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The effect of geometrical parameters on heat transfer coefficient in helical double tube exchangers

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ABSTRACT

Helical coil heat exchangers are widely used in industrial applications ranging from cryogenic processes, air-conditioning, and nuclear reactors to waste heat recovery due to their compact size and high heat transfer coefficient. In this kind of heat exchangers, flow and heat transfer are complicated. This paper reports 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 of 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.

1. Introduction

An increase in the heat transfer coefficient is very important in size reduction, thermodynamic efficiency enhancement, and pumping power reduction. Therefore, different methods were applied for heat transfer rate increasing. It was usually carried out by increasing the heat transfer or/and heat transfer surface area per unit volume. Surveying of heat exchangers and heat transfer issues and its enhancement through inactive methods, requires devising a new scheme. In double tube helical heat exchangers, due to the curvature of tubes and exertion of a centrifugal force on fluid flow, the secondary flow motion was generated which improves heat transfer coefficient substantially. In the analysis of performance of these equipments, several parameters were taken into account such as the Dean number, curvature ratio, tube diameter, and coil pitch.

Many of researchers investigated heat transfer and hydrodynamic characteristics of fluid in double tube heat exchangers in recent years. Rennie and Raghavan [1] simulated heat transfer characteristics in a double tube heat exchanger. They proved that, the flow in the inner tube is the limiting factor of the overall heat transfer coefficient of the heat exchanger. By stabilizing other parameters, the overall heat transfer coefficient will increase. Yadav and Tiwari [2] presented a transient analysis on a double pipe heat exchanger connected to a flat solar collector with counter and parallel flows. Rennie and Raghavan [3] performed an experimental work with two different-sized double pipe helical heat exchangers and investigated both counter and parallel flows. They indicated that there is a small difference between the overall heat transfer coefficient of counter and parallel flows of heat exchangers. In another work, they numerically [4] studied the heat transfer characteristics of helical double tube heat exchanger to determine the effect of fluid thermal properties on the

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heat transfer. They demonstrated that in lower Dean numbers, the Nusselt number was more affected by the Prandtl number than in higher Dean Numbers. Han et al. [5] experimentally investigated heat transfer characteristics of R-134a in double pipe heat exchangers. Their results indicated that the average heat transfer coefficient and the heat transfer coefficient of this refrigerant would increase as the mass flow rate of R-134a increases. Kumar et al. [6] experimentally and numerically studied heat transfer coefficients of a helical double tube heat exchanger. They pointed out that the overall heat transfer coefficient of the heat exchanger increases in the presence of a constant mass flow rate within the annulus by an increase in the Dean number of the inner pipe, and vice versa. Lin and Ebadian [7] experimentally studied pressure drop characteristics of R-134a in curved helical tubes. Results indicated that the Nusselt number of the refrigerant in lower saturation temperatures was so high and increased with increasing mass flow rates of both the refrigerant and the cooling water. They also indicated that the flow of the refrigerant in the annulus can lead to higher-pressure drops than the inner tube. Numerical study of helical double pipe heat exchangers was performed by Kumar et al. [8]. They obtained the overall heat transfer coefficient of a heat exchanger for different flow rates in the inner and outer tubes. Garrido et al. [9] developed a numerical model for heat transfer and dynamic behavior of fluids in helical double pipe evaporators. The inner pipe pressure and the temperature of the annulus are two effective parameters in determination of an increase in the outlet temperature. Xiaowen and Lee [10] experimentally surveyed performance of Window Air Conditioner (WAC) by using a helical double pipe heat exchanger for preheating hot water. Results indicated that the coefficient of performance of WAC increased by the use of a heat recovery system. Xin et al. [11] studied the single-phase flow in a helical double tube heat exchanger in horizontal and vertical arrangements. In their work, the influence of coil geometry and flow rate of air and water on the pressure drop of the single-phase flow was surveyed. Petrakisc and Karahalios [12] studied a viscous fluid flow in coils of a double tube heat exchanger when the pressure gradient along the axis reduces exponentially with time. In another work [13], they obtained the numerical solution of an incompressible viscous fluid flow of water flowing in a curved double tube with a circular cross section. It was indicated that in small radius of core, the change of Dean number has significant impact on fluid's properties whereas the aforesaid phenomenon was not observed in a larger radius. Pressure drop characteristics of R–134a in a helical double tube were surveyed by Han et al. [14]. Beigzadeh and Rahimi [15] presented a model by the use of the artificial neural network to estimate the heat transfer coefficient and friction factor in helical tubes. Their work indicated that the neural network is useful in the prediction of the heat transfer and flow characteristics of helical heat exchangers. Di liberto and Ciofalo [16] studied the heat transfer of a turbulent flow in curved tubes by means of a numerical simulation. They also used this method to fully survey a developed turbulent flow in curved tubes. Results of their investigation indicated that in curved tubes, the turbulent velocity and temperature fluctuations in the outer regions are more drastic than the inner regions. Gabriela and Huminic [17] three dimensionally studied the heat transfer characteristics of nano-fluids in helical double tube heat exchangers. They found that Nano-fluids increase the fluid thermal conductivity and heat transfer.

