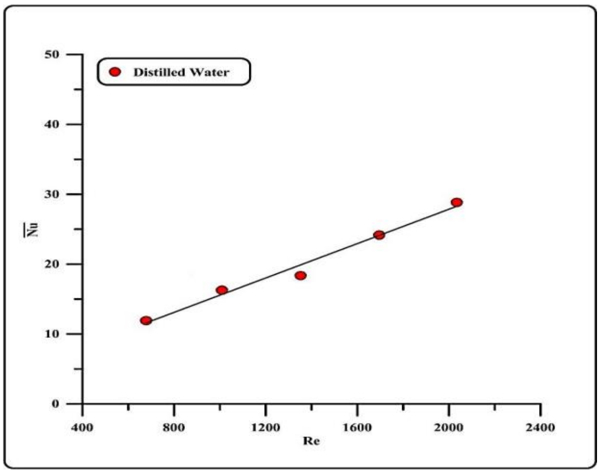

Fig. 8. The effect of (Re) on ($\overline{\mathbf{Nu}}$) for laminar flow.

5.2.2. Hydrodynamic effect of twisted tape

The pressure drop for typical twisted tape in hydrodynamically flow through a circular-finned tube was measured to investigate the flow characteristics for the tube with twisted tape for the twist ratio (TR=1.85) as shown in **Fig. 9**.

Fig. 10 illustrates the variation of friction factor with Reynolds number. From the experimental results, it can be noted that the friction factor tends to decrease with the increase in Reynolds number. Generally, the friction factor decreases as the Reynolds number increases for the whole range of Reynolds.

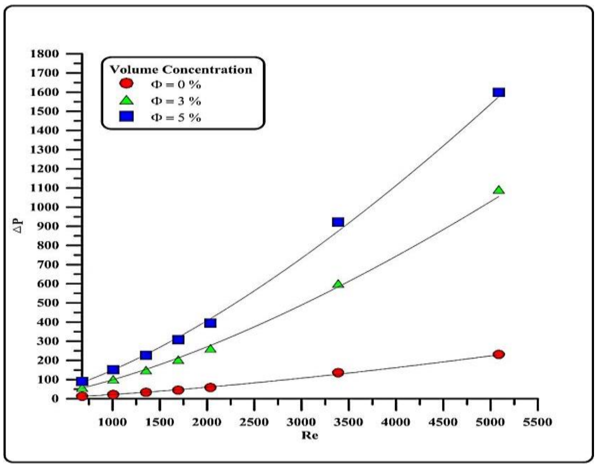

Fig. 9. The effect of Reynolds number on pressure drop.

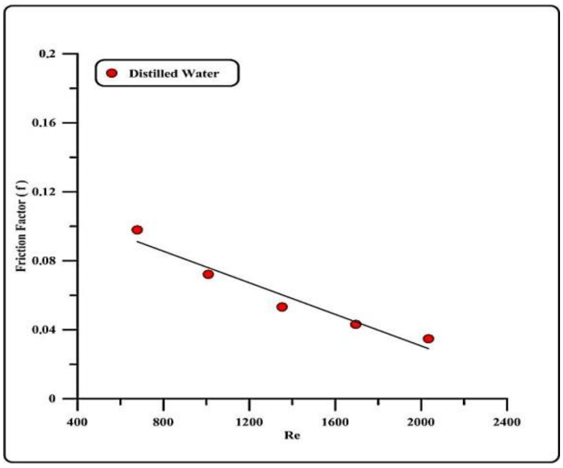

Fig. 10. The effect of (Re) on friction factor for laminar flow.

5.3. Nano Effect on the Thermal and Hydrodynamic Characteristics of the Flow

5.3.1. Thermal Effect of the Use of Nanofluid

Fig. 11 shows the variation of the average Nusselt number ($\overline{Nu}$) with Reynolds number in a finned tube with inserted tape for alumina nanofluid at different concentrations. As the concentration increases, the

($\overline{\mathrm{Nu}}$) increases because the effective thermal conductivity of nanofluid increases with increasing volume fraction of the nanoparticles, which is explained by Brownian motion of the nanoparticles and molecular level layering of the liquid at the liquid - particle interface (wettability). Enhanced thermal conductivity reduces resistance to thermal diffusion in the laminar sub layer of the boundary layer.

