# Modeling and CFD Analysis of Tube in Tube Helical Coil Heat Exchanger

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**Abstract:** *A helical coil heat exchanger has a wide range of application in industries over the straight and shell type heat exchangers because of its greater heat transfer area, mass transfer coefficient and higher heat transfer capability, etc. The relevance of helical coil heat exchanger has been identified in industrial application like turbine power plants, automobile, aerospace, etc. because of above mentioned factors. In this thesis we model a tube in tube helical coil heat exchanger using CATIAV5r18 and done CFD analysis using ANSYS. The thesis shows the deviation of Nusselt Number for different curvature ratio(D/d ratio) and Reynolds Number.CFD analysis has been done for varying inlet condition keeping the heat flux of outer wall constant. The turbulent flow model with counter flow heat exchanger is considered for analysis purpose. Copper was used as the base metal for both inner and outer pipe and simulation has been done using ANSYS 13.0.The software ANSYS 13.0 was used to plot the temperature contour, velocity contour. pressure contour taking cold fluid at constant velocity in the outer tube and hot fluid with varying velocity in the inner one. We also find out the wall shear stress on both inner and outer tube. Water was taken as the working fluid for both inner and outer tube. Result after analysis shows that temperature, pressure velocity contour in the heat exchanger were similar to literature data and It is also visible from the results that Nusselt Number depends on curvature ratio. It is increasing with increase in curvature ratio. In addition, the value of Nu no. was found to increase with increase in mass flow rate (i.e. inlet velocity), with increases in D/d ratio (inverse of curvature ratio) the Nusselt number will decreases; for a particular value of Reynolds number. Nusselt number has maximum value for D/d=10.*

**Keywords**: Helical double tube Heat exchanger, Secondary flow, Curvature ratio, Nusselt number

## 1. Introduction

An HCHE consist of a helical coil fabricated out of a metal pipe that is fitted in the annular portion of two concentric cylinders as shown in fig 1.One fluid flows through the inner pipe and other fluid flows through the annular space between the pipe with heat transfer taking place across the coil wall. The dimension of the both cylinders is determined by the velocity of the fluid in the annulus needed to meet heat transfer requirement". Basically there are two flow arrangements in double pipe helical coil heat exchanger i.e. parallel flow and counter-flow. In parallel flow as the name suggest, The two fluid streams (hot and cold) travel in same direction. The two fluid enter at one end and travel leave at the other end and in a counter flow, The two fluid flow in opposite directions. The hot and cold fluid enter at the opposite ends. It can be arranged in a lot of series and parallel configurations to meet the different heat transfer requirements. Heat transfer in helical coils are higher than as compared to straight coils. Because of its compact size, higher film coefficient, They are widely used in industrial applications like power generation, nuclear industry, process plant, heat recovery system, chemical process industries etc. These heat exchanger are used to control the temperature of the reactors for exothermic reactions. They have less expensive design. Helical geometry allows the effective handling at higher temperatures and extreme temperature differentials without any highly induced stress or expansion of joints. Helical coil heat exchanger consists of series of stacked helical coiled tubes and the tube ends are connected by manifolds, Which also acts as fluid entry and exit locations. The enhancement in heat transfer in helical pipe is due to the complex flow pattern existing inside the pipe. The helix angle and the pitch of the coil results in the torsion of the fluid and due the curvature of the coil centrifugal force is

generated. The centrifugal force develops a secondary flow inside the helical tube which is normal to primary axial direction of flow so circulatory motion is generated, the fluid in the core of the tube move towards the outer wall, then return to the inner portion of tube as shown in fig 3 and enhances the heat transfer per unit length as it reduces the temperature gradient across the cross section of the tubes. Another advantage to using helical coils over straight tubes is that the residence time spread is reduced, allowing helical coils to be used to reduce axial dispersion in tubular reactors (Ruthven,1971). Thus, for design of heat exchangers that contain curved tubes, or helically coiled heat exchangers, the heat transfer and hydrodynamic characteristics need to be known for different configuration of the coil, including the ratio of tube radius to coil radius, pitch, and Reynolds and Prandtl numbers.
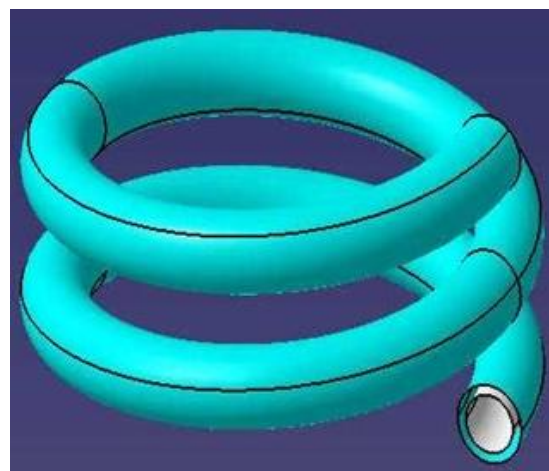

