



Optimisation of Double Pipe Helical Tube Heat Exchanger and its Comparison with Straight Double Tube Heat Exchanger

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Abstract Optimization of double pipe helical coil heat exchanger with various optimizing parameters and its comparison with double pipe straight tube are the prime objectives of this paper. Numerical studies were performed with the aid of a commercial computational fluid dynamics package ANSYS FLUENT 14. In this paper the double pipe helical coil is analysed under turbulent flow conditions for optimum heat exchanger properties. The parameters used for optimization are cross-sectional shape and taper angles. Optimization analysis is being carried out for finding best cross sectional shape of heat exchanger coils by using rectangular, square, triangular and circular cross-sections. The tapered double pipe helical coil is then analysed for best heat transfer and pressure drop characteristics by varying the angle of taper. Finally, an optimum coil on the basis of all the analysis is selected. This optimized double pipe helical coil is compared with double pipe straight tube of equivalent cross-sectional area and length as that of unwound length of double pipe helical coil.

Keywords Double pipe helical coil · Tube in tube heat exchanger · Heat exchanger optimization

Introduction

It has been widely reported in literature that heat transfer rates in helical coils are higher as compared to those in straight tubes. Due to the compact structure and high heat transfer coefficient, helical coil heat exchangers find extensive use in industrial applications such as power generation, nuclear industry, process plants, heat recovery systems, refrigeration, food industry, etc. (Berger et al. 1983; Abdalla 1994; Rao 1994). Due to the extensive use of helical coils in these applications, knowledge about the pressure drop, flow patterns, and heat transfer characteristics are very important. A double pipe helical coil is advantageous because the secondary flow which is responsible for heat transfer increase in helical tubes will be present in both inner and annulus tubes.

Literature Survey

Huttl and Friedrich [1] used direct numerical simulation for turbulent flow in straight, curved and helically coiled pipes to determine the effects of curvature and torsion on the flow patterns. They showed that turbulent fluctuations are reduced in curved pipes compared to the straight pipes. Li et al. [2] numerically studied the turbulent convective heat transfer in the entrance region of a curved pipe with a uniform wall temperature. Lin and Ebadian [3] numerically studied the effects of inlet turbulence intensity on the development of the turbulent flow and heat transfer in helically coiled pipes. Roger & Mayhew [4] studied heat transfer to fluid flowing inside a helical pipe which was heated by steam. Mori and Nakayama [5] studied heat transfer under constant wall temperature boundary condition for the same helical coils and observed that the Nusselt number is remarkably affected by a secondary flow due to

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curvature. CFD study of helically coiled double pipe heat exchangers for laminar flow situations were carried out by Rennie and Raghavan [6, 7]. Goering et al. [8] have studied fully developed laminar convective heat transfer in curved pipes to investigate the dual influence of curvature and buoyancy. Dennis and Ng [9] numerically studied laminar flow through a curved tube using a finite difference method with emphasis on two versus four vortex flow conditions. Kao [10] studied the torsion effect on fully developed flow in a helical pipe using a series expansion method to solve the governing differential equations. Kalb and Seader [11] numerically studied the heat transfer in helical coils in the case of uniform heat flux using an orthogonal toroidal coordinate system. Fully developed laminar flow and heat transfer was studied numerically by Zapryanov et al. [12] using a method of fractional steps for a wide range of Dean (10–7000) and Prandtl (0.005–2000) numbers. The effect of pitch on heat transfer and pressure drop was studied by Austen and Soliman [13] for the case of uniform wall heat flux. Most of the studies in literature are carried out using constant heat flux or constant wall temperature boundary conditions. In this paper the author has considered actual conjugate heat transfer. Also the optimization of double pipe helical coil is considered in this paper which has not been reported in earlier research works.

Objectives of Present Study

In this paper, the parameters used for optimizing helical coils are cross-sectional shape and the coil taper angle. Coil cross-section shape plays an important role in determining heat transfer properties of the coil. So for finding the optimum shape, simulations are carried out with coils of different cross-section. The cross-section used for simulations are triangular, circular, rectangular and square. These cross-section shapes are provided for inner tubes, while outer annular tube cross-sectional shape is kept rectangular for all the coils except round cross-section. The cross sectional area for tube part and also annulus part for all

cross-sections are made same for making fair comparison. A meshed model used in this study and general flow direction used throughout this paper is shown in Fig. 1.

