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*Original Article*

# Efficient shell and helically coiled tube heat exchanger using carbon nanotube-water nanofluid

Arjun Kozhikkatil Sunil* and Rakesh Kumar

*Department of Mechanical Engineering, Indian Institute of Technology (ISM), Dhanbad, 826004 India*

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## Abstract

Diameter and shape of tube, coil pitch and curvature ratio using 0.2% MWCNT-water nanofluid and 0.2-1% SWCNT-water nanofluid, affect the inner Nu and pressure drop in a shell and helically coiled tube heat exchanger, when numerically investigated. Flow configuration and fluid temperatures of both the nanofluids exhibited enormous change in inner Nu and pressure drop, compared with water at laminar (De 1610-2728) and turbulent (De 3019-4050) regime. When pure water in straight tube at the same Re is compared to 0.8% SWCNT-water nanofluid at 335 K with rectangular coiled copper tube of sides 10 mm and 7.85 mm, stretched length of 650 mm, 15 coil turns, coil pitch 15 mm, curvature ratio 0.24 and De 4025 at counter flow configuration, it had 139% higher inner Nu and 56% higher pressure drop. Very high performance index of 1.53 and Colburn factor of 3.25 were obtained using a single phase model.

**Keywords:** nusselt number, carbon nanotube, dean number, curvature ratio, performance index

## 1. Introduction

The enhancement in heat transfer efficiency with minimum pumping power consumption is a currently desirable goal. A coiled or curved tube is a swirl producing geometry where secondary fluid motion is generated by the continuous change in direction. Helically coiled heat exchangers offer advantages like compact size, possibility to accommodate large heat transfer areas within small space when compared with straight tube, higher film coefficients, and high pressure capability, handling high temperatures and extreme temperature differentials without excessive stresses. The helically coiled tubes have a wide range of applications in mixing, mass transfer and heat transfer. Coiled tubes are used as compact heat exchangers, condensers and evaporators.

Laminar heat transfer improvements by 70% and 190% were found for nanofluids containing 0.05 and 0.24 vol% CNT in water at Re=120, and an increase of overall heat transfer coefficient with Re (Wang, Zhu, Zhang & Chen, Y., 2013). MWCNT-water nanofluids at 0.25-0.5% exhibited 105% increase heat transfer at Re 2060 in (Fonseca *et al.*, 2013). A CFD study showed that there is not much difference in heat transfer performances of parallel and counter-flow configurations for a helical coiled double pipe heat exchanger (Behera, 2013). An heat improvements between 9 and 67% were found with optimum concentrations of gum arabic in a concentric tube turbulent flow heat exchanger containing 0.051-0.085 wt% CNT nanofluids (Ong & Walvekar, 2013). Thermal conductivity increases with CNT-water nanofluids have been reported (Fallahiyekta, Nasr, Rashidi, & Arjmand, 2014).

Significant enhancement of the convective heat transfer in a coaxial heat exchanger was found under a laminar flow regime with Re 500–2500 (Halelfadl, Estelle, & Mare, 2014). Homogeneous flow model along a flat plate subjected to Navier slip and uniform heat flux boundary conditions of CNTs found higher skin friction and Nu (Khan, Khan, & Rahi, 2014). Thermophysical effects of SWCNTs and MWCNTs on MHD flow over a stretching sheet found that an engine oil provides higher skin friction and heat transfer rates than water and ethylene glycol (Haq, Khan, & Khan, 2014).

*Corresponding author
Email address: arjunks1000@gmail.com

3% alumina nanoparticles and helical coiling could enhance the heat transfer coefficient by 60% from that of pure water in a straight tube at the same Re and reduce the pressure drop by 80%, while the effect of Re on pressure drop penalty factor was insignificant (Elsayed, Al-dadah, Mahmoud, & Rezk, 2014). The overall heat transfer coefficient increased with the use of nanofluids from concentric tube, shell and tube, to helical heat exchangers respectively (Seyyedvalilu & Ranjbar, 2015). Thermal conductivity enhancement of CNT-water nanofluids is mainly governed by both volume fraction (even at very low value) and temperature increase (Estelle, Halelfadl & Mare, 2015). The rise in temperature of cold water coming from the helical tube in counter flow arrangement is between 7$^{o}$C to 21$^{o}$C, which is more than that with a straight copper tube (Pravin & Kulkarni, 2015).

