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Numerical modeling of nano-fluid flow inside a spiral micro-tube

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Abstract
In this study, numerical methods have been used to simulate the laminarflow of nanofluid inside the spiral micro-tube. Reynolds numbers range from 500 to 2500, helix diameters of 1, 1.5, 2 and 2.5 cm, volume fraction of CuO nanoparticles, 0%, 0.1%, 0.2%, 0.3% and 0.5%, and copper micro tube Whose wall temperature is assumed to be constant. The diameter of the micro-tube is 0.8 mm. In this research, the development of numerical model based on single-phase nanofluid model has been discussed. Then, the validation of the model and the independence of the solution from the mesh are examined. The results show that the use of spiral micro-tubes relative to straight tubes leads to improved heat transfer. The use of nanoparticles in the base fluid also improves the thermal performance of the system. As the diameter of the helix increases, the rate of heat transfer and the coefficient of friction decrease. For different modes, the performance evaluation criterion (PEC) was calculated and compared. It has been shown that a 1 cm diameter micro-tube with 0.05% water-CuO nanofluid has the best thermal performance.
Keywords:*piral micro-tube1, water-CuO nanofluid2, Coil diameter3, , Numerical modeling4*

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I. Introduction

In micro-technology, tubes or ducts with an inner diameter of less than 1 mm are very important due to their small size and good thermal performance. These tubes or channels are called micro-tubes(MT) or micro-channels(MC). Due to the significant advances that have been made in the technology of micro-systems in the last decade, the cooling of macro-electromechanical systems has become a challenge. As a result, many studies have looked at improving cooling and increasing heat transfer in micro-equipment. On the other hand, common cooling fluids such as water have weak thermo physical and thermal properties, which can be overcome by adding metal oxide nanoparticles. In fact, by adding nanoparticles of metal oxide to water fluid, a new type of fluid is obtained, called nanofluid. Nano-fluid has improved thermal properties compared to conventional fluids and thus increases heat transfer [1-12]. Concomitant use of nanofluids and micro-tubes leads to a double increase in heat transfer, and as a result, cooling operations are performed more efficiently. Therefore, a lot of research has been done in this field. Due to the specific behaviour of nanofluid flow, especially inside micro-tubes, many researchers have conducted experimental studies. However, experimental research is often associated with limitations such as high costs, limited measurement of all parameters, time consuming, non-repetition of experiments in the desired number, and so on. With the increasing computational power of computers, the use of numerical modelling methods instead of experimental methods became very common. In this research, it is tried to examine the flow field and temperature of nanofluid inside the helical tube by using numerical modelling method.

Many experimental studies have examined the behavior of nanofluids [13-25]. Hwang et al. [6] performed an experimental study of pressure drop and convective heat transfer in the slow flow of water-aluminum oxide nanofluid in a tube under constant heating flux. They attributed the impact of nanoparticle migration to viscosity gradient, Brownian diffusion, and the increase in convective heat transfer of nanofluid.

Lelea and Nisulescu [26] numerically examined the slow flow and heat transfer of water-aluminum oxide nanofluid inside the micro-tube. They examined the effect of viscous dissipation. They reported that, unlike the macro-scale study, which saw the highest increase in heat transfer in the inlet section, the most heat transfer occurs in micro-tubes near the wall, which is caused by viscous heating. Salman et al [27] investigated the convection heat transfer for different types of nanofluid with ethylene glycol base fluid with nanoparticles in different dimensions and different volume fractions in two dimensions in micro-tubes. Their studies showed that SiO_2 nanoparticles had the best thermal performance compared to other metal oxide nanoparticles. This was also experimentally investigated by Khoshvaght-Aliabadi et al [28]. The effect of micro-tube length on convective heat transfer during turbulent flow was investigated by Minea [29]. They found that as the length of the micro-

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tube increased, so did the heat transfer flux. Salman et al [30] compared the experimental and numerical comparison of water-Al_2O_3 and water-SiO_2 nanofluids within a micro-tube. They showed that if the nanofluid was used, the maximum heat transfer rate would increase by approximately 22%. Dean [32,31] stated that in terms of heat transfer rate, spiral tubes perform better than conventional straight tubes because very strong secondary flows are formed in the spiral tubes, which makes them vortex. They are called. The Vortex later became known as the Dean Vortex. Dean found that the secondary flows in the coil tubes are a function of the Reynolds dimensionless number as well as the d / D ratio. Dean [32] examined the flow of incompressible and steady state fluid inside the helical tube. Mohammed et al [33] numerically examined the effect of geometric parameters of spiral tube heat exchanger with nanofluid on thermal and hydraulic characteristics. The parameters examined by them included the helix radius and the inner diameter of the tube under laminar flow condition. They found that the counter flow configuration performed better than the parallel flow.