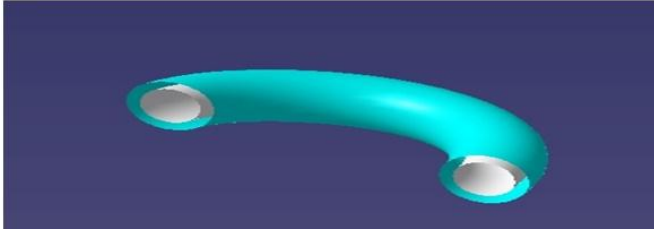
**Figure 1:** Model of tube in tube helical coil

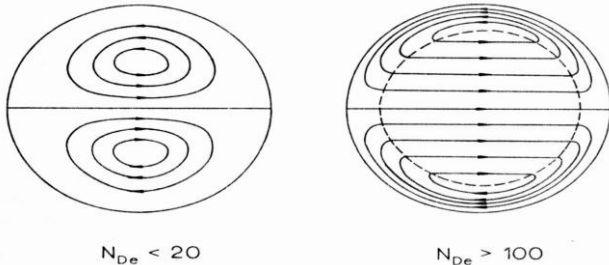**Figure 2:** Close up view of tube in tube helical coil



**Figure 3:** Secondary flow for low and high Dean numbers (Dravid et al., 1971)

## 1.1 Terminology of helically coiled pipes

Figure 4 gives the schematic of the helical coil. The pipe has an inner diameter $2r$. The coil diameter is represented by $2RC$ (measured between the centres of the pipes). The distance between two adjacent turns, called pitch is $H$. The coil diameter is also called as pitch circle diameter ($PCD$). The ratio of pipe diameter to coil diameter ($r/Rc$) is called curvature ratio, $\delta$. The ratio of pitch to developed length of one turn ($H/2\, \pi Rc$) is termed non-dimensional pitch, $\lambda$. Consider the projection of the coil on a plane passing through the axis of the coil. The angle, which projection of one turn of the coil makes with a plane perpendicular to the axis, is called the helix angle, $\alpha$. Consider any cross section of the pipe created by a plane passing through the coil axis. The side of pipe wall nearest to the coil axis is termed inner side of the coil and the farthest side is termed as outer side of the coil. Similar to Reynolds number for flow in pipes, Dean number is used to characterize the flow in a helical pipe.
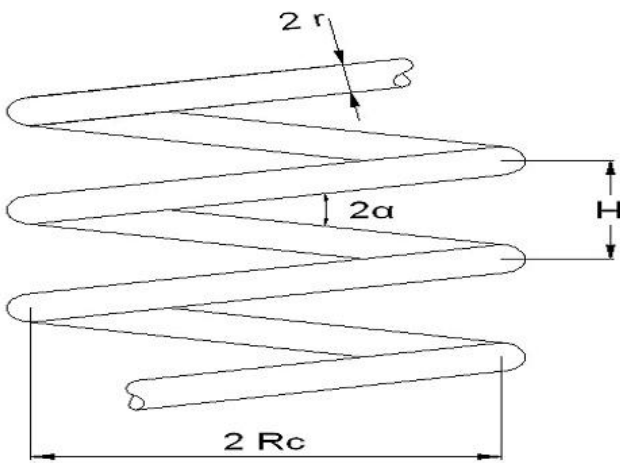


