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Research paper

An experimental study on pressuredrop of CNT/water nanofluid in a triple-tube heat exchanger

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Abstract

In the present study, the pressure drop of the nanofluid flow of carbon-water nanotubes (CNT/water) in a helical three-tube heat exchanger with constant fluid physical properties has been experimentally evaluated. For this purpose, first, the experimental device was designed and manufactured and then the carbon-water nanotube nanofluid with volume percentages of 0.01%, 0.1%, and 0.5% was prepared and stabilized. For the experiment, two triple-tube helical heat exchangers with different geometries are considered, in which the diameter of the middle pipe varies in two geometries. The pitch of the helical coil is 100mm and the helix radius is 9.235mm. The experiment was performed on Dean numbers between 1000 and 5000. The measured and calculated data were according to the available correlation in the literature with an error of less than 4%. It is found that at low volumetric percentages of CNT, the pressure drop is almost equal to that of the base fluid, and with increasing volumetric percentage of nanoparticles, the pressure drop also increases. By changing the geometry of the tube (decreasing the middle diameter of the tube), the pressure drop decreases.

1. Introduction

Study on heat transfer improvement methods is an ongoing process due to its extent. Researchers often seek to introduce new geometries or modify the working fluid thermophysical characteristics to increase the heat transfer coefficient, contact surface, and flow mixing rate while minimizing pressure drop. Therefore, studies on heat exchangers such as helical tubes as well as nanofluids are increasing. Most of these studies have focused on the effects of these methods on heat transfer, however, their effect on pressure drop has not been investigated.

The curvature of helical tubes

Helical tubes have a higher rate of heat and momentum transfer than straight tubes due to their curvature. As a result, the size of the heat exchanger becomes smaller and the heat transfer coefficient increases. When the fluid flows inside the tube in a curved path, centrifugal force is generated due to the curvature of the tube. The secondary motion created by this force has the effective capability to increase the rate of the heat exchange. Zamani *et al.* [1] investigated the role of the geometric and flow parameters such as hot and cold flow rates, the pitch of the helical, and the existence of fins and baffles, in order to increase heat transfer in a helical shell and tube heat exchanger. They showed that as the helical tube pitch increases, the flow velocity decreases, and the heat transfer rate increases. Also, the use of baffles causes an increase of 8.3% in the Nusselt number inside the pipe. In another theoretical study, Nazari *et al.* [2] experimentally investigated temperature variation in a cylindrical tank cooled by a helical coil. Their results show a 42% reduction in tank temperature using a helical tube compared to a straight tube. Pordanjani *et al.* [3] examined shell and tube heat exchangers with ordinary pipes and compared them with helical ones and concluded that the heat transfer rate is improved due to the curved path of the fluid inside the helical tubes. Seyyedvalilu and Ranjbar[4] investigated the effect of geometric parameters on heat transfer and hydrodynamic properties of the helical tube heat exchangers. Their study numerically investigates the effect of various parameters such as coil radius, coil pitch, and pipe diameter on the hydrodynamics and heat transfer properties of the helical double-tube exchangers by computational fluid dynamics. Their results show that heat transfer enhancement occurs with increasing internal Dean Number, internal pipe diameter, and decreasing coil pitch. Nada *et al.* [5] experimentally investigated the heat transfer properties and pressure drop of a multi-tube helical coil heat exchanger. The purpose of their study was to investigate the effect of geometric parameters of the heat exchanger and fluid flow, such as the number of internal pipes, hydraulic diameter, Reynolds numbers, and inlet heat flux, on the performance of the heat exchanger. They tested different coils with a different number of internal pipes. Their result shows that coils with three inner tubes have more heat transfer rate than other coils.