The effects of rectangular cross sections and CNT on heat transfer are very limited for helical coils. No work has reported on the effect of changing flow direction in shell and tube on convective heat transfer in a helically coiled tube by using CNT-water nanofluid. Effect of De in comparing Nu, performance index and Colburn factor is rarely reported and no reports could be found on helical coils. Most studies focus on constant wall temperature or constant heat flux, whereas fluid-to-fluid heat exchange studies are rare. Hence, in the present paper, optimization of efficient shell and helically coiled tube heat exchanger with Nu, pressure drop and Colburn factor using CNT-water nanofluid is carried out and discussed on the above aspects as well as the effects of helical coil tube radius, coil pitch, curvature ratio; and volume fraction and temperature of SWCNT-water nanofluid.

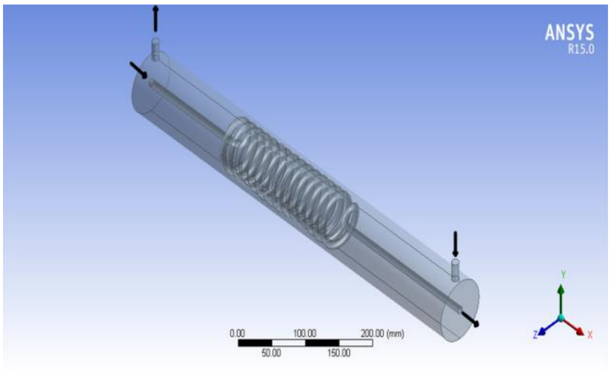


Figure 1. Model of shell and helically coiled tube heat exchanger.

## 2. Device Geometry and Computational Modelling

A helical circular tube of internal diameter 10 mm made of copper with 1.5 mm thickness, 15 coil turns and tube pitch 15 mm was chosen for the 3-D model in ANSYS finite element software, as shown in Figure 1. The inner coil diameter is taken as 50 mm and total stretched length of the tube is 650 mm with the length of coiled tube 225 mm, calming section 175 mm and wake section 250 mm. The shell is made of mild steel with internal diameter 80 mm and thickness 2 mm. 0.2 % MWCNT-water nanofluid is used in parallel flow with De = 2728, nanoparticle diameter 25 nm and fluid to fluid heat transfer is ensured. The flow rate of water in the shell side was kept constant and the nanofluid flow rate in the inner tube was varied. But, in the analysis, 11 mm and 12 mm tube diameters are also included, keeping coil diameter as well as length constant, to see the effect of change in curvature ratio on the inner Nu and pressure drop of helical tube. In the analysis, coil pitch of 10 mm, 20 mm and 25 mm also are tested. The flow directions of shell and coiled tube as counter flow and rectangular helical tube analyses are also performed. The effective thermo-physical properties of the nanofluid were formulated in a UDF subroutine and incorporated into Fluent solver (Rea, McKrell, Hu, & Buongiorno, 2009). Fluid properties are assumed constant with temperature. Insulated outer wall condition is used at outer shell wall and coupled wall boundary condition is used for tube wall for the heat transfer to occur on both sides, and no slip condition is required at each wall. The inlet temperature of the single phase fluid is taken as 295 K and gauge pressure at the outlet as zero atmospheric pressure. The domain was meshed with sweep cells and grid independence was obtained with 345000 and 110000 nodes in the coil fluid region and coil tube, respectively, in terms of average Nu and wall temperature. The assumptions used in this model were: The flow was steady and incompressible, the fluid density, dynamic viscosity and specific heat being constant throughout the computational domain, and the effect of heat conduction through the tube material is small.