Majority of the performed studies in the field of analyzing and testing the helical heat exchangers, were limited to a single helical coil, whereas the current study deals with heat exchangers with several coils that were analyzed. Also, their thermally development status was investigated and also we tried to investigate the effect of geometrical properties on the heat transfer characteristics.

2. Solving Approach

2.1. Characteristics of helical coils

One of the most important characteristics of helical coils is the stretch of flow through the coils. According to figure1 (a), the centrifugal forces are created because of this stretch of flow which leads to a secondary flow. In figure 1(a), it is observed that the maximum velocity is located in central region of inner tube because this region has the maximum distance from the walls and thus from the boundary layers.

Another important characteristic of helical coil heat exchangers is the Dean number. As the Reynolds number is used to analysis of flow in straight tubes, the Dean Number is for curved tubes used which is defined as follows [3]:

$$De_{in} = \left(\frac{Vd_i}{\nu}\right)\left(\frac{d_i}{2R}\right)^{1/2} \tag{1}$$

As helical double pipe heat exchangers are used extensively due to their flow mixing ability (because of secondary flows) in laminar flows, it was decided to choose flow rates in the laminar flow ranges. Therefore, to survey whether flow is laminar, the obtained Reynolds number can be compared with critical number presented by Srinivasan [18]:

$$\mathrm{Re}_{cr}=2100(1+12\,\delta^{1/2}) \tag{2}$$

2.2 Problem description

The heat exchanger model used in this analysis is the experimental model presented by Rennie and Raghavan [3] (figure2). The model is a double pipe heat exchanger

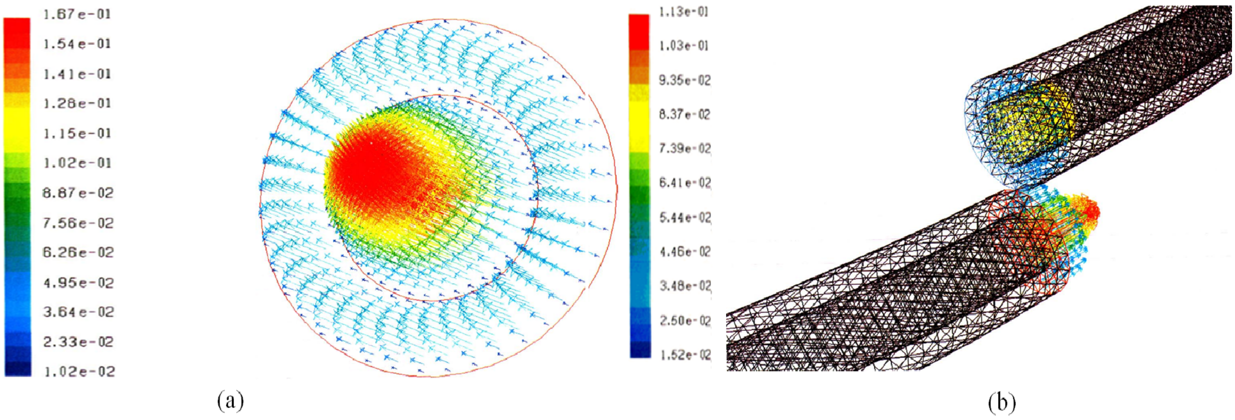

Fig.1 Schematic diagram of the inlet and outlet vectors of a parallel flow in the helical double pipe heat exchanger.

with pipes made of copper that water at $60^{o}C$, as the hot fluid, flows through the inner tube. On the other hand, water at $22.1^{o}C$ flows in the annulus pipe as the cooling fluid, and in the same direction of the hot water. The geometrical properties of the model can be seen in figure 2. It can be noted that, the thickness of both tubes is 0.8mm and each coil with the annulus diameter of 23.59 mm has wrapped up completely.