One way to increase heat transfer is to use nanofluids instead of the conventional working fluid. Nanofluid is obtained by adding very fine solid particles in nanoscale sizes (about 1 to 100 nanometers) to the base fluid. Hosseinipour *et al.* [6] conducted an experimental study on the heat transfer behavior of forced convection and pressure drop of a CNT/water flowing through a horizontal circular tube under fixed wall heat flux boundary conditions. Their results show that by adding a very small amount of carbon nanotubes to water, the heat transfer coefficient significantly increased. Hwang *et al.* [7] measured the pressure drop and the heat transfer coefficient of Al_2O_3/water nanofluid with constant heat flux boundary conditions in a laminar flow regime. Experiments showed an 8% increase in heat transfer coefficient at a concentration of 0.03% compared to the conventional working water. Fotokian and Nasr [8] investigated heat transfer and pressure drop of a very dilute nanofluid CuO/water (less than 0.24% by volume) in a turbulent regime. Their measurements showed that adding small amounts of nano-sized particles to the base fluid significantly increased the heat transfer coefficient. Yu *et al.* [9] performed a laboratory study on flow behavior and heat transfer properties of a nanofluid that is produced by the addition of Al_2O_3 nanoparticles to a mixture of 45% by volume of ethylene glycol and 55% by volume of water. Their studies showed that the increase in heat transfer coefficient should not be attributed only to the increase in thermal conductivity. Nassan *et al.* [10] investigated the heat transfer properties of Al_2O_3/water and CuO/water nanofluids in a square cross-sectional channel under a laminar flow regime with uniform heat flux. Their results yielded heat transfer enhancement for both nanofluids compared to the base fluid. Demir *et al.* [11] investigated the problem of heat transfer in a double tube counterflow heat exchanger experimentally and numerically. Their experimental findings and simulations with Fluent software show an increase in the heat transfer coefficient for nanofluids with increasing nanoparticle concentration. They have taken TiO_2/water nanofluid in turbulent flow with constant heat flux. Pishkar and Ghasemi [12] numerically investigated the effect of nanoparticles on heat transfer in the horizontal channel. They concluded that the use of nanofluids results in better heat dissipation, and with increasing volume fraction of the nanoparticles, heat transfer increases, and this increase is greater at higher Reynolds numbers. Bozorgan *et al.* [13] used Al_2O_3 / Ethylene-Glycol nanofluid as a cooling fluid in a double-tube heat exchanger. They concluded that the higher increase in heat transfer was due to aluminum oxide rod-shaped nanoparticles, because the increase in thermal conductivity of the base fluid with cylindrical particles is more compared to spherical particles.

Due to the importance of using nanofluids as well as helical heat exchangers, the study of heat transfer in a hybrid heat transfer enchantment method is increasing. Nasiri *et al.* [14] investigated the heat transfer performance of Al_2O_3/water and TiO_2/water nanofluids inside the annular channel under a turbulent flow regime. According to their results, the heat transfer coefficient and Nusselt number of nanofluids are higher than the base fluid and this enhancement improved in a higher concentration of nanoparticles. Keyhani *et al.* [15] investigated the heat transfer and pressure drop of the turbulent flow of TiO_2/water nanofluid through a horizontal circular channel with a constant heat flux boundary condition. Their experimental results showed that the heat transfer coefficient increases with increasing the volume fraction of nanoparticles and remains constant with changing the Reynolds number. Harris *et al.* [16 and [17] experimentally investigated the convective heat transfer of a laminar nanofluid stream of Al_2O_3/water into a circular tube at a constant wall temperature. Their results showed that the heat transfer coefficient increases with increasing concentration of nanoparticles in the nanofluid. They also found that the increase in thermal conductivity is not only a potential reason for the increased heat transfer, but it is one of the most important factors. Kumar and Palanisami [18] investigated the heat transfer and pressure drop in a helical shell and tube heat exchanger with nanofluids experimentally. A large difference in Nusselt number in laminar and turbulent flow is observed in their experiments.

Ding *et al.* [19] investigated CNT/water nanofluids flow at the entrance region of the pipe and at different Reynolds numbers in the range of 800<Re<1200. They observed more than 350% increase in heat transfer coefficients at Re=800 for 0.5% by weight of multi-walled CNT. Wang *et al.* [20] experimentally measured the convection heat transfer and pressure drop of CNT/water laminar flow. They reported an increase in heat transfer of 70% and 90% for volumetric concentrations of 0.05% and 0.25% at Reynolds number of 120, respectively; while the increase in thermal conductivity was less than 10%. Ranjbarzadeh *et al.* [21] experimentally investigated the effect of GO/water nanofluid on heat transfer and pressure drop in a copper tube with cross-air flow outside. The experiment was conducted in a wind tunnel. The effect of different concentrations of GO/water nanofluids (0%, 0.05%, 0.1%, and 0.2% by volume) on different Reynolds numbers in a cross-flow tube in the wind tunnel has been evaluated. Their results showed that GO/water nanofluid has a 51% and 21% higher Nusselt number and friction factor in comparison to pure water. According to the results, this nanofluid can be a good alternative in applied equipment such as heat exchangers. Babita *et al.* [22] performed an experimental study of CNT/water nanofluid in a horizontal helical heat exchanger with the coil-to-tube diameter ratio varying between 11.71 to 27.34. The volume percentage of nanofluid varied from 0.003% to 0.051%. From the experimental data, they concluded that the coefficient of friction in the helical tube is larger than the straight one and the concentration of CNT in the nanofluid has a significant impact on the pressure drop.