**Figure 4:** Basic geometry of a helical pipe

The Dean number, $De$ is defined as,
$$De = Re\sqrt{(r/Rc)} \qquad (1)$$
where, $Re$ is the Reynolds number $= 2ru_{av}\rho\,/\mu.$ $\qquad$ (2)

## 2. Literature Review

**Aly et al.(2014)** had done the numerical study to find the heat transfer and pressure drop of nano fluid in coiled tube-in-tube heat exchangers for turbulent flow condition. The numerical study was carried out by computational fluid dynamics (CFD) analysis to find the heat transfer rate and pressure drop characteristics of water-based Al2O3 nano-fluid flowing inside coiled tube-in-tube heat exchangers. The overall performance the heat exchangers was assessed based on the thermo-hydrodynamic performance index. Nanofluid flows inside inner tube side. When he compared the result for the same Re or Dn, the heat transfer coefficient or the rate of heat transfer was increased by increasing the coil diameter and nanoparticles volume concentration. Also, he found that the friction factor increases with the increase in curvature ratio and the pressure drop penalty is negligible with increasing the nano-particles volume concentration up to 2%.For the nano fluid the correlations for predicting average heat transfer and friction factor in turbulent flow regime such as Gnielinski correlation and Mishra and Gupta correlation, respectively, for helical tubes are also valid. In results they found that nano fluids behave like a homogeneous fluid.

**Bai et al.(1999)** experimentally studied turbulent heat transfer from horizontal helical coils. They concluded that as the Reynolds number is increased, the contribution of secondary flow to the heat transfer diminished and the heat transfer approaches that of a straight tube. This is due to the fact that as the Reynolds number increases the boundary layer becomes smaller. It is the large boundary layer that is shed off into the center of the tube by the secondary flow that increases the heat transfer coefficient, and this effect decreases with increasing Reynolds number (Bai et al., 1999).The local heat transfer coefficient on the outer wall can be 3 to 4 times that of the inner wall. They developed a correlation of the Nusselt number as a function of the location on the periphery. They also developed a Nusselt number correlation; however it did not contain the Dean number as only one size of coil was used in the experiment. Comparisons for the heat transfer coefficients between straight tubes and helically coiled tubes immersed in a water bath were performed by Prabhanjan et al. (2002).Findings showed that the heat transfer coefficient were greater in the helically coiled system.

**C.A chaves et al.(2014)** had done the comparative performance study of two different helically coiled heat exchanger with two and three helical coils through a computational fluid dynamics(CFD) simulation for heat transfer characteristics. Numerical study where performed with the assistance of a commercial computational fluid dynamics package(ANSYS-CFX v12).Simulations were performed using various temperatures(hot fluid inlet temperature 25,30,35,40$°C$) and the inlet coil fluid temperature is 20$°C$.Results indicated that performance of both heat exchanger for temperature 25$°C$ (hot fluid inlet) was quit similar, but for temperature 40$°C$ (hot fluid inlet),the heat exchanger with three turn was more efficient than another heat exchanger(two turn).It was shows that the

performance could be increased by increasing hot fluid inlet temperature with two and three helical coils.

**Daniel Flórez-Orrego.(2012)** have studied the characteristics of single phase cone shaped helical coil heat exchanger. They conducted experiments on a prototype of cone helical coil heat exchanger with maximum diameter of 15 cm and minimum diameter as 7.5 cm,3/8 inch pitch and axial length of 40 cm. The flow was in both laminar and turbulent and the range of Reynolds number and Prandtl number were 4300-18600 and 2-6 respectively. According to this study Nusselt number can be found out by $Nu=CRe^m Pr^n$, where C, m are constants that to be determined and n is the Prandtl number index which is taken as 0.4.An empirical correlation was proposed for average Nusselt number, and it was found that there was a maximum deviation of 23%. Inclination of the velocity vector components in the secondary flow was observed unlike in the straight helical coils. These correlations are not reliable and it failed to give any deviation in Nusselt number due to the tapering and the effect of pitch.

**Ferng et al. (2012)** had done the numerical work in a helically coiled heat exchanger. Numerical investigation was focused to predict the effects of Dean number and pitch size of the tube on the thermal and hydraulic characteristics of a helical tube heat exchanger. They had considered three Dean numbers and four sizes of pitch for their study.The turbulent wake around the rear of a coiled tube, the secondary flow within the tube, and the developing flow and heat transfer behaviours from the entrance region, etc was studied by them.