2.3. Mesh generation process

Choosing a suitable mesh generation technique requires a thorough examination and consideration of criteria. Many papers were published with regard to this category, among which Xavier et al. [19] can be mentioned. They suggested tetrahedral meshing owing to its short-term generation and stabilization. For this reason, as it can be seen from figures 1(b) and 3, structured tetrahedral mesh was generated in Gambit software for this model.

To study the effects of geometrical parameters on heat transfer characteristics in helical double tube heat exchangers, the present numerical model should be independent of the grids. Hence, table1 shows the results of mesh generation process to reach a model which is independent of the grids.

According to the results of mesh generation process, a model with 75664 grids has been chosen for the base of present work.

2.4. Governing equations

Fluid properties were assumed variable for a realistic surveying [20]. To model the temperature-dependent properties, the following Polynomial functions (Eqs. (3)–(6)) were programmed in FLUENT. In CFD code, the governing equations are solved with the given fluid properties.

$$\mu(T) = 2.1897e-11T^{4} - 3.055e-8T^{3} + 1.6028e-5T^{2} - 0.0037524T + 0.33158 \quad (3)$$

$$\rho(T) = -1.5629e-5T^{3} + 0.011778T^{2} - 3.0726T + 1227.8 \quad (4)$$

$$k(T) = 1.5362e-8T^{3} - 2.261e-5T^{2} - 0.010879T - 1.0294 \quad (5)$$

$$C_{p}(T) = 1.1105e-5T^{3} - 0.0031078T^{2} - 1.478T + 4631.9 \quad (6)$$

All of physical phenomena, including the fluid flow and heat transfer, can be analysed by using of the continuity, momentum and energy equations:

$$\frac{\partial u_{i}}{\partial x_{i}} = 0 \quad (7)$$

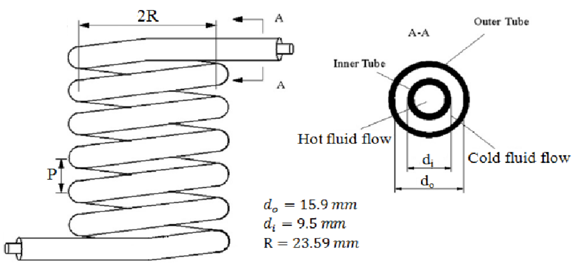

Fig.2 Schematic diagram of the used model

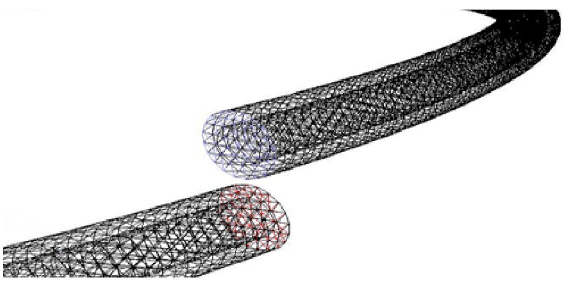

Fig.3 Grids used in the analysis.

Table1. Mesh independency check

Number of generated grids	h_i
30761	496.363
43067	386.936
70810	407.384
75664	403.234
77418	399.989

$$\frac{\partial (u_i u_j)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\nu \frac{\partial u_j}{\partial x_i}\right) - \frac{1}{\rho}\frac{\partial P}{\partial x_j} \quad (8)$$

$$\frac{\partial (u_i T)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\alpha \frac{\partial T}{\partial x_i}\right) \quad (9)$$

2.5. Boundary conditions

The inlet boundary condition for both the inner tube and the annulus is a defined inlet velocity, because the entrance velocities of the flows are known in this case. In the present work, pressure outlet is used for the outlet boundary condition. It assumed that the outlet pressure is equal to the ambient pressure. In the modelled helical double tube heat exchanger, there are two walls. For the outer tube wall, we assum adiabatic boundary condition with zero heat flux, and for the inner tube wall we choose a bilateral wall type and heat transfer is possible on both sides.