The enhancement in the heat transfer ($\overline{\mathrm{Nu}}$), relative to the basefluid, is explained using the Nusselt number ratio ($\overline{\mathrm{Nu}}_{nano} / \overline{\mathrm{Nu}}_{water}$), as shown in **Fig. 12**. It is clear that the Nusselt number ratio increases as the volume concentration increases, and the maximum enhancement in the heat transfer was for ratio (1.68) which occurred at Reynolds number (1353).

The maximum enhancement of convective heat transfer coefficient at 5% volume fraction of nanofluid is (51.1 %) compared with pure distilled water.

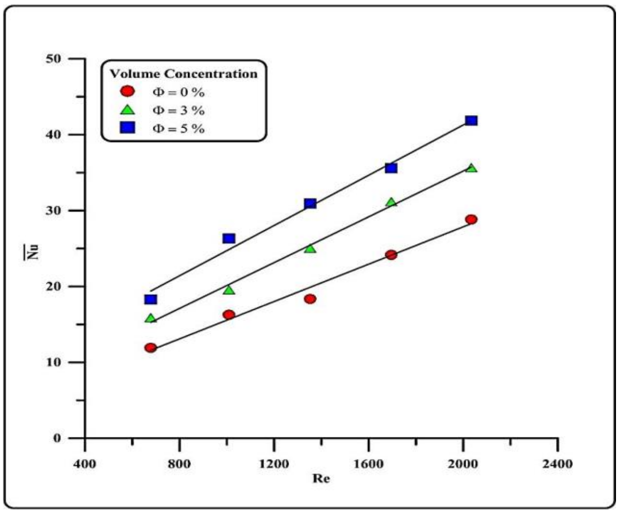

Fig. 11. The effect of (Re) and (φ) on ($\overline{\mathbf{Nu}}$) for laminar flow.

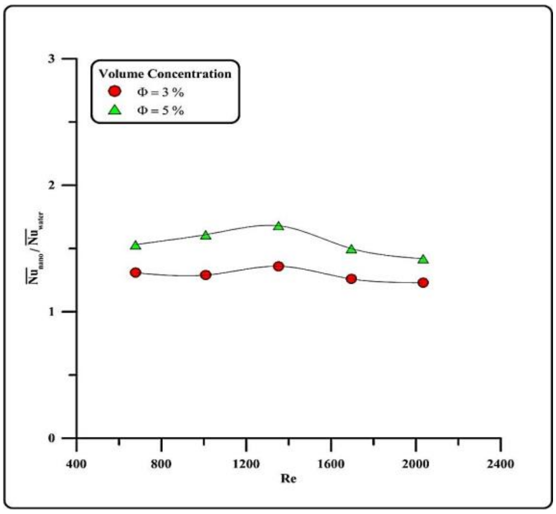

Fig. 12. The effect of (Re) on ($\overline{\mathbf{Nu}}$) ratio for laminar flow.

5.3.2. Hydrodynamic Effect of the Use of Nanofluid

Fig. 13 demonstrates the relationship between the friction factor and Reynolds number of nanofluid flow inside tube with twisted tape insert. For all cases, the friction factor decreases with Reynolds number increasing. At the same Reynolds number, the friction factor increases with increasing the volume concentration value, due to increased viscosity of the working fluid that leads to increase the shear force on the tube wall acted by larger numbers of nanoparticles, leading to increase of the friction factor. The nanofluid with the present range of volume concentration (3 % and 5 %) provides frictions factors higher than the base fluid (water), about (21-44) % and (28-57).

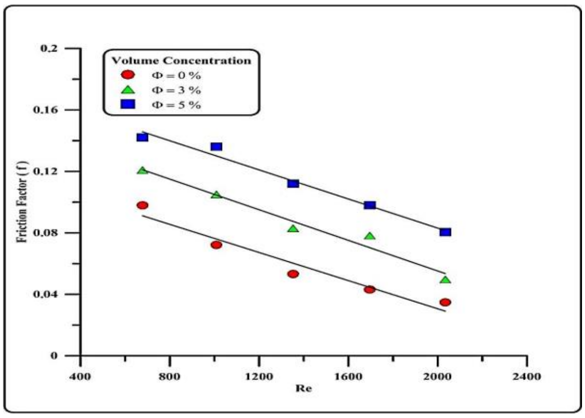

Fig. 13. The effect of (Re) and (φ) on friction factor for laminar flow.