**Genic Srbislav et al. (2012)** had done the experimental work predict the performances of heat exchangers with helical tube coils. In their work they had presented the results of thermal performance measurements on 3 heat exchangers with concentric helical coils. It was found that the shell-side heat transfer coefficient was affected by the geometric parameters. Winding angle, radial pitch and axial pitch are the geometric parameters which affect the heat transfer coefficient. From the results it was concluded that the shell-side heat transfer coefficient is based on shell side hydraulic diameter. Final form of shell-side heat transfer correlation proposed by Srbislav et al. (in which Nusselt and Reynolds numbers are based on hydraulic diameter) is given by,
$$Nu=0.50Re^{0.55}Pr^{1/3}(\eta/\eta_w)^{.14} \quad (3)$$

**Harith Noori Mohammed(2009)** had done the experimental study on steady-state natural convection heat transfer from helical coil tubes. Water was used as a bath liquid without any mixing and air was used as a coolant fluid. A straight copper tube of 13 mm ID, 14 mm OD and 3 m length was bend to fabricate the helical coil. Two coils are used in this experiment has a curvature ratio of 0.1101 and 0.0942.The data were correlated using tube diameter as the characteristic length. The results show that the overall heat transfer coefficient and pressure drop are increased when the flow rate of coolant and curvature ratio increase.

**Inagaki et al. (1998)** studied the outside heat transfer coefficient for helically coiled bundles for Reynolds numbers in the range of 6000 to 22 000 and determined that the

outside Nusselt number could be described by the following relationship for their particular setup:
$$Nu=0.78Re^{0.51}Pr^{0.3} \quad (4)$$

**Jayakumar et al.(2010)** had done the numerical and experimental analysis to find out the variation of local Nusselt number along the length and circumference of a helical tube. They had changed the pitch circle diameter, tube pitch and pipe diameter and their influence on heat transfer rate was found out. They have done the prediction of Nusselt number. The of Nusselt number variation with respect to angular location of the point was also predicted in this literature.

In their conclusion they found that the heat transfer coefficient and hence the Nusselt number is not uniform along the periphery of the helical pipe. They had derived an expression to calculate the Nusselt number at various points along the periphery of the tube in the fully developed region. The effect of pipe diameter was studied and it is found that when the pipe diameter is low, the secondary flow is weak and the mixing of the fluid is less. When the diameter of the coil increases the heat transfer at the outer surface is highest. The PCD influence the centrifugal force of fluid flowing inside the tube, which in turn affects the secondary flow. When the PCD is increased, the curvature effect on flow pattern decreases and the centrifugal force plays a lesser role in flow characteristics.

**Kumar et al. (2006)** had investigated hydrodynamic and heat transfer characteristic of tube in tube helical heat exchanger at pilot plant scale. They had done the experiment in a counter flow heat exchanger. Overall heat transfer coefficients were evaluated. Nusselt number and friction factor coefficient for inner and outer tube was found and compared with numerical value got from CFD package (FLUENT).They observed that the overall heat transfer coefficient increase with inner coil tube Dean Number for constant flow rate in annulus region.

**Kharat et al. (2009)** had done the experiments to study the heat transfer rate on a concentric helical coil heat exchanger and develop the correlation for heat transfer coefficient. Heat transfer coefficient has improved for the tube containing flue gas of the heat exchanger by using CFD simulation and the experimental study. The effect of different operating variables was studied. The variables they had considered are gap between the concentric coils, diameter of tube and coil diameter. The heat transfer coefficients are affected by the coil gap and the tube diameter. They found that the heat transfer coefficient decreases with the increase in coil gap. With increase in tube diameter the heat transfer coefficient increases.

**Seyed Faramarz Ranjbar et al(2014)** had done a numerical investigation of the influence of different parameters such as coil radius, coil pitch and diameter of tube on the characteristics of heat transfer in helical double tube heat exchangers using the well-known Fluent CFD software. Modeling of the study was implemented based on principles heat transfer, fluid mechanics, and thermodynamics. By imposing boundary conditions and selecting of an appropriate grid, whereby the results are independent of meshing, the

obtained results were compared and validated with existing experimental results in the open literature. The results indicate that heat transfer augments by increasing of the inner Dean number, inner tube diameter, curvature ratio, and by the reduction of the pitch of the heat exchanger coil.

