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Heat and Fluid Flow in a Tube with Helical Tape Around Rod

السريان وانتقال الحرارة فى انبوب بداخلة شريط لولبى

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فى هذه الدراسة المعملية تم تصميم وانشاء دائرة اختبار معملية لدراسة تأثير العوامل المختلفة على كل من معامل انتقال الحرارة بالحمل الجبرى وكذلك الإنخفاض فى الضغط على طول اللأنبوب. وقد تم دراسة تأثير كل من الحرارة المضافة ونوع مائع التشغيل وكذلك تأثير الجزيئات متناهية الصغر فى مائع التشغيل الأساسى. حيث اجريت التجارب عند قيم متعددة من الحرارة المضافة من ٢١٠٠ الى ٣٣٠٠ وات وتغيير موضع الشريط الولبى داخل الماسورة بالنسبة للمركز وكذلك تغيير رقم رينولد من ١٣٨٠ الى ٤١٥٠. وكذلك تغيير تركيز الجزيئات المتناهية الصغر (أكسيد الألومنيوم) فى مائع التشغيل الأساسى من ٠.٠٢٥% الى ٠.٣٥%. وتمت دراسة تأثير كل منهم على معامل انتقال الحرارة الكلى وكذلك الفقد فى الضغط وايضا رقم نسلت. وقد أظهرت النتائج المعملية ارتفاع ملحوظ فى رقم نسلت يبدأ من ٣.٧٨ %عند تركيز ٠.٠٢٥% وصل الى ١٦.١٧ % عند تركيز ٠.٢% فى حالة استخدام مائع مكون من ماء مضاف الية اكسيد الألومنيوم عنه فى حالة الماء فقط. ومن اجل التأكد من صحة النموزج العملى تمت مقارنة النتائج المعملية مع نتائج معملية لأبحاث سابقة وقد وجد انها مطابقة الى حد ما. وقد تمت صياغة علاقة لرقم نسلت من النتائج المعملية معتمدا على رقم برانتدل وكذلك رقم رينولد. وتمت مقارنة النتائج الناتجة من هذه العلاقة مع الموجودة من النتائج المعملية وجاءت متقاربة.

Abstract

Forced convective heat transfer and friction factor for nanofluid flows inside circular horizontal tube with inserting helical tape, is experimentally studied. Electric heater is wrapped around the outer surface of the tube to obtain a constant and uniform heat flux at the tube wall. Experiments are conducted with adding nano-particles (Al_2O_3) to water up to 0.35% by volume to obtain different concentrations of nanofluids. An experimental test loop equipped with the required measuring instruments was designed and constructed to assess the effects of nano-particles concentration, mass flow rate, and applied heat flux on the convection heat transfer process and pressure drop. The tested tube fitted with screw helical tape inserts to evaluate its effects on heat transfer rate and friction factor with nanofluid and pure water as a working fluid. The measurements of temperature, flow rate, applied volt and pressure drop are recorded and manipulated to calculate the convective heat transfer coefficient and friction factor.

The obtained experimental results show that, wall temperature is reduced by using nanofluid compared with pure water. Accordingly, the convection heat transfer coefficient increased when using nanofluid and it also increased with increasing heat flux and mass flow rate (Reynolds number). Higher rates of heat transfer and pressure drop are obtained from the tube fitted with screw helical tape insert compared to flow in a plain tube under similar conditions. This improvement in the convection heat transfer coefficient characterized by a swirling flow as a result of the secondary flow of the fluid flow inside tube with helical tape and the area of the tape inserted. The average value for the thermal–hydraulic performance (η) for tube with inserting helical tape was 1.4 for water only, and 1.8 for nanofluid at concentration 0.2%. Comparison with the previous work shows fairly agreement. A correlation is obtained between Nusselt number and operating parameter.

Key words: Nanofluid; Convective heat transfer, helical tape around rod inserted, Horizontal tube.

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1. Introduction

Low thermal conductivity of conventional heat transfer fluids such as water, oil, and ethylene glycol is a serious limitation in improving the performance and compactness of many engineering equipments such as heat exchangers and electronic devices. To overcome this disadvantage, there is strong motivation to develop advanced heat transfer fluids with substantially higher thermal conductivity. An innovative way of improving the thermal conductivities of fluids is to suspend small solid particles in the fluid. Various types of powders such as metallic, non-metallic and polymeric particles can be added into fluids to form slurries. The thermal conductivities of fluids with suspended particles are expected to be higher than that of common fluids. In conventional cases the suspended particles are of micrometer or even millimeter dimensions. However, such large particles may cause severe problems such as abrasion and clogging. Therefore, fluids with suspended large particles have little practical application in heat transfer enhancement. Nanofluids are a new kind of heat transfer fluids containing a small quantity of nano-sized particles (usually less than 100 nm) that are uniformly and stably suspended in a liquid. The dispersion of a small amount of solid nanoparticles in conventional fluids changes their thermal conductivity remarkably. Compared to the existing techniques for enhancing heat transfer, the nanofluids show a superior potential for increasing heat transfer rates in a variety of cases. The heat transfer rate can be improved also by introducing a disturbance in the fluid flow (breaking the viscous and thermal boundary layers), but in the process, pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, several techniques have been proposed in recent years and are discussed in the following section. The great attempt on utilizing different methods is to increase the heat transfer rate through the compulsory forced convection. Meanwhile, it is found that this way can reduce the sizes of the heat exchanger device and save up the energy. In general, enhancing the heat transfer can be divided into two groups. One is the passive method, without stimulation by the external power such as a surface coating, rough surfaces, extended surfaces, swirl flow devices, the convoluted (twisted) tube, additives for liquid and gases. The other is the active method, which requires extra external power sources, for example, mechanical aids, surface-fluid vibration, injection and suction of the fluid, jet impingement, and use of electrostatic fields.

Swirl flow is generated inside tubes fitted with tape inserted is one of the most important members of this group. These tubes are the most widely used in several heat transfer applications; for example air conditioning and refrigeration systems and chemical reactors and heat recovery processes. Many papers or researchers are studied the swirl flow inside the tubes [1 to 14]

Sivashanmugam, and Suresh [1, and 2], studied experimentally the heat transfer and friction factor characteristics of circular tube fitted with full-length helical screw element of different twist ratio, with uniform heat flux under laminar and turbulent flow conditions. The experimental data obtained were compared with those obtained from plain tube. The maximum Nusselt number for the twist of 1.95 was obtained, indicating heat transfer coefficient increases with the twist ratio and friction factor also increases with the twist ratio.

