Energy Equip. Sys./ Vol. 7/No. 1/March 2019/ 11-22

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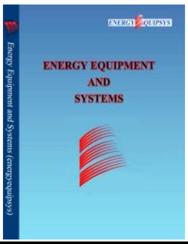

Comparison of convective heat transfer of turbulent nanofluid flow through helical and conical coiled tubes

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Article history:

Received : 8 March 2018
Accepted : 6 October 2018

ABSTRACT

Application of nanofluid and coiled tubes are two passive methods for increasing the heat transfer. In the present study, the turbulent flows of water and nanofluid in coiled tubes heat exchanger were numerically studied. CuO-water nanofluid containing 1 vol% copper oxide nanoparticles was used and single-phase approach was considered for nanofluid flow. The effect of different geometrical parameters on Nusselt number was also investigated for both helical and conical coiled tubes. Water properties were defined as temperature dependent function, and constant wall temperature was employed as wall boundary condition. Simulation results were validated by available experimental and numerical data of heat transfer coefficient and pressure drop inside the helically coiled tube. The results show that for the helically coiled tubes, Nusselt number increased by tube diameter enhancement, while the increase of pitch circle diameter resulted in its decline. However, the variation of helical pitch exhibited a slight effect on the heat transfer. Moreover, for conical coiled tubes, increase helical pitch and cone angle reduced Nusselt number; while this parameter increased as the tube diameter increases. Addition of 1% copper oxide to water led to 8% heat transfer augmentation in the helically coiled tube. Also, using CuO-water nanofluid, the performance index was enhanced by 3.5 and 5% for helical and conical coiled tubes, respectively.

Keywords: Single-Phase Approach, Conical and Helical Coiled Tubes, Nanofluid, Heat Transfer, Turbulent.

1. Introduction

Heat exchangers have been extensively used in various industries such as chemical reactors, air conditioning systems, and boilers. Enhancement of heat exchangers' efficiency and reduction of their size are of crucial importance. Active and passive methods have been developed to increase heat transfer.

Active methods require external power and energy consumption; while in passive methods, it is possible to increase the heat transfer without using external forces. Extended surfaces, coiled tube and addition of particles to base fluid are some passive strategies. Helically coiled tubes occupy less space in comparison to the straight pipes and improve the thermal performance due to the presence of secondary flow at the cross-section. Furthermore, the developing length is increased in the coiled tube which increases the heat transfer [1]. The low thermal conductivity of conventional fluids such as water, ethylene glycol, and oil has limited their efficiency. Nowadays, the addition of nanoparticles to the base fluids has improved their thermal

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conductivity [2]. These nano colloids are named nanofluid. Aly [3] investigated the heat transfer and pressure drop in turbulent nanofluid flow through a double-tube coiled heat exchanger. Enhanced wall treatment was used with the realizable k-ε model for turbulence modeling with a single-phase approach. The results showed that for constant Reynolds, the heat transfer coefficient increased with the increase of pipe diameter and nanoparticle concentrations. Also, the friction coefficient enhanced with the increase of the curvature ratio, the increase of nanofluid pressure drop up to 2% was however negligible. Numerical simulation of laminar water-alumina nanofluid flow through a helically coiled tube was carried out by Bizham and Abbasi [4]. They investigated the effects of different parameters such as curvature ratio, Reynolds number, and nanoparticle concentration. Their results indicated that for a given nanoparticle concentration, the overall heat transfer coefficient experienced more enhancement in lower Reynolds numbers. Rakhsha et al. [5] surveyed the turbulent flow of nanofluid in the helically coiled tube heat exchanger by adopting a single-phase approach. k-ε turbulence model was used for turbulent flow of CuO-water nanofluid. They found out that the single-phase model predicts only 6-7% increase in the heat transfer coefficient for 1% nanoparticle concentration, while the enhancement was 16-17% in the experimental results with the same condition. By variation of the coil ring diameter, the formation of fully developed flow throughout the pipe can be prevented which could be attributed to the change in centrifugal forces due to coil ring radius alteration. Studies on conical coiled tube heat exchangers are more limited than spiral and helically coiled tube heat exchangers. Bagherzadeh et al. [6] simulated the laminar nanofluid flow through the helically coiled tube by a four-equation nonhomogeneous model which involved the effects of Brownian motion and thermophoresis. These effects caused a non-uniform nanoparticle concentration distribution across the flow field. Comparison of the simulation outcomes with experimental data indicated that the results of the nonhomogeneous model were closer to the experimental data as compared with the homogenous one. Recently, Purandare et al. [7] experimentally analyzed the thermal performance of conical coil heat exchanger. Fifteen conical coils with different cone angles (from helical to spiral) and tube sizes with similar average coil diameter and tube length were examined. The heat transfer coefficient, Nusselt number, effectiveness and friction factor were estimated for the heat exchanger. Their results showed Nu enhancement by the increase of tube side flow rate whereas it is reduced with the elevation of shell side flow rate, cone angle and tube diameter.

