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Convective heat transfer of MWCNT / HT-B Oil nanofluid inside micro-fin helical tubes under uniform wall temperature condition

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Abstract. Experiments are performed to investigate the single-phase flow heat transfer augmentation of MWCNT/HT-B Oil in both smooth and micro-fin helical tubes with constant wall temperature. The tests in laminar regime were carried out in helical tubes with three curvature ratios of $2R/d$=22.1, 26.3 and 30.4. Flow Reynolds number varied from 170 to 1800 resulting in laminar flow regime. The effect of some parameters such as the nanoparticles concentration, the dimensionless curvature radius ($2R/d$) and the Reynolds number on heat transfer was investigated for the laminar flow regime. The weight fraction of nanoparticles in base fluid was less than 0.4%. Within the applied range of Reynolds number, results indicated that for smooth helical tube the addition of nanoparticles to the base fluid enhanced heat transfer remarkably. However, compared to the smooth helical tube, the average heat transfer augmentation ratio for finned tube was small and about 17%. Also, by increasing the weight fraction of nanoparticles in micro-fin helical tubes, no substantial changes were observed in the rate of heat transfer enhancement.

Keywords: convective heat transfer; nanofluid; helical tube; micro-fin tube; enhancement

1. Introduction

In general, the enhancement techniques are divided into two groups: active and passive techniques. The active techniques require external forces, e.g., electric field, acoustic and surface vibration while passive techniques require special surface geometries or fluid additives. Both techniques have been used for improving heat transfer in heat exchangers. Due to their compact structure and high heat transfer coefficient, curved tubes were introduced as one of the passive heat transfer enhancement techniques. Curved tube are an essential component of nearly all industrial processes, ranging from power production, chemical and food industries, electronic, waste heat recovery, manufacturing, air-conditioning and refrigeration. The use of curved tubes in continuous processes is an attractive alternative to conventional agitation since similar and sometimes better performance can be achieved at lower energy consumption and reduced maintenance requirement because of no moving parts (Ghobadi *et al*. 2013, Salimpour *et al*. 2008). Hence, curved tubes such as spiral and helical coiled tubes are utilized in order to enhance the heat

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transfer (Akhavan Behabadi *et al*. 2012). Moreover, micro-fin tubes, known as another passive heat transfer enhancement technique, are efficiently contributing the fluid heat transfer and are used widely to make more compact heat exchangers. Researchers have also tried to increase the thermal conductivity of base fluids by suspending micro- or larger-sized solid particles in fluids since the thermal conductivity of solids is typically higher than that of liquids. Compared with the existing techniques for enhancing heat transfer by adding millimeter or micrometer-sized particles in fluids, nanofluids are expected to be ideally suited for practical application with incurring little pressure loss since the nanoparticles are so small and the nanofluids behave like a pure fluid or single phase liquid. The term nanofluid refers to the kinds of fluids that are stable suspension of nanoscale particles inside a base fluid (Choi and Eastman 1995, Lee *et al*. 1999). Several published articles show that the heat transfer coefficient of nanofluids is much higher than that of the common-base fluid and their application is accompanied with little or no penalty in pressure drop (Hwang *et al*. 2009, Mansour *et al*. 2011, Heris *et al*. 2006). Recently, Lazarus Godson *et al*. (2010) conducted a literature review on the general heat transfer characteristics of nanofluids. Based on the previous works, they concluded that nanofluids have great potential for heat transfer enhancement and are highly suited for application in practical heat transfer processes.

Single-phase heat transfer characteristics in the helically coiled tubes have been studied by researchers both experimentally and theoretically. Due to some complexity of heat transfer process in the helically coiled tubes, experimental studies are very difficult to handle and numerical investigations are also needed. Yang *et al*. (1995) presented a numerical model to study fully developed laminar convective heat transfer in a helical pipe having a finite pitch. The effects of the Dean number, torsion, and the Prandtl number on the laminar convective heat transfer were discussed. The results revealed that the temperature gradient increased on one side of the pipe wall and decreased on the other side with increasing torsion. Chen and Zhang (2003) studied the combined effects of rotation (coriolis force), curvature (centrifugal force), and heating/cooling (centrifugal type buoyancy force) on the flow pattern, friction factor, temperature distribution, and Nusselt number. Sillekens *et al*. (1998) employed the finite difference discretization to solve the Parabolized Navier-Stokes and energy equations in a helically coiled heat exchanger. The effect of buoyancy forces on heat transfer and secondary flow was considered. The results showed that for the helically coiled tube with constant wall temperature, secondary flow induced by centrifugal and buoyancy forces affected the heat transfer rate. Xin and Ebadian (1997) considered the effects of the Prandtl number and geometric parameters on the local and average convective heat transfer characteristics of a flow inside helical pipes. The results showed that for the laminar flow region the peripheral Nusselt number changed significantly as the Prandtl and the Dean numbers increased. Naphon (2011) studied the turbulent convective heat transfer in the spiral-coil tube both numerically and experimentally. The results showed that the induced centrifugal force in the spiral-coil tube has significant effect on the enhancement of heat transfer.