In summarizing the above literature review, the number of studies related to MTs using nanofluids is very limited. On the other hand, curved geometries of the MTs like helical coil have not been examined in the past, when a nanofluid has been applied as working media. Accordingly, this motivates this investigation. Thus, the main objective of this study is to characterize the water based nanofluid containing copper nanoparticles across the helical micro tubes (HMTs). The nanofluids are prepared in three weight concentrations (0.0%, 0.1%,0.2%, 0.3% and 0.5%) and they are tested in a highly precise test loop with the ability to produce a constant wall temperature. The wall of the Tube is considered at a constant temperature. A single-phase model is also used to simulate nanofluid flow.

II. Materials and methods

As mentioned earlier, the nanofluid used is water-CuO. The specifications of water-based fluid and CuO nanoparticles are shown in Table (1).

Table1. Thermophysical properties of water and copper oxide at $20^{o}C$[38]

	$\rho\left(\frac{kg}{m^3}\right)$	$C_p\left(\frac{J}{kg.K}\right)$	$k\left(\frac{W}{m.K}\right)$	$\beta\times10^{-5}\left(\frac{1}{K}\right)$	$\mu\left(\frac{N.S}{m^2}\right)$
Base fluid (water)	997.1	4179	0.613	21	1.003×10^{-3}
Nanoparticle (CuO)	6500	540	18	0.85	-

In this study, a steady, two-dimensional flow is assumed. The effects of radiation and viscose heat loss are also neglected. The flow for the Reynolds number range is 500 to 2500invistaged. The k-ε model is used to model turbulence. Nanofluid is considered to be a continuous medium with thermal equilibrium between the base fluid (water) and the solid spherical nanoparticles (CuO). The flow of Newtonian fluid is considered to be incompressible. The length of the tube is assumed to be 60 cm, which is always more than developing length, and the Helical coil diameter is assumed to be 1, 1.5, 2 and 2.5 cm. The inside diameter of the tube is 0.8 mm.

1. Governing equations

Navier-Stokes equations are used to model the flow inside a micro-tube. Thus, the governing equations are the same as the equations of continuity, momentum, and energy in cylindrical coordinates. Explain that these equations are defined for a nanofluid.

Continuity equation:

$$\frac{1}{r}\frac{\partial(rv)}{\partial r}+\frac{\partial u}{\partial x}=0 \quad (1)$$

Momentum equation in x direction

$$\rho_{nf}\left[u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial r}\right]=-\frac{\partial p}{\partial x}+\mu_{nf}\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right)+\frac{\partial^2 u}{\partial x^2}\right] \quad (2)$$

Momentum equation in r direction

$$\rho_{nf}\left[u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial r}\right]=-\frac{\partial p}{\partial r}+\mu_{nf}\left[\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(rv)\right)+\frac{\partial^2 v}{\partial x^2}\right] \quad (3)$$

Energy equation

$$\left(\rho C_p\right)_{nf}\left[\frac{\partial(uT)}{\partial x}+\frac{1}{r}\frac{\partial(rvT)}{\partial r}\right]=k_{nf}\left[\frac{\partial^2 T}{\partial x^2}+\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)\right] \quad (4)$$

By defining the dimensionless variables below

$$u^*=\frac{u}{u_c},\quad v^*=\frac{v}{u_c},\quad R^*=\frac{r}{R},\quad x^*=\frac{x}{R\,Re\,Pr},\quad P^*=\frac{p}{\rho_{nf}u_c^2},\quad Pr=\frac{(\mu C_p)_f}{k_f},\quad Re=\frac{\rho_f u_c D}{\mu_f},\quad \theta=\frac{T-T_c}{T_{h1}-T_c} \quad (5)$$

In the above relations, u_c indicates the fluid velocity at the inlet of the micro-tube, R is the radius of the micro-tube and D is the diameter of the micro-tube, which is twice the radius. T_c is the temperature of the input nanofluid and T_{h1} is the temperature of the hot wall of the micro-tube. By applying the introduced dimensionless parameters, the dimensionless equations are obtained as follows:

The dimensionless momentum equation in x direction

$$\frac{\partial}{\partial x^*}\left[u^*u^* - \left(\frac{\mu_{nf}}{\mu_f}\frac{\rho_f}{\rho_{nf}}\frac{2}{Re^2 Pr}\frac{\partial u^*}{\partial x^*}\right)\right] + Re.Pr\frac{1}{R^*}\frac{\partial}{\partial R^*}\left[R^*\left(u^*v^* - \left(\frac{\mu_{nf}}{\mu_f}\frac{\rho_f}{\rho_{nf}}\frac{2}{Re}\frac{\partial u^*}{\partial R^*}\right)\right)\right] = -\frac{\partial P^*}{\partial x^*} \quad (6)$$

The dimensionless momentum equation in r direction

$$\frac{1}{Re.Pr}\frac{\partial}{\partial x^*}\left[u^*v^* - \left(\frac{\mu_{nf}}{\mu_f}\frac{\rho_f}{\rho_{nf}}\frac{2}{Re^2 Pr}\frac{\partial v^*}{\partial x^*}\right)\right] + \frac{1}{R^*}\frac{\partial}{\partial R^*}\left[R^*\left(v^*v^* - \left(\frac{\mu_{nf}}{\mu_f}\frac{\rho_f}{\rho_{nf}}\frac{2}{Re}\frac{\partial v^*}{\partial R^*}\right)\right)\right] = -\frac{\partial P^*}{\partial R^*} - \frac{\mu_{nf}}{\mu_f}\frac{\rho_f}{\rho_{nf}}\frac{2}{Re}\frac{v^*}{R^{*2}} \quad (7)$$

Dimensionless energy equation

$$\frac{\partial y}{\partial x^*}\left[u^*\theta - \left(\alpha_{nf}\frac{\rho_f}{\mu_f}\frac{2}{Re.Pr}\frac{\partial \theta}{\partial x^*}\right)\right] + Re.\mathrm{Pr}\frac{1}{R^*}\frac{\partial}{\partial R^*}\left[R^*\left(v^*\theta - \left(\alpha_{nf}\frac{\rho_f}{\mu_f}\frac{2}{Re}\frac{\partial \theta}{\partial R^*}\right)\right)\right] = 0 \quad (8)$$

III. Boundary conditions

Due to the uniformity of the input speed and temperature, the boundary conditions at the input of the micro-tube are calculated as follows:

$u^* = 1, \quad v^* = 0, \quad \theta = 0$(9)

For heat transfer that is applied linearly to the wall of the micro-tube:

$$u^* = v^* = \frac{\partial\theta}{\partial R^*} = 0 \qquad 0 < x^* < \frac{l_1}{R\ Re\ Pr}, R^* = 1$$

$$u^* = v^* = 0,\ \theta = 1 + G\ Re(x^* - L_1)\ \frac{l_1}{R\ Re\ Pr} < x^* < \frac{l_1 + l_2}{R\ Re\ Pr}, R^* = 1 \quad (10)$$

$$u^* = v^* = \frac{\partial\theta}{\partial R^*} = 0 \qquad \frac{l_1 + l_2}{R\ Re\ Pr} < x^* < \frac{L}{R\ Re\ Pr}, R^* = 1$$

In the above relations, the parameter G indicates the changes in the surface temperature gradient, which are defined as $G = \frac{A\,\mathrm{Pr}\,R}{T_{h1} - T_c}$

According to the definition of fully developed velocity profile can be written:

$\frac{\partial u^*}{\partial x^*} = 0, \quad v^* = 0, \frac{\partial\theta}{\partial x^*} = 0$(11)

The local nusselt number is calculated as follows:

$Nu(x) = \frac{hD}{k_f}$(12)

Using Newton's cooling law, the convection heat transfer coefficient is written as follows:

$h = \frac{q''}{T_{h1} - T_c}$(13)

According to the Fourier law, the conduction coefficient is equal:

$K_{nf} = -\frac{q''}{\left(\frac{\partial T}{\partial r}\right)}$(14)

By combining equations (12), (13) and (14) and using dimensionless parameters, the local Nusselt number for nanofluid is defined as follows:

$Nu(x) = -\frac{2k_{nf}}{k_f}\frac{\partial\theta}{\partial R^*}|_{R^*=1}$(15)

Average Nusselt is obtained by integrating localized Nusselt along a portion of the micro tube wall that is subject to linear temperature gradient:

$Nu_m = \frac{1}{l_2}\int_{l_1}^{l_1+l_2} Nu(x)dx$(16)

In order to solve the equations governing, the thermo physical properties of nanofluid are required, which can be calculated from the following relations [21,14]:

$\rho_{nf} = (1 - \emptyset)\rho_f + \emptyset\rho_s$(17)

In the above relation,$\emptyset$ indicates the volumetric percentage of nanoparticlesρ_f is the base fluid density and $\rho_{nf}$is the nanofluid density.

