Parametric Analysis of Tube in Tube Helical Coil Heat Exchanger at Constant Wall Temperature

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Abstract

Working towards the goal of saving energies and to make compact the design for mechanical and chemical devices and plants, the enhancement of heat transfer is one of the key factors in design of heat exchangers. In this process without application of external power we can enhance the heat transfer rate by modifying the design by providing the helical tubes, extended surface or swirl flow devices. Helical tube heat exchanger finds applications in automobile, aerospace, power plant and food industries due to certain advantage such as compact structure, larger heat transfer surface area and improved heat transfer capability. In this project numerical studies of helical coil tube-in-tube heat exchanger for find out heat transfer at different D/d ratio. The turbulent flow model with counter flow heat exchanger is considered for analysis purpose. The effect of D/d ratio on heat transfer rate is found out for constant outer wall temperature boundary conditions. The D/d ratio is varied from 10 to 25 with an interval of 5. Nusselt number, friction factor and LMTD variation of inner fluid with respect to Reynolds number is finding out for different D/d ratio.

Keywords: Counter Flow Heat Exchanger, D/d ratio, Friction factor, Nusselt number, Friction factor, LMTD

I. INTRODUCTION

A heat exchanger is a device that is used to transfer thermal energy between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single or multicomponent fluid streams. The objective may be to recover or reject heat, crystallize or control a process of fluid. Helical coil heat exchangers are universally used in various industrial applications ranging from heat exchangers, power plant, electronics, environmental engineering, manufacturing industry, air-conditioning, waste heat recovery, cryogenic processes, to chemical processing, because of their compact size and high heat transfer performance. The flow and convective heat transfer in a helical coiled tube are complicated as compared with the straight tube, because strongly depend on the behavior of secondary flow.

II. LITERATURE REVIEW

Naphon and Wong wises et al. (2005) [1] Was Study of the heat transfer characteristics of a compact spiral coil heat exchanger under wet-surface conditions. They had done the numerical and experimental studies to find out the heat transfer rate and predict the presentation of a spiral coil heat exchangers. Cooling and dehumidifying conditions were used for investigation.

Kumar et al. (2006) [2] 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.

Jayakumar et al. (2010) [3] had done the numerical and experimental analysis to find out the variation of local Nusselt number along the length and perimeter 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 calculation of Nusselt number. The Nusselt number variation w.r.t angular location of the point was also predicted in this literature.

Naphon (2011) [4] was Study on the flow and heat transfer characteristics in a spiral-coil tube. He did both the numerical and experimental study on a horizontal spiral-coil tube to calculate the flow characteristic. The standard k–ε two-equation turbulence model was used to simulate the turbulent flow and heat transfer characteristics of the fluid. The heat transfer rate or heat transfer coefficient had affected by the centrifugal force. Conversely, the pressure drop also increases.

Ghorbani et al. (2010) [5] had done the experimental study to assume the behavior on the mixed convection heat transfer in a coil-in-shell heat exchanger. They chose the operating parameters for the analysis are Reynolds, Rayleigh numbers and also the tube-to-coil diameter ratios. They had done the steady-state analyses and they had done the experiments for both laminar and turbulent flow. It was found that the mass flow rate of tube-side to shell-side ratio was effective on the axial temperature profiles of heat exchanger.

Yang et al. (2011) [6] had done experimental work to predict the characteristics of convective heat transfer in heat exchanger. They consider a heat exchanger with membrane helical coils and membrane serpentine tubes. The efficiency of the power generating system was affected by heat transfer performance of syngas cooler. They had done the experimental investigation on heat transfer in convection cooling section of pressurized coal gasifies with the membrane helical coils and membrane serpentine tubes under high pressure. They found that the heat transfer coefficient increases due to the increase of gas pressure and velocity.