Eiamsa-ard, and Promvonge [3], Studied experimentally the influence of helical tapes inserted in a tube on heat transfer enhancement. A helical tape is inserted in the tube with a view to generating swirl flow that helps to increase the heat transfer rate of the tube, with changing the mass flow rates. The swirling flow devices consisting of full-length helical tape with or without a centered-rod, and the regularly-spaced helical tape, Experimental results confirmed that the use of helical tapes leads to a higher heat transfer rate over the plain tube. The full-length helical tape with rod provides the highest heat transfer rate about 10% better than that without rod but it increased the pressure drop. The regularly spaced helical tape inserts yields to the highest Nusselt number which is about 50% above the plain tube.

Roy et al, [4], studied numerically the hydrodynamic and thermal behaviors of nanofluids flowing inside a uniformly heated tube. Results have clearly shown that the inclusion of nanoparticles has produced a considerable increase of the heat transfer with respect to that of the base liquid. Such heat transfer enhancement, which appears to be more pronounced with the augmentation of the particle volume concentration, is accompanied, however, by a major drawback on the wall shear stress. Among the nanofluids studied, it has been shown that the ethylene glycol offers, so far, a better heat transfer enhancement than the water γ-Al_2O_3 nanofluid. It is also the one for which a more pronounced adverse effect on the wall shear stress has been observed. For the turbulent flow regime, results have also shown that the heat transfer enhancement due to the presence of nanoparticles becomes more important with the increase of the Reynolds number.

Roy et al, [5] studied numerically heat transfer enhancement capabilities of coolants with suspended metallic nanoparticles inside typical radial flow cooling systems. The laminar forced convection flow of these nanofluids between two coaxial and parallel disks with central axial injection has been considered using temperature dependent nanofluid properties. Results clearly indicate that considerable heat transfer benefits are possible with the use of these fluid/solid particle mixtures. For example, a Water/Al_2O_3 nanofluid with a volume fraction of nanoparticles as low as 1% can produce a 25% increase in the average wall heat transfer coefficient when compared to the base fluid alone (water). Furthermore, results show that considerable differences are found when using constant property nanofluids (temperature independent) versus nanofluids with temperature dependent properties. The use of temperature-dependent properties make for greater heat transfer predictions with corresponding decreases in wall shear stresses when compared to predictions using constant properties. With an increase in wall heat flux, it was found that the average heat transfer coefficient increases whilst the wall shear stress decreases for cases using temperature-dependent nanofluid properties.

Abu-Nada et al, [6], studied the heat transfer enhancement in horizontal annuli using nanofluids. Water-based nanofluid containing various volume fractions of Cu, Ag, Al_2O_3 and TiO_2 nanoparticles is used. The addition of the different types and different volume fractions of nanoparticles were found to have adverse effects on heat transfer characteristics. For high values of Rayleigh number, nanoparticles with high thermal conductivity cause significant enhancement of heat transfer characteristics. On the other hand, for intermediate values of Rayleigh number, nanoparticles with low thermal conductivity cause a reduction in heat transfer.

Behzadmehr et al, [7], studied numerically the turbulent forced convection heat transfer in a circular tube with a nanofluid consisting of water and 1% Cu using two phase mixture model and single phase model, for comparison with the mixture model. Comparisons of the Nusselt number predicted by these two models with corresponding experimental results show that the mixture model is more accurate than the single phase model. However, it seems that the accuracy of mixture model could improve by using suitable effective physical properties for nanofluid instead of volume weighted average of particle and fluid properties. Adding 1% nanoparticles (Cu), increases the Nusselt number more than 15% while it did not have any significant effect on the skin friction. It was shown that the particles concentration at high values of Reynolds number may not be uniform.

Ding, and Wen [8], studied experimentally the convective heat transfer of nanofluids, made of γ-Al_2O_3 nanoparticles and de-ionized water, flowing through a copper tube in the laminar flow regime. The results showed considerable enhancement of convective heat transfer using the nanofluids. The enhancement was particularly significant in the entrance region, and was much higher than that solely due to the enhancement on thermal conduction. Migration of nanoparticles and the resulting disturbance of the boundary layer were proposed to be the main possible reasons for the heat transfer enhancement.

Eiamsa-ard et al, [9], studied experimentally the effect of the tapes twisted in clockwise and counterclockwise arrangement on heat transfer and friction factor characteristics in a double pipe heat exchanger. The experiments were undertaken for various ranges of Reynolds number range and for several twisted-tape pitch ratios, the mean heat transfer rates obtained from using C-CC twisted-tape arrangement and original twisted-tape arrangement are found to be 219% and 204%, respectively over the plain tube. The increase in friction factor changed between 1.5 and 4.7 times in comparison with those for the original twisted-tape and the plain tube, respectively, depending on Reynolds number and pitch ratio.

Veysel et al, [10] studied numerically the heat transfer enhancement in a tube with the circular cross sectional rings. The rings were inserted near the tube wall. Five different spacings between the rings were considered. Uniform heat flux was applied to the external surface of the tube and air was selected as working fluid. Numerical calculations were performed with FLUENT 6.1.22 code, in the range of Reynolds number 4475–43725. The results obtained from a smooth tube were compared with those from the studies in literature in order to validate the numerical method. Consequently, the variation of Nusselt number, friction factor and overall enhancement ratios for the tube with rings were presented and the best overall enhancement of 18% was achieved for Re = 15,600 for which the spacing between the rings is 3D.

Zeinali et al, [11, and 12], studied experimentally the laminar flow forced convection heat transfer of Al_2O_3/water nanofluids inside a circular tube with constant wall temperature. The Nusselt numbers of nanofluids were obtained for different nanoparticles concentrations as well as various Peclet and Reynolds numbers. Experimental results emphasized that the enhancement of heat transfer due to the nanoparticles presence in the fluid. Heat transfer coefficient was increasing by increasing the concentration of nanoparticles in Nanofluids. The increase in heat transfer coefficient due to presence of nanoparticles was much higher than the prediction of single phase heat transfer correlation used with nanofluid properties.