Singh *et al.* [23] studied the pressure drop and rate of heat transfer in helical exchangers with CNT/water nanofluid. Their experimental observation indicates that the heat transfer rate of nanofluid is 62.62% more than the base fluid at Re=50. They also concluded that the pressure drop would increase if the Reynold number increases. Omri *et al.* [24] presented a novel microchannel heat exchanger configuration based on CNT/water nanofluid by applying constant wall temperature boundary conditions. It was found that the performances of the heat exchanger are significatively improved using the CNT nanofluid and the triangular fins. Anitha and Pichumani [25] studied the rate of heat transfer in a conventional double-tube heat exchanger using CNT/water nanofluid. The effectiveness, pumping power, and pressure drop of the whole heat exchanger are investigated and reported in this work. They showed that the overall heat transfer coefficient of the heat exchanger is improved by 25% with the usage of Newtonian hybrid nanofluid.

In the present study, the pressure drop in a CNT/water nanofluid flow inside a triple-tube helical coil heat exchanger is investigated experimentally. Nanofluid flows through the tubes and pressure drop is measured. The volume percentage of the nanofluid was selected to be 0.01%, 0.1% and 0.5. The copper helical tube has two turns with a pitch of 100mm and a radius of 235.9mm. The test is performed under the boundary conditions of constant heat flux and constant physical properties of the fluid. The nanofluid temperature is considered to be 25°C.

2. Method and materials

2.1. Nanofluid preparation

Carbon nanotubes with outer diameter of 10-30 nanometers, length of 10 micrometers and purity of 99% have been used to prepare CNT/water nanofluid. The morphology characteristics of nanoparticles are tested by a Transmission Electron Microscope (TEM). Fig. 1 depicts the hollow structure of the CNT. In order to produce nanofluid, first, a certain amount of CNT nanoparticles is added to mixture of water and surfactant. For each liter of mixture 25 grams of CNT nanoparticles is added to produce a nanofluid with 2.5% weight concentration. Then a magnetic blender is used for half an hour to change the suspension to a solution. After that, the same amount of CNT would be added to the solution (to produce a nanofluid with 5% weight concentration) and the blending process goes on for 40 minutes. By means of an ultrasonic device, the uniform dispersion of the nanoparticles is achieved. The samples prepared with this method would remain stable for some months. In order to use lower concentrations, a mixture of water and surfactant is added.
The density and specific heat of the final nanofluid could be calculated according to the following relationships which are presented by Choi correlations [26]:

$$\rho_{nf} = \varphi\rho_p + (1-\varphi)\rho_f \tag{1}$$

$$C_{nf} = \frac{\varphi\rho_p C_p + (1-\varphi)\rho_f C_{pf}}{\rho_{nf}} \tag{2}$$

where φ is the CNT concentration, ρ is density, and C is the specific heat. The indices nf, p, and f are for nanofluid, nanoparticles, and base fluid, respectively. The Brinkman correlation is used to estimate the viscosity of the nanofluid [27]:

$$\mu_{nf} = \frac{1}{(1-\varphi)^{2.5}}\mu_f \tag{3}$$

where μ is viscosity.

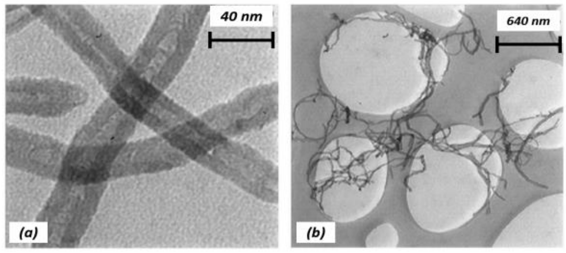

Fig. 1. TEM photographs of CNT used in this study; (a) hollow structure of CNT and (b) longitudinal size of CNT.

2.2. Helical tube fabrication

Important geometrical parameters in a helical tube are tube diameter (d), coil diameter (D), and coil pitch (b). Two non-dimensional variables of δ and λ for the ratio between tube and coil diameters and the ratio between pitch and length of a single turn of helix could be introduced, respectively.

$$\delta = d/D \tag{4}$$

$$\lambda = b/\pi D \tag{5}$$

Straight copper tubes of different sizes are used to fabricate the tripe-tube helical coil. First, a direct concentric triple-tube is made by entering the small tubes into the larger ones. Then, all tubes are filled with sand to prevent cracks during the bending process. The tubes are then wrapped around a compressed Teflon module on which the pitch length and a coil diameter of the pipe in question have already been machined. After altering the straight tubes to the helical ones, the whole process is washed carefully. The photographs of the made triple-tubes with their Teflon modules are given in Fig. 2. Refractory Teflon pieces are also placed between the turns of the helical tubes, because the helical tubes may not be able to maintain their pitches over time due to their special structure and external forces. Fig. 3 depicts the drawing of the resulted triple-tube helical heat exchanger.
In this study two different helical triple-tubes are fabricated. The geometrical properties of each case are presented in Table 1.

