"Examination of Helical Coil Heat Exchanger at continuous heat flux on outer wall"

[1]Manish Kuvadiya
[1]Assistance Professor,
[1]Department of Mechanical Engineering,
[1]Parul Polytechnic Institute, Vadodara, India

Abstract:* Functioning to the aim of corresponding energies and to create compact the proposal on behalf of automatic and organic equipment and industrial plant, the improvement of heat transfer is unique vital elements in proposal of heat exchangers. In design procedure deprived of knowing the use of energy we can increase the heat transfer amount by adapting the plan by providing the helically pipes, protracted outward or eddy stream equipment. tube in tube helical coil heat exchanger finds applications for automobile area, aero-space area, energy producing area and diet productions owing assured benefit such as compressed design, more heat transfer superficial zone and enhanced heat transfer ability. Here in the project mathematical studies of tube-in-tube helically loop heat exchanger on behalf of determine heat transfer at changed D/d relation. The counter stream heat exchanger through turbulent stream model is measured for study perseverance. The outcome of different D/d ratio on heat transferal amount is determined aimed at different boundary situations. The D/d relation is various since 10 to 25 through an intermission of 5. Nu, f, LMTD difference of hot liquid through Reynolds number is finding out on behalf of dissimilar D/d relation.

Index Terms - **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 among fluids; a hard surface and solid particulates, at changed heats and thermal connection for any exchanger, here commonly no connection between heat and work . Usual uses consist of heating as well as fluid cooling and vaporization or reduction of particular or multi element of fluid flow. For some application like recover and reject heat, sterilize, sterilize, fractionate, and condense, distillate, manifest. For some exchanger, the fluids heat transfer through straight interaction. But in other heat exchangers, heat transfer between fluids through in a transient mode, also in various heat exchangers, the liquids are divided by a heat transfer surface .Such heat exchangers are discussed to undeviating transmission kind. In compare, exchangers in which there is discontinuous heat exchanger among the hot fluid and cold fluid through heat energy storing and discharge over the exchanger apparent or medium are discussed to as unintended transfer type, or basically regenerators, such exchanger ordinarily have unsolidified leak from one fluid to the further, by the difference of pressure and atmosphere variation/regulator interchanging. Mutual cases of exchanger are shell and tube exchangers, radiator of automobile, condenser, evaporator and air preheater. When no phase variation arises in exchanger, it is occasionally mentioned to as a practical heat exchanger. Here inner heat energy causes in the heat exchangers, such as in electrical warmers and atomic energy features. Burning and biochemical response might take place inside the exchanger, like in reservoirs, enthusiastic heaters, and fluidized-bed type heat exchangers. Automated devices may be used in certain exchangers such as in rubbed surface exchangers, disconcerted vessel, and motivated tank devices. However, in a heat tube heat exchanger, the heat tube not individual performances as a splitting wall, but also helps for the of heat transfer by condensation, vaporization, and conveyance of the occupied fluid inside the heat pipe in general, When the fluids are immiscible, the splitting partition can be removed, and the boundary among the fluids substitutes a heat transfer superficial in a direct-contact heat exchanger. Heat exchanger contains of heat transmission element such as a basic or atmosphere contracting the heat transfer apparent, and unsolidified delivery component like heaters, reservoirs and jets or tubes. Typically, here no one is stirring quantities in a heat exchanger silent; here omissions, like a rotational renewing exchanger or a rubbed superficial heat exchanger.

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 TiO2 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/Al2O3 (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.

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.