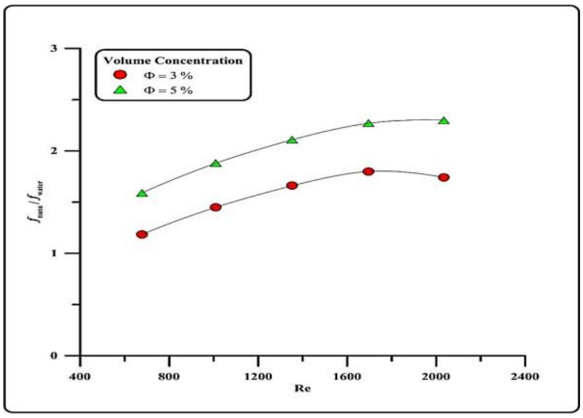

Fig. 14. The effect of (Re) and (φ) on (f) ratio for laminar flow.

Fig. 14 represents the relation between friction factor ratio and Reynolds number for the typical twisted tape (TR=1.85), at different studied parameters. From these figures, it is clear that the friction factor ratio increases as Reynolds number increase, except at (Re =2033) and increases as volume concentration increase for laminar flow regime.

The maximum increase in the pressure drop (friction factor) was for friction factor ratio (2.3) which is found at Reynolds number (2033), and the minimum increase in the friction factor was for friction factor ratio (1.4) which occurred at Reynolds number (678).

5.3.3. Thermal Performance Factor Evaluation

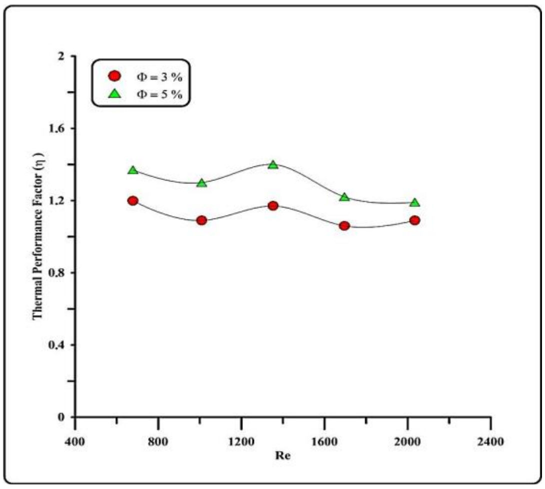

Fig. 15. The effect of (Re) and (φ) on (η) for laminar flow.

In general, the thermal performance factor above unity indicates that the effect of heat transfer enhancement due to enhancing device is more dominant than the effect of rising friction and vice versa. **Fig. 15** shows the variation in thermal performance factor with Reynolds number and volume concentration.

From this figure, the maximum thermal performance factor for typical twisted tape was (1.4) at Reynolds number (1353), and (φ = 5 %).

5.4. Correlations

Empirical correlations of Nusselt number and friction factor were estimated from the experimental results. This is accomplished by using multiple regression analysis method. The obtained data are related with Re, Pr, φ, and TR (y) with (Al_2O_3-distilled water) nanofluid through the following correlations:

<u>1- Correlations for tube with inserted twisted tape:</u>

For typical twisted tape, the equation is:

$$\overline{Nu} = C\, Re^{m}\, Pr^{n}\, (1+\varphi)^{a}\, y^{f} \tag{21}$$

The fitted values of average Nusselt number is represented by the equation (22). The fitted values are compared with experimental results as shown in **Fig. 16**. The fitted values are agreeing with the experimental data within ±13%.

$$\overline{Nu} = 0.492\, Re^{0.45}\, Pr^{0.33}\, (1+\varphi)^{0.158}\, y^{-0.223} \tag{22}$$