Based on the above discussion, taking advantage of both microfinned tube and the helical tube’s strong secondary flow characteristic may result in significant increase in flow heat transfer. This new type of helical tube can bring a breakthrough into the energy related industries in terms of designing more efficient and compact heat exchangers. Going through the previous work of the authors, no study was found investigating heat transfer characteristics of nanofluid inside microfinned helical tube for isothermal boundary condition. Since this behavior has not been reported, experiments with carbon nanotubes were planned to see if the same behavior was observed. As a result, in the present study, the effect of simultaneous utilization of microfinned helical tube and nanotubes as passive techniques on heat transfer has been investigated experimentally. The effects of various relevant parameters such as nanotube concentration and helical tube diameter on the heat transfer will be investigated. Finally, the results obtained for the microfinned helical tube is compared against those of the smooth helical tube.

2. Nanofluid preparation

Preparing stable suspension of nanoparticles in the base liquid is the first step in applying nanofluids for heat transfer enhancement. In order to prepare nanofluids by dispersing the nanoparticles in a base fluid, proper mixing and stabilization of the particles are required. Normally, there are three effective methods used to attain stable suspension of the nanoparticles, which are controlling the pH value of the suspensions, adding surface activators or surfactants and using ultrasonic vibration. All of these techniques aim to change the surface properties of suspended nanoparticles and suppress the formation of clustering particles in order to obtain stable suspensions. In this study, an ultrasonic vibrator is used to disperse the nanoparticles into the base oil and no surfactant is used since it might affect the physical or/and thermal property of the fluid. Multi-Wall Carbon nanotubes (MWCNT) with an average diameter of 10-30 nm prepared in Research Institute of Petroleum Industries (RIPI) were used as nanoparticle. Thermophysical properties of these nanoparticles calculated at 30^{o}C are shown in Table 1 as provided by manufacturer. Nanofluid was prepared by dispersing MWCNT nanoparticles in heat transfer oil (HT-B oil). Thermophysical properties of HT-B oil calculated at 38^{o}C are shown in Table 2 as provided by manufacturer. The nanofluids with three different nanoparticle weight concentrations of 0.1%, 0.2% and 0.4% were prepared and used to study enhanced heat transfer. In addition, pure base oil was used for the sake of comparison. Nanoparticles were mixed with heat transfer oil by electrical mixer for ten minutes. Following this, the nanofluids were sonicated continuously for 2-3 h using an ultrasonic vibrator (UP400S-Hielcher) in order to ensure complete dispersion. No agglomeration was observed for 12 hours after the dispersion process.

After preparation of the nanofluids, their thermophysical properties such as thermal conductivity, viscosity, specific heat and density are measured. Density for the base oil and nanofluids with different weight concentrations are measured through SVM3000 instrument. A differential scanning calorimeter (DSC F3 maia, manufactured by NETZSCH-Germany) was used to measure the specific heat of the nanofluids. The rheological behavior and viscosity of the nanofluids were measured by Brookfield visometer with a temperature controlled bath, supplied by Brookfield engineering laboratories of the USA. In addition, thermal conductivity of the nanofluids with different weight concentrations is measured using a KD2 thermal properties analyzer. More details on thermophysical properties of MWCNT/Oil nanofluid with the same concentrations as we used, are well documented by FakoorPakdaman *et al*. (2012). They developed some empirical correlations to calculate thermo physical properties of MWCNsT/Oil nanofluids which were compared against some well-known theoretical models for predicting nanofluid properties. Study on rheological behavior of nanofluids with concentration up to 0.4% showed that it exists linear relation between shear stress and shear rate, and therefore the nanofluids can be considered as Newtonian fluids. In addition, they reported an increase of about 67% in kinematic viscosity of 0.4% wt nanofluid compared to pure oil at T=40°C. They also concluded that the heat capacity of nanofluids is remarkably less than that of the base fluid. Heat capacity of 0.4 wt.% nanofluid was reported to be almost 42% less than that of the base fluid at about T=40°C. They also showed that thermal conductivity of nanofluid is increasing with

