Local & Average Heat Transfer and Friction Factor Characteristics of a Flow through a Helical Wire Coil for Turbulent Flow Conditions

Rahul Chaudhari, Prashant W. Deshmukh *, Vishal Bhalla, Subhash V. Lahane

Department of Mechanical Engineering, College of Engineering Pune, Wellesley Road, Shivajinagar-411005, Maharashtra, India

* Corresponding author e-mail: pwdeshmukh25@gmail.com

Abstract. Heat transfer augmentation is essential for numerous industrial applications. The flow through a helical coil is one of the most efficient and effective techniques of heat transfer enhancement due to the provision of a large surface area. The present study is focused on the thermo-hydraulic performance of turbulent flow through a helical coil, made up of copper with a curvature ratio of 0.0519 (d/D) and pitch equal to 50 mm using water as a test fluid. The objective of the present study is to reveal the influence of secondary flow, centrifugal force, and Reynolds number (Re) on local velocity, local heat transfer coefficient in a helical coil. The values of average heat transfer and friction factor at different Re for a flow through a helical coil are compared with corresponding values for a flow through a smooth tube. The numerical results indicate that there is a circumferential variation of local velocity and local Nusselt number (Nu) values concerning the inner and outer sides of the helical coil. The values of average heat transfer are observed to be 150 to 200% more than the smooth tube values with increased pressure drop.

Keywords. Helical coil, Turbulent flow, Thermo-hydraulic performance, Local heat transfer

1. Introduction

Heat transfer augmentation is essential for many industrial applications. There are different types of heat transfer augmentation techniques: active and passive. Inactive techniques external forces such as external vibration, surface vibration, the electric field are applied to the smooth tube to increase the heat transfer. In passive techniques specific geometries (such as helical coil, conical coil, twisted tape), tube inserts, fluid additives are used to increase the heat transfer. The helical path of the tube increases the residence time of fluid over the heated surface. The major disadvantage of a smooth straight tube is that it creates thermal expansion under heat transfer applications. The thermal expansion associated with the helical coil is minimum due to the generation of secondary flow within the mainstream flow that causes enhancement in heat transfer. However, the secondary flow generated due to the helical path of the fluid within the coil increases pressure drop and hence increases the pumping power in comparison with the straight tube [1].
Grindley et al. [2] and Eustice J. [3] reported the experimental data for pressure drop for the flow of water through flexible helical coiled tubing of various radii of curvature. The authors reported that the pressure drop is increased due to increases in the flow length of the fluid. Dean W.R. [4,5] reported the theoretical study for the flow of the incompressible Newtonian fluids in the torus. The authors concluded that the pressure drop is observed to depend on the flow rates, the tube curvature. The authors presented the results of isothermal pressure drop in terms of dimensionless quantity Dean number.

Few researchers reported the study of flow through a helical coil for understanding the physical phenomenon of induction of helical fluid flow path. The fundamental studies explore the mechanism of helical flow inside helical as well as curved tubes. Mishra and Gupta [6] reported the study of flow characteristics for different geometrical parameters of the curved tubes. The various geometrical parameters for helical coils such as coil mean diameter, pitch, tube diameter, the thickness of tube that affect the thermo-hydraulic performance of the system. The authors also reported that changes occur within the fluid and flow characteristics significantly differ for configuration when pitch exceed the coil mean diameter, whereas when the pitch is smaller than the coil mean diameter the influence on the flow is observed to be negligible.

Seban and McLanghin [7] reported that the tube curvature, torsion, and secondary flow create difficulty in the measurement of thermo-hydraulic performance of the coil and flow. Bai et al. [8] reported that the curvature ratio of the helical affects the pressure drop, which further alters the intensity of the secondary flow generated within the mainstream flow. Many researchers [9-13] reported the effect of variation of Re on the pressure drop and average heat transfer values in helical coils. Ito [14] carried out a hydrodynamics study of fluid flow through coiled tubes for estimation of friction factor at different flow rates corresponding to turbulent flow regime. The authors concluded that the friction factor gradually reduces with an increase in Re of the flow. Hardik et al. [15] collected the experimental analysis of flow through the helically coiled tube, and the effects of several geometrical parameters like tube curvature, and flow parameters like Re on heat transfer with working fluid like water. A brief assessment of work carried out on for a flow through a helical flow.

It is observed from the literature review that there is a complexity for the measurement of local Nu value at different locations of the helically coiled tube. Therefore, the results of average heat transfer and pressure drop are available in the literature. There is a need to measure the local wall temperature to estimate the circumferential variation of local heat transfer values over the surface of the helically coiled tube. The present study is focused on the thermo-hydraulic performance of turbulent flow through a helical coil, made of copper with a curvature ratio of 0.0519 (d/D) and pitch equal to 50 mm. The water is used as a working fluid.