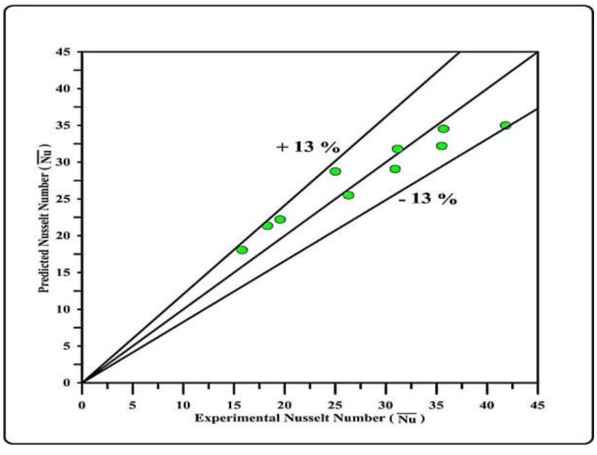

Fig. 16. Comparison of experimental and predicted ($\overline{\mathbf{Nu}}$) for laminar flow.

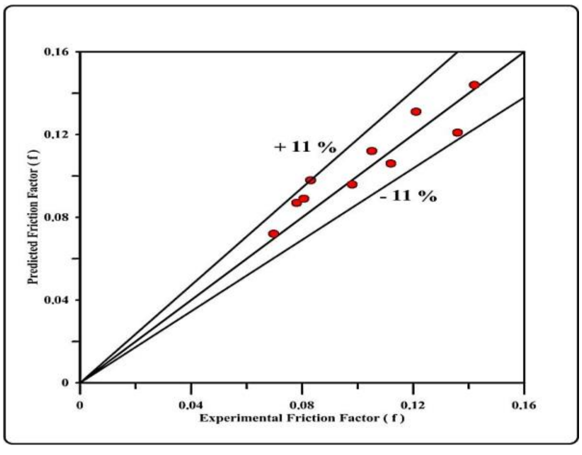

Fig. 17. Predicted versus experimental (f) under laminar flow.

2- Correlations for Friction Factor:

The friction factor is correlated generally by:

$$f = C\,Re^{m}\,(1+\varphi)^{a}\;y^{f} \quad (23)$$

The fitted values of friction factor are represented by the equations (24). **Fig. 17** gives the representation of this correlation. The fitted values are agreeing with the experimental data within ±11%.

$$f = 3.29\,Re^{-0.44}\,(1+\varphi)^{0.174}\,y^{-0.479} \quad (24)$$

6. Conclusion

This study presents the experimental investigation of Convective heat transfer performance of Al_2O_3/water nanofluid for 3% and 5% volume concentration for laminar flow regime under constant inlet temperature conditions. It was found that:

1- The nanofluid enhances the heat transfer higher compared with the base fluid (pure water) in general, with increase of pressure drop. The heat transfer enhanced with increasing nanoparticles concentration, and (φ =5 %) gives higher heat transfer enhancement among the studied concentrations.
2- The combined use of the nanofluid and twisted tape gives higher heat transfer enhancement compared to the individual use of each one and shows that the maximum enhancement in the heat transfer (2 times Nusselt number of the distilled water at Re = 1695) and thermal performance factor was (1.4) for Al_2O_3 with typical twisted tape and twist ratio (TR=1.85) at Reynolds number (1353) and concentration (φ =5 %).
3- The temperature of the flow increases and becomes more homogenous with increasing flow rate and concentration.
4- An empirical correlation has been developed for Nusselt number based on the experimental data for water and nanofluid flowing through the circular finned-tube with twisted tape insert.
5- Addition of agitated impeller to with ultrasonic cleaner device have significant effect to reducing preparation time.
6- There is significant increase in pressure drop and friction factor in comparison to water at the same twist ratio.
7- Results show that Hong & Bergles and Donevski and Kulesza theoretical models are in good agreement with the experimental data related to the friction factor and the observed uncertainty is low.
8- The Convective heat transfer coefficient enhancement decreases with increasing axial distance.
9- The ratio of (h_{nf}/h_{bf}) increases with increasing Re number for both volume concentrations. The average heat transfer coefficient enhanced about 19-46% with 3% volume concentration and increased about 21-51% with 5% volume concentration compared to distilled water.
10- The average ratio of (f_{nf}/f_{bf}) was about 2.3 for 5% volume concentration.

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