Previous studies did not consider the thermal performance of nanofluid flow in a conical coiled tube heat exchanger. Therefore, the present study aims to investigate the effects of two passive methods of heat transfer increase (i.e., coiled tube and nanoparticle addition to the base fluid) on thermal performance of a conical coiled tube heat exchanger and compare them with those of helically coiled tube. The effects of various geometric parameters of the helical and conical coiled tubes on thermal performance were also investigated in this study.

Nomenclature

Cp	Specific heat capacity
d	Pipe diameter
D	Pitch circle diameter
h	Heat transfer coefficient
f	Friction factor
k	Turbulence kinetic energy
Nu	Nusselt number
P	Helical pitch
T	Temperature
y^+	Dimensionless wall distance

Greek letters

ρ	Density
μ	Dynamic viscosity
φ	Volume fraction
δ	Curvature ratio
τ	Torsion ratio
θ	Cone angle
η	Performance index

Subscripts

f	fluid
eff	effective
t	turbulence
w	wall
b	bulk
p	particle

2.Governing equations and thermophysical properties

Nanofluids are generally diluted solid-liquid mixtures containing very fine particles (less than

100nm). Hence, it has been suggested that these very fine particles may be easily fluidized and consequently, can behave more like a fluid. Moreover, by assuming no slip between the particles and the continuous phase and the thermal equilibrium conditions, the nanofluid may be then considered as a conventional single-phase fluid with effective thermophysical properties. Thus, in the present study, the nanofluid was treated as a single-phase fluid with the following assumptions: incompressible flow, no chemical reactions and negligible external forces, viscous dissipation and radiative heat transfer. Moreover, the nanoparticles and the base fluid are in local thermal equilibrium. Under these assumptions, the governing equations can be expressed as [8]:

Continuity.

$$\nabla.(\rho_{eff}\vec{V}) = 0 \tag{1}$$

Momentum

$$\nabla.(\rho_{eff}\vec{V}\vec{V}) = -\nabla P + \nabla.(\mu_{eff}\nabla\vec{V}) + \nabla.(\mu_T\nabla\vec{V}) \tag{2}$$

Energy

$$\nabla.\left(\rho_{eff}\vec{V}C_{p_{eff}}T\right) = -\nabla p.V + \nabla.\left\{\left(k_{eff} + \frac{c\mu_T}{Pr_T}\right)\nabla T\right\} \tag{3}$$

Water is assumed to be incompressible with temperature-dependent thermophysical properties. The properties of CuO nanoparticles are listed Table 1.

$$\mu(T) = 2.1897 \times 10^{-11}T^4 - 3.055 \times 10^{-8}T^3 + 1.6028 \times 10^{-5}T^2 - 0.0037524T + 0.33158 \tag{4}$$

$$\rho(T) = -1.5629 \times 10^{-5}T^3 + 0.011778T^2 - 3.0726T + 1227.8 \tag{5}$$

$$k(T) = 1.5362 \times 10^{-8}T^3 - 2.261 \times 10^{-5}T^2 + 0.010879T - 1.0294 \tag{6}$$

$$C_p(T) = 1.1105 \times 10^{-5}T^3 - 0.0031078T^2 - 1.478T + 4631.9 \tag{7}$$

It is assumed that nanoparticles have a spherical shape with the same diameter. The thermophysical properties of nanofluids were computed by below equations [8,9,10,11].

Density and specific heat capacity:

$$\rho_{eff} = \varphi\rho_p + (1-\varphi)\rho_f \tag{8}$$

$$C_{p_{eff}} = \frac{(1-\varphi)(\rho C_p)_f + \varphi(\rho C_p)_p}{\varphi\rho_p + (1-\varphi)\rho_f} \tag{9}$$

Thermal conductivity is presented by Koo and Kleinstreuer [12], the model is based on Brownian motion and predicts the nonlinear behavior of thermal conductivity at low particles concentration, as follows [2, 13]:

$$k_{eff} = k_{static} + k_{Brownian} \tag{10}$$

$$k_{static} = k_f\left[\frac{k_p + 2k_f - 2(k_f - k_p)\varphi}{k_p + 2k_f + (k_f - k_p)\varphi}\right] \tag{11}$$

$$k_{Brownian} = 5 \times 10^5\beta\varphi\rho_f C_{p,f}\sqrt{\frac{K_B T}{2\rho_p r d_p}}f(T,\varphi) \tag{12}$$