All the governing equations were solved by using ANSYS FLUENT 15.0 finite volume method. Second order upwind scheme was used for solving momentum and energy equations. The convergence criterion was fixed such that the residual value was lower than $10^{-6}$. The pressure correction approach using the SIMPLE algorithm was used. Relaxation factors were at their default values. The turbulence model applied for present main stream flow analysis was Shear Stress Transport k-ε model with blending function and SST *k*-ω turbulence model with standard wall function for the analysis of flow near the boundary layer where the gradient is much steeper. SWCNT-water nanofluid with the same particle diameter is also used for analysis at the volume concentrations 0.02, 0.05, 0.08 and 0.1 at 335 K temperature. The inner Nu and pressure drop in the laminar regime (inner De 1610 and 2728) and the turbulent regime (inner De 3019 and 4025) are also analysed along with performance index. Colburn factors of SWCNT-water nanofluid and MWCNT-water nanofluid are analysed at the above De. Validation of simulated inner Nu and pressure drop values as a function of De in laminar and turbulent regimes using water used analytical values at the same De for which experimental values are reported (Kumar, Maheshwari, & Tripathi, 2015).

Applying boundary conditions, the governing equations for convective heat transfer are as follows:

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \tag{1}$$

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=\rho X-\frac{\partial p}{\partial x}+\frac{1}{3}\mu\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\mu\nabla^2 u \tag{2}$$

$$\rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\left(u\frac{\partial p}{\partial x}+v\frac{\partial p}{\partial y}+w\frac{\partial p}{\partial z}\right)+k\nabla^2T+\mu\emptyset \tag{3}$$

$$\emptyset=2\left[\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial w}{\partial z}\right)^2\right]+\left[\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2+\left(\frac{\partial w}{\partial x}+\frac{\partial u}{\partial z}\right)^2\right]-\frac{2}{3}\left[\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right] \tag{4}$$

$$De=(V\,d/v)(d/2R)^{1/2} \tag{5}$$

$$Nu=\frac{-\frac{\partial T}{\partial x}d_h}{T_w-T_f} \tag{6}$$

$$\Delta p=f\rho L v^2/2d \tag{7}$$

$$j=\frac{Nu}{RePr^{1/3}} \tag{8}$$

$$Re=\rho\,vd/\mu \tag{9}$$

$$Pr=C_p\mu/k \tag{10}$$

$$\eta=\frac{Nu_{nf}/\Delta p_{nf}}{Nu_{bf}/\Delta p_{bf}} \tag{11}$$

## 3. Validation

The simulated data is validated by experimental data on the inner Nu and pressure drop in shell and helically coiled tube heat exchanger using water at different De in laminar and turbulent regimes (Kumar, 2012). In Figure 2, both the cases of predicted results match accurately the experimental results with MSE of 6.099 *(*$R^2$=0.995***)* for inner Nu and MSE of 1.619 *(*$R^2$=0.973**)* for pressure drop.

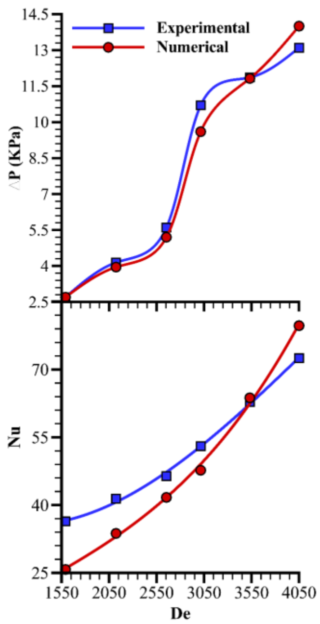


Figure 2. Validation of simulation with experimental data (Kumar, 2012) on Nu and Δp.