Table 1 Thermophysical properties of MWCNT nanoparticles

Mean diameter (nm)	10-30
Length of tube(μm)	10
Specific surface area(m^2/g)	270
Density (kg/m^3)	1474.5
Thermal conductivity (W/m K)	1500
Specific heat (J/kgK)	711

Table 2 Thermophysical properties of heat transfer oil (HT-B)

Kinematic Viscosity (m^2/s)	10-30
Density (kg/m3)	10
Thermal conductivity (W/m K)	270
Specific heat (J/KgK)	1474.5

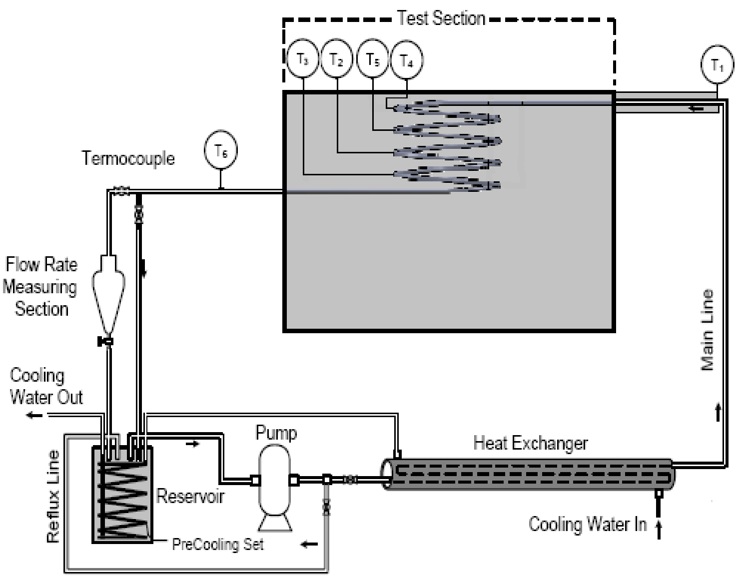

Fig. 1 Schematic diagram of experimental set-up

temperature and with nanoparticle concentration and reported the maximum enhancement of 15% in thermal conductivity for 0.4 wt.% nanofluid at 70℃ compared to the pure oil at the same temperature.

3. Experimental apparatus and procedure

An experimental setup is built to study the flow and convective heat transfer feature in the curved tube. As shown schematically in Fig. 1 the experimental system mainly includes a reservoir tank, a gear pump, a flow loop, a test section, a cooler and steam supplier tank. The transparent

Table 3 The geometrical characteristics of the coil tubes

Outer diameter of helical tube (mm)	9.52
Inner diameter of smooth helical tube, d (mm)	8.32
Inner diameter of micro-fin helical tube, d (mm)	8.42
Diameter of helical coil, 2R	210, 250, 290
Helical pitch (mm)	50
Number of coil turns	4.5
Curvature ratio (2R/d)	22.1, 26.26,30.46

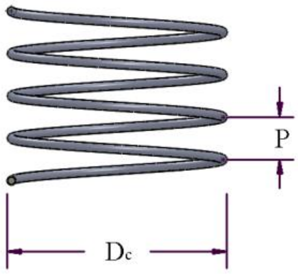

Fig. 2 Schematic of test section

plastic reservoir tank of 6 liter is manufactured to reserve the nanofluid and to monitor the sedimentation and level surface of nanofluid. The geometrical characteristics of the copper test sections including four coil tubes are listed in Table 3. Also, Fig. 2 shows schematic diagram of a test section. Four K-type thermocouples were mounted on the copper tube wall at equal interval by mercury welding to measure the wall temperature. The other two K-type thermocouples are inserted at the entrance and exit of test section to read the bulk temperature. The first 80 cm of the copper tube was thermally isolated from its upstream with fiber glass to minimize the heat loss and to guarantee hydro-dynamically fully-developed condition. The flow rate is controlled with two adjusting valves, one at the end of test section and the other at the by-pass line (by-pass line is shown in the figure as reflux line). The cooler is a shell and tube heat exchanger which is used to reduce temperature of the nanofluid at the exit of test section. The steam supplier tank of 120 liter capacity contains water and 8 KW element heaters to prepare steady saturated vapor. To reach constant wall temperature, entire test section is surrounded by saturated vapor. In order to minimize the heat loss from steam tank supplier to the ambient the whole tank is thermally isolated on the outside with a layer of fiber glass. One liter glass vessel with a drain valve was used for calculating flow rate. The stopwatch with accuracy of 0.01 was employed to measure the time required to fill the vessel and flow rate then was calculated.