2.6. Calculation of the heat transfer coefficient

The overall heat transfer coefficient is obtained using the given inlet and outlet temperatures, and the following equations (10-11):

$$U = \frac{q}{A_o \Delta T_{LMTD}} \quad (10)$$

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\Delta T_2 / \Delta T_1\right)} \quad (11)$$

where we have for ΔT_1 and ΔT_2 :

$$\Delta T_1 = T_{h,i} - T_{c,i}$$

$$\Delta T_2 = T_{h,o} - T_{c,o}$$

where $T_{h,i}$ and $T_{h,o}$ are temperatures of the hot fluid at the inlet and outlet.

2.7. Model validation

As already mentioned, the numerical method is used to investigate the model in this work. It should be noted that the numerical method is not suitable enough to get the trusted results. Hence, at the first and before solving the problem with numerical methods, we compare our results with experiments [3]. In figure 4, results of problem's solution and experimental work [3] for a laminar parallel flow are compared. The problem is solved for annulus mass flow rates of 100, 300 and 500 cm^3/min and different Dean numbers for the inner tube. The overall heat transfer coefficient is obtained by using of equations 10 and 11. Figure 4 shows a good agreement between the numerical work and experimental results.

The numerical method is not a trusted one to study the effects of different parameters. Hence, the results obtained from this method have some differences with the actual results, because of various factors such as sediment factor participated in the calculation of heat exchangers which may not be considered in the numerical analysis. The average error related to the inner Dean number of 208.1 is equal to 2.3 percent; hence, this inner Dean number has been chosen as the basis of future calculations in the current study.

3. Result and Discussions

For augmentation of heat transfer between two fluids, secondary flow plays the main role in helical heat exchangers. Therefore, the effects of several geometrical parameters on the heat transfer characteristics are surveyed in this study. Obtained results can be presented as the curvature ratio effect, increase of loop pitch of heat exchanger, tube diameter, and influence of number of loops on heat transfer characteristics.

3.1. Comparison of velocity contour in straight and helical double tube heat exchanger

In figures 5 and 6, x is the distance from the inlet of the heat exchanger. As winding of the loop is 360^o, x values of 0.37, 0.74, and 1.11m represent angles of 90^o, 180^o, and 270^o, respectively. By comparing these three contours, it can be seen that the flow becomes disordered in the helical double tube heat exchanger. According to figure 1(a,b), flow is stretched and the stretch in all of three cases is from the inner wall towards the outer wall in the helical double tube heat exchanger. This stretch can be attributed to centrifugal forces in helical double tube heat exchangers. The secondary flow is a result of the centrifugal forces.

3.2. Effect of curvature ratio on heat transfer

Curvature ratio is one of the most important parameters in helical double tube heat exchangers and is presented by $d_o/2R$. The Dean number includes the curvature ratio and it has to be noted that both parameters of d_o and R can change the curvature ratio. Here, to avoid the variation of Reynolds number in the annulus, only the radius of loop is changed. In figure 7 it is observed that the increase of curvature ratio increases the heat transfer coefficient significantly. It means that an increase in the loop's radius reduces the overall heat transfer drastically.

As the radius of the loops increases, fluid's torsion behavior approaches to a linear behavior and also the helical tube turns to the straight one. To explain and illustrate the influence of radius changes, we use figures

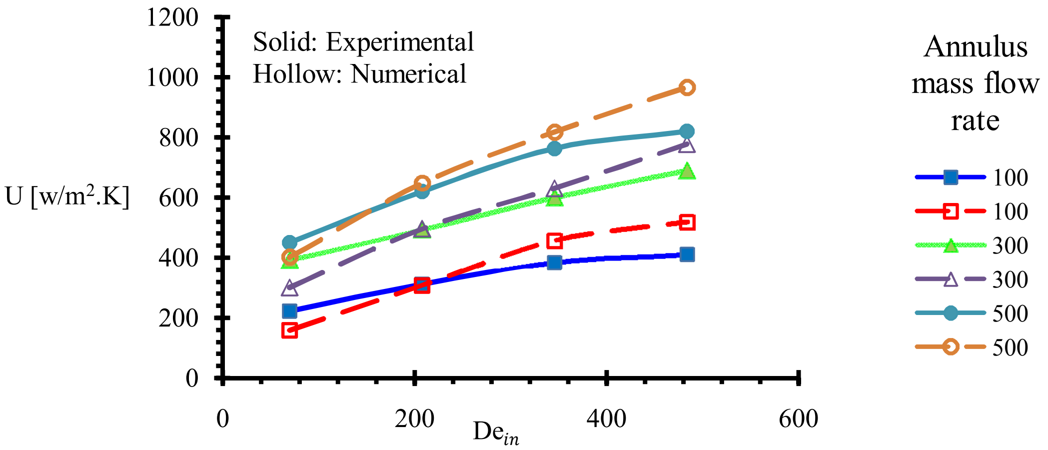

Fig.4 Comparison of numerical and experimental [3] results for the overall heat transfer coefficient at different mass flow rates