## 4. Results and Discussion

### 4.1. Effect of coil tube radius

As the coil tube radius increases from 5mm to 6mm; the Nu decreases by 3% and the corresponding decrease in pressure drop is 5% (Figure 3), using 0.2% MWCNT-water nanofluid at De = 2728. On increasing the coil tube radius, the effect of secondary flow diminishes and the fluid behaves like in straight pipe flow**.** As the helical tube radius becomes smaller, the effect of the secondary flow is intensified, because the centrifugal force is significant. This finding is in agreement with (Jayakumar, Mahajani, Mandala, Iyer, & Vijayan, 2010). The rate of heat transfer was increased by increasing the coil tube diameter for turbulent flow condition (Wael, 2014).

### 4.2. Effect of coil pitch

It is seen in Figure 4 that the inner Nu decreases when increasing coiled tube pitch from 15mm to 25mm, while it increased on change from 10mm to 15mm. The inner Nu of 0.2% MWCNT-water nanofluid at 25mm tube pitch is 14% less than that with 15mm tube pitch at De = 2728 and that at 15mm is higher than the inner Nu at 10mm, with a pressure drop decrease by 17%. This is because of the lower heat transfer coefficient when tube pitch is at 25mm and lower heat transfer rate with higher temperature difference between inner wall and bulk nanofluid at 25mm coiled tube pitch. The coiled tube at 15mm pitch gives better thermal performance than the 25mm coiled tube pitch. This is because the coiled tube is modified from close coiled condition (15mm) to loose coiled condition (25mm). In loose coiled condition, the coiled tube tends to approach the straight tube thermal behaviour. Further,

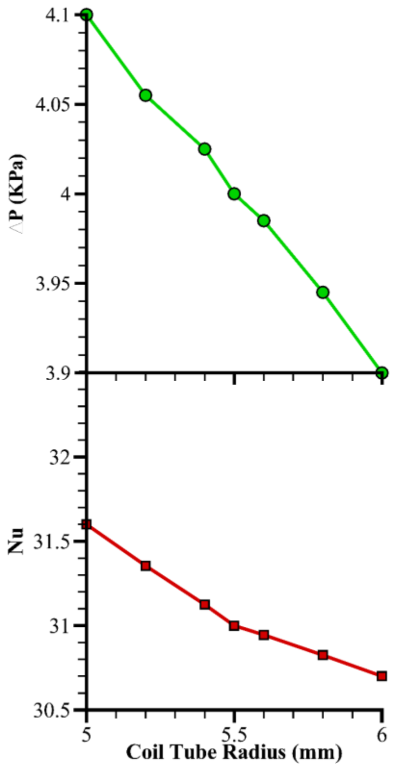


Figure 3. Effect of coil tube radius on Nu and Δp.

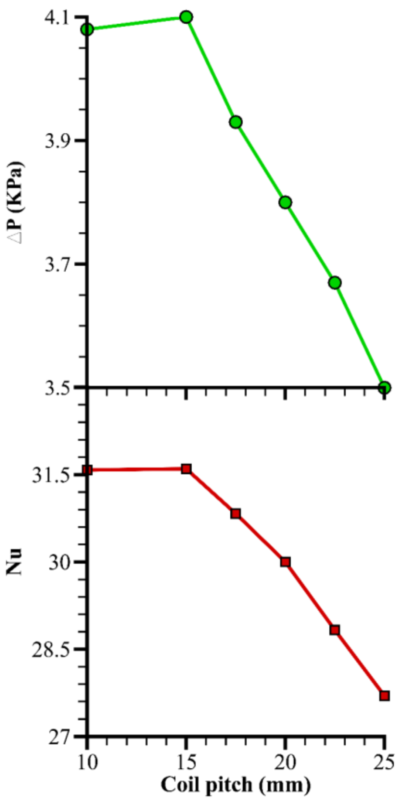


Figure 4. Effect of coil pitch on Nu and Δp.

the coiled tube at 25mm pitch generates relatively weak centrifugal force and mixing in the fluid. In close coiled condition, the curvature effect intensifies fluid mixing, resulting strong secondary flow, and this increases heat transfer. These results are in a good agreement with (Kumar, 2012).