Dutta et al, [13], studied experimentally heat transfer and pressure drop in a circular tube fitted with regularly twisted tape elements. Laminar swirl flow of viscous fluid having intermediate prandtle number range was considered. The swirl was generated by regularly spaced twisted tape elements with single twist in the tape module and connected by thin circular rods. The tape width and the rod diameter were both varied. Heated as well as isothermal friction factor data has been generated. The heat transfer test section was heated electrically imposing axially and circumferentially constant wall heat flux boundary condition. Reynolds number, Prandtle number, and twist ratio, tape width, rod diameter and phase angle govern the characteristics. Higher than zero phase angles is no use; rather it increases the manufacturing complexity. The difference of heated friction factor and isothermal friction factor for the periodic swirl flow is substantially less than that in case of straight flow through plain tube.

The present work aims to study the convection heat transfer coefficient and friction factor of water and Al_2O_3/water nanofluid convection heat transfer in a circular tube fitted with helical tape around a rod under constant heat flux, and different nanofluid concentration, and different eccentricity. This is made at different mass flow rate, and heat flux.

2. EXPERIMENTAL TEST LOOP

The experiment test loops has been designed and build to study the effect of different operating parameters on the convective heat transfer and friction factor for the working fluids (nanofluid with different concentrations and pure water). The operating parameters are heat flux, changing eccentricity, concentration and mass rate of the working fluid, which flows inside a horizontal tube with inserting helical tape. A schematic diagram for the experimental test loop is shown in Fig. (2). It consists of nanofluid cooling circuit, circulating pump, the tested tube and its heating unit.

The circulating pump is used to circulate the working fluid through the test loop. The specified amount of working fluid is controlled to flow inside the tested tube and then passes to the tank. Paddle wheel is used to stir constantly the nanofluid inside the tank during the experiments to prevent deposition or sedimentation of the nanoparticles. The concentration of nanofluid is varied from 0 % to 0.35% by volume. Working fluid is heated in the tested tube to study the convection heat transfer and friction pressure drop at different operating parameters. The working fluid is cooled before entering the tested tube in the heat exchanger. City water is used in the heat exchanger to cool the working fluid nearly to the initial temperature of working fluid. Then the working fluid returned back to the tank to complete the cycle. Mass flow rate is varied and the corresponding values of Reynolds number ranging from 1380 to 4150.

The tested tube is made from copper and the inner diameter is of 38 mm and it is of length 1000 mm with thickness 1.0 mm. Uniform heat flux condition is achieved by heating the outer wall of the tested tube by an electric heater coil wrapped around it. A thin layer of electric insulation (Mica sheet) is found between the electric coil and the tested tube. The supplied power to the heating coil is changed by using step less variable output power supply (Variac). Throughout all experiments, the resistance of heating coil has constant value of 10.8 Ω.

The geometrical configuration of helical screw tape inserts is shown in Fig. 2. The helical screw tape inserts with specified twist ratio (equal to (12/17)). Its tooth of 3 mm width and 3 mm thickness over a 17 mm rod with pitch 12 mm, and coated with chromium by electroplating to prevent corrosion. The twist ratio defined as the ratio of length of one full twist (360 degree) to diameter. the helical tape inserted inside the tested tube at different eccentricities first at 0.0, then 0.158, and 0.316 from the center of the test tube as shown in figure (1).

To ensure minimum heat loss to the surroundings, a layer of 50-mm of glass wool thermal insulation followed by additional aluminum foil sheet is wrapped on outer surface of the tested tube.

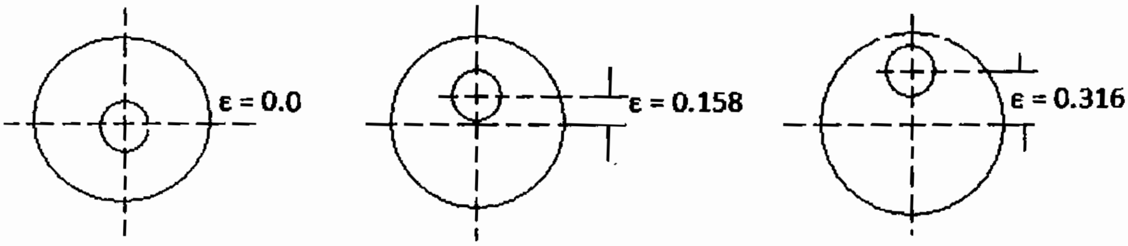

Figure (1) The eccentricity of inserted tape at different location

3. EXPERIMENTAL PROCEDURE

Before starting a new run, the test loop is checked for leakage of working fluid (Al_2O_3/water). The experimental procedure for each run can be described in the following steps:

1. Switch on the pump motor to circulate nanofluid. Air is extracted from the U-tube manometer through a vent valve.
2. Adjust the discharge valve to obtain the desired flow rate for circulating nanofluid. The circulating nanofluid is then heated up through the tested tube by the heating coil.
3. The hot nanofluid is passed through the heat exchanger to remove the heat absorbed by the nanofluid in the tested tube using cooling city water, and then send it to the tank containing the paddle wheel.
4. Before recording any data, each experiment is carried out for 30 minutes to insure steady condition is reached (the fluctuation in temperature was about ±0.3 °C)
5. The required measurements are recorded for each experiment.
6. Repeat the run for different values of the operating parameters.

4. EXPERIMENTAL MEASUREMENTS

To compute the total input heat, the applied volt to the electrical coil was measured by voltmeter with minimum readable value ±0.1 Volt.

Also the temperatures were measured by temperature recorder with accuracy ±0.1 °C. Two thermocouples of K-type are used to measure inlet and outlet working fluid temperatures. Outer surface temperatures for the tested tube are measured along its length at the following locations: 0.150, 0.350, 0.550, 0.750, and 0.950 mm. Also inlet and outlet temperatures of the working fluid entering and out of the test tube are measured. Inlet and outlet temperatures for the city water which flows inside the heat exchanger are also measured to control the cooling process for working fluid. Also, ambient and insulation temperature are measured.