2.3. Test section and other components

To study the pressure drop of the CNT/water nanofluid flow in the helical triple-tubes under constant wall temperature conditions, a test system according to Fig. 4 has been designed and manipulated. In this system, a steam bath is used to apply the constant temperature boundary condition of the tube wall. The system consists of various parts such as the nanofluid reservoir, pump, test section, cooling heat exchanger, a U-tube manometer, and a flow meter. To circulate the nanofluid in the circuit, a centrifugal pump is used, which is coupled to a single-phase electric motor with a power of 0.5 hp. This pump has a nominal capacity of pumping fluid with a flow rate of 40 lit⁄min. The test section consists of pipes, U-shaped manometers, thermocouples, and steam bath tanks. To measure the flow pressure drop, from the beginning and the end of the test section, two branches perpendicular to the main flow path are connected to a U-shaped manometer. As mentioned, steam was used to apply the constant temperature boundary condition. To produce steam, a galvanized rectangular cube tank 106 x 40 x 30 cm with a thickness of 1 mm was used on the test tube. The tank walls were insulated to prevent energy wastage through the tank walls. At the bottom of this tank, there are 5 heating elements, each has a power of 2000 watts. To start, all heating elements would be turned on. After a while, when water starts boiling, 3 outer heating volumes would be turned off to achieve a stable condition. The inlet stream enters the test section into a spiral tube, which is completely surrounded by steam to provide a boundary condition for a constant surface temperature. A U-shaped manometer is used to measure the pressure difference between the two sides of the test tube. In this flowmeter, the fluid flow passes vertically, from bottom to top, through a transparent conical tube. The measuring range of this flow meter is 2-18 liters per minute and its maximum measurement error is 4%. A five-liter plastic graduated container is used as a nanofluid storage tank.

2.4. Experiments

To measure different parameters in the experiment, the system must reach a stable state. For this purpose, we first fill the tank of the test section and the feed tank of the circuit with nanofluid. By turning on the heating elements, we wait for the water to boil and the resulting steam to create a constant wall temperature condition for the test tube. After the water boils, it is enough for two of the heating elements to be on to maintain a stable condition. Then, by operating the pump and flowing the nanofluid in the circuit using two built-in valves, one in the return line to the tank and the other in the main line, and before entering the test section, we adjust the desired flow rate in the circuit.

Fig. 2. Triple-tube helical heat exchanger with their Teflon module.

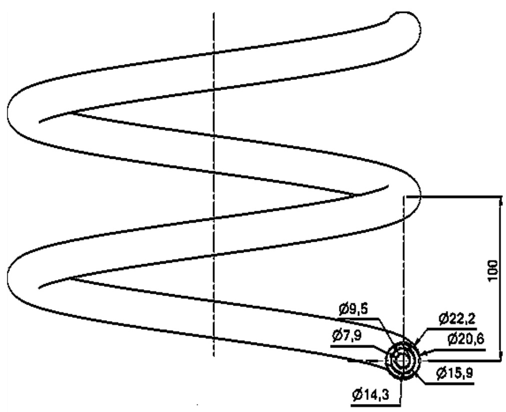

Fig. 3. Triple-tube helical heat exchanger with their Teflon module (drawing).

Table 1. Geometrical properties of triple-tube helical heat exchangers.

	Pitch (mm)	Coil diameter (mm)	Inner tube diameter (mm)	Middle tube diameter (mm)	outer tube diameter (mm)
Case I	100	235.9	7.9	14.3	20.6
			9.5	15.9	22.2
Case II	100	235.9	4.8	14.3	20.6
			6.4	15.9	22.2

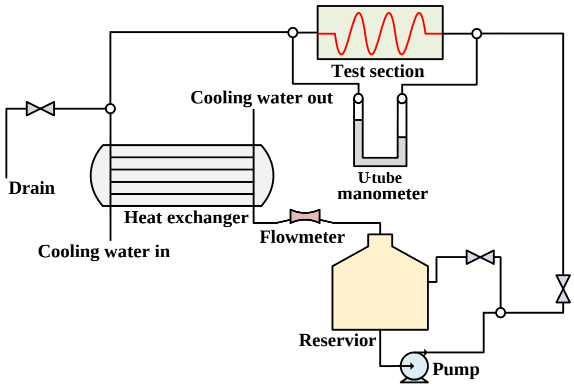

Fig. 4. Test system components.