Nano-fluid thermal capacity:

$(\rho C_p)_{nf} = (1 - \emptyset)(\rho C_p)_f + \emptyset(\rho C_p)_s$(18)

Nanofluid thermal diffusion coefficient:

$\alpha_{nf} = \frac{k_{nf}}{(\rho C_p)_{nf}}$(19)

For the effective dynamic viscosity of nanofluid in various sources, different relationships have been presented. In this study, Brinkman equation [39] is used:

$\mu_{nf} = \mu_f (1-\emptyset)^{-2.5}$(20)

For the effective thermal conductivity of a nanofluid, the model provided by Patel [40] has been used. The following relationship is used for nanofluid with suspended spherical particles:

$k_{nf} = k_f \left(1 + \frac{k_s A_s}{k_f A_f} + C k_s Pe \frac{A_s}{k_f A_f}\right)$(21)

In the above relation, C is an experimental constant value for CuO nanoparticles equal to 25,000.

The following equation is used to calculate the ratio of $\frac{A_s}{A_f}$:

$\frac{A_s}{A_f} = \frac{d_f}{d_s}\frac{\emptyset}{1-\emptyset}$(22)

In the above relation, d_s is the nanoparticle diameter, which in this study is equal to 40nm and d_f is the molecular size of water, which is equal to$2^o A$.

The following equation is used to calculate Pe:

$Pe = \frac{u_s d}{\alpha_f}$(23)

In the above relation, Pe indicates the peclet number, u_s is the brownian velocity of nanoparticles, and α_ff is the base fluid diffusion coefficient. brownian's velocity can be calculated from the following equation:

$u_s = \frac{2k_b T}{\pi \mu_f d_s^2}$(24)

In the above relation, k_b is the Boltzmann constant, which is equal to$1.3807 \times 10^{-23} \left(\frac{J}{K}\right)$.

IV. Geometry, meshing and validation

The geometry studied, as mentioned, includes twisted micro-tubes with different internal diameters, different helical diameters and pitchs, and the micro-tube wall has a constant temperature. An overview of the geometry can be seen in Figure (1).

First, solution independence of the mesh is examined. For this purpose, for each of the five different meshes, the amount of Nusselt number is calculated for Reynolds 1500 and compared with each other. The results are shown in Figure (2) and Table (2).

As can be seen in Table2 and Figure2, with the increase in the number of elements from 1150214 to 1450127, although the computational cost increases sharply, the final answer changes only by about 0.51%. Therefore, it can be concluded that the solution is independent of the mesh.

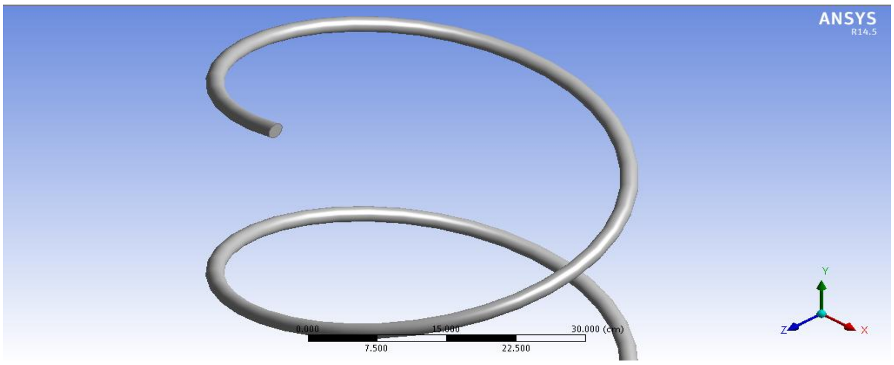

Figure1. Schematic view of the helical micro tube

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Table2. Solution independence of mesh

No	Element number	Nu	Error (%)
1	541245	5.51	5.32%
2	842145	5.67	2.57%
3	1001020	5.74	1.37%
4	1150214	5.79	0.51%
5	1450127	5.82	-