In the next part the double pipe helical coil is incorporated with taper angles to find out the effect on heat transfer. These results are to be compared with ordinary helical coil. Then the simulations are carried out by using different taper angles to figure out optimum angle with respect to heat transfer and pressure drop properties. Based on the above two analyses an optimum coil is to be selected.

Numerical Solution Procedure

The governing equations used in the computational analysis are Energy equation, Continuity equation, Momentum equation, k-epsilon equation for turbulence model. Pressure velocity coupling was done using the SIMPLEC algorithm with a PRESTO (PREssure STaggering Option) scheme. The Second Order Upwind algorithm was employed in the discretization of the equations because of its accuracy and iterating efficiency. For momentum, turbulent kinetic energy and turbulent dissipation rate the Power law scheme was used. For the energy equation, second order upwind was employed. A convergence criterion of $1.0e^{-05}$ was used for continuity and x , y and z velocities. The convergence criterion for energy equation was $1.0e^{-08}$, while that for the k and ϵ was $1.0e^{-04}$. To accelerate the convergence the under relaxation factor given for pressure is 0.3, temperature is 0.9, k and epsilon was 0.7.

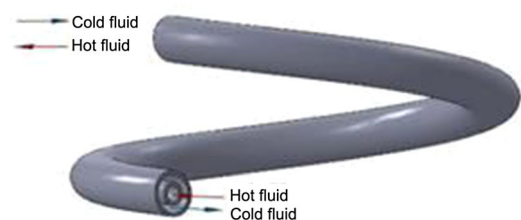
Model Validation

The simulations are validated by comparing the results of Nusselt number for the flow through helical coil in the simulation with that of literature. In literature Nusselt number correlations are available only for boundary conditions of either constant heat flux or constant wall temperature. Actual boundary conditions differ from both these conditions. So for validating model, trials are performed with both these

Fig. 1 Figures showing meshed model and boundary conditions used for analysis



Meshed model used in this study



General flow boundary condition used

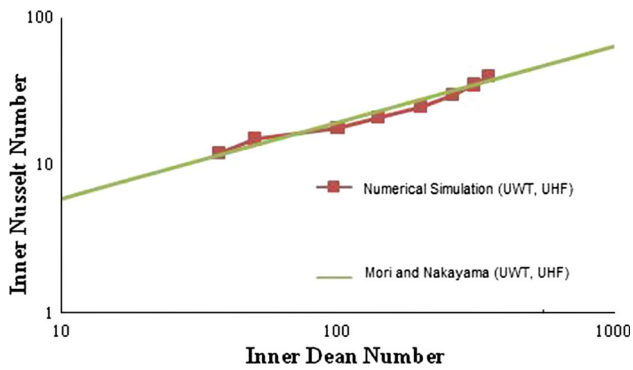


Fig. 2 Comparison of current model with that of literature for uniform wall temperature and uniform heat flux condition

conditions separately. From the literature, Mori and Nakayama [5] stated that the difference in the Nusselt numbers in the inner coil for the constant wall heat flux and the constant wall temperature, for all practical purposes are negligible. Mori and Nakayama [5] developed the following correlation and stated that it could be used for both constant wall temperature and constant wall heat flux boundary conditions.

$$Nu_{cur} = \frac{0.864}{\zeta} \sqrt{De} \left(1 + \frac{2.35}{\sqrt{De}} \right) \quad (1)$$

where, De is the Dean number, ζ is the boundary layer thickness ratio

$$\zeta = \frac{2}{11} \left\{ 1 + \sqrt{\left(1 + \frac{77}{4} \frac{1}{Pr^2} \right)} \right\} \quad (2)$$

Figure 2 shows that for both UWT and UHF, the actual simulation data points coincide so it is shown as one. These points almost lie in range of theoretical data points as proposed by Mori and Nakayama.