The heat transfer performance of nanofluid was defined in terms of the convective heat transfer coefficient that can be expressed as

$$h_{nf}\left(\exp.\right) A_p (T_w - T_b)_{LMTD} = \dot{m} C_p (T_{b_{out}} - T_{b_{in}}) \tag{1}$$

$$h_{nf}\left(\exp.\right) = \frac{(\rho C_p)_{nf} \cdot A \cdot V (T_{b_{out}} - T_{b_{in}})}{\pi \cdot d \cdot L (T_w - T_b)_{LMTD}} \tag{2}$$

Where $T_{b_{in}}$ and $T_{b_{out}}$ are the bulk temperature of fluid flow in the inlet and outlet, respectively and $(T_w-T_b)_{LMTD}$ is the logarithmic mean temperature difference in which T_w is the wall temperature that is the average of four measured temperatures on tube wall at different positions. ρ_{nf} and C_{pnf} are the density and specific heat transfer of nanofuild respectively which were well documented by Fakoor *et al.* (2012). In Eq. (2), L is the length of coil tube and A is the tube cross-section area. The convective heat transfer coefficient is usually converted to Nusselt number (Nu) by

$$Nu_{nf}\left(\exp.\right)=\frac{h_{nf}(\exp.)\cdot d}{k_{nf}} \tag{3}$$

Where d is the tube diameter (inside diameter) and K_{nf} is the fluid thermal conductivity. Traditionally, the Nu is related to the Reynolds number and the Prandtl number defined as

$$\mathrm{Re}_{nf}=\frac{Vd}{\upsilon_{nf}} \tag{4}$$

$$\mathrm{Pr}_{nf}=\frac{\upsilon_{nf}}{\alpha_{nf}} \tag{5}$$

$$\alpha_{nf}=\frac{k_{nf}}{(\rho C_p)_{nf}} \tag{6}$$

Where υ_{nf} and α_{nf} are the kinematic viscosity and fluid thermal diffusivity, respectively. Three main parameters involved in calculating heat transfer rate of the nanofluid are heat capacity, viscosity, and thermal conductivity, which may be quite different from those of the pure fluid.

The rheological and physical properties of the nanofluid were calculated at the mean temperature. Then the Nusselt number and convective heat transfer coefficient at different concentration were calculated.

4. Results and discussion

As shown in Fig. 3 the experimental Nusselt number of straight tube for pure oil is compared with the theoretical data at thermal entrance region of the flow (Kays *et al*. (1993)). Good agreement of the experimental data with theoretical ones within the error of ±10% illustrates the accuracy of the instrumentation and the experimental apparatus. The experimental mean heat transfer coefficient for nanofluid flow with weight concentration of 0.4% and for different dimensionless curvature ratio is shown in Fig. 4.

From Fig. 4, it can be clearly seen that with decreasing of dimensionless curvature ratio, the mean heat transfer coefficient increases at constant Reynolds number. An enhancement of mean heat transfer coefficient is about 25% for *Re*=1721 at 0.4%wt concentration. It seems that the decreasing of curvature at constant Reynolds number causes increasing of centrifugal forces that leads to the augmentation of secondary flow. The centrifugal force drives part of the fluid with higher velocity toward the concave part of the curve channel, while the fluid close to convex part is slowing down.

In order to balance the pressure gradient in cross section, a double-vertical motion happens that causes a secondary flow in the main flow. This secondary flow increases the mixing and