The working fluid flow rate is measured by using flow meter. The accuracy of the used flow meters was 2% of the full scale. The pressure drop for working fluid flows through the tested tube was measured by using U-tube mercury manometer.

The experimental apparatus was allowed to operate until the fluctuation in temperatures was about ±0.3 °C. Then, steady state condition was reached and the required measurement of the applied Volt, temperature, flow rate and pressure drop were taken. The root –mean-square random error propagation analysis was carried out in the standard fashion using the measured experimental uncertainties of the basic independent parameters. The experimental uncertainties associated with these measurement techniques were estimated to be approximately equal to 4.5% and 5.6% for convection factor respectively.

5. DATA REDUCTION

The basic measurements were analyzed using a computer reduction program to calculate the friction factor and convection heat transfer coefficient for working fluid flow inside the tested tubes.

5.1 Pressure Drop

The measured pressure drop includes fictional, entrance and exit components. The entrance and exit components were small and may be neglected. Therefore, the measured pressure drop (ΔP) for nanofluid flow inside the tested tube is then directly related to the frictional pressure drop. The friction factor (f) was obtained from the pressure drop measurements through the tested tube as;

$$f = \left(\frac{D_i}{L}\right)\left(\frac{2\,\Delta P}{\rho_{nf}\,U_m^2}\right)$$

Where D_i, L, U_m and ρ_{nf} are the tested tube inner diameter and length, mean velocity and density for the working fluid.

The mean velocity can be calculated by dividing the measured volume flow rate by cross sectional area of the tested tube without inserts. If the blockage due to helical tape insert was considered.

Before assembly the test loop, the volume of water filling the tested tube with and without helical tape insert can be measured experimentally. Then, the corresponding cross section area with blockage was obtained. The mean velocity can be calculated by dividing the measured volume flow rate by this obtained area.

5.2 Heat Transfer

At steady state, the total input heat from the electric heater coil wrapped around the tested tube (Q_t) divided into useful heat which flows through the tube wall to working fluid (Q_{us}) and the remaining amount of heat transferred to the surrounding as heat loss (Q_{loss}). The total input heat can be determined as:

$$Q_t = \frac{V^2}{R}$$

Where: V, and R are the applied voltage across electric heater, and electric resistance of electric heater respectively. The useful heat transfer to the nanofluid can be calculated as:

$$Q_{us} = m_{wf}\,Cp_{nf}\,(T_{wf,o} - T_{wf,i})$$

Where: m_{wf}, Cp_{wf}, $T_{wf,i}$ and $T_{wf,o}$ are mass flow rate, specific heat, inlet and outlet temperatures of the working fluid respectively.

The amount of heat loss from outer surface of insulation to the surrounding air (Q) can be determined as:

$$Q_{loss} = Q_t - Q_{us}$$

Heat flux (q'') can be calculated from the following equation as:

$$q'' = Q_{us}/A$$

Where; A is the inside surface area for tested tube and equal ($A = \pi\, D_i\, L$). Also, D_i is the inner diameter of the tested tube and L is the tested tube length.

To calculate the local average temperature along the tested tube, one can divide the tube into set of elements and carry out the heat balance for each element. The outlet temperature at each element, see in Fig.2b can be determined using the following relation:

$$T_i = T_{i+1} + \left(\frac{Q_{us}\,\Delta x}{m_{nf}\,Cp_{wf}\,L}\right)$$

Where; T_i, T_{i+1}, and Δx, and m are inlet and outlet temperature for each element and the thickness of the element and mass flow rate respectively. And then the bulk temperature T_b can be calculated as

$$T_b = \frac{T_{i+1} + T_i}{2}$$

The local heat transfer coefficient can be calculated with the aid of the measured local wall temperatures and the above calculated bulk temperature as:

$$h = \frac{q}{T_w - T_b}$$

Where; T_w is the local wall temperature along the tested tube, with neglecting the thermal resistance of the tube wall so the inner wall temperature is approximately equal the outer wall temperature.

The average heat transfer coefficient can be calculated as:

$$h_{av} = \frac{1}{N}\sum_{i=1}^{N} h_i$$

Based on the hydraulic diameter of the tested tube, Local Nusselt number, average Nusselt number, and Reynolds number are evaluated using the following relations:

$$Re_x = \frac{\rho_{wf}\, v\, x}{\mu_{wf}} \qquad Re = \frac{\rho_{wf}\, v\, D_H}{\mu_{wf}}$$

$$Re = \frac{4\, m}{\pi\, (D_i + d)\mu_{wf}}$$

$$Nu_{av} = \frac{h_{av}\, D_H}{K_{nf}}$$

Where K_{wf}, μ_{wf} are the thermal conductivity, dynamic viscosity of working fluid.
Thermal-hydraulic performance (η) is defined by Chang et al. (2007) as follow;

$$\eta = \frac{(Nu_{in}/Nu_p)}{(f_p/f_{in})^{0.33}} \times \frac{A}{A_p}$$

One can assume that the nanoparticles are well dispersed within the base-fluid, so the effective physical properties are described by classical formulas which mentioned by Mansour et al. (2007) as;

$$\rho_{nf} = (1 - \Phi)\, \rho_w + \Phi\, \rho_{np}$$

$$Cp_{nf} = (1 - \Phi)\, Cp_w + \Phi\, Cp_{np}$$

Where, Φ is the nano-particle volume fraction.

The effective dynamic viscosity of nanofluids can be calculated using different existing formulas that have been obtained for two-phase mixtures. The following relation is the well known (Einstein's equation) for a viscous fluid containing a dilute suspension of small, rigid, spherical particles [15].

$$\mu_{nf} = \mu_w\, (123\Phi^2 + 7.3\Phi + 1)$$

$$K_{nf} = K_w\, (4.97\Phi^2 + 2.72\Phi + 1)$$

The relevant thermo-physical properties of the solid nanoparticles (Al_2O_3/water) used in the present study are specified as;

Cp_{np}=773 J/kg. °C, ρ_{np}=3880 kg/m^3 and k_{np}=37 W/m. °C, and K_w = 0.613 W/m.°C

A correlation between operating parameter and average Nusselt number is

$$Nu_{av} = 0.578\, Re^{0.5456}\, Pr^{0.358}$$

6. RESULTS AND DISCUSSION

Experimental runs are performed to study the effects of heat flux, Reynolds number for pure water, nanofluid (with different concentrations, ϕ) flow inside tube with inserting helical tape around rod, and changing eccentricity.