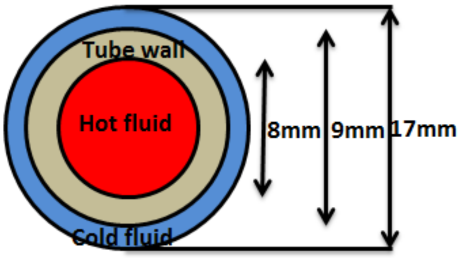


Fig.1 Front view of tube in tube helical coil heat exchanger

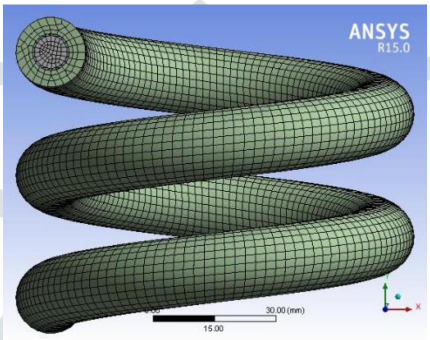


Fig.2 meshing for tube in tube helical coil heat exchanger

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 no 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

Boundary conditions

The outer wall of the heat exchanger has taken constant wall heat flux condition 60000/m^2, 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

All of the equations can be described as follows.

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.

1. **Continuity equation:**

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

2. Navier Stokes equation:

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

3. **Energy equation**:

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

Where;

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

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

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

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

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

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

Local Nusselt number is given by;

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

Or it can also be represented by following equation

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

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

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

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

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

Length of pipe is given by following relation;

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

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

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

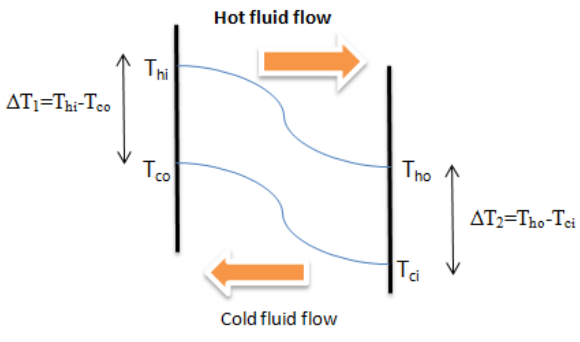


Fig.3 Log mean temperature difference of hot and cold fluid

Where $\Delta T1 = Thi - Tco$

And $\Delta T2 = Tho - Tci$

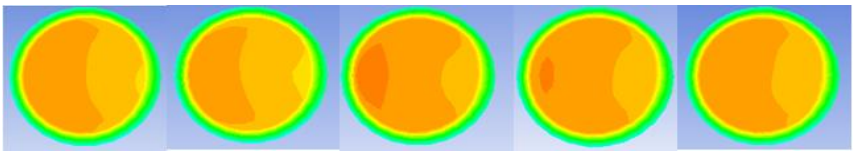
To find the outlet fluid temperature we can use the energy balance equation,

$$Q = m\,Cp\,(T1 - T2) = h\,As\,(Tw - Tf) \quad \text{…… (13)}$$

IV. Results and Discussion

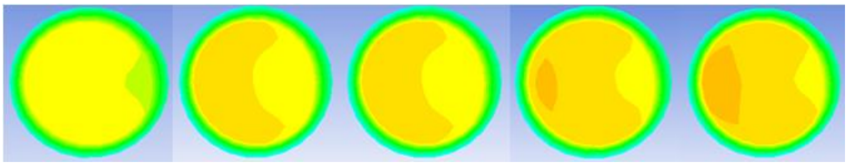
Constant Outer Wall 60000W/m^2 **Boundary condition**

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.WALL



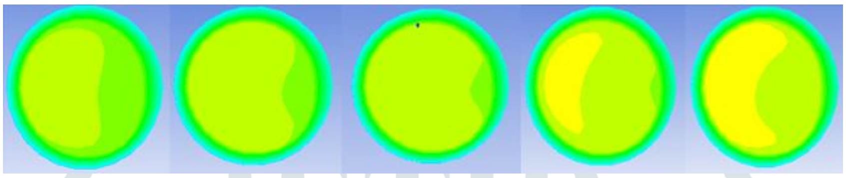
Re = 12000 Re = 15000 Re = 18000 Re = 21000 Re = 24000