In which $K_s = 1.3807 \times 10^{-23} J/K$ is Boltzmann constant. Modeling functions $f(T,\varphi)$ and β were defined for CuO nanoparticles $1\% \leq \varphi \leq 6\%$ as [14]:

$$\beta = 9.881(100\varphi)^{-0.9446} \tag{13}$$

$$F(T,\varphi) = (2.8217 \times 10^{-2})\varphi + (3.917 \times 10^{-3})\left(\frac{T}{T_0}\right) + (\varphi(-3.0699 \times 10^{-2}) - (3.91123 \times 10^{-3})) \tag{14}$$

Table. 1. Thermophysical properties of CuO nanoparticles

	k [W/m K]	C_p [J/kg K]	ρ [kg/ m^3]	d_p [nm]
CuO	32.9	535.6	6320	68

The dynamic model was adopted for calculating the dynamic viscosity of nanofluid as below [15]:

$$\mu_{eff} = \frac{\mu_f}{(1-\varphi)^{2.5}} + \mu_{Brownian} \quad (15)$$

$$\mu_{Brownian} = 5 \times 10^5 \beta\varphi\rho_f \sqrt{\frac{K_B T}{\rho_p d_p}} f(T,\varphi) \quad (16)$$

$$F(T,\varphi) = (-134.63 + 1722.3\varphi) + (0.4705 - 6.04\varphi)\left(\frac{T}{T_0}\right) \quad (17)$$

$$\beta = 0.0011(100\varphi)^{-0.7272}, \varphi > 0.01 \quad (18)$$

The realizable k-ε model was adopted for turbulent modeling. In comparison with standard k-ε and RNG models, this model is more precise for flows involving rotation, secondary flow, and separation. Wall function was used for including the viscosity effect between the wall and fully turbulent regions. In this regard, the enhanced wall treatment function was applied [3].

3.Numerical simulation

In this study, conical coiled tubes were simulated. The geometry was created by Autodesk Inverter software. The geometric characteristics of the coiled tubes are listed in Tables 2 and 3. The CFD simulation of nanofluids thermal characteristics was performed by ANSYS FLUENT software. The absolute convergence criterion for continuity and velocity was set to 10^{-6} while for energy and turbulent equations, 10^{-8} and 10^{-5} were considered, respectively. Appropriate under-relaxation factors were used for stabilizing and damping the residuals oscillation. The total CPU time for simulations varied from 1181.5 to 1344.2 seconds on a computer with processor Intel® Core™ i7-4790k CPU@4.00GHz. The user-defined function (UDF) was used to specify the thermophysical properties of nanofluid. The computational grid was generated using ANSYS-Meshing with considering that the distance of the first node should satisfy the criterion of $y^+ \approx 1$ [3]. The computational grid is shown in Fig. 1.

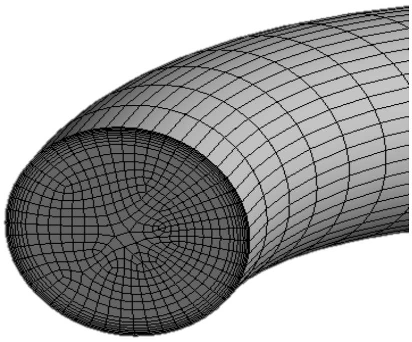

Fig. 1. Computational grid

The characteristics of helical and conical coiled tubes are listed in Tables 2 and 3, respectively.

Table 2. Characteristics of helically coiled tubes

Coil number	d (mm)	P (mm)	D (mm)	δ
1	8	45	120	0.0677
2	10	25	120	0.0833
3	10	45	70	0.1429
4	10	45	120	0.0833
5	10	45	170	0.0588
6	10	65	120	0.0833
7	12	45	120	0.1
8	8	20	110	0.0727

Table 3. Characteristics of conical coiled tubes

Coil number	d (mm)	P (mm)	θ (degree)
1	8	45	45
2	10	25	45
3	10	45	30
4	10	45	45
5	10	45	60
6	10	65	45
7	12	45	45

For all cases, coils had three turns. The small diameter of the conical coiled tube was 70 mm. Schematics geometries are depicted in Fig. 2. The curvature and torsion ratios are dimensionless parameters with effects on thermal and hydraulic performance of the curved tubes. These parameters are defined as [16]:

$$\delta = \frac{d}{D} \tag{19}$$

$$\tau = \frac{P}{2\pi D} \tag{20}$$

Dean number in a curved tube describes flow characteristics and is effective on heat transfer and friction coefficient [17].

$$De = Re\sqrt{\delta} \tag{21}$$

Choosing appropriate boundary conditions is necessary. At the pipe inlet, fluid enters with constant temperature and velocity corresponding to Reynolds number. The turbulence intensity was assumed to be 5%. Moreover, the tube wall was assumed as a no-slip wall with a constant temperature. Pressure outlet was set for the pipe outlet with a pressure gauge of zero.