### 4.3. Effect of fluid flow configuration, shape of cross section of coil and single/multi walled carbon nanotubes

On comparing the parallel flow and counter flow configurations, there is a significant impact of the flow configuration on inner Nu when MWCNT-water nanofluid is circulated (Figure 5) with inner De of 2728 while keeping the coiled tube pitch and coiled tube position same. The counter flow inner Nu as well as pressure drop are 10% higher than with parallel flow. Moreover, the length of the shell is 650mm and there is ample difference in inlet and outlet temperatures in shell side flow in the counter flow configuration in comparison to the parallel flow configuration. Counter-flow configuration produced better results than parallel-flow configuration (Mohammed & Narrein, 2012). The inner heat transfer coefficient of counter flow is 4-8% higher than that of parallel flow for 0.4% nanofluid, and 5-9% higher than that of parallel flow for 0.8% nanofluid in a helical coil tube (Kumar, 2012). Such enhancement can be achieved because the cold fluid (water) can approach the highest temperature of the hot fluid (nanofluid) at the entrance region of the hot fluid and due to the larger temperature difference between the two working fluids.

Figure 5 also depicts the effect of coil tube cross section shape on the heat transfer. As the shape of coil tube is changed to rectangular from circular, the Nu is increased by 3% with a corresponding 2% increase in pressure drop. The flow boiling in straight rectangular channels is better than in circular ones due to the holdup of liquid at the corners of the channel and thinning of the liquid film causing better heat transfer (Thome, 2004).

The changes of Nu and pressure drop for water based single or multi walled CNTs are presented in Figure 5. Using SWCNT-water nanofluid made a thermal improvement by 7% with an associated lower comparative increase of pressure drop of 2% in comparison to MWCNT-water nanofluid, due to the enhanced thermal conductivity of SWCNT-water

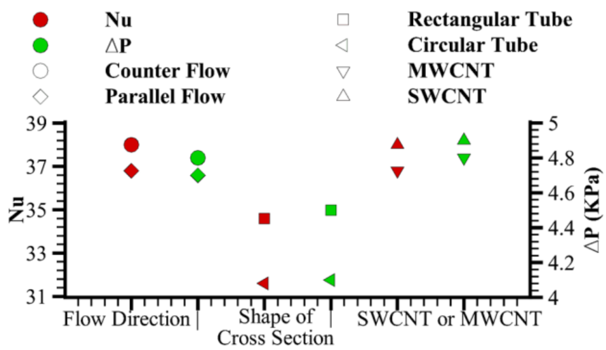


Figure 5. Effect of fluid flow configuration, shape of cross section of coil and single/multi walled carbon nanotubes on Nu and Δp.

nanofluid over MWCNT-water nanofluid. SWCNT-water nanofluids offer higher skin friction and Nu than MWCNT-water nanofluids over a static/moving wedge, according to a study under the influence of magnetic field (Khan, Culham, & Haq, 2015).

### 4.4. Effect of curvature ratio

Increasing the curvature ratio from 0.2 to 0.24 increases the Nu to the tune of 6% (Figure 6). Also, this causes comparatively lower pressure drop of 4%, though it increases. The maximum curvature ratio reported in literature is 0.125, although reports on ratios above 0.1 are rare. With decreased curvature ratio, the secondary forces acting on a fluid element due to flow inside helical tube will decrease. Due to decreased secondary forces, the turbulent mixing of fluid will also decrease, which reduces the heat transfer rate along with the Nu. The present findings are in agreement with (Wael, 2014).

### 4.5. Effect of volume fraction of SWCNT in water nanofluid

Both Nu and pressure drop improvements by 3% are achieved by adding different volume fractions of nano-particles (0.02-0.1) with an unfavourable increase after 0.08% volume fraction (Figure 7). The Nu increase from 0.08 to 0.1 volume fraction was 0.5%, whereas the pressure drop was significantly improved by 17.5%, proving that the ideal volume fraction of SWCNT in water nanofluid in a helical coil tube was 0.08%. Because the nanoparticles provide large surface area for molecular collisions, momentum transfer is improved with the concentration of nanoparticles. This momentum carries and transfers thermal energy more efficiently and enhances heat absorption by the coolant, causing its temperature to increase. Viscosity increases with increasing particle volume concentration and this suppresses the particle motion, which leads to higher friction factor. Nu increases with increased volume fraction in helical coil tube with $SiO_2$-water nanofluid (Maheshwari, Sahu, & Kumar, 2015) and increase in the nanoparticle volume fraction intensifies the interactions and collisions of nanoparticles (Kumar, Maheshwari, & Tripathi, 2015).