Computational Domain

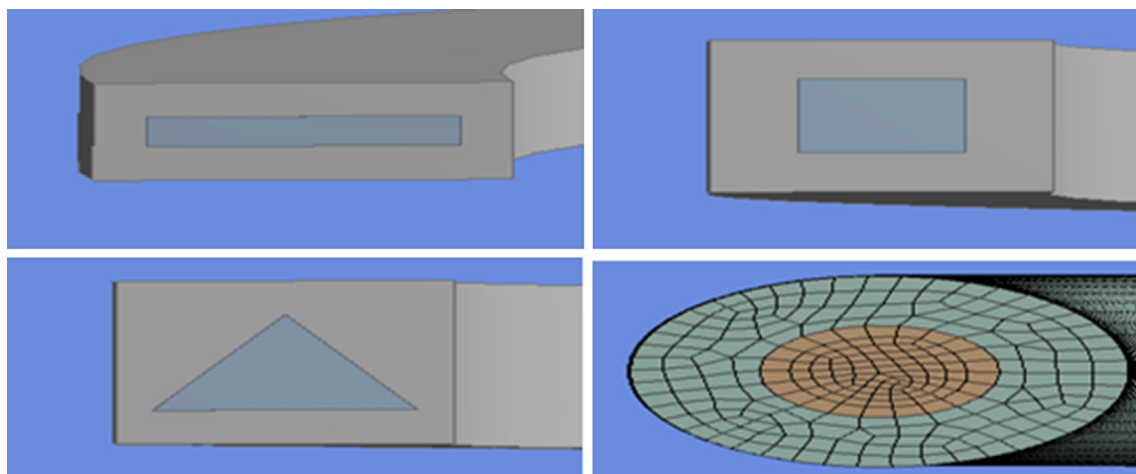
Four helical coil cross-sectional shapes (Fig. 3) were analysed for heat transfer and pressure drop characteristics. The shapes used were circular, rectangular, triangular and square. Part modelling is done on Pro-E and is imported into fluent. The cross-sectional areas of all the 4 coils were made same. The coil consists of 4 turns in order to make sure that flow inside the coil is fully developed. Grid refinement tests were conducted starting with initial coarse mesh size of $1e^{-06}$ and finally mesh size of $1e^{-03}$ was selected. Inner coil diameter was 23 with 2 mm wall thickness and annulus coil diameter was 58 mm.

Boundary conditions

The mass flow rate used in annulus tube was twice that of inner tube, as from the earlier analysis it was clear that the $m_i/m_o = 0.5$ condition has higher heat transfer rate. Mass flow rate used in the inner tube was 0.1, 0.15, 0.2, 0.25, and 0.3 and in the annulus tube was 0.2, 0.3, 0.4, 0.5, and 0.6 respectively. The temperatures at hot and cold fluid inlets are 360 K and 290 K respectively.

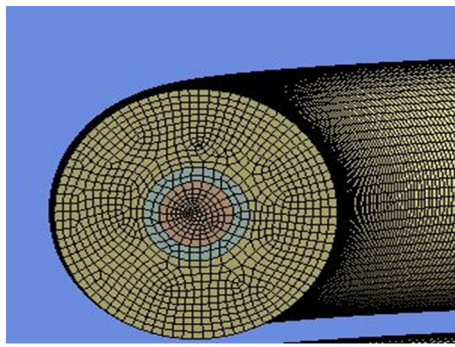
Grid Independency Tests

The first step in a CFD solution is generation of a grid (also called a mesh) that defines the cells on which flow variables are calculated throughout the computational domain. Mesh should be refined to such an extent that, results should not vary on further refining. The error in heat transfer is the criteria for finalising grid size. The final mesh size used for analysis is shown in Fig. 4.



Cross- sections used for optimization study

Fig. 3 Cross- sections used for optimization study



Cross section with refined final mesh

Fig. 4 Cross section with refined final mesh

Initial coarse mesh statistics	Refined mesh statistics
Number of nodes: 376554	Number of nodes: 1476416
Number of elements: 258044	Number of elements: 1347864

Results and Discussions

Heat Transfer Characteristics

In the analysis it was found that round cross-section which is usually used in the heat ex-changer has lowest heat transfer, but is easy to manufacture. Also it was found that rectangular cross-section have the highest heat transfer followed by triangular and square cross-sections. A graph (Fig. 5) is plotted for heat transfer variation with inner mass flow rate for the inner tube. In all the cases the outer tube mass flow rate was twice the inner flow rate.