Srbislav et al. (2012) [7] 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 had been 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,

$Nu = 0.50 Re^{0.55} Pr^{1/3} (\eta/\eta_w)^{0.14}$

Jamshidi et al. (2013) [8] had done experimental work enhance the heat transfer in shell and helical tube heat exchanger. In the helical tube section of the heat exchanger hot water flows. The cold water flows in the shell side of the heat exchanger. The heat transfer coefficients are determined using Wilson plots. Taguchi method is used to find the optimum condition for the desired parameters in the range of $0.0813 < Dc < 0.116$, $13 < Pc < 18$, tube and shell flow rates from 1 to 4 litter per minute. From their results it is found that the higher coil diameter, coil pitch and mass flow rate in shell and tube can enhance the heat transfer rate for this type of heat exchanger. Contribution ratio obtained by using Taguchi method ; shows that shell side flow rate, coil diameter of helical coil, tube side flow rate and coil pitch are the most important design parameters in coiled heat exchangers.

Huminic et al. (2011) [9] had numerically investigated the heat transfer characteristics in double tube helical coil heat exchangers. Nano fluids were used as working fluid in the exchanger. They consider laminar flow condition. CuO and TiO_2 are used as Nano particles in the working fluid. The concentration of Nano particles affects the heat transfer rates. The Dean number which is a function of curvature ratio also affects the heat transfer coefficients in helical heat exchanger. They came to know that by the use of Nano particles as working fluid the heat transfer rate can be improved from that of pure water.

Ferng et al. (2012) [10] 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 behaviors from the entrance region, etc. was studied by them.

Jamshidi et al. (2012) [11] had done numerical work to optimize design parameter of Nano fluids inside helical coils. In their study they used water/Al_2O_3 (aluminum oxide) Nano fluid in helical tubes. The fluid flow was assumed to be laminar. The outer wall of exchanger was maintained at constant wall temperature. Thermo physical properties of Nano fluids are depend on particles volume fraction and temperature. Numerical simulations are used to investigate the effect of fluid flow and geometrical parameters. Taguchi method is also used to optimize the geometrical parameter of heat exchanger. From their results they found that the thermal–hydraulic performance of helical tubes was improved by the Nano fluids. But the Nano fluids don't change the optimized shape factors.

Yang San et al. (2012) [12] had numerically investigated the heat transfer characteristics of a helical heat exchanger. The performance of a helical heat exchanger was investigated on the basis of heat transfer rate. The cross section of the tube was made rectangular section with two cover plates. They found that the friction factor was increased with the increase in spacing of the channel. The friction factor decrease with increase in the Reynolds number. They found Nu increases with increase in Re and channel spacing.

III. Model and Analysis

The heat exchanger to be numerically modeled consists of two coiled tubes, placed one inside the other. In the present study the hot fluid flows in the inner-coiled tube, while the cold fluid was flowing in the opposite direction annulus region formed by two coiled tubes. The outer-coiled comprised of semicircular plates to support the inner-coiled tube and to provide turbulence in the annulus region. The simulation package used was ANSYS FLUENT 15, which makes use of the control volume finite difference method (CVFDM). The governing equations were solved for flow, temperature and pressure values at every cell.

A. Mathematical Formulation:

In the study, the double tube helical coil heat exchanger or tube in tube helical coil heat exchanger with two (2) numbers of turns is considered. For simplification in numerical analysis only two turns are considered 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. The inner tube diameter (d1) was 8mm. the thickness (t) of the tube was taken 0.5mm. The outer tube diameter (d2) was taken 20mm. In this study, the tube diameter (both inner and outer diameter) of the heat exchanger is fixed and the coil diameter of the tube is varied 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.