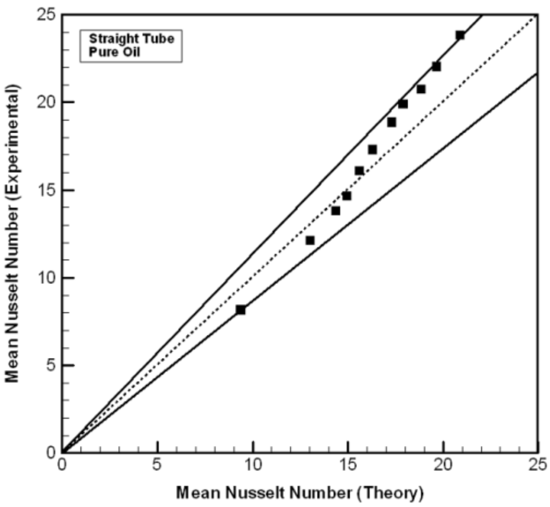

Fig. 3 Experimental mean Nusselt number versus theoretical Nusselt number

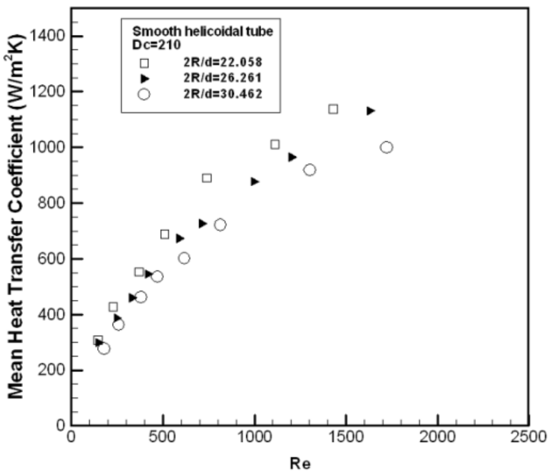

Fig. 4 Mean heat transfer coefficient of 0.4% wt nanofluid versus Reynolds number for different curvature ratios

turbulence of flow. Main consequences of this phenomenon are that thermal boundary layer thickness decreases and temperature profile at cross section of the tube becomes more flat. As a result the temperature gradient at interface of the fluid and tube increases, and then the mean heat transfer coefficient along the tube enhances.

The experimental mean heat transfer coefficients at constant dimensionless curvature ratio of $2R/d$=22.058 are shown in Fig. 5. This figure shows that with increasing the concentration of nanofluid, the mean heat transfer coefficient increases at constant Reynolds number. At $2R/d$=22.058 and Re=1720 an enhancement of mean heat transfer coefficient is about 41% with respect to the base fluid. Based on the results, it seems that the secondary flow generates more powerful mechanism for nanoparticles motion in flow. Increasing the concentration of nanofluid enhances conductivity. Moreover, good thermal balance that is due to the heat transfer from hotter

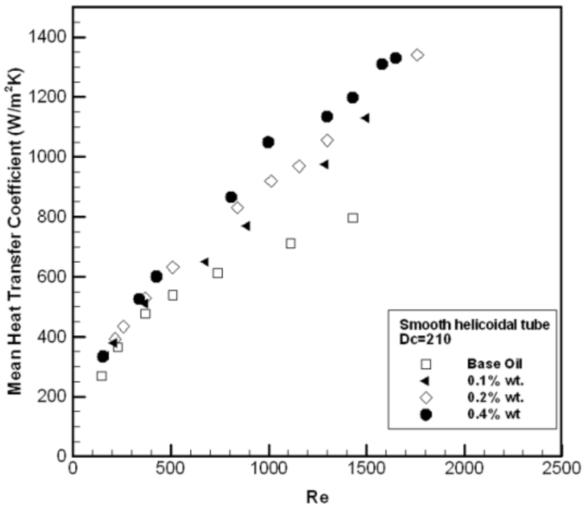

Fig. 5 Mean heat transfer coefficient versus Reynolds number for different nanofluids flow in tube with 2R/d=22.058

part to cooler part of the flow by Brownian motion and micro convection mechanisms causes the reduction of thermal boundary layer thickness and more flattened temperature profile at cross section of the tube. As a result, the temperature gradient at interface between fluid and tube increases which results in higher heat transfer coefficients.

Also, increasing the Reynolds number, which is the result of increasing mass flow rate, is making the secondary flow stronger. In addition, the augmentation of the secondary flow leads to the better mixing and dispersing of nanoparticles in flow and as indicated before it increases the mean heat transfer coefficient.

The experimental mean heat transfer coefficients in micro-fin helically coiled tube are shown in Fig. 6 for laminar region. As it can be seen in this figure, with the addition of nanoparticle to the base fluid the heat transfer coefficient increases substantially in micro-fin tube. However an increase in the weight concentration of the nanoparticle has no considerable effect on heat transfer enhancement. It is believed that for the nanofluid at constant Reynolds number, the effect of increasing secondary flow on heat transfer performance is much stronger than the turbulence effect of micro-fin tube. In such a case, the flow pattern is complex and there exist not only the effects of centrifugal and Coriolis forces on both the secondary flow and main flow, but also the disturbing effect of the internal finned surface. It is worth noting that this type of complex flow study has not been seen in the literature.