Fig.4 Temperature contour for D/d=10 at Constant Outer Wall 60000W/m^2



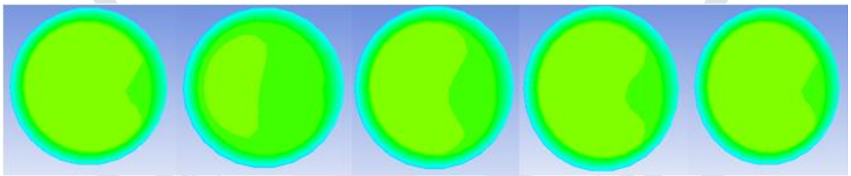
Re = 12000 Re = 15000 Re = 18000 Re = 21000 Re = 24000

Fig.5 Temperature contour for D/d=15 at Constant Outer Wall 60000W/m^2



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

Fig.6 Temperature contour for D/d=20 at Constant Outer Wall 60000W/m^2

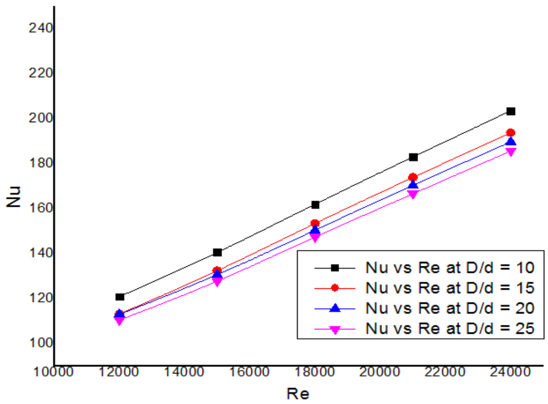


Re = 12000 Re = 15000 Re = 18000 Re = 21000 Re = 24000

Fig.7 Temperature contour for D/d=25 at Constant Outer Wall 60000W/m^2

In the above Fig. temperature contour of outlet of hot fluid for different Reynolds number for Constant Outer Wall 60000W/m^{2}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.



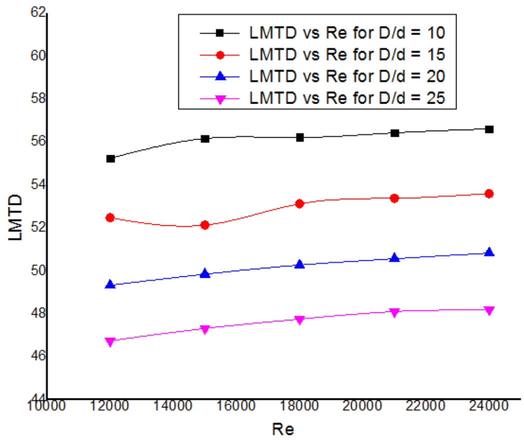


Fig.8 variation of Nu with Re for different D/d ratio

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

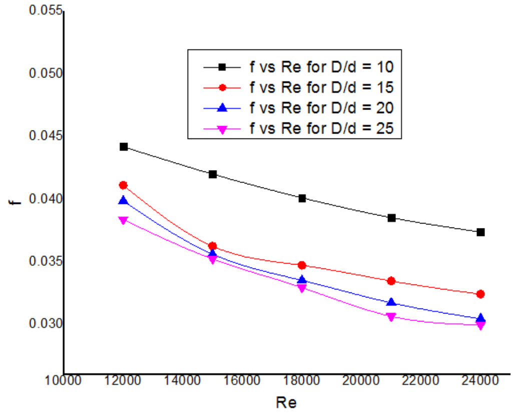


Fig.10 variation of friction factor (f) with respect to Re for different D/d ratio

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

Mathematical imitation has been sanctioned for TUBE IN TUBE helically coil heat exchanger exposed to dissimilar borderline situations. Nu numbers, Darcy friction factor, LMTD variant through Re on behalf of dissimilar D/d relation is strategized. In actual use dissimilar borderline situations prescribed on the outside wall of exchangers are continuous heat flux situations in any type of power plant, modern boiler, condenser and evaporator etc. insulated outside wall situation in general situation of heat exchanger recycled in workshop and instructive organizations, and convective heat transmission state in foodstuff, auto parts and procedure of productions. Subsequent are the conclusion of overhead mathematical learning;