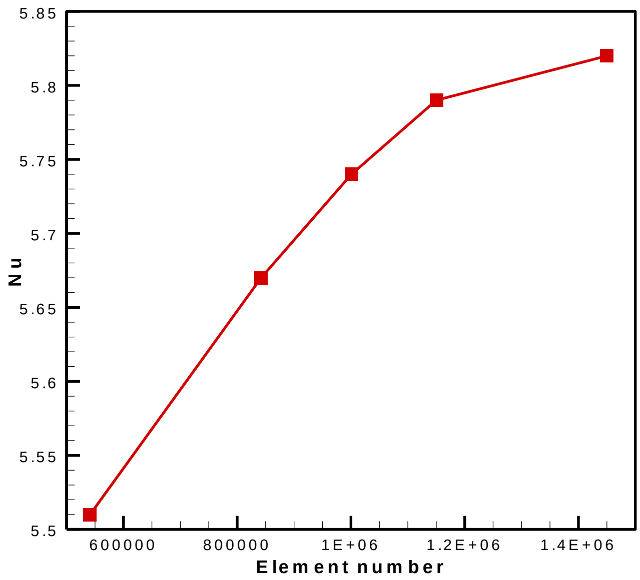

Figure2. Solution independence of mesh

The data provided by Yang and Lin [41] are used to validate the numerical model developed in this study. They experimentally tested the flow of water in a straight microwave with a length of 0.2 meters for the range of Reynolds numbers 700 to 2000. The results of their experimental data are compared with the data obtained from the numerical model developed in this paper in Figure (3).

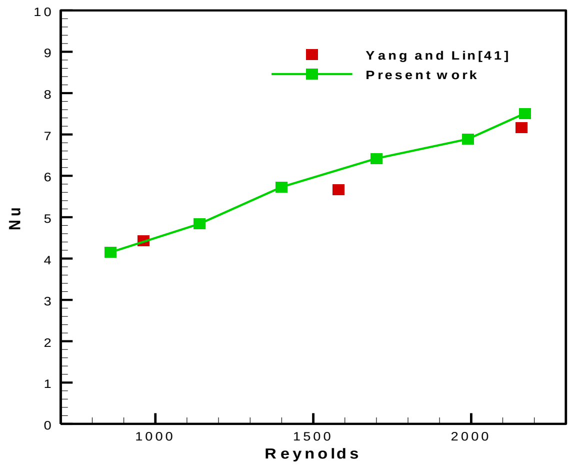

Figure3. Comparison between Yang and Lin experimental results [41] and numerical data of the present study

V. Results

In general, the three main forces, which are the centrifugal , pressure, and the viscous force, affect the nanofluid flow inside the spiral micro-tube. Centrifugal forces play a very important role and among the three mentioned forces, it has the greatest impact on the characteristics of the velocity field and the temperature field. Centrifugal forces are directly related to square of velocity and inverse spiral diameter. The centrifugal force controls the flow and directs the maximum point of the flow velocity from the central region to the wall [42]. This leads to effective convective mixing between the cold region in the middle zone and the hot fluid near the wall. Therefore, it is expected that with increasing fluid velocity and decreasing helical diameter, centrifugal forces will increase, resulting in the formation of vortices and an increase in heat transfer rate.

Figures (4) and (5) show the changes in heat transfer coefficient and Pressure drop per Reynolds number for different values of helical diameter for water-CuO nanofluid with a volume fraction of 0.1%.