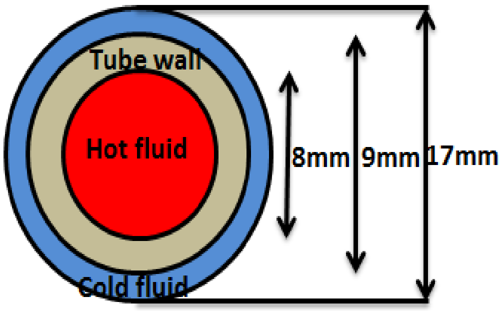


Fig. 1: Front view of tube in tube helical coil heat exchanger

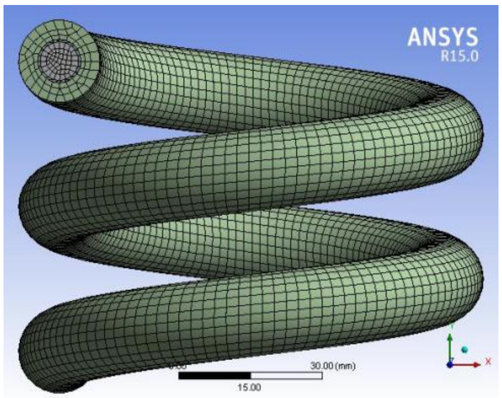


Fig. 2: meshing for tube in tube helical coil heat exchanger

The mathematical models used to investigate the flow and heat transfer characteristics in a helically coiled-tube heat exchanger include continuity, momentum, energy equations, turbulence model, and appropriate boundary conditions. Constant properties are assumed for the cold water in the coiled tube.

Table – 1:
Parameters of helical coil heat exchanger

Parameters	*Dimensions*
Diameter of outer tube	*17 mm*
Diameter of inner tube	*8 mm*
Pitch of the coil	*30 mm*
Thickness of the tube	*0.5 mm*
Number of coil turns	*2*
Coil diameter	*80-240*
Heat exchanger wall material	*Copper*
Fluid	*Water*

B. Boundary Conditions:

The outer wall of the heat exchanger has taken constant wall temperature of 330K, It can be expressed numerically by; at d=17mm; T=330K

For inner wall conjugate heat transfer, boundary condition was taken. In this condition the heat is transferred from one fluid to the other fluid via a solid (wall of the tube) so that the cold fluid get warmer and the hot fluid get colder.

For inlet of hot fluid velocity inlet condition was taken. Here the velocity of the fluid was varied by changing the Reynolds number (Re). For turbulent fluid flow the critical Re was found out by using the Schmidt correlation as given by equation no- 10. The Reynolds number of the working was fluid assumed at the inlet are 12000, 15000,18000,21000,24000. Respectively. As the Reynolds number changes the mass flow rate of the hot fluid also changes and maximum for Re=24000. The hot fluid temperature at inlet was taken 355 K. representing this condition in mathematical form we have;

At x, y, z=0; ux=uy=0 and uz=1.55072, 1.884, 2.26081, 2.6376, 3.01442 m/sec respectively and Thi=355K

Similarly for the cold fluid at the inlet velocity inlet condition was also taken. For the cold fluid the fluid flow rate was assumed to be constant. The Reynolds number for the outer fluid was taken 25000 for all the condition of fluid flow. The mass flow rate of the cold fluid was found to be 0.512105 kg/sec. The temperature of the fluid at the inlet is taken 290 K.

At exit, Re=25000 or ux=uy=0 and uz=-3.14002m/sec and Tci=290

C. All of the Equations Can Be Described As Follows:

1) *Governing Equations:*

The governing differential equation for the fluid flow is given by Continuity equation or mass conservation equation, Navier Stokes equation or momentum conservation equation and energy conservation equation.

a)Continuity Equation:

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \qquad \text{….. (1)}$$

b)Navier Stokes Equation:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=\rho x-\frac{\partial p}{\partial x}+\frac{1}{3}\mu\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\mu\nabla^2 u$$
$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=\rho y-\frac{\partial p}{\partial y}+\frac{1}{3}\mu\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\mu\nabla^2 v$$
$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=\rho z-\frac{\partial p}{\partial z}+\frac{1}{3}\mu\frac{\partial}{\partial z}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\mu\nabla^2 w \qquad \text{….. (2)}$$

a) Energy Equation:

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\left(u\frac{\partial p}{\partial x}+v\frac{\partial p}{\partial y}+w\frac{\partial p}{\partial z}\right)+k\nabla^2 T+\mu\emptyset \qquad \text{….. (3)}$$