Pressure Drop Characteristics

It was found that the circular section have the lowest pressure drop which is a favourable property when considering pumping power. The rectangular cross-section have higher

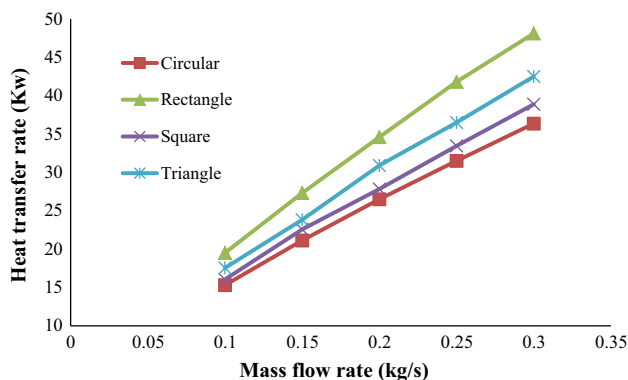


Fig. 5 Heat transfer rate in the inner tube for different cross-sections under varied mass flow rates

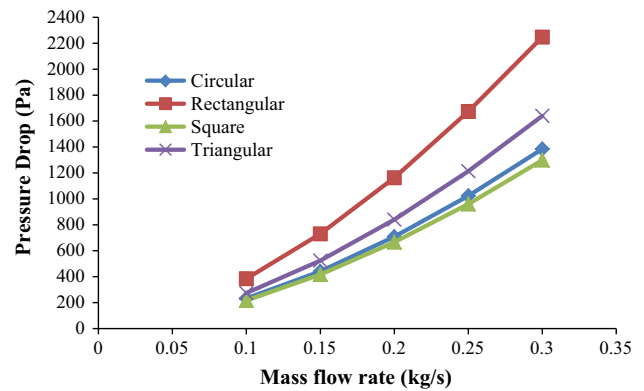


Fig. 6 Pressure drop variation with mass flow rate for different cross-sections

pressure drop and friction factor, followed by triangular and square cross-sections. In the entire cases pressure drop increased with increasing mass flow rate. At higher mass flow rates the difference between pressure drop of rectangular and other coils increases tremendously. A graph (Fig. 6) is plotted showing the variation of pressure drop with mass flow rate for all the cross-sections used in the study.

Rectangular Cross-section Coil

From the analysis carried out it was found that rectangular cross-section was having higher heat transfer property, while that of square was near to circular cross-section. The difference in heat transfer property between rectangular and square cross-sections can be explained through aspect ratio.

Aspect ratio

Aspect ratio is the ratio of longer side to shorter side. For square cross-sections, aspect ratio is 1. Simulations are carried out by modelling coils with aspect ratio 1, 2, 3, 4, 5 and 6. A graph (Fig. 7) is plotted showing the heat transfer rate variation with different aspect ratios. It was found that as the aspect ratio is increasing, the heat transfer rate increases considerably. But this impose a constriction in applying optimum aspect ratio, because if it goes too high, manufacturing difficulty arises as the coil becomes too thin to manufacture.

Comparison of Tapered Helical Coil and Normal Helical Coil

Computational Domain

The taper angle is incorporated in a double pipe helical coil (Fig. 8) in such a way that the unwounded length of tapered coil and ordinary non tapered coil are same.