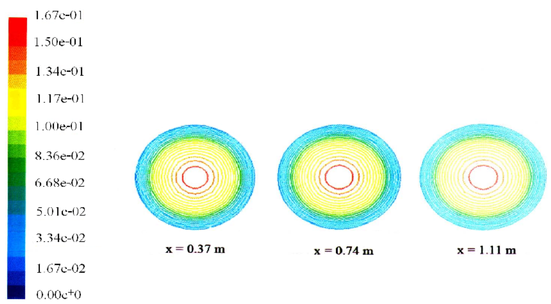

Fig.5 velocity contour in the inner tube of the parallel flow in a straight double tube heat exchanger, inner tube flow rate is $0.018 m^3/s$

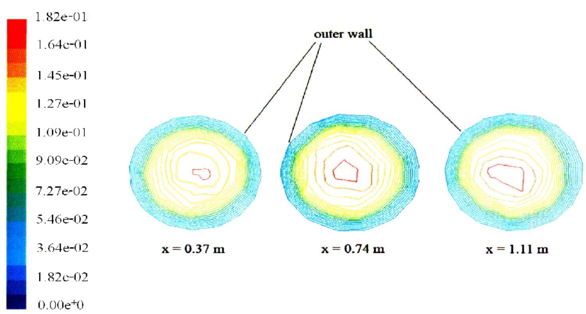

Fig.6 velocity contour in the inner tube of parallel flow straight double tube heat exchanger, inner tube flow rate is $0.032 m^3/s$

8-10. With these three-dimensional diagrams, we can compare simultaneously the effects of Reynolds number through the inner tube and the mass flow rate of the annulus on the overall heat transfer coefficient. These figures indicate that an increase in the annulus mass flow rate and the internal Dean number increases the heat transfer coefficient between these two fluids.

3.3. The Influence of pitch of coil on heat transfer

Influence of tube's pitch on the overall heat transfer coefficient of heat exchanger is surveyed in this section.

It must be noted that in this study, pitches of 1.6, 2, and 2.6 are used. Figure 11 indicates the influence of tube pitch size at different inner Dean number on the overall heat transfer coefficient in laminar flow regime.

Increasing the pitch of coils decreases the torsion behaviour of flow as well as the heat transfer coefficient. But, as it can be seen from figure 11, this influence is so negligible. It can be said that the pitch of the loops doesn't have a significant impact on the heat transfer in helical exchangers.

3.4. Influence of the tube diameter change on heat transfer characteristics

Another important parameter surveyed in this work, is the diameter of tubes used in heat exchanger. As it can be observed from figure 12, an increase in the inner tube diameter and the inner Dean number increases the Nusselt number. Consequently according to the Nusselt number's definition (Eq.14), the heat transfer will increase.

$$Nu = \frac{h_o D}{k} \tag{14}$$

The inner and outer heat transfer coefficients, that are usually obtained from the overall thermal resistance consists of three resistances in series: the convective resistance in the inner surface, the conductance resistance of the pipe wall, and the convective resistance on the outer surface given by the equation 15:

$$\frac{1}{U} = \frac{A_o}{A_i h_i} + \frac{A_o \ln\left(\frac{d_o}{di}\right)}{2\pi k L} + \frac{1}{h_o} \tag{15}$$

where d_o is the diameter of the outer tube; d_i is the diameter of the inner tube; k is the thermal conductivity of the wall and L is the length of the inner tube.