After that, with the flow of coolant in the cooler, which is the same as ordinary city water, all the preparations for the start of the test will be provided. After performing the above steps, the system should work for a while to reach a stable state. Steady state is established when significant changes in the amount of pressure drop are measured and the average temperatures of the fluid and the pipe wall do not occur. Repeated experiments showed that after adjusting each flow, the time required for the system to reach a steady state is 30 minutes.

Experiments were performed for pure water and different concentrations of nanofluids at different flow rates and different patterns of spiral tubes. It was repeated at least once for each test sample. It should be noted that after performing all the tests related to each sample, the corresponding nanofluid is discharged from the system for stabilization. To prevent the remaining nanoparticles from being impregnated in the test cycle the next time, after the tests are completed, the cycle path is flushed with water and then a strong pressure wind is used to completely drain the water from the system. To measure the flow pressure drop, the difference between the inlet and outlet pressure of the test tube can be calculated by measuring the difference in the liquid level of the two sides of the U-shaped manometer. The pressure drop in each test is measured three times at the beginning, middle, and end of the test to ensure its accuracy. To use the device, the correctness of its performance is first evaluated. For this purpose, in the nanofluid tank, distilled water is first poured and flows in the direction of the helical tube. By collecting the information generated from the experiment and also calculating the experimental Nusselt number with the energy balance and comparing it with the information generated from the common Nusselt number estimation relationships, the accuracy of the performance is ensured.

2.5. Experiment errors

Estimated errors in experimental works, according to their origin, are grouped into two general categories: definite errors and indefinite errors. Certain errors are those that have definite values and can be measured and calculated. Normally in this case we create an adjustment curve (calibration) and we take the number from the experiment done on it and get its real value. Indeterminate errors are caused by the expansion of a measurement system into its maximum. These errors cannot be well identified and their definite value can be determined by measuring them; instead, these errors fluctuate randomly. It is not possible to list all sources for a given error, but it can be determined that the source of these errors is the personal errors of the tester, the device error in the instrument used, and the errors resulting from the test method or other combinations of them. In order to reduce certain errors in the experiments, in this study, the experiments were performed at least twice with the most accurate laboratory equipment.

3. Results and discussion

3.1. Certification of results

The thermophysical properties of nanofluids with different CNT nanoparticle concentration are presented in Table 2. Two first rows are related to the properties of CNT and pure water at working temperature of the experiment.

Friction factor is calculated according to the Darcy-Weisbach relationship [28]:

$$f = \frac{2\,\Delta p\, d}{\rho\, u^2\, l} \tag{6}$$

where f is friction factor, Δp is the measured pressure drop, d is hydraulic diameter, ρ is density, u is average velocity, and l is the length of tube. Instead of Reynolds number in helical coils, the Dean number that is a modification on Reynolds number is important. The Dean number is [28-30]:

$$De = Re\sqrt{\frac{r}{R}} \tag{7}$$

where De is Dean Number, Re is Reynolds number, r is the radius of the tube, and R is the radius of the helix. The Reynolds number itself is:

$$Re = \frac{\rho u d}{\mu} \tag{8}$$

where μ is the viscosity of the fluid.
In order to certify the result, the experiment is conducted first for pure water and then the result is compared to the available correlations in the literature. The critical Reynolds for the flow inside a helix is:

$$Re_{cr} = 20000\delta^{0.32} \tag{9}$$

where δ is the ratio between tube's radius and helix's radius. The friction factor for such a system is:

$$f\left[\frac{R}{r}\right]^{\frac{1}{2}} = 0.029 + 0.304[Re\left(\frac{r}{R}\right)^{2}]^{\frac{1}{4}} \tag{10}$$

Fig. 5 illustrates the variation of friction factor vs Dean number for the current experimental study and correlation 10 which was according to the literature reference [30]. As this figure depicts, there is a very good match between the tested samples and the reference data. As Dean number increases, the friction factor decreases and the maximum value of the f occurs at Dean Number of 1000. This figure also shows the difference (error) between the results of the current study and the corresponding reference data. As the Dean number increases, the percentage of test error relative to the reference increases. The highest percentage of error is related to the Dean number of 5000 and the lowest percentage of error is related to the Dean number of 1000. The difference between the highest and lowest error is 1.5%. This diagram shows that the experiments are in good agreement with the reference correlation.

3.2. Effect of different geometries

As stated below, two different geometries are tested in this study. The dimension of each case is presented in Table 1. Fig. 6 shows the friction factor of water in terms of Dean Number in the first and second geometry of the triple-tubes. As can be seen, as the Dean number increases, the friction factor decreases. In case I, this reduction rate is uniform, but in case II, as Dean number increases from 1000 to 2000, the friction factor is reduced with a greater slope and for other Dean Numbers, it varies continuously.

