Numerical investigations on the heat transfer characteristics of tube in tube helical coil heat exchanger

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Abstract. Heat transfer characteristics of a tube in tube helical heat exchanger in the laminar flow regime is analysed numerically. Numerical scheme is validated with the experimental results reported in literature. The effect of helix angle on the overall heat transfer coefficient is analysed. The overall heat transfer coefficient increases with Dean number and its rate of increase at higher Dean number is low. The rate of heat transfer is maximum at a helix angle of 27°. Simulation studies show that the secondary flows that originate in a coiled tube due to curvature and torsion produce an additional transport of the fluid over the cross section of the pipe. The overall heat transfer coefficient increases with increase in inner diameter.

1. Introduction

Energy and materials saving considerations, as well as economic incentives, have led to efforts to produce more efficient heat exchange equipment. Common thermo-hydraulic goals are to reduce the size of a heat exchanger required for a specified heat duty, to upgrade the capacity of an existing heat exchanger, to reduce the approach temperature difference for the process streams, or to reduce the pumping power. Due to their compact structure and high heat transfer coefficient, curved tubes have been introduced as one of the passive heat transfer enhancement techniques and are widely used in various industrial applications. Helical and spiral coils are well known types of curved tubes which have been used in a wide variety of applications, for example, heat recovery processes, air conditioning and refrigeration systems, chemical reactors, food and dairy processes. Due to the curvature of the tubes, as fluid flows through curved tubes, centrifugal force is generated. A secondary flow induced by the centrifugal force has significant ability to enhance the heat transfer rate as it moves fluid across the temperature gradient. Thus, there is an additional convective heat transfer mechanism, perpendicular to the axial flow, which does not exist in straight tube heat exchangers.

Dravid et al. [1] numerically investigated the effect of secondary flow on laminar flow heat transfer in helically coiled tubes both in the fully developed region and in the thermal entrance region. Patankar et al. [2] discussed the effect of the Dean number on friction factor and heat transfer in the developing and fully developed regions of helically coiled pipes. Yang et al. [3] presented a numerical model to study the fully developed laminar convective heat transfer in a helicoidal pipe having a finite
pitch. The effects of the Dean number, torsion, and the Prandtl number on the laminar convective heat transfer were discussed. Rabin and Korin [4] developed a new simplified mathematical model for thermal analysis of a helical heat exchanger for long-term ground thermal energy storage in soil for use in arid zones. Acharya et al. [5] numerically studied the phenomenon of steady heat transfer enhancement in coiled-tube heat exchangers due to chaotic particle paths in steady, laminar flow with two different mixings. Zhang et al. [6] applied a control-volume finite difference method having second-order accuracy to solve the three-dimensional governing equations. Chen and Zhang 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.

The laminar forced convection and thermal radiation in a participating medium inside a helical pipe were analysed. Rennie and Raghavan [7] simulated the heat transfer characteristics in a two-turn tube-in-tube helical coil heat exchanger. Various tube-to-tube ratios and Dean numbers for laminar flow in both annulus and in-tube were examined. Prabhanjan et al. [8] compared the heat transfer rates between a helically coiled heat exchanger and a straight tube heat exchanger. Xin and Ebadian [9] considered the effects of the Prandtl number and geometric parameters on the local and average convective heat transfer characteristics in helical pipes. The heat transfer characteristics of a tube-in-tube helical geometry was studied by Timothy J Rennie and Vijaya G S Raghavan [14]. This is the relevant geometry of this analysis. Rennie and Raghavan conducted experimental and numerical studies separately. They used Wilson [17] plots to determine the heat transfer coefficients in the experimental study. However, the numerical results showed significant divergence from the experimental values for smaller coils. They assumed that Wilson plots may be the reason behind this divergence. Most of the works reported in literature on heat transfer in helical coil tubes have been carried out considering wall boundary conditions of either constant heat flux or constant wall temperature. Physically realistic fluid to fluid forced convection heat transfer studies are to be done to get more insight to the problem.

Nomenclature

| Symbol | Description                          | Units        | Notes                           |
|--------|--------------------------------------|--------------|---------------------------------|
| A      | Cross sectional area, m²             | q            | Heat transfer rate, W           |
| L      | Length of heat exchanger, m          | U            | Overall heat transfer coefficient, W/m²K |
| D      | Diameter of outer tube, m            | ΔT           | Temperature difference, K       |
| d      | Diameter of inner tube, m            | k            | Thermal conductivity, W/m K     |
| R      | Pitch circle radius of helix, m      | h            | Heat transfer coefficient, W/m²K |
| p      | Axial pitch of helix, mm             | C            | Specific heat, J/kg K           |
| Q      | Volume flow rate, ml/min             | Re           | Reynolds number                 |
| U      | Velocity of flow, m/s                | De           | Dean number                     |
| ṁ      | Mass flow rate, kg/s                 | Nu           | Nusselt number                  |
| T      | Temperature, K                       | Pr           | Prandtl number                  |