2. Numerical Modelling Methodology
A three-dimensional model of the helical coil is modeled in the Solid Edge V19 packages shown in Figure 1. The geometric specifications of the tube such as the outer diameter of tube d_o, thickness of the tube t, inner diameter of tube d_i, and the coil such as coil diameter D, pitch P, number of turns n, and length L are listed in Table 1. The material of the helical coil used is copper.

![Figure 1. 3D geometry of the helical coil.](image-url)
2.1. Governing equations
In the present work, the investigation for friction coefficient and heat transfer for helical turbulent flow conditions for the flow of water with the Re range from 5000-20000 is undertaken in the step increment of 2500. The corresponding inlet velocity ranges from 0.834-3.336 m/sec. The selection of the Re number range for the present study is closely matched with most of the thermal applications. The study presented is for the steady, three-dimensional turbulent flow of Newtonian fluid i.e. water.

The rate equations of continuity, momentum, energy and standard k-ε model are:

Continuity equation
\[
\frac{\partial (\rho u)}{\partial x} = 0
\]  

Momentum equation for turbulent flow
\[
\frac{\partial (\rho u)}{\partial t} + \frac{\partial (\rho u u_i)}{\partial x_i} = - \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \mu \left( \frac{\partial u_i}{\partial x} + \frac{\partial u_i}{\partial x_j} \right) - \frac{2}{3} \mu \frac{\partial u_i}{\partial x_j} \delta_{ij} \right)
\]  

Energy equation
\[
\frac{\partial}{\partial x_j} \left( \rho u_i C_i T - k \frac{\partial T}{\partial x_j} \right) = \mu \left( \frac{\partial u_i}{\partial x} + \frac{\partial u_i}{\partial x_j} \right) - \frac{2}{3} \mu \frac{\partial u_i}{\partial x_j} \delta_{ij}
\]  

Standard k-ε model (Lauder and Spalding)
Turbulent kinetic energy rate equation (k)
\[
\frac{\partial (\rho k)}{\partial t} + \text{div} (\rho k v_i) = 2 \mu_t S_{ij} . S_{ij} + \text{div} \left[ \mu_t \sigma_k \text{grad} k \right] - \rho \epsilon
\]  

Turbulent dissipation energy rate equation (ε)
\[
\frac{\partial (\rho \epsilon)}{\partial t} + \text{div} (\rho \epsilon v_i) = C_{1\epsilon} \frac{\epsilon}{k} \mu_t S_{ij} . S_{ij} + \text{div} \left[ \frac{\mu_t}{\sigma_s} \text{grad} \epsilon \right] - C_{2\epsilon} \frac{\epsilon^2}{k}
\]  

2.2. Meshing
The meshing decides solution accuracy in computational analysis. The goodness of the mesh not only depends on the size of the element but also depends on the parameters like aspect ratio, skewness, wrapping angle, orthogonal quality. The most affected parameter on the goodness of the mesh is the orthogonal quality. The suggested value of the orthogonal quality is nearly equal to one. The present study uses the mesh refinement grid with tetrahedrons elements as shown in Figure 2. The grid details are listed in Table 2.
Figure 2. Mesh of the model (a) orthogonal quality (b) whole grid

Table 2. Mesh details

| Parameters          | Statistics  |
|---------------------|-------------|
| Nodes               | 2430414     |
| Elements            | 1701661     |
| Element type        | Tetrahedrons|
| Orthogonal quality  | 0.8         |

2.3. Boundary conditions
At the outer surface of the helical coil the constant heat flux condition is applied. In the present study the temperature drop between the outlet and inlet of the fluid is maintain nearly equal to 27°C. At the tube inlet the correspondence velocity is given as boundary condition. The operating conditions used in the simulations are mentioned in Table 3.

Table 3. Operating conditions

| Sr. No. | Reynolds number, $Re$ | Velocity, $V$ (m/sec) | Heat Flux, $q''$ (kW/m²) |
|---------|-----------------------|------------------------|---------------------------|
| 1       | 5000                  | 0.834                  | 58.5                      |
| 2       | 7500                  | 1.251                  | 87.77                     |
| 3       | 10000                 | 1.668                  | 117.03                    |
| 4       | 12500                 | 2.085                  | 146.326                   |
| 5       | 15000                 | 2.502                  | 175.59                    |
| 6       | 17500                 | 2.919                  | 204.85                    |
| 7       | 20000                 | 3.336                  | 234.12                    |

2.4. Computational scheme
The pressure-based steady solver is used in the present study. All the governing equations listed above are solved by the boundary conditions using the finite volume method. The terms turbulent kinetic
energy, turbulent dissipation rate, momentum, are modeled in the standard k-ε turbulent model using a second-order upwind scheme.