Where;

$$\emptyset=2\left[\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial w}{\partial z}\right)^2\right]+\left[\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2+\left(\frac{\partial w}{\partial x}+\frac{\partial u}{\partial z}\right)^2\right]-\frac{2}{3}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)^2 \qquad \text{….. (4)}$$

The governing differential equation for solid domain is only the Energy equation which is given by;

$$\nabla^2 T=0 \qquad \text{….. (5)}$$

Heat transfer coefficient is obtained by equating the conduction heat transfer to the convection heat transfer;

$$Q_{cond}=Q_{conv}$$

$$h=\frac{-k\frac{\partial T}{\partial x}}{T_W-T_r} \qquad \text{….. (6)}$$

Local Nusselt number is given by;

$$Nu_x=\frac{hD}{k} \qquad \text{….. (7)}$$

Or it can also be represented by following equation

$$Nu_x=\frac{-\frac{\partial T}{\partial x}d_h}{T_W-T_f} \qquad \text{….. (8)}$$

Then the average Nusselt number can be found by following relation;

$$Nu_{avg}=\frac{1}{L}\int_0^L Nu_x\,dx \qquad \text{….. (9)}$$

Critical Reynolds number as per the Schimidt correlation (1967);

$$Re_{cr}=2300[1+8.6\,(d/D)^{0.45}] \qquad \text{….. (10)}$$

Length of pipe is given by following relation;

$$L=\ n\sqrt{H^2-(\pi D)^2} \qquad \text{..… (11)}$$

Log Mean Temperature Difference for counter flow heat exchanger can be presented by following relation

$$LMTD=\frac{\Delta T_1-\Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}} \qquad \text{…... (12)}$$

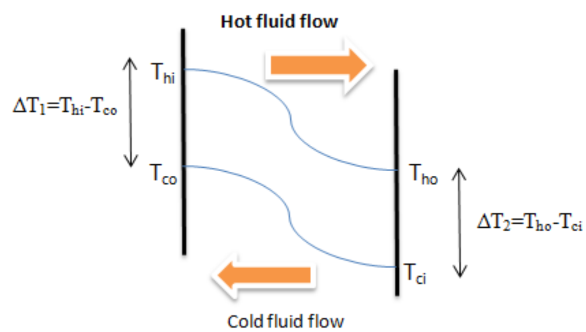


Fig. 3: Log mean temperature difference of hot and cold fluid

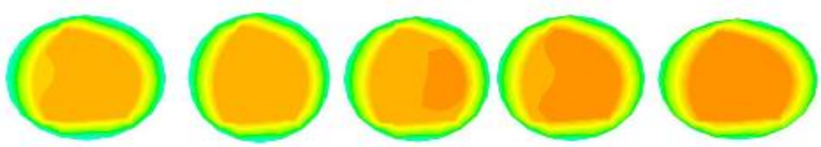
Where $\Delta T1 = Thi - Tco$
And $\Delta T2 = Tho - Tci$
To find the outlet fluid temperature we can use the energy balance equation,

$$Q = m\,Cp\,(T1 - T2) = h\,As\,(Tw - Tf) \quad \ldots\ldots (13)$$

IV. Results and Discussion

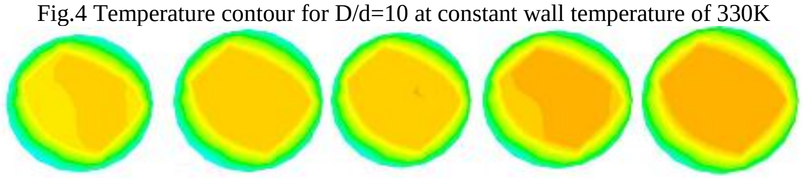
A. Constant Outer Wall Temperature Boundary Condition:

For constant outer wall condition the temperature contour of hot fluid outlet is shown in below fig.The inner Nusselt number, friction factor, LMTD are calculated subsequently with respect to Reynolds number.