3.3. Effect of CNT nanoparticle concentration

Table 3 shows the pressure drop of the nanofluid in terms of the Dean number in different volumetric percentages of the nanofluid and water in the first geometry of the triple-tubes (case I). As it can be seen, in volumetric percentages up to 0.1%, with increasing the Dean number, the pressure drop increases slightly, which is almost equal for water and nanofluids with a volume percentage of 0.01%. For nanofluids with a volume percentage of 0.1%, it increases slightly and its maximum value is at De=5000. At a volume percentage of 0.5%, with increasing the Dean number, the pressure-drop increases. Therefore, it can be concluded that with increasing the volume percentage of nanofluid, the pressure drop also increases. Table 4 presents the pressure drop of the nanofluid in terms of the Dean number in different volumetric percentages of the nanofluid and water in the second geometry of the triple-tubes (case II). As it can be seen, in volumetric percentages up to 0.1%, with increasing the Dean number, the pressure drop increases slightly, which is almost equal for water and nanofluids with a volume percentage of 0.01%.

Table 2. Density, viscosity, and specific heat of nanofluid with different nanoparticle concentration.

	ρ (kg/m^3)	μ (mPa.s)	c_p (W/m^2°C)
CNT	2100	-------	530
ϕ = 0.00	1024	1.080	4001
ϕ = 0.01	1035	1.107	3966
ϕ = 0.10	1132	1.405	3654
ϕ = 0.50	1562	6.109	2266

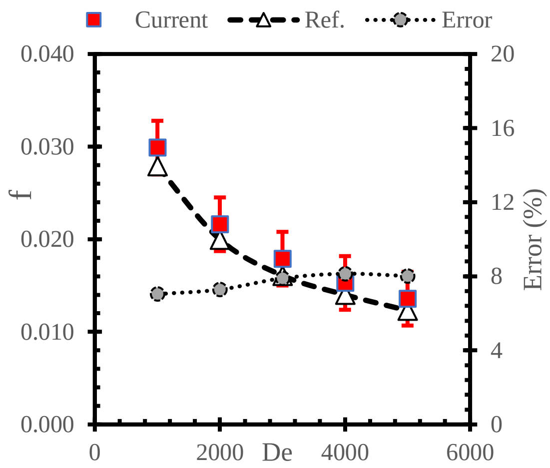

Fig. 5. Comparison between calculated friction factor (according to the measurement) and Ref. [30] correlation.

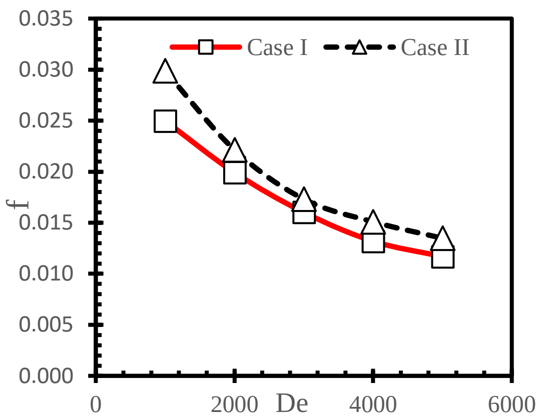

Fig. 6. Friction factor vs Dean number for different geometries**.**

Table 3. Measured pressure drop (Pa) for nanofluids with different nanoparticle concentration (geometry case I).

De	φ=0.00	φ =0.01	φ =0.1	φ =0.5
1000	60.7	78.4	127.0	1919.0
2000	194.4	222.5	360.4	5362.1
3000	349.9	409.6	670.2	9633.5
4000	513.2	627.0	1021.7	14740.7
5000	704.6	853.3	1396.3	20388.3

Table 4. Measured pressure drop (Pa) for nanofluids with different nanoparticle concentrations (geometry case II).

De	φ =0.00	φ =0.01	φ =0.1	φ =0.5
1000	16.3	20.4	33.1	473.3
2000	48.6	61.2	99.2	1420.0
3000	85.8	104.6	169.9	2449.0
4000	132.5	163.3	264.6	3786.6
5000	185.3	226.8	375.8	5144.9

For nanofluids with a volume percentage of 0.1%, it increases slightly and its maximum value is in the De=5000. At a volume percentage of 0.5%, with increasing the Dean number, the pressure-drop increases. Therefore, it can be concluded that with increasing the Dean number, the pressure-drop increases.