### 4.6. Effect of temperature of SWCNT-water nanofluid

Figure 8 illustrates the effects of SWCNT-water nanofluid temperature on Nu and pressure drop. As the nanofluid temperature increases from 295K to 335K; the Nu increases by 16% and the corresponding increase in pressure drop is 29%. The heat transfer enhancement at higher temperatures is from increased thermal conductivity at higher temperature of nanofluid. This result is in agreement with (Halelfadl, Estelle, Aladag, Doner, & Mare, 2013).

### 4.7. Effect of Dean number on Nusselt number and pressure drop

The Figure 9 indicates that increasing the De from 1610 to 4025 results in increasing Nu and pressure drop by

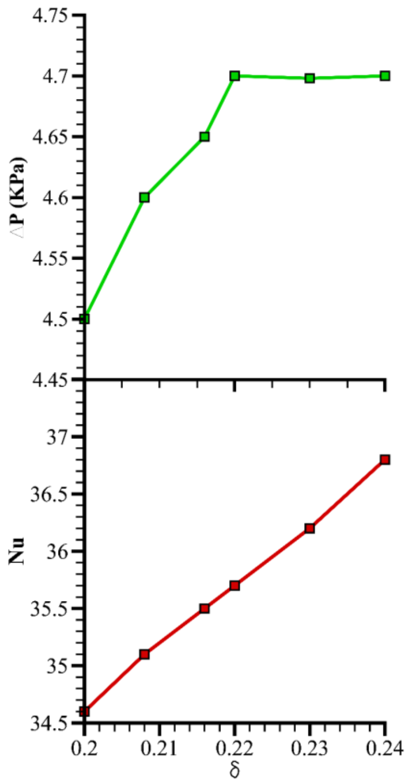


Figure 6. Effect of curvature ratio on Nu and Δp.

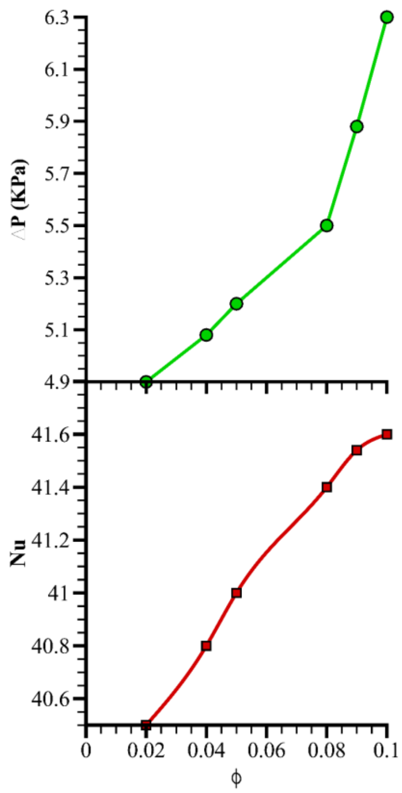


Figure 7. Effect of volume fraction of SWCNT in water nanofluid on Nu and Δp.

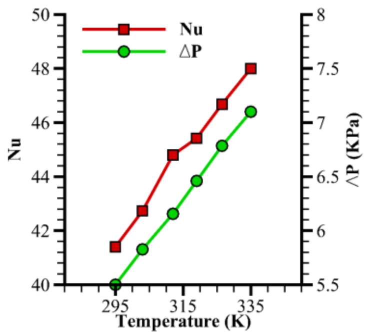


Figure 8. Effect of temperature of SWCNT-water nanofluid on Nu and Δp.