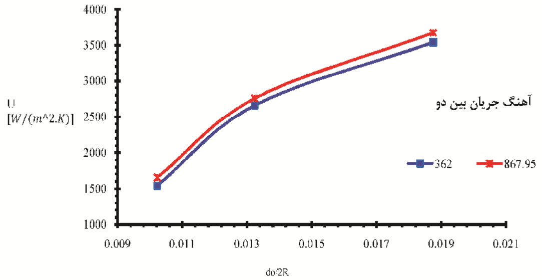

Fig.7 Influence of the curvature ratio on the overall heat transfer coefficient

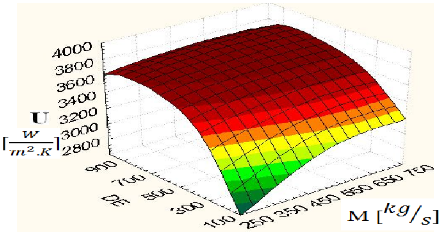

Fig.8 Influence of the variation of annulus mass flow rate on the overall heat transfer coefficient, R=24cm

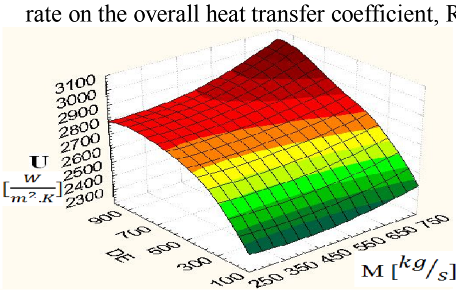

Fig.9 Influence of the variation of annulus mass flow rate on the overall heat transfer coefficient, R=34 cm

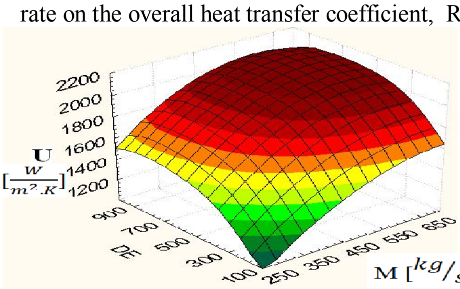

Fig.10 Influence of the variation of annulus mass flow rate on the overall heat transfer coefficient, R=44 cm

Heat transfer coefficients between the two tubes, ho and for the inner tube hi, were calculated using traditional Wilson plot technique [3, 6]. For the calculation of the outer heat transfer coefficient in Eq. (14), the mass flow rate in the annulus side was kept constant; and assumed that the inner heat transfer coefficient is constant. The outer heat transfer coefficient was assumed to behave in the following manner with the fluid velocity in the tube side, u_o:

$$h_o = Cu_o^n \tag{16}$$

Eq. (16) was put into Eq. (15) and the values for the constant C and the exponent n were determined through the curve fitting. The inner and outer heat transfer coefficients could then be calculated. Similar procedure was adopted for the calculation of the inner heat transfer coefficient.

3.5. Influence of the number of coils on the overall heat transfer coefficient

Increasing the number of coils is equal to lengthening the heat transfer path. So, according to the figure 13, when the number of coils increases, the amount of the heat transfer coefficient decreases significantly. For the number of coils larger than n=6, it approaches to a constant amount.

Since the Prandtl number for water is larger than one (Pr>1), according to the Eqs. 17 and 18, the length of the thermal entrance will be greater than hydraulic entrance length.

$$L_H = 0.058.\mathrm{Re}_D.D \tag{17}$$

$$L_T = 0.058.\mathrm{Re}_D.D.\mathrm{Pr} \tag{18}$$

As we know, until the fluid has not passed the thermal entrance length, flow was developing and the heat transfer coefficient decreases during the thermal development. The least value will occur in the fully developed region. Increasing the number of coils results in longer path, and this indicates that the flow will be fully developed.

4. Conclusions

The present study has numerically conveyed the heat transfer in a helical double tube heat exchanger with variable fluid properties. In this investigation, after the validation of the obtained numerical results compared