6.1 Friction factor

Pressure drop experiments are performed for adiabatic condition to obtain the friction factor for nanofluid and pure water flow inside tube with inserting helical tape compared with plain tube. Figure (2) shows the variation of friction factor (f) versus Reynolds number (Re) for both tubes. It is observed from figure that, friction factor decreases with increasing Reynolds number for pure water flow inside both tubes. Also, it is clear from figure that, friction factor in tube with inserting helical tape has higher values than that for the plain tube. The increase in the friction factor can be explained by the swirling flow as a result of the secondary flows of the fluid. This hydraulic behavior for tube with inserting helical tape is expected because it promotes more friction. Comparison between the values of friction factor for nanofluid with concentration=0.2 % and pure water flow inside tube with inserting helical tape is shown also in Figure (3). It is clear from figure that, friction factor for nanofluid is higher than that for water only. Nano-particles suspended in the base fluid changes the transport properties than the base fluid and cause an increase for friction factor compared with water only. The increase of kinematics viscosity makes the increase of friction factor.

6.2 Performance of Heat Transfer using Water as Working Fluid

Local surface temperatures along the tested tube is measured and plotted in figure (4) for tube with inserting helical tape for pure water for experimental runs. But the local surface temperatures along the test tube with nanoparticles is presented in figure (5) for different values of heat flux and Reynolds number (Re=2750) at 0.05% Concentration. It is observed that surface temperature increases along tube length and increases with increasing heat flux and also with adding nanoparticles as shown in the two figures. Local surface temperatures along the tested tube with inserting helical tape around the rod in the tube with zero eccentricity takes lower values than of 0.158 eccentricity than of 0.316 eccentricity for 3000 W heat rate and Reynolds number of (Re = 4120) as shown in figure (6). This decrease in the surface temperature for the tube with inserting helical tape around the rod can be explained by the swirling flow near the tube wall as a result of the secondary flow of the fluid behind the surface and also for the non-uniform flow than make non-uniform disturbance in the flow stream.

Local convection heat transfer coefficient along the tested tube was calculated for each surface temperature and plotted in figures (7) for tube with inserting helical tape around the rod for experiment of pure water. But as it compared with the local heat transfer coefficient values for nanofluid of concentration 0.05 % as shown in Fig (8) shows that it has lower values; this means that adding nanoparticles to the water enhance the heat transfer rates due to their higher thermal conductivities.

Nanoparticles leads to increase thermal conductivity for nanofluid, therefore convection heat transfer coefficient increases. Also, other factors such as dispersion and chaotic movement of nanoparticles, and particle migration, particle fluctuations and interactions may play role in heat transfer augmentation.

Figure (9) shows the variation of the local Nusselt number versus local Reynolds number. It is observed from figure that, local Nusselt number decreases with increasing local Reynolds number and the higher values are obtained for nanofluid as compared with water only as shown if fig (5)

Figure (10) shows the variation of average values of Nusselt number versus average values of Reynolds number for pure water at different eccentricity which increases with increasing eccentricity as shown in the figure

Figure (11) shows the variation of the average Nusselt number versus nanofluid concentration of the nanofluid from the figure the average Nusselt number increases with increasing the eccentricity and take the highest value of Nu = 62 at $\varepsilon = 0.0$ at $\varepsilon = 0.15$ %, then it decreases with increasing concentration.

Figure (12) Show the variation of average Nusselt number and Eccentricity at different Concentrations for Reynolds (Re = 2750), which shows that the average Nusselt number increase with increasing eccentricity and also with increasing the concentration.

CONCLUSIONS

Forced convective heat transfer and friction factor for nanofluid flows inside circular horizontal tube with and without inserting helical tape around a rod, was studied experimentally. Experiments are conducted, with adding nanoparticles (Al_2O_3) to the water up to 0.35 % by volume, with changing the mass flow rate and also the eccentricity, to study the effect of all these parameters on the heat transfer and friction factor performance. The obtained experimental results show that the convection heat transfer coefficient and friction factor for the tube with inserting the helical tape around the rod take higher values compared with the plain tube. This increase can be explained by the swirling flow as a result of the secondary flows of the fluid. Thermal hydraulic performance for the tube with inserting tape was 1.4 when using water only, and 1.8 when using nanofluid at concentration 0.20%.

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NOMENCLATURE

A:	Surface area of test tube, m^2
Cp:	Specific heat at constant pressure, J/kg K
f:	Friction factor
h:	Convective heat transfer coefficient, W/m^2 K

d: Core-rod diameter, m
D: Tube inner diameter, m
m Mass flow rate, kg/s
Q Heat transfer rate, W
Q Heat flux (W/m^2)
t Tape thickness, m
T Temperature, °C
U Mean velocity in tube, m/s