- Through rise in the Re, the Nu on behalf of the internal pipe rise. Conversely, by rises in mass stream rate commotion among the fluid component rises which will raise the contribution of the fluid and finally the Nu or the heat transmission amount rises.
- Through rises in D/d relation the Nu will reductions; on behalf of a specific rate of Re. Nu has supreme rate on behalf of D/d = 10.
- The external wall boundary situation does not have any main consequence on the internal Nu, which can be established since the outcomes.
- Roughness (friction) factor reductions with rise in Re owing to relative roughness of superficial, and velocity of graceful liquid.
- Through rise in Re at a steady rate log mean temperature difference also increases.
- The heat removal is disturbed since the warm fluid for every boundary situation can be expected on behalf of outside wall of outside pipe since it does not affect significantly the heat transfer amount.

Future scope:
More precise technique must be established to conclude the extreme situation. Beside through mathematical investigation experimental validation must be compulsory. For this problem particular phase stream outline is measured however in upcoming dual or multiphase stream might be deliberated. The optimization condition should be developed for different boundary condition.

VI. REFERENCES

[1] Naphon Paisarn., Wongwises Somchai., A study of the heat transfer characteristics of a compact spiral coil heat exchanger under wet-surface conditions, Experimental Thermal and Fluid Science, vol.-29 (2005) 511–521
[2] Kumar V., Saini S., Sharma M., Nigam K.D.P., Pressure drop and heat transfer in tube in tube helical heat exchanger, Chemical Engineering Science, vol.-61 (2006): 4403–4416.
[3] Jayakumar J.S, Mahajani S.M., Mandal J.C., Iyer Kannan N., Vijayan P.K., CFD analysis of single-phase flows inside helically coiled tubes, Computers and Chemical Engineering, vol.- 34 (2010) 430–446.

[4] Paisarn Naphon., Study on the heat transfer and flow characteristics in a spiral-coil tube, International Communications in Heat and Mass Transfer, vol.-38 (2011) 69–74.

[5] Ghorbani Nasser., Taherian Hessam., Gorji Mofid., Mirgolbabaei Hessam., An experimental study of thermal performance of shell-and-coil heat exchangers, International Communications in Heat and Mass Transfer, vol.-37 (2010) 775–781.

[6] Yang Zhen., Zhao Zhenxing., Liu Yinhe., Chang Yongqiang., Cao Zidong., Convective heat transfer characteristics of high-pressure gas in heat exchanger with membrane helical coils and membrane serpentine tubes, Experimental Thermal and Fluid Science, vol.-35 (2011) 1427–1434.

[7] Genic Srbislav B., Jacimovic Branislav M., Jaric Marko S., Budimir Nikola J., Dobrnjac Mirko M., Research on the shell-side thermal performances of heat exchangers with helical tube coils, International Journal of Heat and Mass Transfer, vol.-55 (2012) 4295–4300.

[8] Jamshidi N., Farhadi M., Ganji D.D., Sedighi K., Experimental analysis of heat transfer enhancement in shell and helical tube heat exchangers, Applied Thermal Engineering, vol.-51 (2013) 644-652.

[9] Huminic Gabriela., Huminic Angel., Heat transfer characteristics in double tube helical heat exchangers using nanofluids, International Journal of Heat and Mass Transfer, vol.-54 (2011) 4280–4287.

[10] Ferng Y.M., Lin W.C., Chieng C.C., Numerically investigated effects of different Dean number and pitch size on flow and heat transfer characteristics in a helically coil-tube heat exchanger, Applied Thermal Engineering, vol.-36 (2012) 378-385.

[11] JamshidiNaghmeh., FarhadiMousa., SedighiKurosh.,GanjiDavood Domeiry. Optimization of design parameters for nanofluids flowing inside helical coils, International Communications in Heat and Mass Transfer, vol.-39 (2012) 311–317.

[12] San Jung-Yang., Hsu Chih-Hsiang., Chen Shih-Hao., Heat transfer characteristics of a helical heat exchanger, Applied Thermal Engineering, vol.-39 (2012) 114-120.