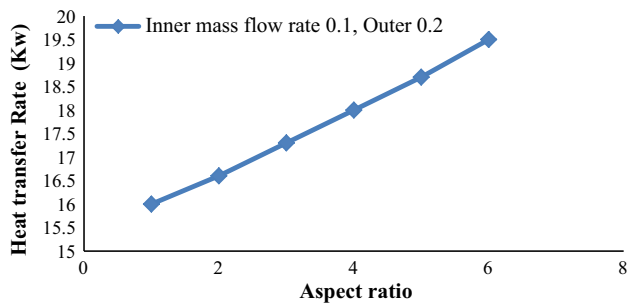
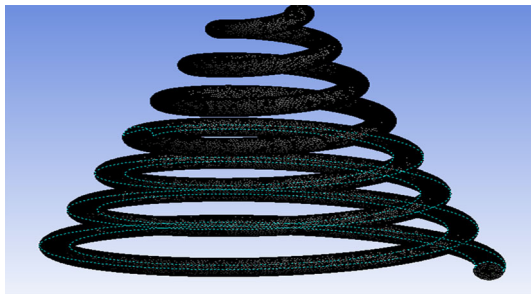


Fig. 7 Heat transfer rate variation with aspect ratio in a rectangular coil



Meshed model of a tapered double pipe helical coil

Fig. 8 Meshed model of a tapered double pipe helical coil

Boundary Conditions

The inlet properties of hot fluid is made velocity inlet condition for both coils and the outlets are made based on pressure outlet condition. For cold fluid, the inlet and outlet is assigned mass flow inlet and pressure outlet boundary condition. The velocity of hot fluid in both the coils are varied within 0.09 to 0.16 m/s, while the mass flow rate of cold fluid in the annulus is made constant at 0.5 kg/s. The hot fluid enters with an inlet temperature of 360 K while that of cold fluid is 290 K.

Results and Discussions

Overall Heat Transfer Coefficient

A graph (Fig. 9) is plotted for overall heat transfer coefficient versus inner Dean Number for inner tube. Overall heat transfer coefficient shows considerable increase for tapered coil especially for higher Dean Numbers. The velocity in the inner tube was varied from 0.09 to 0.16 m/s and the mass flow rate in the annulus tube was fixed at 0.5 kg/s. As the taper angle decreases the curvature ratio increases and it increases the Dean number. As the Dean Number increases, Nusselt number and therefore heat transfer also increase.

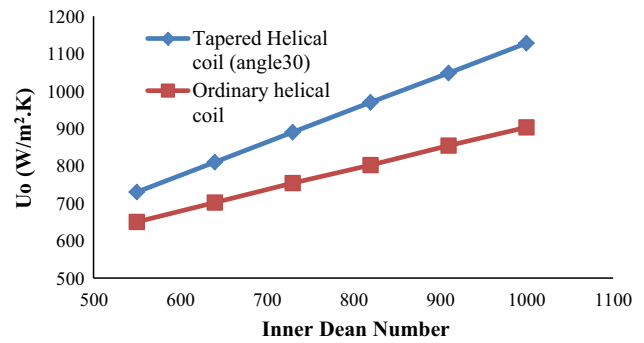


Fig. 9 Comparison of overall heat transfer coefficient with inner Dean Number for tapered and non tapered helical coil

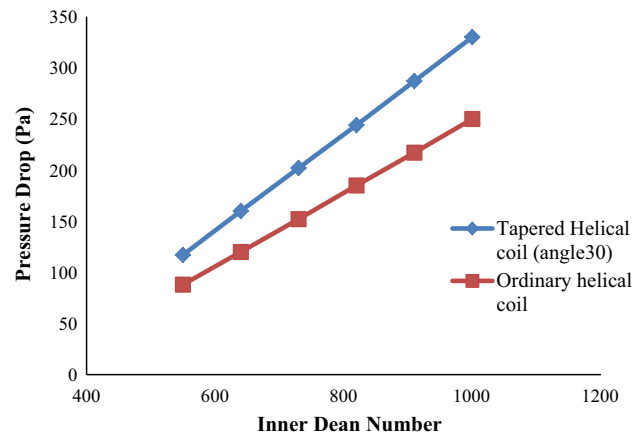


Fig. 10 Variation of pressure drop with Inner Dean Number

Pressure Drop

Pressure drop is higher for tapered helical coil than that of normal helical coil. The graph (Fig. 10) shows the comparison of pressure drop for both tapered and normal helical coils. The taper angle used in this study was 30°. The boundary conditions used for simulation were inner tube velocity of range 0.09 to 0.16 m/s, annulus tube mass flow rate 0.5 kg/s. The inlet temperature of hot fluid was 360 K, and that of cold fluid inlet was 290 K.