## 3. Model and Analysis

In my study I consider the double tube helical coil heat exchanger or tube in tube helical coil heat exchanger with two (2) numbers of turns. For simplification in numerical analysis I consider only two turns but in practical problems it may be large number of turns depending on the requirements. The coil diameter (D) was varying from 80mm to 240mm in an interval of 40mm that is120mm,160mm,200mm respectively. As the coil diameter increases the length of the exchanger (L) also increases. The inner tube diameter (d1) was 8mm.the thickness (t) of the tube was taken 0.5mm.The outer tube diameter (d2) was taken 17mm.In my study I fixed the tube diameter (both inner and outer diameter) of the heat exchanger and vary the coil diameter of the tube to see the effect of curvature ratio (d/D) on heat transfer characteristics of a helical coil heat exchanger. The pitch of the coil was taken 30mm that is the total height of the tube was 60mm. The heat exchanger was made of COPPER. The fluid property was assumed to be constant for analysis. In this study I considered the counter flow heat exchanger as it has better heat transfer rate compared to parallel heat exchanger. The cold fluid and the hot fluid flow in opposite directions in their respective tube. In this study for analysis, turbulent fluid flow was considered. Both the hot fluid and cold fluid flow with a velocity, for which Reynolds number is greater than critical Reynolds number as per the correlation calculated by Schmidt,(1967).The flow velocity of cold fluid is remained constant and the hot fluid flow rate varied to find the heat transfer rate, friction factor and optimize the heat exchanger to have minimum pressure loss and maximum heat transfer.

After creating the geometry on CATIA V5R18 and doing the meshing in ANSYS 13 the problem was analyzed in ANSYS 13 (FLUENT) for different boundary conditions as specified later. For analysis of the problem turbulent fluid flow condition was considered.

### 3.1 Mathematical Relations

Heat transfer coefficient is obtained by equating the conduction heat transfer to the convection heat transfer;
$q_{cond} = q_{conv}$ (5)

$$h= \frac{-k\frac{\partial T}{\partial X}}{T_W-T_f} \qquad (6)$$

Local Nusselt number is given by;

$$Nu\ x=hD/k \qquad (7)$$

Or it can also be represented by following equation

$$Nu= \frac{\frac{\partial T}{\partial X}d_h}{T_W-T_f} \qquad (8)$$

**Correlation for estimation of Nusselt number**

The correlation for Nusselt number already consists of pipe diameter in terms of Reynolds number and curvature ratio. Hence the correlation can be of the form,

$$\mathbf{Nu=CRe^nPr^{0.4}\delta^m} \qquad (9)$$

where *C, n* and *m* are to be evaluated. If we use Dean number in the formulation, the curvature ratio term needs to be included twice. Hence Reynolds number is chosen in the general form of the equation for estimation of Nusselt number.

Correlation for estimation of Nusselt number for constant wall heat flux boundary condition given by Jayakumar et al**.**

$$\mathbf{Nu=0.085Re^{0.74}Pr^{0.4}\delta^{0.1}} \qquad (10)$$
where,$3<Pr>5$

**Correlation for Friction factor by Guo et al.**

$$f=2.552Re^{-0.15}(d/D)^{0.51} \qquad (11)$$

### 3.2 Boundary conditions

**Table 1:** Properties of copper

| Description | Value | Units |
|---|---|---|
| Density | 8978 | kg/m3 |
| Specific heat | 381 | J/kg-K |
| Thermal conductivity | 387.6 | W/m-K |

**Table 2:** Properties of water

| Description | Symbol | Value | Units |
|---|---|---|---|
| Density | ρ | 1000 | kg/m3 |
| Dynamic Viscosity | μ | 0.001003 | kg/ms |
| Specific Heat | Cp | 4182 | J/kgK |

**Table 3:** Boundary Condition for Inner Fluid

| Inlet condition | Inlet velocity (varies from 0.9942to1.8842 m/s) |
|---|---|
| Outlet condition | Pressure outlet |
| Outlet condition | Zero Pascal |
| Inlet temperature | 283k |