4.Grid independence and validation

To test the grid-independency of the numerical results for the different grid sizes, the computations were carried out using different number of elements. The details of different computational grids are presented in Table 4. Variation of the convective heat transfer coefficient along the tube for different mesh sizes is presented in Fig. 3. The second grid was selected as the computationally optimal gird for numerical simulationse.

Validation of the present study was carried out by experimental data of Rakhsha et al. [5] for water and nanofluid flow inside a helically coiled tube. The coil number 8 of Table 2 was created for this purpose. The water flow with temperature-dependent properties as well as the nanofluid flow was simulated using a single-phase model. The heat transfer coefficient and pressure drop were compared with experimental and numerical data. The present simulations resulted in acceptable predictions as shown in Figs. 4 and 5.

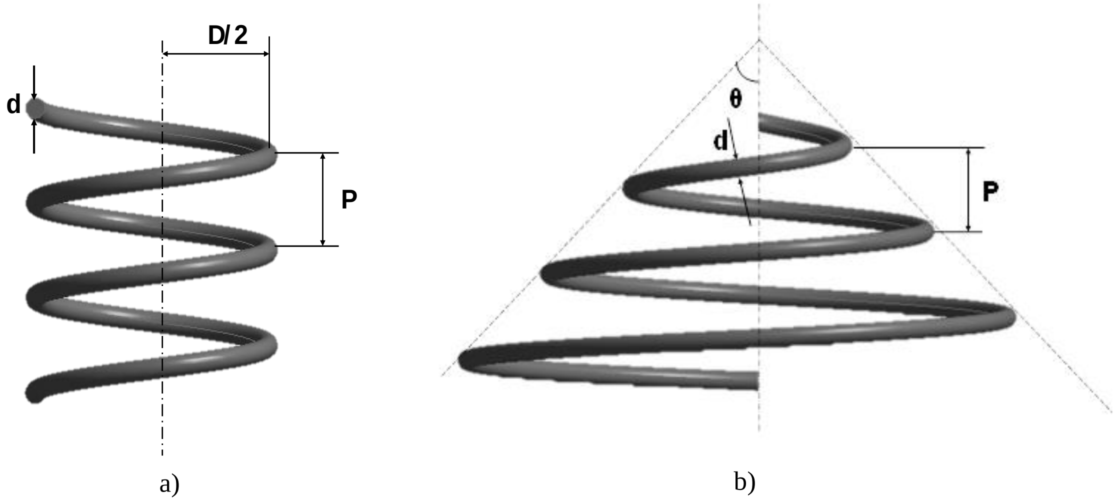

Fig. 2 . Schematic of coiled tubes, a) helical and b) conical

Table 4. Computational grids

Grid number	nl	nr	$n\theta$	Number of elements
1	400	11	50	345200
2	456	13	57	517900
3	520	13	65	776100
4	456	15	57	544300

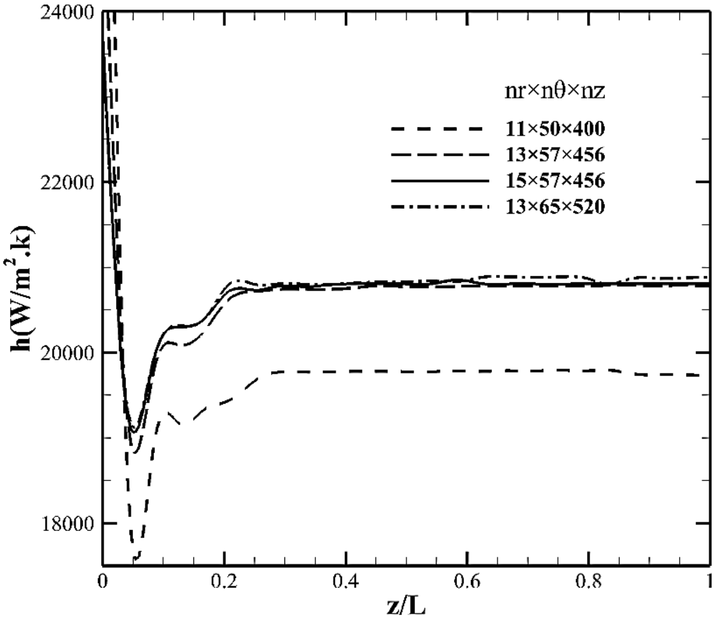

Fig. 3 .Variation of heat transfer coefficient along the helically coiled tube for different grids