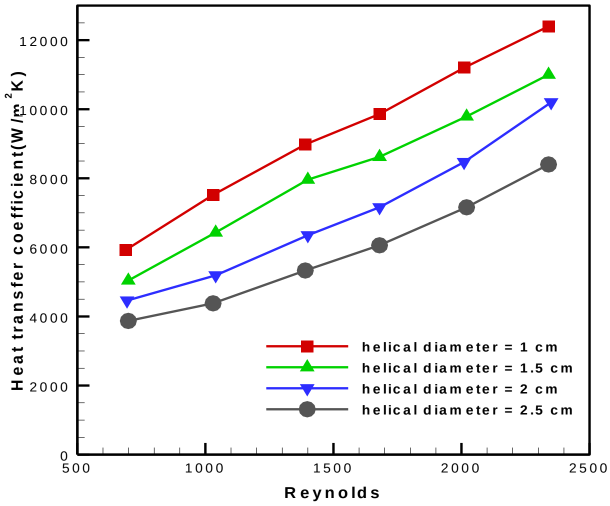

Figure4. Heat transfer coefficient - Reynolds number

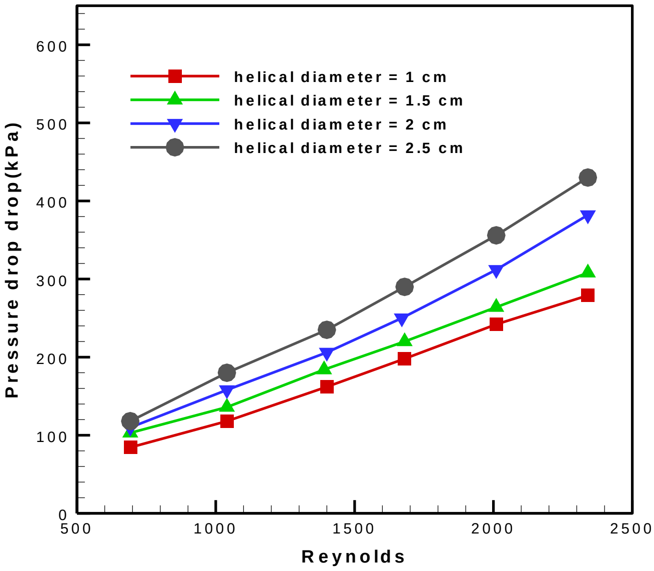

Figure4. Pressure drop - Reynolds number

As can be seen, the centrifugal force increases with decreasing helical diameter, and as a result the heat transfer coefficient increases, so in Figure (4) the heat transfer coefficient decreases with decreasing helical diameter. For this reason, the heat transfer coefficient of micro-tubes with a helix diameter of 1 cm, 14.5%, 34.3% and 60.9%, respectively, is higher than the heat transfer coefficient for micro-tubes with a helix diameter of 1.5, 2 and 2.5 cm. As mentioned, centrifugal force is also directly related to the square of velocity. Increasing the Reynolds number means increasing the fluid flow rate, so as the Reynolds number increases, the heat transfer rate also increases sharply. More attention is needed to analyze the effect of helix diameter on pressure drop. In previous studies, such as Rakhsha et al. [43] It has been shown that if the tube length is constant, the pressure drop increases with decreasing helical diameter. As the diameter of the helix decreases, the centrifugal forces intensify, resulting in more casualties, but the present study appears to have yielded different results. As the diameter of the helix decreases, the pressure drop decreases. Because in the present study, the length of fixed micro-tubes has not been considered. Therefore, with increasing helical diameter, the length of the micro-tube increases and as a result, its pressure drop increases. The results show that for micro-tubes whith 2.5 cm helical diameters, the pressure drop is about 64%, 28% and 11.5%, respectively, more than tubes with 1, 1.5 and 2 cm helical diameters.

The results of the use of nanofluid with different volume fractions on the ascending number and the coefficient of friction in different Reynolds numbers are shown in Figures (6) and (7).