![Figure 3](image_url)

**Figure 3.** Variation of Nu with Re for various turbulence models

The turbulent intensity was selected as 5%. The temperature and heat flux terms are taken into account using an energy equation. The convergent criterion for all the residual variables wise x velocity, y velocity, z velocity, continuity is 10⁻³ and for energy, the equation is 10⁻⁶. The pressure and velocity are coupled by using a COUPLED algorithm to avoid the limitation regarding the temperature that occurred in the case of SIMPLE, PISO algorithms.

To decide the suitable turbulence model which gives the correct result, the coil is simulated for three different turbulence models viz. realizable k-ε, standard k-ε, and standard k-ω. Figure 3 shows that the variation of Nu concerning Re for various turbulence models. It is concluded from Figure 3 that the standard k-ε model can be used as the variation is observed to be less than 5% in comparison with the other two models of turbulent flow.

2.5. **Data reduction equations**

The required thermo-hydraulic parameters are calculated from the obtained computational results using data reduction equations are listed below.

The friction coefficient for straight, \( f_s \) and helical coiled flow, \( f_c \):

\[
f_c = \frac{\Delta P}{2 \rho V^2 L}
\]  
(6)

Friction coefficient (Blasius) and Nu(Dittus-Boelter correlation) for turbulent flow:

\[
f_s = 0.079 \times Re^{-0.25} \quad Nu = 0.023 \times Re^{0.8} \times Pr^{0.4}
\]

(7)

Reynolds number,
\[ Re = \frac{\rho \times v \times d_1}{\mu} \]  
(8)

Heat transfer,

\[ Q = mC_p(T_{in} - T_w) \]  
(9)

Heat flux,

\[ q = \frac{Q}{\pi dL} \]  
(10)

Heat transfer coefficient,

\[ h = \frac{q}{T_i - T_m} \]  
(11)

Nusselt number,

\[ Nu = \frac{hd}{K_w} \]  
(12)

3. Results and Discussion

The numerical results are presented in the form of thermohydraulic performance of the flow through a helical coil are as follows.

3.1. Variation of friction coefficient

Figure 4 depicts the variations of the friction coefficient concerning \( Re \). The friction coefficient and average \( Nu \) values for the straight tube are obtained using equation (7). The average friction coefficient of the present study for the flow through the wire coil of geometrical parameters mentioned in table 1 is shown in Figure 4. The figure also includes the results of Ito[14] for the coil of the same configuration. It is observed that the friction coefficient decreases with increases in the \( Re \). It is to be noted that the friction coefficient is nearly 13 times more than the straight tube which shows higher pumping power associated with the helical coil than the straight tube. The values of the friction coefficient of the present study are comparatively less than the values specified by Ito [14].

![Figure 4](image)

**Figure 4.** Deviation of friction coefficient with \( Re \).
3.2. Visualization of secondary flow
The secondary flow is the key phenomenon to understand the mechanism of the helical flowpath of the fluid particles. It is shown in Figure 5. The streamlines are used for the visualization of induced secondary flow. The gap between the two halves of streamlines provides the evidence of the generation of the secondary flow.

![Figure 5. Secondary flow streamlines for Re = 17500](image)

3.3. Variation of velocity
Figure 6 depicts the velocity variation of the fluid particles in the circumferential direction due to induced centrifugal force as a result of the helical path of the fluid. It is observed clearly from figure 6 that the fluid particles at the outer side of the tube have more velocity in comparison with the fluid particle at the inner side of the tube.

![Figure 6. Circumferential variation of velocity for Re= 17500](image)
The variation of velocity is observed in a circumferential direction however, the velocity of fluid particles almost remains constant in the axial direction. The average velocity at the outer side of the helical coil is observed to be around 20% more than the mean velocity. The variation of velocity at the inner side, at the outer side, and the mean velocity concerning Re is shown in Figure 7.

3.4. Variation of local heat transfer
The variation of the local wall heat transfer coefficient is shown in Figure 8. It is observed that the increase in heat transfer at the outer side of the helical coil is more than that of the inner side. This is due to cooler mainstream fluid from the central region of the tube cross-section directed more vigorously towards the outer side of the helical coil due to the centrifugal force. It causes more transport of heat at the outer side than the inner side of the helical coil.