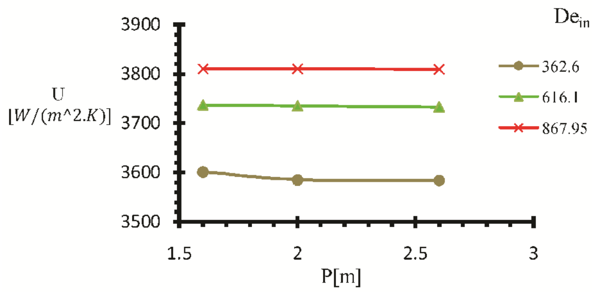

Fig.11 Influence of the pitch increase on the overall heat transfer coefficient

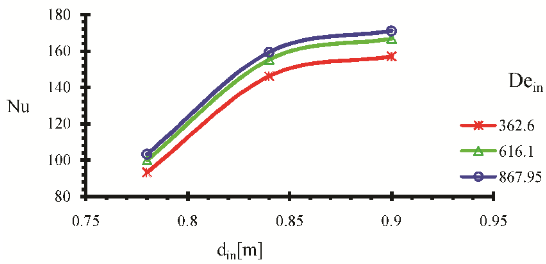

Fig.12 Influence of inner tube diameter on Nusselt Number

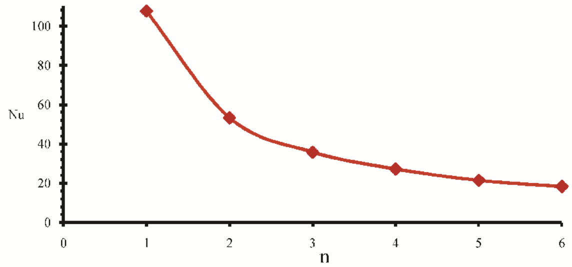

Fig.13 Influence of the number of coils on the overall heat transfer coefficient

with the available experiment [3], it was indicated that the present model is in a good agreement with the experimental work. By surveying different parameters, following results were obtained:

It is observed that, the maximum velocity is located in central region of the inner tube because this region has the maximum distance from the boundary layers.

The Dean number is an important and effective parameter in helical double tube heat exchangers. By increasing of the inner tube diameter, the overall heat transfer coefficient of heat exchanger increases. It should be mentioned that with an increase in the annulus mass flow rate (that leads to an increased Dean number) the rate of heat transfer will increase.

The Nusselt number has a direct relevance to the inner tube diameter and it means that increasing the heat exchanger coil radius leads to a drastic decrease in the overall heat transfer coefficient. It is argued that as the coil radius increases, torsional behavior of fluid will approach to the linear behavior and a helical tube will become similar to a straight tube.

The heat transfer coefficient is augmented considerably by increasing the curvature ratio. It can be argued that with more radius increments, the torsional behavior approaches towards the linear one.

Increasing the pitch of the heat exchanger, leads to decrease of its overall heat transfer coefficient. But, this effect is negligible. So, it can be concluded that the tube pitch is a parameter that doesn't have a great influence on the analysis trend.

Increasing the number of coils means a longer heat transfer path. The observed decrease in the Nusselt number due to an increase in the number of coils indicates that the flow will be fully developed in a larger portion of the path.

Nomenclature

A_o	outer tube surface area (m^2)
A_i	inner tube surface area (m^2)
C_p	specific heat ($J\ kg^{-1}K^{-1}$)
De_{in}	inner tube Dean number
d_i	diameter of inner tube (m)
d_o	diameter of outer tube (m)
h	convective heat transfer coefficient ($Wm^{-2}\ K^{-1}$)
k	Thermal conductivity (W/mK)
L_H	length of hydraulic entrance (m)
Nu	Nusselt number
L_T	length of thermal entrance (m)
M	mass flow rate of annulus tube ($kg\ s^{-1}$)
pr	Prandtl number, υ_f/α_f
n	number of coils
P	Pressure (Pa)
P	pitch of the coils (m)
$\dot{q}$	heat flux per unit of area (W/m^2)
R	radius of coil (m)
Re_{cr}	critical Reynolds number
t	Time (s)
T	Temperature (K)
ΔT_1	fluid inlet temperature difference (K)
ΔT_2	fluid outlet temperature difference (K)
ΔT_{LMTD}	Log-mean temperature difference (K)
u_i, u_j	Velocity in x, y, and z direction (ms^{-1})
U	overall heat transfer coefficient
V	average velocity (m)
x_i	x, y, and z directions (m)
L_H	length of hydraulic entrance (m)
Nu	Nusselt number
Greek symbols	
ρ	Density (kgm^{-3})
μ	dynamic viscosity ($kgm^{-1}s^{-1}$)
δ	curvature ratio $\left(\frac{d}{2R}\right)$
α	thermal diffusivity $\left(\frac{k}{\rho c_p}\right)$
ν	kinematic viscosity
Subscript	
c	cold
cr	critical
h	hot
i	inner side(tube)
o	outer side(tube)

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