**Table 4:** Boundary Condition for Annulus Fluid

| Inlet condition | Inlet velocity 0.9942m/s |
|---|---|
| Outlet condition | Pressure outlet |
| Outlet condition | Zero Pascal |
| Inlet temperature | 348k |

## 4. Result

### 4.1. Contour

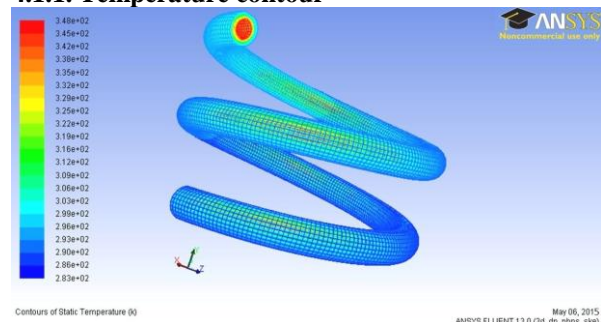#### 4.1.1. Temperature contour



**Figure 5:** Temperature Contour

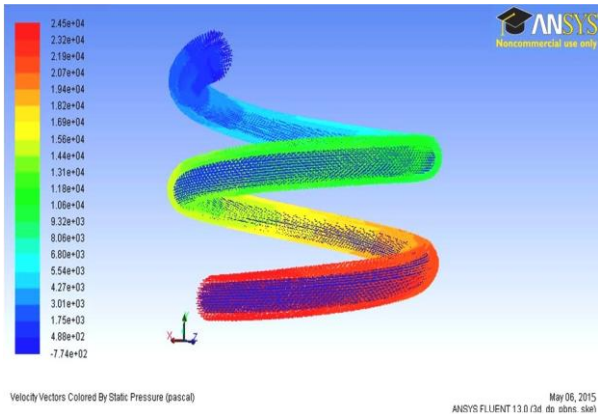### 4.1.2. Pressure contour



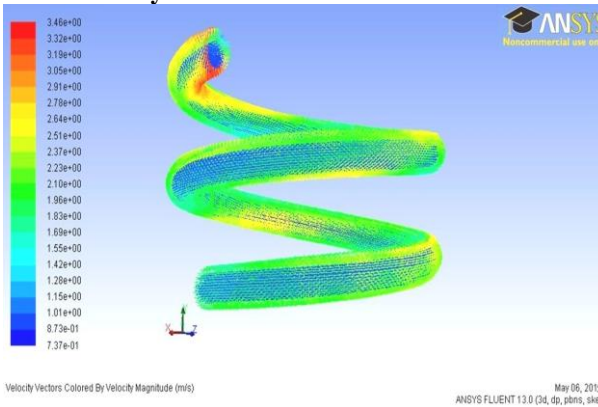**Figure 6:** Pressure Contour

### 4.1.3. Velocity Contour



**Figure 7:** Velocity Contour

### 4.2. Tabulation

**Value of Nu and f for different D/d value**

| D/d | V(m/s) | Nu | F |
|-----|--------|--------|---------|
| 10 | 1 | 95.558 | 0.198 |
| 10 | 1.2 | 109.36 | 0.1927 |
| 10 | 1.4 | 122.575 | 0.1883 |
| 10 | 1.6 | 135.305 | 0.184 |
| 10 | 1.8 | 147.628 | 0.181 |
| 15 | 1 | 90.799 | 0.1602 |
| 15 | 1.2 | 103.914 | 0.1559 |
| 15 | 1.4 | 116.470 | 0.1523 |
| 15 | 1.6 | 128.567 | 0.14935 |
| 15 | 1.8 | 140.276 | 0.14674 |
| 20 | 1 | 89.15 | 0.139 |
| 20 | 1.2 | 102.037 | 0.1353 |
| 20 | 1.4 | 114.336 | 0.1323 |
| 20 | 1.6 | 126.244 | 0.1296 |
| 20 | 1.8 | 137.74 | 0.1273 |
| 25 | 1 | 87.19 | 0.1241 |
| 25 | 1.2 | 99.785 | 0.1207 |
| 25 | 1.4 | 111.842 | 0.1180 |
| 25 | 1.6 | 123.458 | 0.1156 |
| 25 | 1.8 | 134.702 | 0.1136 |
| 30 | 1 | 85.53 | 0.1125 |
| 30 | 1.2 | 97.884 | 0.1095 |
| 30 | 1.4 | 109.711 | 0.107 |
| 30 | 1.6 | 121.106 | 0.104 |
| 30 | 1.8 | 132.135 | 0.103 |