A comparison of the mean heat transfer coeficient of the flow inside the smooth and micro-fin helicoidal tube is made in Fig. 7. Heat transfer augmentation ratio is defined as the ratio of heat transfer coefficient of the flow inside microfin helicoidal tube to that of the flow inside smooth helicoidal tube. From Fig. 7, it can be seen that the heat transfer augmentaion ratio is between 1.12 and 1.20 at larger Reynolds number in laminar region. In other words, as the Reynolds number increases, the effect of micro-fin structure of the tube on heat transfer enhancement will slighltly increase. This slight enhancement is beacuase by increasing the mass flow rate, the convective heat transfer on the micro-fin surface is greatly suppressed due to the centrifugal and coirolis forces acting on the internal wall.

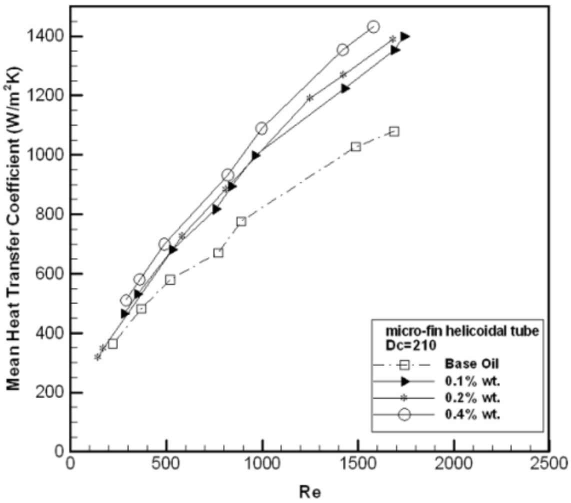

Fig. 6 Mean heat transfer coefficient versus Reynolds number for fluids flow inside the tube with 2*R/d*=22.058

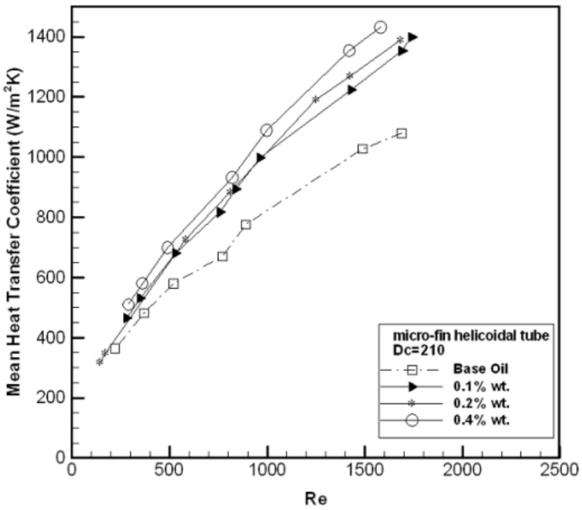

Fig. 7 Comparison of mean heat transfer coefficients of smooth and micro-fin helical tubes

4. Conclusions

The MWCNT/HB-Oil nanofluid convective heat transfer in laminar region inside a micro-fin helical tube was investigated experimentally. The nanofluids with nanoparticles weight concentrations less than 0.4% and also various helical tubes with three curvature ratios were used. The maximum value of 63% increase in heat transfer coefficient of nanofluid flow with 0.4% weight fraction and for 2*R/d*=22.058 at the Reynolds number of 1720 was observed.

The experimental results for smooth helical tubes indicated that heat transfer coefficient of nanofluids enhances with increasing the nanoparticle concentration as well as decreasing the curvature ratios. However, increasing the nanoparticles concentration did not show sensible effect on heat transfer of the flow inside micro-fin helical tube. Within the applied range of Reynolds number, compared with smooth helical tube, the average heat transfer augmentation ratio for the finned helical tubes was 17%. Moreover, it is concluded that the secondary flow is of first order of importance in the rate of enhancement attributed to the nanofluid flow inside helical micro-fin tubes.

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Nomenclature

A	Tube cross section area (m^2)
Cp	Specific heat capacity (KJ/kg.K)
d	Inside diameter of tube (m)
h	Heat transfer coefficient (W/m^2K)
K	Thermal conductivity (W/mK)
L	Tube length (m)
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
R	Diameter of helical coil (m)
T	Temperature (^oC)
V	Velocity (m/s)
Greek symbols	
P	Density (kg/m^3)
Y	Kinematic viscocity (m^2/s)
A	Thermal diffusivity (m^2/s)
Subscripts	
B	Bulk
exp.	Obtained experimentally
Nf	Nanofluid
In	Inlet
Out	Oulet
P	Solid nanoparticle
W	Wall