Effect of Various Taper Angles on Overall Heat Transfer Coefficient

As the taper angle is increased the height of the coil is increased subsequently in order to keep the unwound length of all coils same. The taper angles used for the analysis are 15°, 20°, 25°, 30° and 35°. It is found that the heat transfer coefficient increases with increasing taper angle. The taper angle effect on heat transfer coefficient is larger when the velocity in the inner tube increases. The graph (Fig. 11) shows the variation of Overall heat transfer coefficient with different taper angles of helical coil.

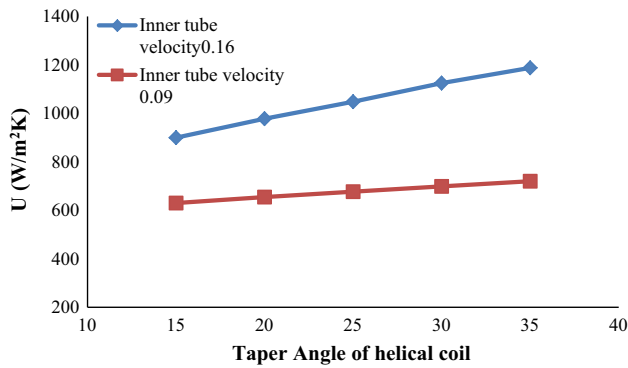


Fig. 11 Overall heat transfer coefficient for different taper angles of double pipe helical coil

Comparison of Optimized Double Pipe Helical Coil with Straight Double Pipe

The Fig. 12 shows the optimized coil (optimum cross section) found out by analysis discussed earlier. The taper angle used here is 30° and is not optimum, as heat transfer gets better when taper angle is increased.

Inner and Annulus Nusselt Number

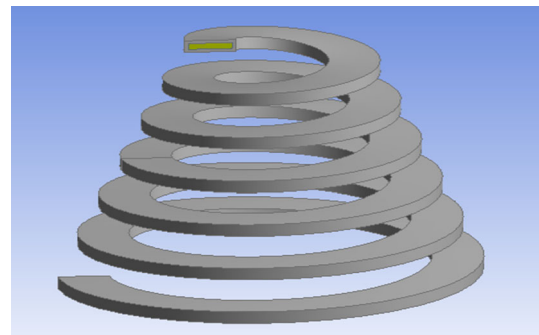
It can be seen from Figs. 13 and 14 that Nusselt number in both inner and outer tubes of both straight and helical coils are increasing with Reynolds Number. Nusselt number and heat transfer coefficients in the helical coil are nearly twice higher than straight tube. This increase in heat transfer may be due to secondary flow developing in the helical coil.

Inner Friction Factor

The friction factor is higher in helical coil which implies more pressure drop in helical coil than in straight tubes. As the inner tube Reynolds number increases, the friction factor decreases for both helical and straight tubes, but there is considerable decrease in helical tube as compared to straight tube. Figure 15 shows the comparison of friction factor variation with inner Reynolds Number for both helical and straight inner tube.

Outer Friction Factor

The annulus tube friction factor is higher for both straight and helical tubes than the inner one. This may be due to wall resistance as outer flow is having contact with both inner part of annulus and outer part of inner tube. Figure 16 shows the comparison of friction factor variation with outer Reynolds Number for both helical and straight annulus tube.



Optimized coil used for comparison with straight tube

Fig. 12 Optimized coil used for comparison with straight tube

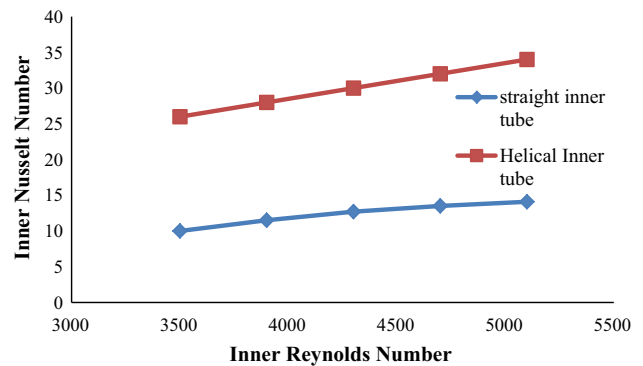


Fig. 13 Variation of Nusselt Number with Reynolds Number for helical and straight inner tubes