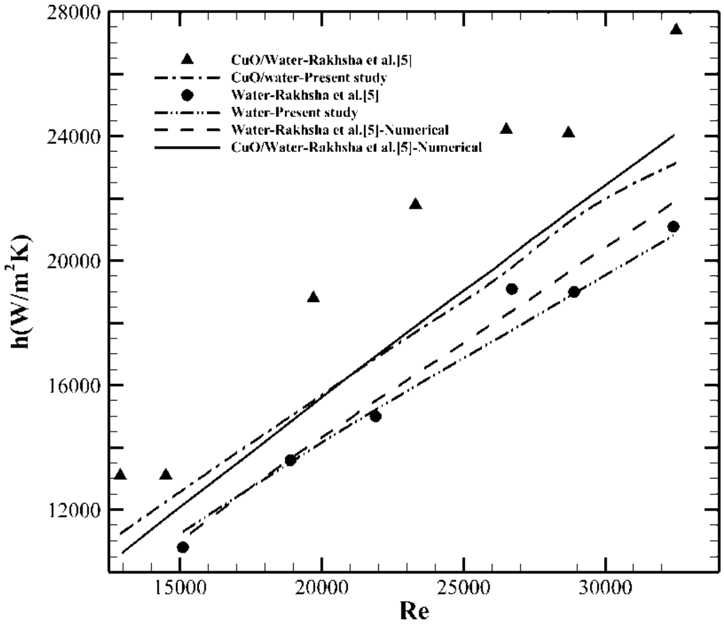

Fig .4. Variation of heat transfer coefficient for different Reynolds number

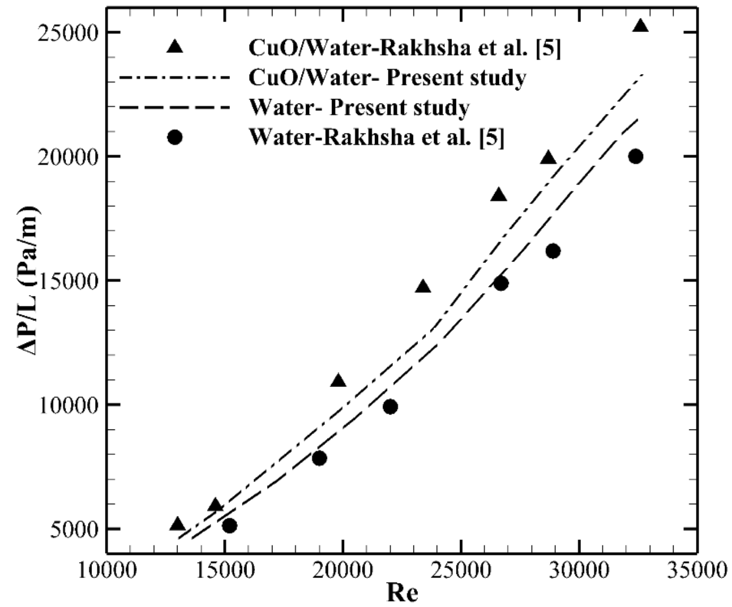

Fig. 5. Variation of pressure drop for different Reynolds number

7. Results and discussions

In the present study, the thermal performance of helical and conical coiled tube was examined for turbulent flows of water and CuO-water nanofluid. Single-phase approach for CuO-water nanofluid with 1% particles volume fraction was studied under constant wall temperature boundary condition. Moreover, the effects of coil geometric parameter such as tube diameter and coil pitch were taken into account for both shapes. Also, pitch coil diameter and cone angle were discussed for helical and conical coiled tubes, respectively. The simulations were conducted for the fully turbulent region with Re=20000;

5.1. Effects of pitch circle diameter and helical pitch for helically coiled tube

As the helical pitch increased, the rotational force on fluid flow increased as well; causing more streamlines twisting. Therefore, the mixing of fluid in the tube cross-section was enhanced as well as the heat transfer coefficient. As shown in Fig. 6, Nusselt number showed a slight enhancement by the increase of helical pitch. The pitch circle