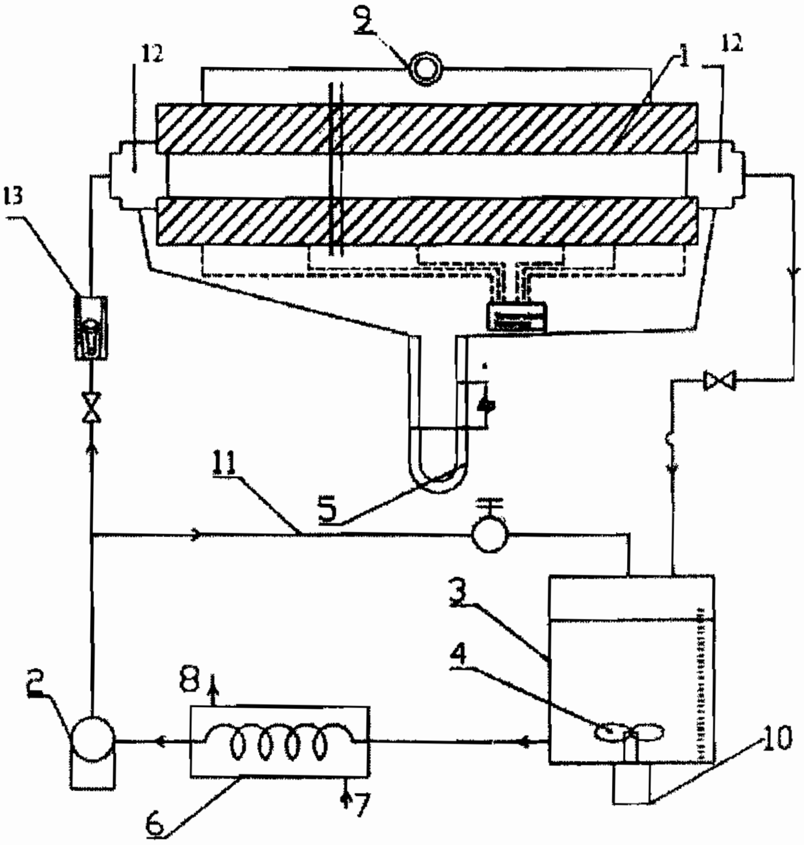
GREEK SYMBOLS

Φ: The nanoparticles volume fraction
ρ: Density, kg/m^3
η: Hydraulic thermal efficiency
μ: Dynamic viscosity, kg/m s
ε: Eccentricity.

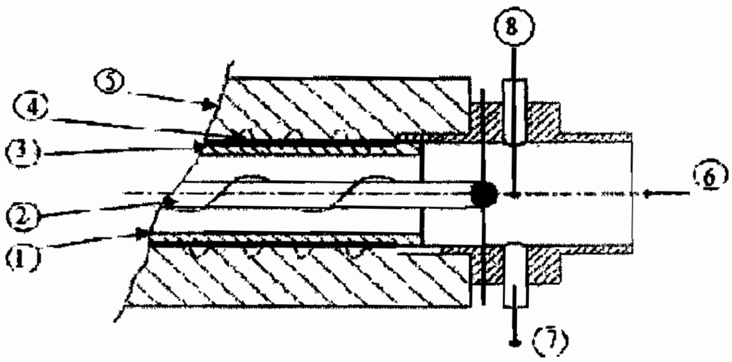
SUBSCRIBT

av Average
b Bulk
c Cold
h Hot
L: Tube length, m
i Inner
o Outer
w Water
p Plain tube
in Tube with helical tape around rod inserted
np Nano-particles
nf Nanofluid
w Tube wall
m Mean
t Total
us Useful
wf Working fluid

K: Thermal conductivity, W/m K



1	Tested tube
2	Circulating Pump
3	Working Fluid tank
4	Paddle Wheel
5	U-Tube Manometer
6	Heat Exchanger
7	Inlet City Water
8	Outlet City Water
9	Variac
10	Fan Motor
11	Bypass line
12	Flow Meter
13	Temperature



1	Tested tube
2	Helical Tape insert
3	Mica Sheet
4	Electric Heater Coil
5	Insulation
6	Inlet Working Fluid
7	To Manometer
8	Temperature

Figure (2) Experimental Test loop and sectional view in test section and support and sectional view in test section and support

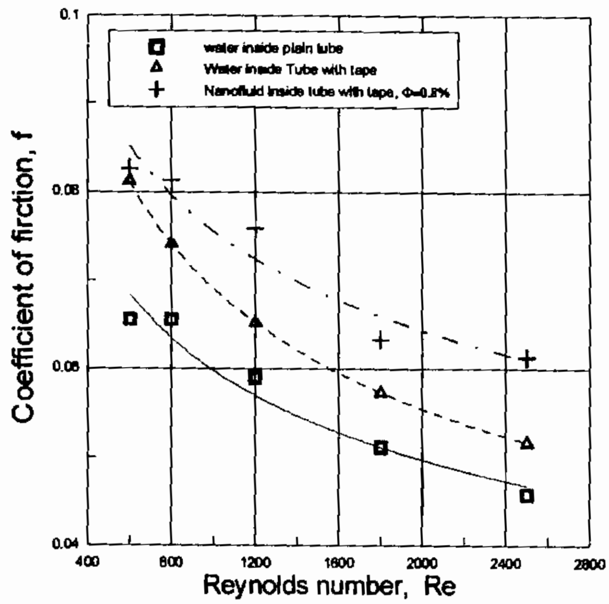


Fig. (3) Friction factor versus Reynolds number inside tube with inserting tap

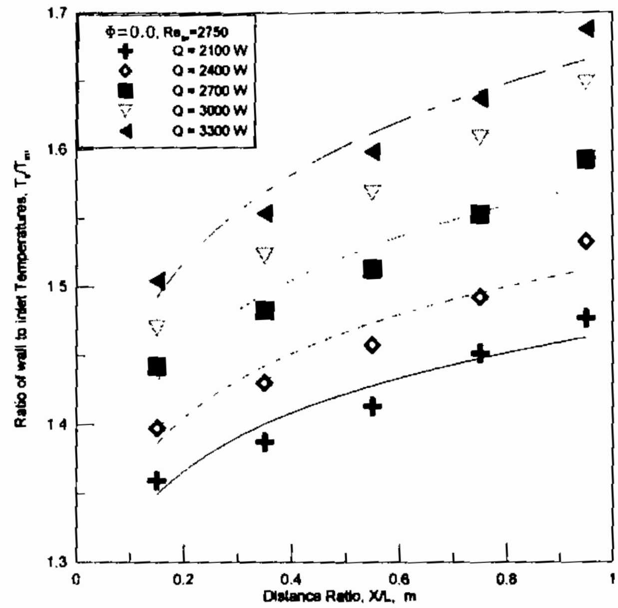


Fig. (4) Distribution of wall temperature along the tube length at different heat rates, for pure water