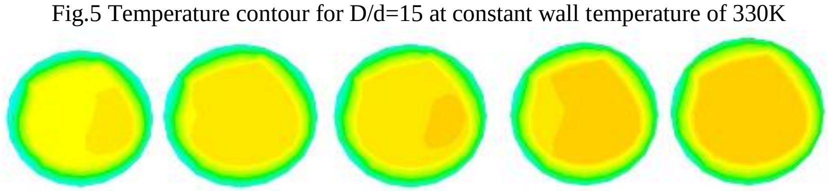
Re = 12000 Re = 15000 Re = 18000 Re = 21000 Re = 24000

Fig.4 Temperature contour for D/d=10 at constant wall temperature of 330K



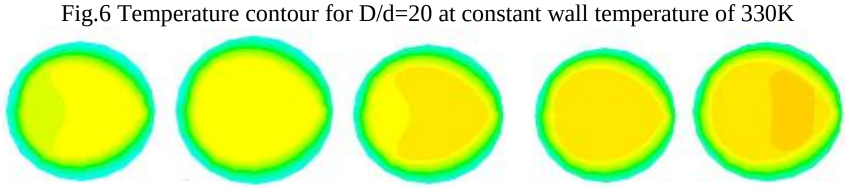
Re = 12000 Re = 15000 Re = 18000 Re = 21000 Re = 24000

Fig.5 Temperature contour for D/d=15 at constant wall temperature of 330K



Re = 12000 Re = 15000 Re = 18000 Re = 21000 Re = 24000

Fig.6 Temperature contour for D/d=20 at constant wall temperature of 330K



Re = 12000 Re = 15000 Re = 18000 Re = 21000 Re = 24000

Fig. 7: Temperature contour for D/d=25 at constant wall temperature of 330K

In the above Fig. temperature contour of outlet of hot fluid for different Reynolds number for constant outer wall temperature of 330K is shown. In Fig 4 temperature contour of outlet of hot fluid for D/d=10 shown, where we conclude that with increase in velocity of flow or the Reynolds number, mean temperature at outlet increases. This is because; with increase in Reynolds number velocity of flow increase and with increase in velocity of flow time available for heat transfer between two fluid decreases. We have the flow rate of cold fluid in all case is same but the flow rate of hot fluid increases, so the hot fluid flow past the inner tube with high velocity and not find enough time to transfer heat to the cold fluid. Fig 5 shows the temperature contour of outlet of hot fluid for D/d=15. By comparing the fig 4 and 5. Also find that for same Re (for example Re=12000), outlet temperature for D/d=15 is less than outlet temperature for D/d=10. This is due to the reason that with increase in D/d ratio the length of the tube of the exchanger increases, which increases the surface area of contact of heat exchanger. With increase in area of contact the heat transfer rate between two fluid increases. So the outlet temperature of the hot fluid decreases with increases in D/d ratio.

Below Fig shows the Nusselt number variation with respect to Reynolds number for different curvature ratio (d/D ratio). It is obvious that with increase in Reynolds number Nusselt number increases. With increases in Reynolds number the flow become more turbulent and mixing of the fluid between two layers occurs more rapidly which will enhance the heat transfer rate. For a particular value of Reynolds number (for example Re=18000or 21000 etc) with decreases in curvature ratio Nusselt number decreases. With decreases in curvature ratio the secondary forces which will act on the fluid element due to flow inside helical tube will decreases.