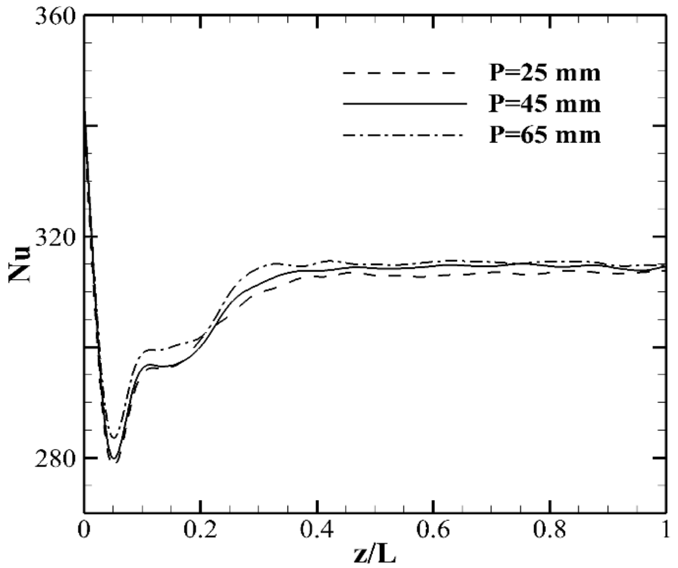

Fig. 6. Variation of Nusselt number along the helically coiled tube for different helical pitch

diameter could affect centrifugal force on fluid flow. Therefore, the intensity of the secondary cross-sectional flow could also be influenced. The entrance length reduced as a result of the increase in pitch circle diameter. Figure 7 shows the variation of Nusselt number along the helically coiled tubes. Clearly, Nusselt number was dramatically increased by reduction of pitch circle diameter to 70 mm. Enhancement of the curvature and torsion ratio led to the increase of Nusselt number, but the effect of curvature ratio on Nusselt number was more profound as compared with torsion ratio.

5.2.Effect of pipe diameter on Nusselt number

Pipe diameter increase gave rise to enhancement of curvature ratio in helical and conical coiled tubes. As a result, the entrance length will be increased; this can delay the fluid flow development and increase the heat transfer. In fact, for lower pipe diameters, the secondary flow is weak and mixing in cross section is low. The variations of Nusselt number along the helical and conical coiled tubes are demonstrated in Figs. 8 and 9, respectively. The increase of pipe diameter enhanced Nusselt number.

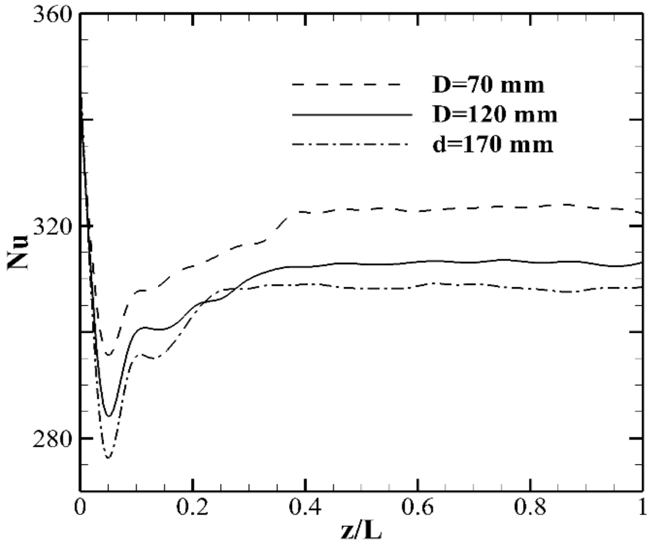

Fig. 7. Variation of Nusselt number along the helically coiled tube for different pitch circle diameter

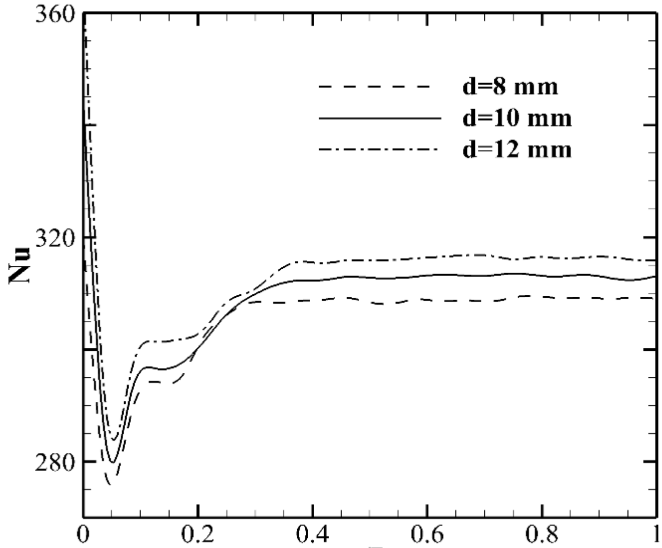

Fig. 8. Variation of Nusselt number along the helically coiled tube for different tube diameter