Suffixes:

- i: Inner
- o: Outer
- h: Hot fluid
- c: Cold fluid
- l: Inlet
- 2: Outlet

2. Numerical Setup

2.1. Geometry modelling and meshing

The Geometric modelling of the heat exchanger is done using AutoCAD. Two helically coiled concentric tubes were created one over the other to form a double pipe helical heat exchanger. Both tubes have the wall thickness of 0.8 mm. All the dimensions are shown in Table 1. The mesh was created...
using meshing module of the ANSYS 14.5 WORKBENCH. Triangular elements were used in the Map type to mesh the faces with a spacing of 0.8. Tet/Hybrid meshing is applied to create volume meshes with a spacing of 0.8 in T Grid type. This resulted in 2,81,869 volume elements. The geometry for numerical modelling is shown in figure 1.

| Table 1 Dimensions of coil used in the present study |
|-----------------------------------------------------|
| **Dimensions**                                    | **Values** |
| Outside diameter of inner tube                    | 9.5 mm     |
| Inside diameter of inner tube                     | 7.9 mm     |
| Wall thickness of both inner and outer tubes      | 0.8 mm     |
| Outside diameter of outer tube                    | 15.9 mm    |
| Inside diameter of outer tube                     | 14.3 mm    |
| Radius of curvature of the helical coil           | 235.9 mm   |
| Pitch of the helical coil                         | 15.9 mm    |
| Length of the coil                                | 1.48 m     |

The geometry created and the mesh generated for the validation of the numerical scheme are shown in figure 1 and figure 2 respectively. Counter flow configuration was used. Both the hot and cold inlets were provided with mass flow inlet boundary condition. Both hot and cold fluid outlet boundaries were provided with pressure outlet boundary condition. The outer surface of the outer tube and the end concentric surfaces of the tubes were provided with wall boundary condition and the heat transfer is set to zero value.

Simulations were performed using five different mass flows in the inner tube and the annulus region. The flow rates used were 0.3 lpm, 0.5 lpm, 0.7 lpm, 0.8 lpm and 0.9 lpm. This resulted in 25 simulations. Water and copper were selected in materials selection tab for the fluid and solid domains respectively. Temperature dependent properties like thermal conductivity (k), viscosity (μ), and specific heat (Cp) are given.

The realizable k-ε turbulence model with enhanced wall functions was used in the analysis. The SIMPLE algorithm is used for pressure velocity coupling with second order upwind scheme for the discretization of the momentum and energy equations. Boundary conditions are used according to the need of the model. The inlet and outlet conditions are defined as mass flow inlet and pressure outlet. The outer surface of the outer tube and the end concentric surfaces of the tubes were provided with wall boundary condition and the heat transfer is set to zero value. Hybrid initialization method was used for initializing the flow variables which is found to have faster convergence as compared to standard initialization in this particular case.

A convergence criterion of 1.0e-04 was used for continuity k, ε and x, y and z velocities while for energy equation a convergence criterion of 1.0e-06 was found to be satisfactory. Grid independence study is conducted wherein the grid is refined successively using edge sizing and inflation growth rates. This is repeated until a tolerance of about 1% difference in two successive refinements. The results of grid independence study is shown in figure 3.
2.2. Data reduction

Overall heat transfer coefficient, $U_{io} = \frac{Q}{A_{io} LMTD}$  

Outside surface area of the inner tube, $A_{io} = \pi d_o L$  

Where $L$ is the length of the tube and $d_o$ is the outer diameter of the inner tube.

Log mean temperature difference, $LMTD = \frac{(\Delta T_2 - \Delta T_1)}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$  

Where, $\Delta T_1 = T_{h1} - T_{c2}$ and $\Delta T_2 = T_{h2} - T_{c1}$

Heat transfer coefficients for the inner and annulus were calculated using the average bulk fluid temperature and the average temperature of the coil.

3. Result and discussion

Numerical investigations have been carried out to study the effect of variations in Reynold’s number, Dean number, helix angle and inner diameter on the heat transfer characteristics of the helical coil.

3.1 Grid independent study

Figure 3 shows the grid independent study performed for the physical model to obtain the optimum number of elements. The number of elements is varied from 1.4 million to 3.9 million. The result from the calculation using 3.2 million cells agrees well with the case using 3.9 million cells. Thus, the mesh with 0.45 million cells has been chosen as the suitable mesh for further investigations.
3.1. Effect of Dean number on heat transfer

Figure 4 shows the effect of flow rate through the inner tube on overall heat transfer coefficient. The experimental results reported by Rennie et al. [14] is also shown for comparison. The overall heat transfer increases with increase in flow rate through the inner tube. Further, there is very good agreement between the results obtained from numerical studies and the experimental results reported.