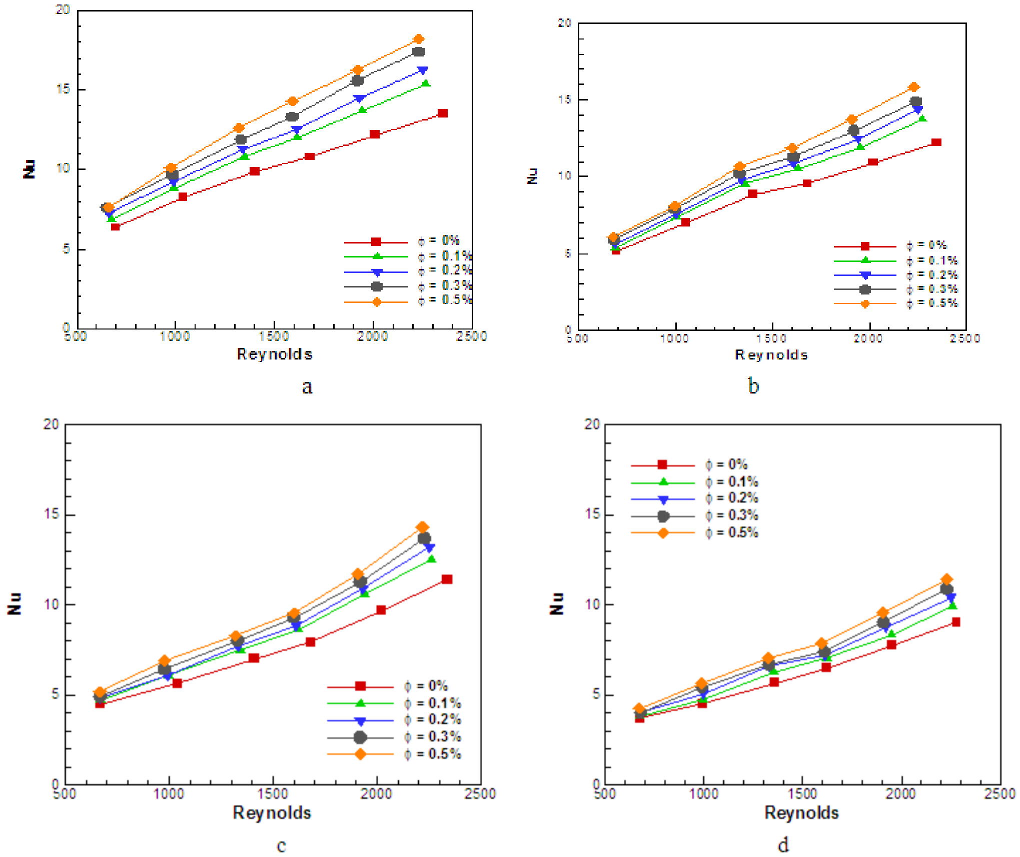

Figure 6. Nusselt number variations versus Reynolds number for CuO-water nanofluid inside HMTs (a) 1 cm, (b) 1.5 cm, (c) 2 cm, (d) 2.5 cm.

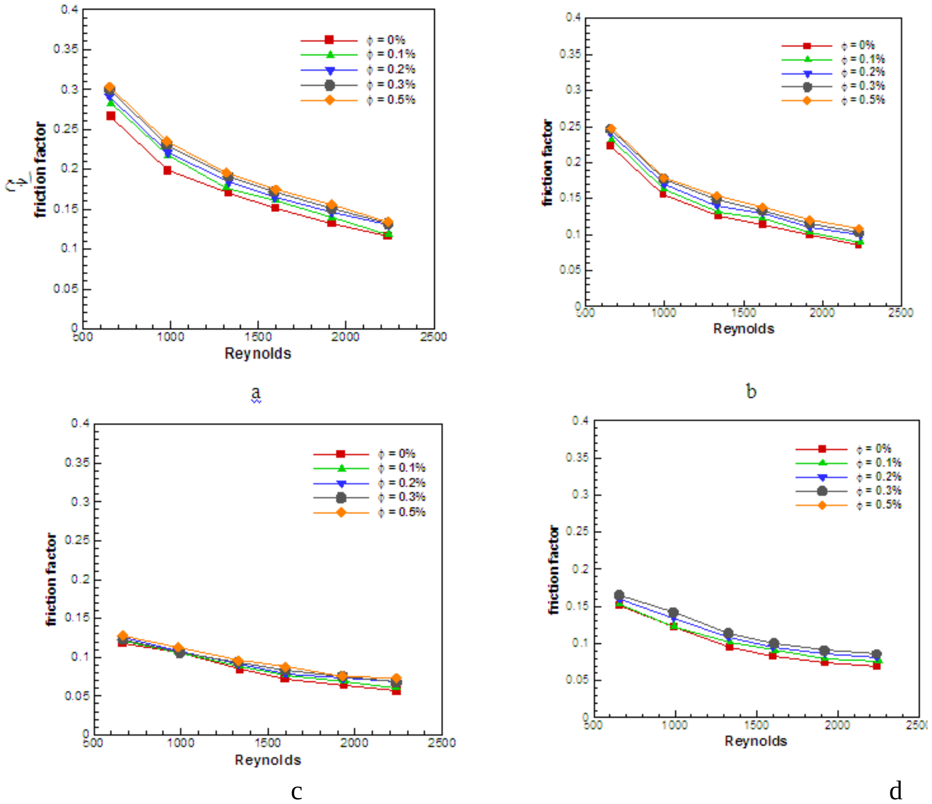

Figure7. friction factor variations versus Reynolds number for CuO-water nanofluid inside HMTs (a) 1 cm, (b) 1.5 cm, (c) 2 cm, (d) 2.5 cm.