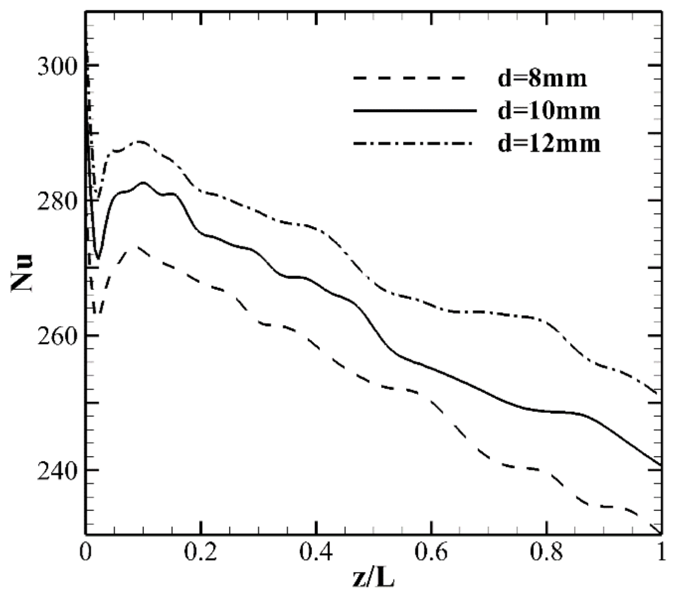

Fig. 9. Variation of Nusselt number along the conical coiled tube for different helical pitch

5.3. Effect of cone angle and helical pitch for conical coiled tube

In a conical coiled tube, the curvature ratio, as well as Dean number, changed along the tube. Fluid flow at the inlet had the highest Dean number. By the increase of pitch circle diameter along the tube, Dean number declined and reached its minimum at the outlet. Reduction of Dean number along the tube led to a reduction in Nusselt number. The helical pitch and cone angle the conical coiled tube can affect the fluid flow by influencing the pitch circle diameter. Therefore, these parameters could have a significant effect on hydraulic and thermal performance. The rate of change in pitch circle diameter increased by enlargement of the coil angle. Thus, the intensity of secondary flow can be reduced through the conical coiled tube by enhancing the coil angle. Figure 10 shows the effect of cone angle on Nusselt number. In the conical coiled tube, the variation of pitch circle versus helical pitch is linear.

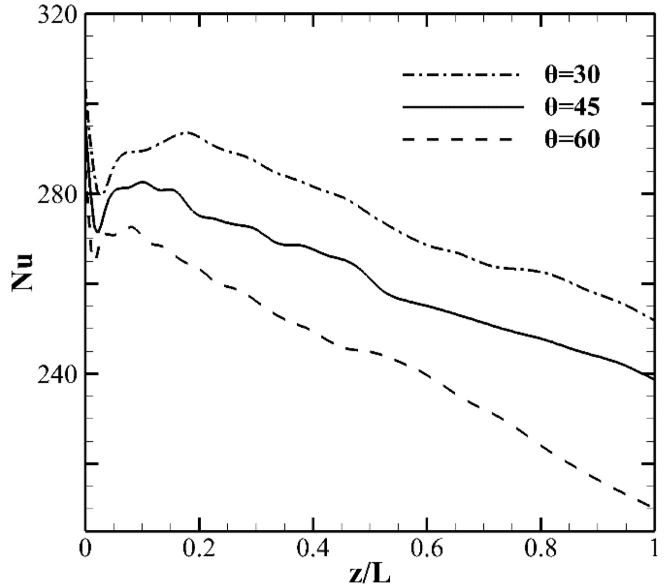

Fig. 10. Variation of Nusselt number along the conical coiled tube for different cone angle

Therefore, pitch circle diameter will be enhanced by the increase of helical pitch leading to lower secondary flow intensity. As shown in Fig. 11, Nusselt number of the conical coiled tube is sharply declined with increase the helical pitch.

5.4.Comparison of heat transfer coefficient for water and nanofluid flow

Single-phase approach was used to simulate nanofluid flow. In this content, copper oxide (CuO) nanoparticles (1 vol%) were used. The results for helical coiled tube #4 are shown in Fig. 12. Naturally, in comparison with water, nanofluids could increase the average heat transfer coefficient by 8%.

5.5.Performance index evaluation

The performance index of heat exchangers is usually defined as the ratio of heat transfer coefficient enhancement to pump power enhancement, which can be simplified as [13]:

$$\eta = \left(\frac{h}{h_0}\right) / \left(\frac{\Delta P}{\Delta P_0}\right) \qquad (22)$$

where, h and ΔP represent the heat transfer coefficient and pressure drop for the applied heat transfer enhancement method, respectively. Moreover, h_0 and ΔP_0 denote the heat transfer coefficient and pressure drop for the base fluid flow in a straight pipe with the same hydraulic diameter and length. The performance indices of helical and conical coiled tubes with different geometric characteristics are listed in Tables 5 and 6 for water and nanofluid flow, respectively. Application of coiled tubes increased the heat transfer coefficient as well as the performance index. Addition of nanoparticles to the base fluid improved the performance index of both helical and conical coiled tubes. This enhancement was however more profound in conical coiled tubes.