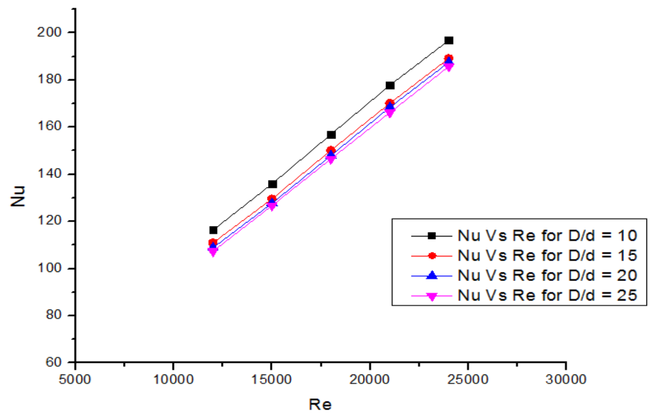


Fig. 8: variation of Nu with Re for different D/d ratio

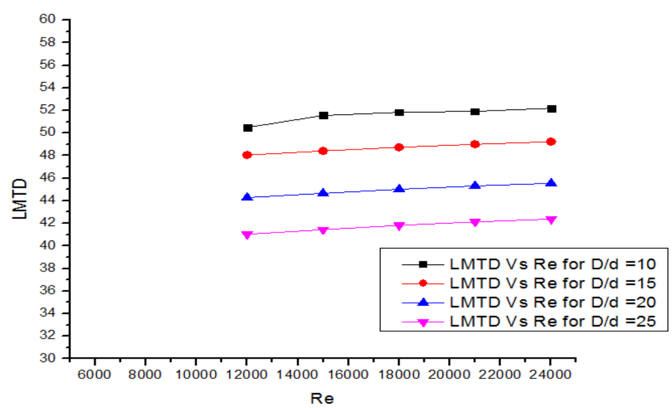


Fig. 9: variation of LMTD with Re for different D/d ratio

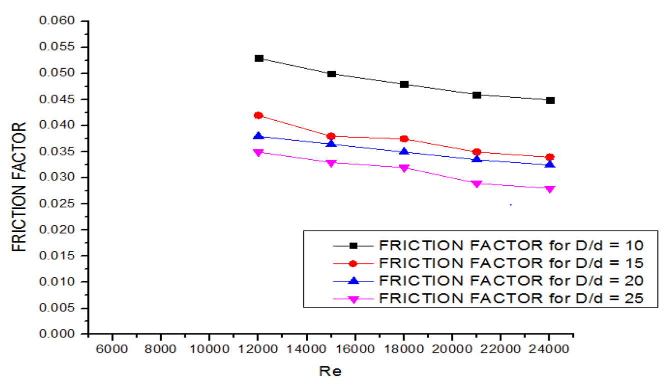


Fig. 10: variation of friction factor (f) with respect to Re for different D/d ratio

Due to decreases in secondary forces the turbulent mixing of fluid will also decreases which will reduces the heat transfer rate or the Nusselt number. As the curvature ratio decreases the variation of Nusselt number for a particular value of Reynolds number is less or insignificant; which can be shown in Fig.8

Fig.9 shows the variation between Log mean temperature differences with respect to Reynolds number for different D/d ratio. With increases in Reynolds number the LMTD increases. We know that larger the LMTD; lager will be the temperature driving force, which will increases the heat transfer. For D/d=10 has maximum LMTD for all case of Reynolds number. The increases in value of log mean temperature difference with respect to Reynolds number is due to less time of contact with in the heat exchanger.

Fig.10 shows the variation of friction factor with respect to Reynolds number for different D/d ratio. The friction factor will decreases with increases in Reynolds number, which is quite obvious. The variation of friction factor for a particular Reynolds number is maximum between D/d=10 to D/d=15. As we increase the curvature ratio the variation will decreases.

V. CONCLUSION

Numerical simulation has been carried out for tube in tube helical coil heat exchanger subjected to different boundary conditions. Nusselt numbers, Darcy friction factor, Log mean temperature difference variation with respect to Reynolds number for different D/d ratio is plotted. In practical application different boundary conditions imposed on the outer wall of exchangers are constant heat flux conditions in power plant boiler, condenser and evaporator etc. insulated outer wall condition in general case of exchanger used in laboratory and educational institutions, and convective heat transfer condition in food, automobile and process industries. A heat transfer behavior for constant outer wall boundary conditions is predicted. Following are the outcome of above numerical study;

- With increase in the Reynolds number, the Nusselt number for the inner tube increase. However, with increases in flow rate turbulence between the fluid element increases which will enhance the mixing of the fluid and ultimately the Nusselt number or the heat transfer rate increases.
- 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.
- Friction factor decreases with increase in Reynolds number due to relative roughness of surface, and velocity of flowing fluid.
- Log mean temperature difference increases at a steady rate with increase in Reynolds number.

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