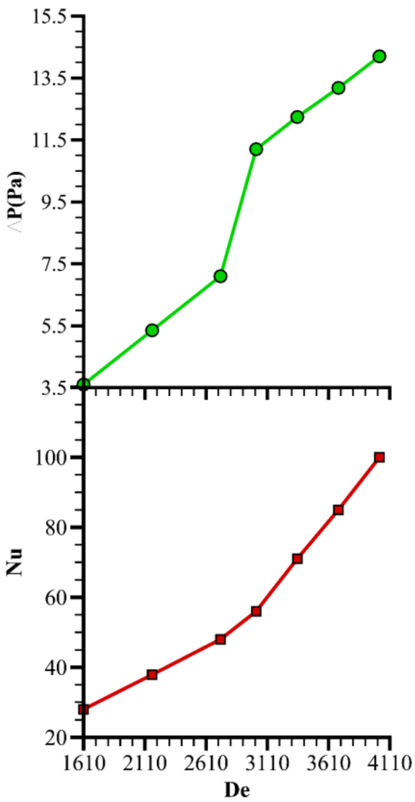


Figure 9. Effect of Dean number on Nu and Δp.

257% and 294% respectively, using SWCNT-water nanofluid. The increases in the laminar regime (De 1610 and 2728) were 71% and 97%, whereas in the turbulent regime (De 3019 and 4025) they were 79% and 27%; even though there was 11% higher variability in De range in the laminar regime than in the turbulent regime. There is a remarkable decrease of pressure drop by approximately 31% when flow switched to turbulent from laminar, and this is important as the power requirements are reduced. A higher Nu enhancement in the turbulent regime was found using SWCNT-water nanofluid than in the laminar regime. This might be due to the fact that De is directly related to the velocity of the flow. At higher flow rates, mixing fluctuations intensify due to the dispersion effects and chaotic movements of the nanoparticles and this increased Nu. $SiO_2$-water nanofluid possesses higher Nu on increase of De in a double pipe helical coil heat exchanger (Maheshwari, Sahu, & Kumar, 2015). The inner Nu and pressure drop at turbulent flow conditions with De 4025 for water were found to be 72 and 13 KPa and in a straight tube of length equivalent to the stretched length of the helical coil these were 41.9 and 9.1 KPa. The contributions of 0.8% SWCNT in water nanofluid were thus 39% and 9%, respectively, to inner Nu and pressure drop over the base fluid. The contributions of helical tube were 72% and 43% over the straight tube, and the overall increments to inner Nu and pressure drop were 139% and 56%, respectively, with the performance index being 1.53. Also, 55% higher inner Nu and 20% pressure drop were obtained by the use of 0.8% $Al_2O_3$-water nanofluid in a similar helical tube exchanger at De 4050 (Chougule & Sahu, 2014). Thus, by the use of SWCNT-water nanofluid instead of $Al_2O_3$-water nanofluid, 84% additional thermal enhancement could be obtained with a pressure drop penalty of 35%; when the shape of tube changed to rectangular from circular of same volume, temperature of nanofluid increased by 26 K and curvature ratio changed from 0.097 to 0.24. For 1% MWCNT-water nanofluid, the enhancement in heat transfer was by 91% over water in an automobile radiator (Chougule & Sahu, 2014).

### 4.8. Effect of Dean number on Colburn factor on using different nanofluids

Figure 10 proves that increasing De from 1610 to 4025 increased the Colburn factor of both SWCNT-water nanofluid and MWCNT-water nanofluid by 43% and 46, respectively. The corresponding changes in the laminar regime (De 1610 and 2728) were by 1% and 2%, whereas in the turbulent regime (De 3019 and 4025) they were 34% and 35; a remarkable increase in the Colburn factor by approximately 8% when flow switched to turbulent from laminar. The corresponding increase in Nu was only double. The increase in Colburn factor with De is due to stronger secondary flow when velocity is increased, which decreases thermal boundary layer thickness and migration of nanoparticles. In terms of the Colburn factor, when used in place of Nu, the variation in heat transfer was reduced to approximately one sixth. The magnitude of difference in cube root of Prandtl numbers of SWCNT-water nanofluid and MWCNT-water nanofluid is lower than the magnitude of difference of their Nu. This is because the ratio of momentum and energy molecular transport coefficients for SWCNT-water nanofluid is higher than that for MWCNT-water nanofluid, and also owing to higher thermal conductivity of the former. Different methods of calculation can produce changes in Re as large as 15%, and this corresponds to a difference in the Colburn factor of over 12% (Pike, 2012).