8.Conclusion

Owing to their less space occupancy along with appropriate thermal performance, coiled tubes have found extensive applications. In the present study, the performance of helical and conical coiled tubes is investigated. The results for helically coiled tubes showed an increase of Nusselt number by enhancement of helical pitch and tube diameter as well as the decline of pitch circle diameter. For conical coiled tubes, Nusselt number showed a decline by the increase of helical pitch and cone angle as well as the decrease in tube diameter. Use of

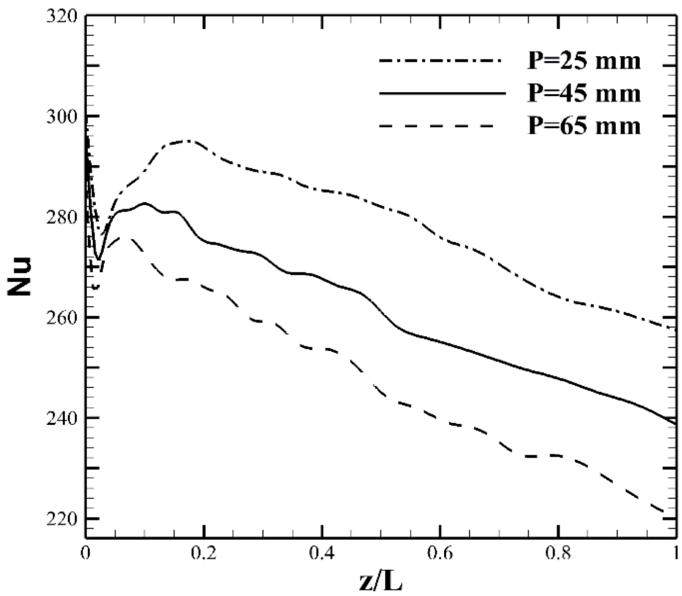

Fig. 11. Variation of Nusselt number along the conical coiled tube for different helical pitch

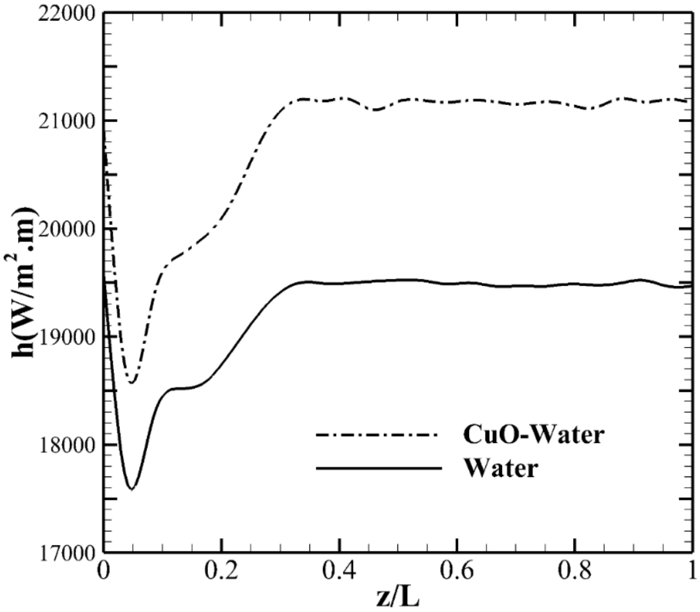

Fig. 12. Variation of heat transfer coefficient along the helically coiled tube

Table 5: Performance index using different methods for helically coiled tubes

Base fluid / Geometry	Water	CuO-water 1%
Helical 1	1.41	1.46
Helical 2	1.41	1.46
Helical 3	1.43	1.49
Helical 4	1.41	1.47
Helical 5	1.41	1.46
Helical 6	1.42	1.47
Helical 7	1.43	1.48

Table 6: Performance index using different methods for conical coiled tubes

Base fluid / Geometry	Water	CuO-water 1%
Conical 1	1.40	1.46
Conical 2	1.42	1.48
Conical 3	1.43	1.50
Conical 4	1.41	1.47
Conical 5	1.33	1.41
Conical 6	1.39	1.46
Conical 7	1.41	1.48

nanofluid significantly increased the heat transfer coefficient. Addition of 1% copper oxide nanoparticle to water increased the heat transfer coefficient by 8%. It also could result in improvement of performance index in both helical and conical coiled tubes. This enhancement of performance index was however more profound in conical coiled tubes.

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