![Figure 4 Variation of overall heat transfer coefficient with inner mass flow rate](chart1)

Overall heat transfer coefficients for various annulus flow rates is presented in figure 5. The trends are typical for a fluid-to-fluid heat exchanger with the overall heat transfer coefficient increasing with both inner and annulus flows. However, the rate of increase of the overall heat transfer coefficient is less at higher inner Dean numbers. For a given annulus flow rate, increasing the inner flow rate results in an eventual asymptotic overall heat transfer coefficient.

![Figure 5 Variation of overall heat transfer coefficient with inner Dean number](chart2)

Temperature contours at different outer flow rates are shown in figure 6. The secondary flows that originate in a coiled tube due to the curvature and torsion produce an additional transport of the fluid over the cross section of the pipe. From the temperature contours at different flow rates, it is clear that the high temperature contours are moved to the outer wall due to centrifugal force. As the flow rate increases the temperature contour symmetry distorts. The temperature gradient near the outer wall region increases and near the inner wall region temperature gradient decreases. As the Dean number increases,
these variations in the temperature gradients in the outer and inner wall regions of the coil changes and which lead to high rates of heat transfer.

![Temperature contour](image)

Figure 6: Temperature contour a) 0.3lpm, b) 0.7lpm and c) 0.9lpm

3.2. Effect of helix angle on heat transfer

Studies have been conducted to see the effect of helix angle on the heat transfer characteristics. The helix angle is varied from $5^\circ$ to $45^\circ$ in steps of $5^\circ$. Figure 7 indicates the variation of overall heat transfer coefficient with helix angle.

![Overall heat transfer coefficient](image)

Figure 7. Variation of Overall heat transfer coefficient with helix angle
The heat transfer rate increases with helix angle, reaches a maximum value and then decreases. As the helix angle is increased, the torsion of the liquid also increases which may be the cause for the increase in heat transfer rate. It is currently accepted that the effect of coil curvature is to suppress the turbulent fluctuations arising in the flowing fluid, smoothing the emergence of turbulence and increasing the value of the Reynolds number required to attain a fully turbulent flow, with respect to a straight pipe. The above effect of turbulent fluctuations suppression enhances as the coil curvature increases. Torsion, on the other hand, is believed to destabilize the flow; reducing the Reynolds number at which turbulence emerges at values lower than the ones characteristic of straight pipe flow. The above destabilizing effect first increases, as torsion increases, reaches a maximum and then decreases with further increase in torsion. Variation of effectiveness with helix angle is shown in figure 8.

As the pitch of the coil increases, additional torsional flow patterns were observed. The torsional effect increases with increase in coil pitch. Therefore, an increase in heat transfer coefficient is expected. Maximum overall heat transfer coefficient is observed near a helix angle of 27°.

3.3. Effect of Inner tube diameter on heat transfer

In order to analyse the effect of inner tube diameter, simulations were performed for three different inner tube diameters. The dimensions of these three configurations are shown in Table 2.

| Table 2. Dimensions for study of the effect of inner tube diameter |
|------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|
|                  | do (mm)          | di (mm)          | t (mm)           | Do (mm)          | Di (mm)          | R (mm)           | p (mm)           |
| First coil       | 12.7             | 11.1             | 0.8              | 15.9             | 14.3             | 235.9            | 15.9             |
| Second coil      | 9.5              | 7.9              | 0.8              | 15.9             | 14.3             | 235.9            | 15.9             |
| Third coil       | 6.35             | 4.75             | 0.8              | 15.9             | 14.3             | 235.9            | 15.9             |

Figure 9 and figure 10 indicate the effect of annulus flow rate on overall heat transfer coefficient and heat transfer rate respectively. Heat transfer rate and overall heat transfer coefficient are maximum for bigger tube since its surface area for heat transfer is the largest. Since the radius of curvature of the coils is the same, coil with larger inner diameter have larger curvature ratio which results in greater secondary flow effects and hence higher rate of heat transfer. Increase in tube size, increases the rate of heat transfer. When the annulus flow rate is high and the inner flow rate is low, the limiting heat transfer is in the inner tube. Hence any change in the inner flow rate will have significant effects on the overall heat transfer coefficient.
Increase in the inner tube diameter reduces the velocity of flow which resulted in lower Reynolds number. Increase in the inner tube diameter at the same radius of curvature increases the curvature ratio and increased secondary flow. This results in the decrease of inner Dean number and lower heat transfer coefficient.

4. Conclusion
Numerical study of a double-pipe helical coil heat exchanger was performed. A very good conformance with the experimental results reported in literature was observed. The following conclusions are derived out of this study. The Dean number and torsion have significant effect on the heat transfer characteristics. The overall heat transfer coefficient increases with inner Dean number. The rate of increase of the overall heat transfer coefficient is less at higher inner Dean numbers. For this heat exchanger the optimum value of helix angle is 27°. An inner tube to outer tube diameter ratio of about 0.88 leads to optimum performance of this helical heat exchanger.

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