The addition of nanoparticles would result in a fluid with higher viscosity. Therefore, any increase in the nanoparticle concentration causes a higher pressure drop. However, the variation of the pressure drop vs nanoparticle concertation is not linear. For geometry in Case I, nanofluid with φ =0.01 has a pressure drop about 21% more than base water, nanofluid with φ =0.1 has a pressure drop about 97% more than water, and nanofluid with φ =0.5 has a pressure drop 2788% more than water. However, there is a small fluctuation in these values regarding the Dean number which is negligible and it could be concluded that the effect of nanoparticles on pressure drop enhancement does not depend on Dean number. For geometry in Case II: the values of pressure drop enhancement are a little bigger. There is 24%, 102%, and 2763% enhancement in the pressure drop in comparison to pure water when φ =0.01, φ =0.1, and φ =0.5 respectively.

Fig. 7 shows the coefficient of friction in terms of Dean number for water and nanofluids with different volume percentages in the first geometry of the pipe. As can be seen, with increasing volume percentage of nanofluid, the coefficient of friction increases but the amount of this increase is small. For water fluid, the coefficient of friction decreases with increasing number of Dean numbers, and for nanofluids, the coefficient of friction decreases with increasing Dean number, but this decrease is not uniform and continuous. Between 1000 and 2000, the coefficient of friction is reduced with a high slope. Between 2000 and 3000, this slope decreases, and between 3000 and 4000, the slope increases again, and between 4000 and 5000, the slope decreases again.

Fig. 8 shows the coefficient of friction in terms of Dean number for water and nanofluids with different volume percentages in the second geometry of the tube.

As can be seen, with the increasing volume percentage of nanofluid, the coefficient of friction increases but the amount of this increase is small.

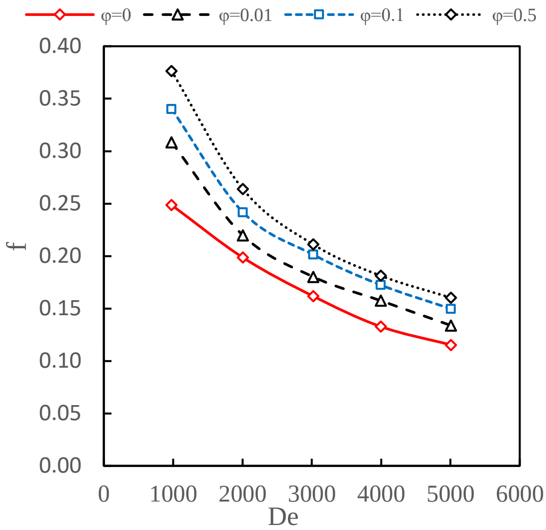

Fig. 7. Friction factor vs Dean number for nanofluid with different CNT concentration in geometry case I.

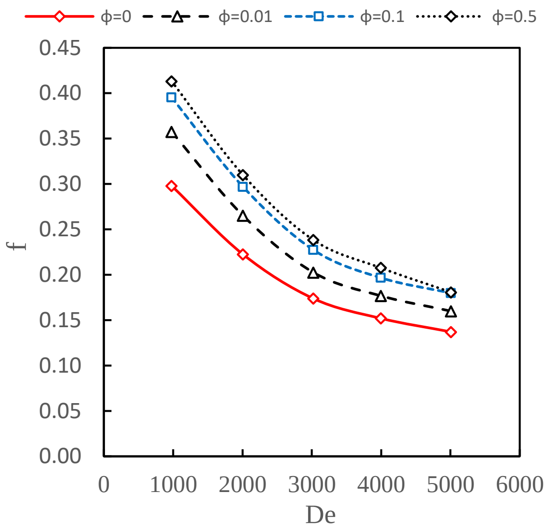

Fig. 8. Friction factor vs. Dean number for nanofluid with different CNT concentration in geometry case II.

For water fluid, the coefficient of friction decreases with increasing Dean number, and for nanofluids, the coefficient of friction decreases with increasing Dean number, but this decrease is not uniform and continuous. For nanofluids with different volume percentages, in Dean numbers between 1000 and 3000, the coefficient of friction decreases with more slope, and in Dean numbers between 3000 to 5000, the coefficient of friction decreases with less slope, and in volume percentages of 0.1% and 0.5%, the changes in the coefficient of friction in terms of the Dean numbers are very close to each other and almost equal, and in Dean number 5000, they are exactly the same.