The non-monotonic variation in trend ( change in direction of slope) observed in Figures 4, 6, 7, 9 and 10 might be due to a change in the underlying flow structure. There might be a change in the flow behaviour from primary Dean vortex to secondary Dean vortex, as the local curvature might induce viscous dissipation. For channels with weaker or stronger streamwise curvature, the critical De increases almost linearly, and for strongly curved channels there is a direct transition from laminar to vortex flow with 4 vortices (Debus, Mendoza, & Herrmann, 2015).

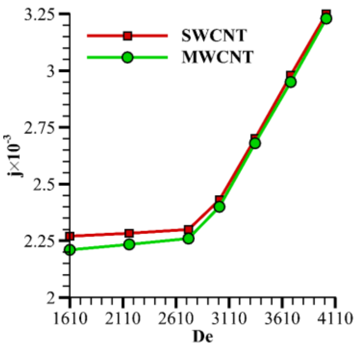


Figure 10. Effect of Dean number on Colburn factor CNT-water nanofluids.

## 5. Conclusions

The inner Nu and pressure drop increase with increasing De, volume concentration and temperature of SWCNT-water nanofluid and decreasing diameter of coil tube, coiled tube pitch and curvature ratio.

Counter flow configuration was beneficial over parallel flow configuration, rectangular tube was better than circular tube, and SWCNT nanofluid was thermally better than MWCNT-water nanofluid.

Increasing De increased the Colburn factor of both SWCNT and MWCNT-water nanofluids by 34-35% (3 fold increase in Nu) in the turbulent case, and little in the laminar regime.

The performance index for SWCNT-water nanofluid was higher than that for MWCNT-water nanofluid (higher temperature gave higher Nu, but with lesser performance index).

The ideal volume fraction of SWCNT in water nanofluid in a helical coil tube is 0.8% at 335 K, and ideal tube diameter, curvature ratio and coil pitch were found to be 10 mm, 0.24 and 15 mm respectively.

The MSE between exprimental and CFD inner Nu was found to be 6.1 with $R^2$ of 0.995 and that of pressure drop was 1.6 with $R^2$ of 0.973.

The overall improvements of inner Nu and pressure drop were by 139% and 56%, respectively, with a high performance index of 1.53 at De 4025, relative to base fluid and straight tube. The contributions of SWCNT-water nanofluid and helical tube were 39% and 9%, and 72% and 43%, respectively.

Use of SWCNT-water nanofluid instead of $Al_2O_3$-water nanofluid, changing tube shape of same volume from circular to rectangular, increasing temperature of nanofluid by 26 K and changing curvature ratio from 0.097 to 0.24 effected 84% additional thermal performance with 35% pressure drop penalty.

## Nomenclature

| | |
|---|---|
| Cp | Specific heat, kJ/kg K |
| d | Diameter of tube, mm |
| D | Coil diameter, mm |
| dh | Hydraulic diameter, mm |
| H | Pitch of coil, mm |
| j | Colburn factor |
| k | Thermal Conductivity, W/mK |
| p | Kinematic pressure, kPa |
| R | Radius of tube curvature, mm |
| n | Number of turns |
| u, v, w | Velocity (X, Y & Z axes), m/s |
| V | Fluid velocity, m/s |
| μ | Viscosity, kg/m.s |
| ρ | Fluid density, kg/m$^3$ |
| υ | Fluid kinematic viscosity, kg/m.s |
| η | Performance index |
| φ | Rayleigh dissipation function |

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