Figure 8. Contour plot of heat transfer coefficient for Re = 17500
Figure 9. Nu variation with Re

The variation of Nu at the outer, inner side, and their average is illustrated in Figure 9 at different Re. The average value reported by Hardik et al. [15] is also shown in Figure 10 to ensure the correctness of the results of the present study. Figure 9 also contains the Nu values of smooth tube specified by Dittus-Boelter correlation at different Re. In all the cases it is observed that the heat transfer increases at higher flow rates. The figure shows the variation of the outer side, inner side, and average Nu values of the present study. The outer side Nu value for the flow through a wire coil is observed to be around 23% more than the average Nu value of the coil whereas the inner side Nu value is observed to be around 22% less than the average Nu value. It is to be noted that the enhancement in heat transfer is observed to be very poor at the inner side of the tube as the Nu is less than that of the smooth tube flow. The average heat transfer enhancement of 20% is observed for the geometry used in the present work.

4. Conclusion
The numerical work is presented for the study of the effect of Re and constant Prandtl number (Pr) equal to 5.0 at a coil curvature ratio (d/D) of 0.0519 on local Nu, and friction coefficient with water as working fluid. The friction coefficient is obtained from the pressure drop values while the Nu and local temperature distribution is obtained from temperature contours and wall heat transfer coefficient contours. The following inferences are extracted from the present study.

- The friction coefficient in the helical coil was more than the smooth tube for turbulent flow.
- There is a variation of the velocity of the fluid in the circumferential direction. The outer side fluid velocity in the helical coil is observed to be 20% more than the mean inlet velocity while the inner side velocity is less than the mean inlet velocity of the fluid.
- The average Nu value is observed to be 20% more than the corresponding smooth tube values.
- The outer and inner side local Nu values are observed to be 23% more and 22% less than the average Nu value.
Nomenclature

| Symbol | Description |
|--------|-------------|
| $A$    | The surface area of the coil, (m$^2$) |
| $V, U$ | Mean velocity of the fluid, (m/sec) |
| $\mu$ | Dynamic viscosity, (Pa.s) |
| $\varepsilon$ | Turbulent dissipation energy rate, (m$^2$/s$^3$) |

References

[1] Vashisth S., Kumar V., Nigam K.D.P. 2008 A review on the potential applications of curved geometries in process industry. *Ind. Eng. Chem. Res.* 47, 3291-3337.
[2] Grindley J.H., Gibson, A.H. 1908 On the frictional resistance to the flow of air through a pipe. *Proc. R. Soc. London, Ser. A* 80, 114–139.
[3] Eustice J. 1910 Flow of water in curved pipe. *Proc. R. Soc. London, Ser. A* 84, 107–118.
[4] Dean W.R. 1927 Notes on the motion of fluid in a curved pipe. *Philos. Mag.* 4, 208–233.
[5] Dean W.R. 1928 The streamline motion of fluid in a curved pipe. *Philos. Mag.* 5, 673–695.
[6] P. Mishra, S.N. Gupta, Momentum transfer in curved pipes. 1. Newtonian fluids, 1979 *Indust. Eng. Chem. Process Des.* Dev. 18130–137.
[7] Seban R.A., McLaughlin E. 1963 Heat transfer in tube coils with laminar and turbulent flow, *Int. J. Heat Mass Transfer* 6 387–395.
[8] Bai B., Guo L., Feng Z., Chen X. 1999 Turbulent heat transfer in a horizontal helically coiled tube, *Heat Transfer Asian Res.* 28 395–403.
[9] Kubair V., Kuloor N.R. 1966 Heat transfer to Newtonian fluids in coiled pipes in laminar flow, *Int. J. Heat Mass Transfer* 9 63–75.
[10] Pawar S.S., Vivek K.S. 2013 Experimental studies on heat transfer to Newtonian and non-newtonian fluids in helical coils with laminar and turbulent flow, *Exp. Thermal Fluid Sci.* 44 792–804.
[11] Pimenta T.A., Campos J.B.L.M. 2013 Heat transfer coefficients from Newtonian and non-newtonian fluids flowing in laminar regime in a helical coil, *Int. J. Heat Mass Transfer* 58 676–690.
[12] Dravid A.N., Smith K.A., Merrill E.W., Brain P.L.T. 1971 Effect of secondary fluid on laminar flow heat transfer in helically coiled tubes, *Am. Inst. Chem. Eng. J.* 17 1114–1122.
[13] Janssen L.A.M., Hoogenwdoorn C.J. 1978 Laminar convective heat transfer in helical coiled tubes, *Int. J. Heat Mass Transfer* 21 1197–1206.
[14] Ito H. 1959 Friction factors for turbulent flow in curved pipes, *Trans. Am. Soc. Mech. Eng.* D 81 123–134.
[15] Hardik B.K., Baburajan P.K., Prabhu S.V. 2015 Local heat transfer coefficient in helical coils with single phase flow *Int. J. Heat Mass Transfer* 89 522-538.