Fig. 9 shows the coefficient of friction in terms of different volumetric percentages of nanofluids and water in different Dean numbers and the first geometry of the tubes. As can be seen, with increasing volume percentages of nanofluids, the coefficient of friction increases, but this increase is not continuous and uniform. From volumetric percentages of 0 to 0.01%, with high slope and in volumetric percentages of 0.01% to 0.1%, with lower slope and in volumetric percentages of 0.1% to 0.5%, with very low slope (approximately close to the straight line), the coefficient of friction increases. For water and nanofluids with different volume percentages, the coefficient of friction decreases with increasing number of Dean numbers. From 1000 to 2000, with a large distance and Dean numbers 3000 to 5000, with a shorter distance, the coefficient of friction decreases and the lower the volume percentages of nanofluid, the more the coefficient of friction decreases, so that for fluid and water number 5000 has the largest reduction in coefficient of friction.

Fig. 10 shows the coefficient of friction in terms of different volumetric percentages of nanofluid and water in different Dean numbers and the second geometry of the tubes. As can be seen, with increasing volume percentages of nanofluids, the coefficient of friction increases, but this increase is not continuous and uniform.

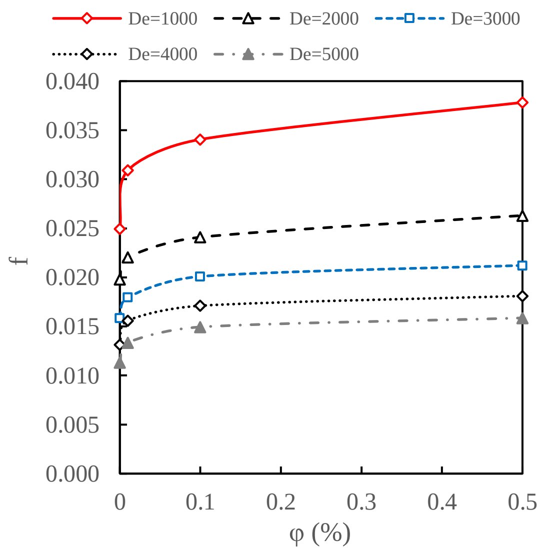

Fig. 9. Friction factor vs. nanofluid CNT concentration in geometry case I.

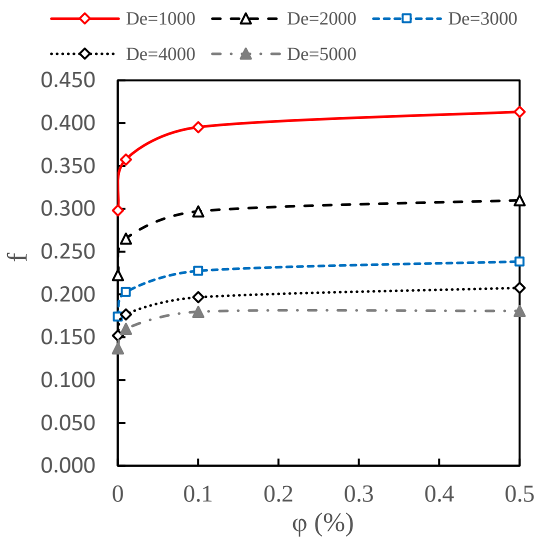

Fig. 10. Friction factor vs. nanofluid CNT concentration in geometry case II.

From volumetric percentages of 0 to 0.01%, with high slope and in volumetric percentages of 0.01% to 0.1%, with lower slope and in volumetric percentages of 0.1% to 0.5%, with very low slope (approximately close to the straight line), the coefficient of friction increases.
For water and nanofluids with different volume percentages, the coefficient of friction decreases with increasing Dean number. From 1000 to 2000, with a large distance, and Dean numbers 3000 to 5000, with a shorter distance, the coefficient of friction decreases, and the lower the volume percentages of nanofluid, the more the coefficient of friction decreases, so that for fluid and Dean number 5000 has the largest reduction in coefficient of friction.

4. Conclusions

Effect of CNT nanoparticle concentration on friction factor in a helical three-tube heat exchanger is investigated experimentally. The process of nanofluid preparation and experiment set-up is introduced. Two other factors are also examined which are the effect of heat exchanger geometry and the effect of mass flow rate (Reynolds number). A combination of these two parameters is presented by Dean number. From this experimental analysis, it is concluded that:

- with decreasing the diameter of the middle tube, the coefficient of friction increases, but its value is small.
- By reducing the middle diameter of the tube, the cross-sectional area of the nanofluid increases.
- As the cross-section of the nanofluid increases, the coefficient of friction increases with increasing Dean number, but this increase is not continuous and uniform.
- With increasing Dean number and volume percentage of nanofluid, pressure drop increases. In volume percentages less than 0.1%, it increases with low slope and in higher percentages, with more slope.
- The increase in pressure drops due to the addition of the nanoparticle do not depend on Dean number significantly. There is 24%, 102%, and 2763% enhancement in pressure drop in comparison to pure water when φ =0.01, φ =0.1, and φ =0.5 respectively

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