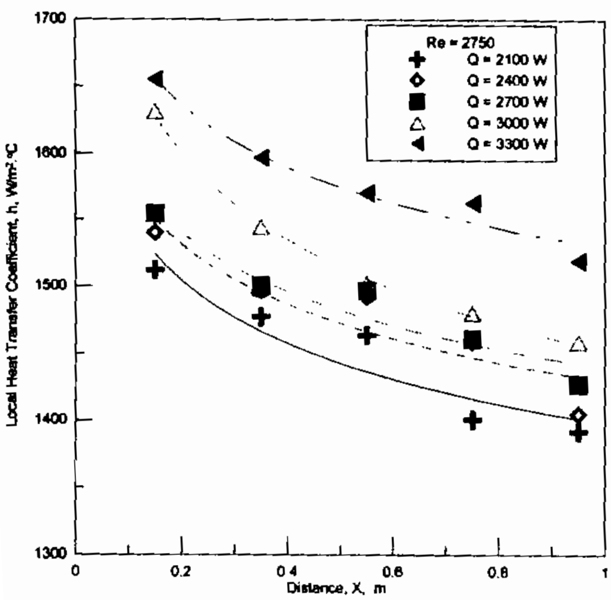


Fig. (5) Relation between Local heat transfer coefficient and axial tube length for different heat rates for water

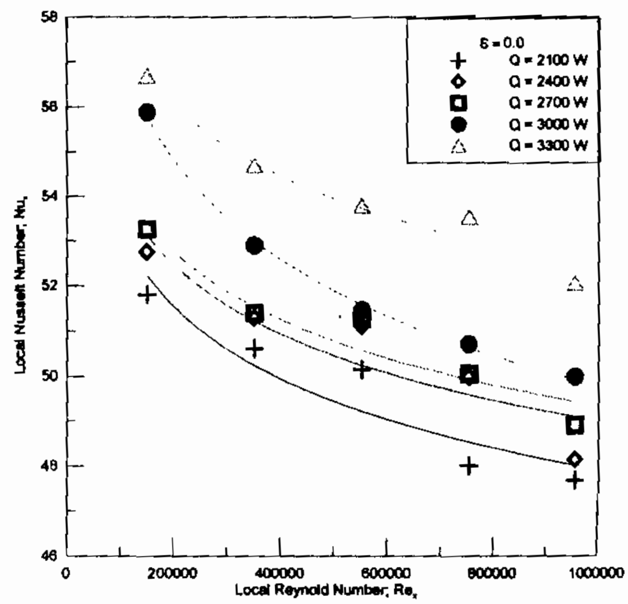


Fig. (6-a) Relation between Local Nusselt Number and Local Reynolds Number.

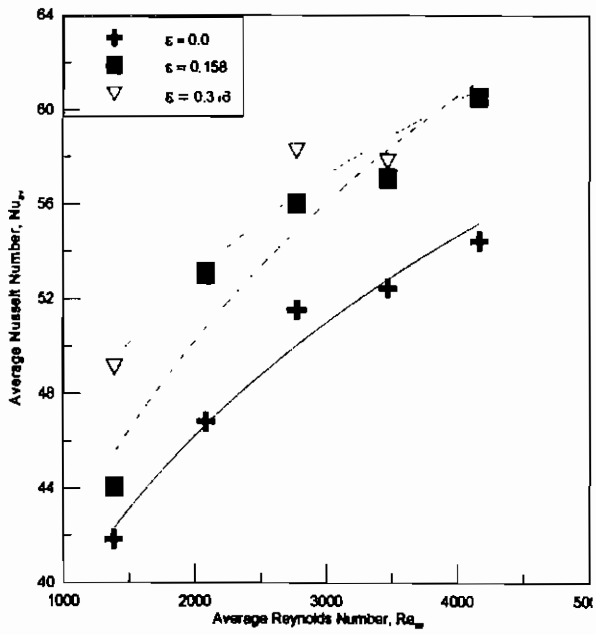


Fig. (6-b) Average Nusselt Number and Reynolds number at different Eccentricity without Nanoparticles

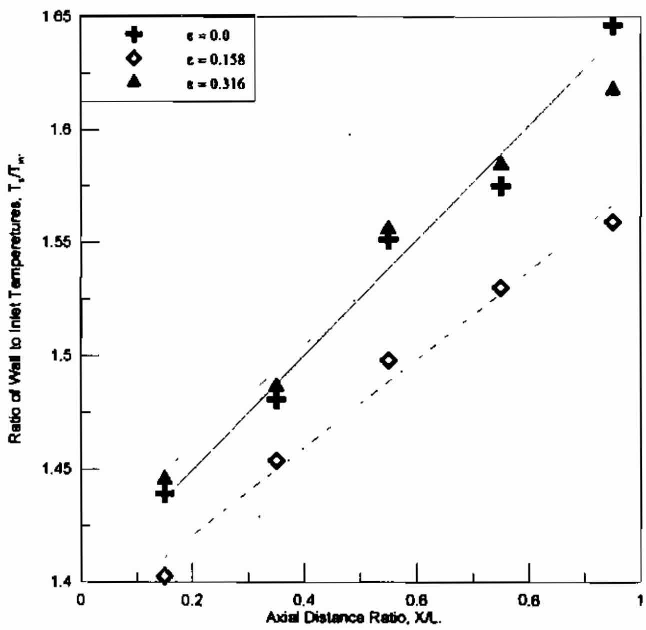


Fig. (7) Wall temperature distribution along the axial distance of the tube length at different eccentricities for pure water

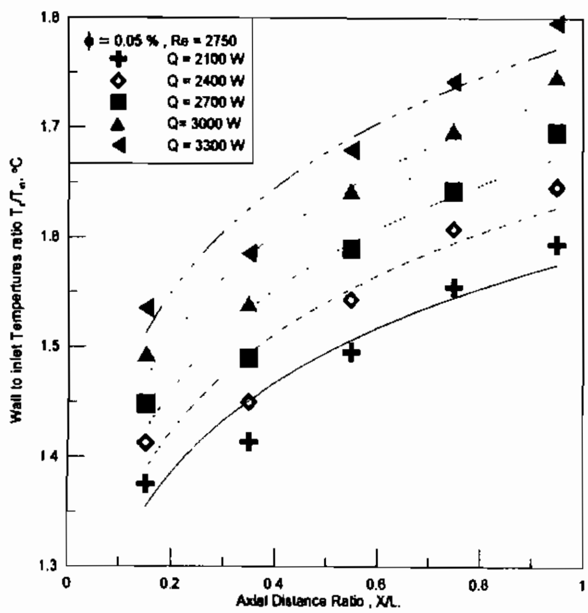


Fig. (8) Local surface Temperature change with the tube length with 0.05 % Nanoparticles at Re = 2750.