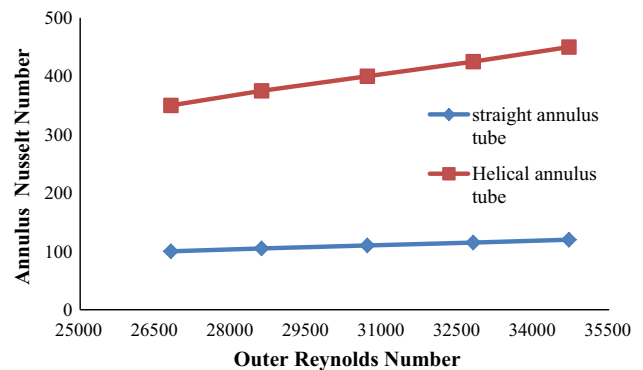


Fig. 14 Variation of Nusselt Number with Reynolds Number for helical and straight annulus tubes

Conclusions

Simulations are carried out for finding out optimum cross-sectional shape for double pipe helical coil using 4 shapes which are circular, rectangular, triangular, and square. It was found that rectangular cross-section was having best heat transfer properties followed by triangular, square and

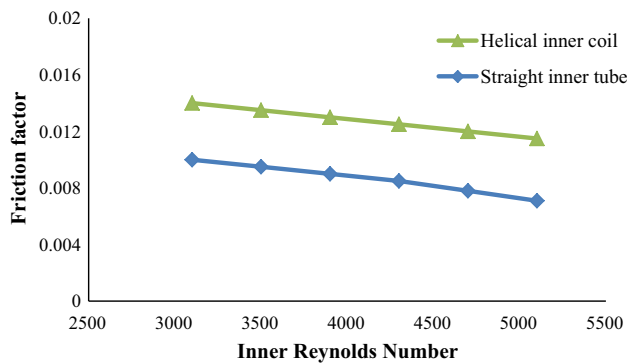


Fig. 15 Graph showing variation of friction factor with Inner Reynolds Number

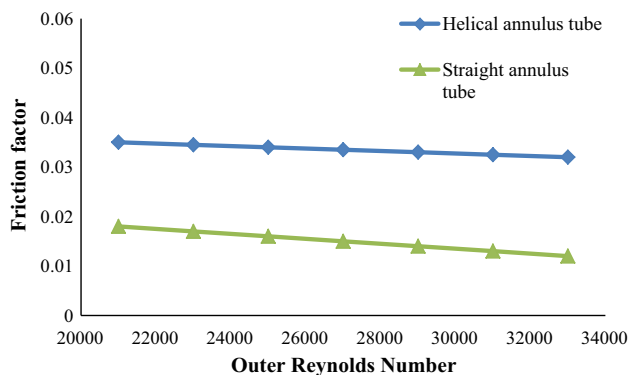


Fig. 16 Graph showing variation of friction factor with Outer Reynolds Number

circular. But pressure drop was highest for rectangular shape and least for circular shape.

Various simulations are carried out for finding out how the taper angle on helical coil influences heat transfer and pressure drop properties. It was found that as the coil taper angle increases, the heat transfer rate increases considerably, while pressure drop also increases with taper angle.

After carrying out various simulations for finding out optimum coil, it was found out that a rectangular cross-section coil (Fig. 12) with maximum possible taper angle can give best heat transfer characteristics. The extent of taper angle is limited by coil length increase which would demand more space, and also the pressure drop and

pumping power. This rectangular cross-section coil with a fixed taper angle of 30° is used for comparison with double pipe straight tube. It was observed that heat transfer in the optimized helical coil was nearly twice more than that of straight tube. The friction factor was higher in the helical coil than straight tube which implies more pumping power requirement for helical coil.

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