As you can see, adding nanoparticles to the base fluid increases the heat transfer rate. The increase in heat transfer rate is due to the improvement of the thermal conductivity of nanofluids compared to the base fluid, the migration of nanoparticles and brownian movements, as well as the reduction of the thickness of the thermal boundary layer for nanofluids. Therefore, improving the thermo physical properties of nanofluid compared to base fluid cannot be considered as the only factor improving heat transfer. For example, for water-CuO nanofluid, with a volumetric fraction of 0.1%, the thermal conductivity ratio to water has improved by only 7.5%, while the Nusselt number increases by about 14.8%. Therefore, the effect of nanoparticle diffusion and migration on increasing heat transfer rate is also important. As the Reynolds number increases, the effect of Brownian motions and nanoparticle diffusion becomes more apparent. For nanofluids with a volume fraction of 0.1% and for the lowest number of Reynolds numbers studied, which is equal to 500, the difference between the Nusselt number for water and nanofluid is only about 7%, this difference increases to more than 14% for the maximum Reynolds number, which is 2500. As the volume fraction of nanoparticles increases, heat transfer increases. The effect of using nanofluid in the micro-tube with a smaller helical diameter is more obvious, which can be attributed to the increase in centrifugal forces and its effect on the movement of nanoparticles.

In all systems, the goal is to maximize heat transfer while reducing pressure drop. Therefore, a criterion is defined as Performance evaluation criterion (PEC), which indicates the ratio of increasing heat transfer to pressure drop in different conditions. The higher the value of this criterion, the more favorable those conditions will be. Figure (8) shows the PEC criteria for different study conditions.

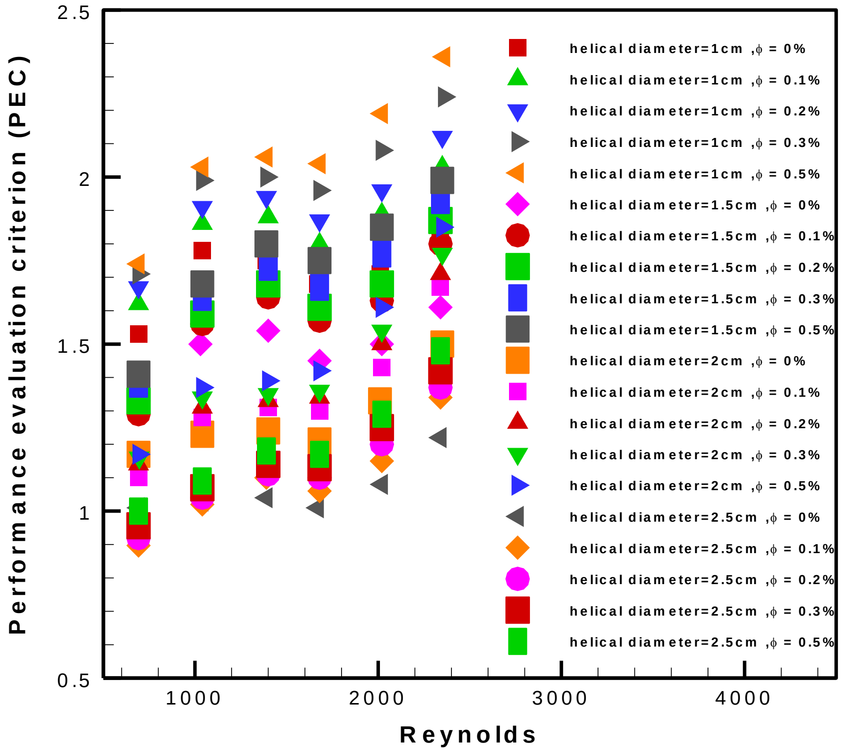

Fig8. Performance evaluation criterion — Reynolds number for different working fluids inside HMTs.

VI. Conclusion

In this study, the laminar flow of nanofluid inside the spiral micro-tube was investigated assuming a constant wall temperature. The effect of volumetric nanoparticle fraction in base fluid, Reynolds and helix diameters on heat transfer coefficient, Nusselt number, pressure drop and friction factor was investigated and PEC value was calculated in each case. The results can be categorized as follows.

• As the number of Reynolds and the volume fraction of nanoparticles in the base fluid increases, increases the Nusselt number and the pressure drop.

• By reducing the helical diameter and increasing flow velocity, centrifugal forces are increased, followed by an improvement Nusselt number.

• Adding nanoparticles in three ways to improve fluid thermo physical properties, particle migration, and Brownian motions leads to increased heat transfer. The effect of particle migration and the Brownian motions in the Reynolds numbers studied is as much as the effect of improving fluid thermo physical properties.

• The maximum PEC value for the helix diameter is 1 cm and the volume fraction is 0.5%.

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