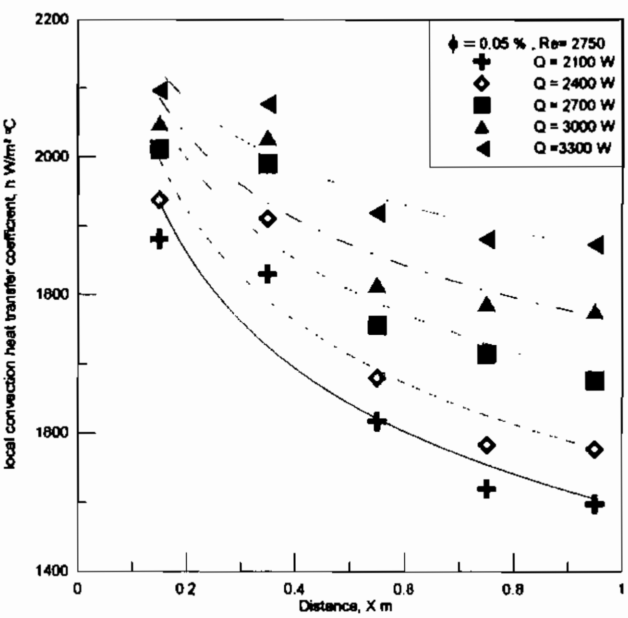


Fig. (9) Local heat transfer coefficient change with the tube length with 0.05 % Nanoparticles and Re = 2750.

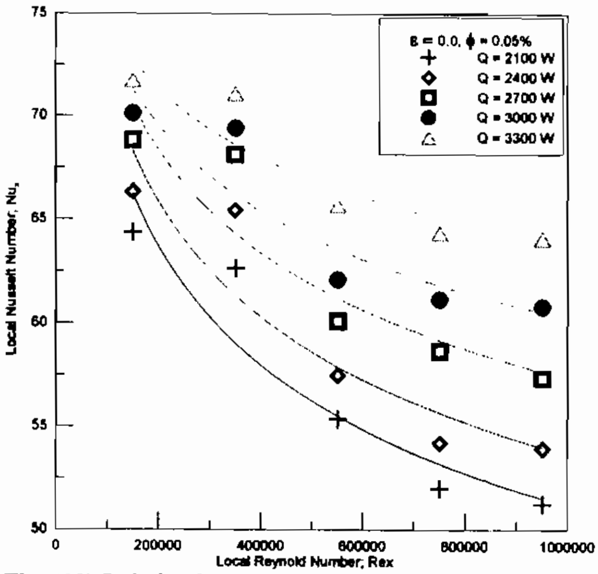


Fig. (10) Relation between local Nusselt number and Local Reynolds number at □ = 0.05%.

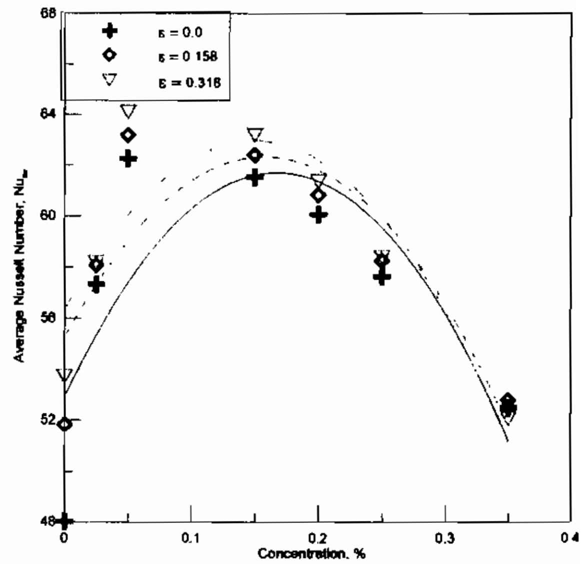


Fig. (11) Variation of average Nusselt Number and Concentration at different Eccentricities, and Re_{av} = 2750

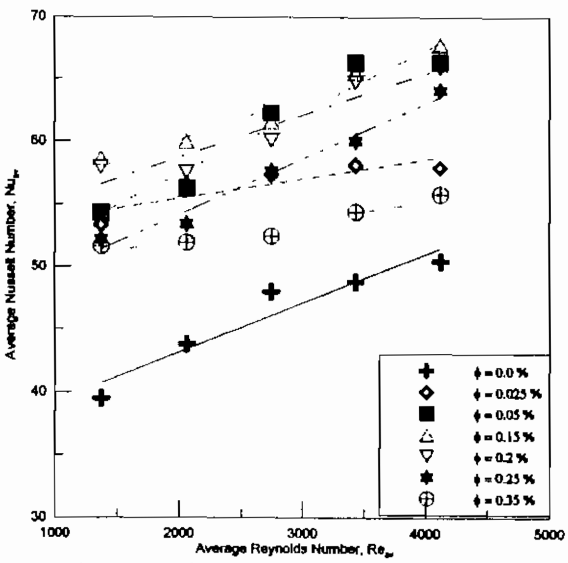


Fig. (12) Relation between Average Nusselt Number and Reynolds Number for different Concentrations ,ϕ

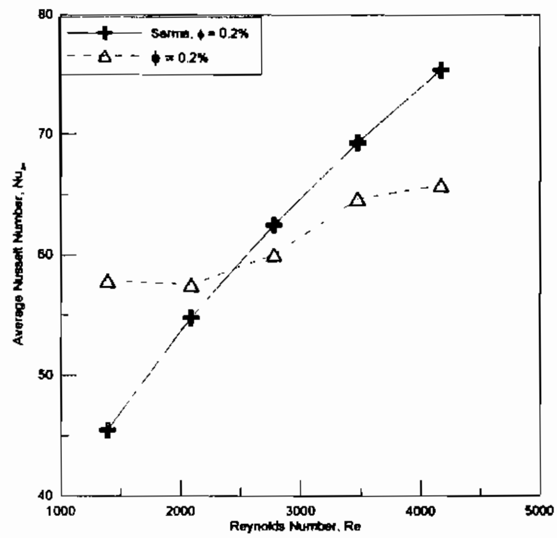


Fig. (13) Comparison between measured Nusselt Number and that calculated from Sarma equation at ϕ= 0.2% Nanofluid