### 4.3. Plots
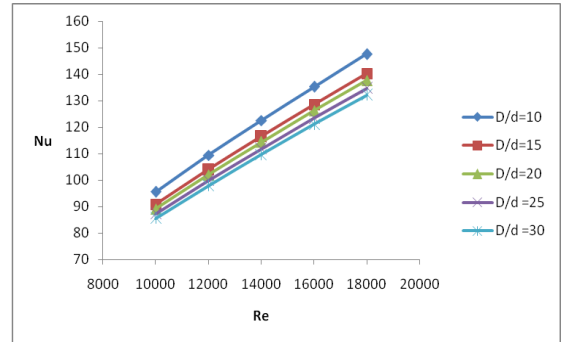


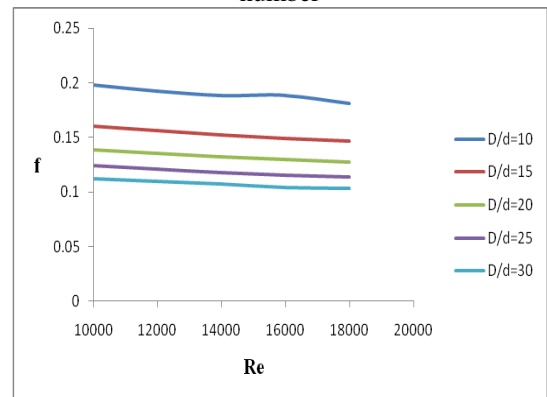**Figure 8:** Variation of Nusselt Number with Reynolds number



**Figure 9:** Variation of Friction factor with Reynolds number

## 5. Conclusion

A CATIAV5r18 and CFD package (ANSYS FLUENT 13.0) was used for modeling and CFD study of heat transfer characteristics of a helical coiled double pipe heat exchanger for counter flow. Characteristics of the fluid flow were also studied for the constant temperature and constant wall heat flux conditions. Several important conclusions could be drawn from the present simulations and would be presented as follows. The fluid particles were undergoing an oscillatory motion inside both the pipes. Along the outer side of the pipes the velocity and pressure values were higher in comparison to the inner values. The shear stress at wall of inner pipe is greater than the wall of outer pipe. It is visible from the results that Nusselt Number depends on curvature ratio. It is increasing with increase in curvature ratio. In addition, the value of Nu no. was found to increase with increase in mass flow rate (i.e. inlet velocity), With increases in D/d ratio (inverse of curvature ratio) the Nusselt number and frictional factor will decreases; for a particular value of Reynolds number. Nusselt number and frictional factor has maximum value for D/d=10 and minimum value for D/d=30.

## 6. Acknowledgements

## 7. Nomenclature

| | |
|---|---|
| A | Area of heat transfer (m$^2$) |
| h | Heat transfer coefficient (Wm$^{-2}$ K$^{-1}$) |
| p | tube pitch (m) |
| K | Thermal conductivity (Wm$^{-1}$ K$^{-1}$) |
| Q | Heat transferred (W) |
| U | Overall heat transfer coefficient (Wm$^{-2}$ K$^{-1}$) |
| Re | Reynolds number |
| Nu | Nusselt number |
| Pr | Prandtl number |
| L | Length of the pipe (m) |
| v | Lelocity (m s$^{-1}$) |
| r | Inner radius of the tube (m) |
| $d_1$ | Diameter of inner tube, mm |
| $d_2$ | Diameter of middle tube or thickness tube, mm |
| $d_3$ | Diameter of outer tube, mm |
| D | Coil diameter, mm |
| De | Dean Number |
| f | Friction factor |
| H | Pitch of coil, mm |
| k | Thermal conductivity, W/m-K |

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