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COMPUTATIONAL ANALYSIS TO MAXIMIZE THE HEAT TRANSFER RATE OF DOUBLE TUBE HELICAL COIL HEAT EXCHANGER

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***Abstract*: 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 paper Computational study of helical coil tube-in-tube heat exchanger is done for different boundary conditions and maximizes condition of heat transfer is found out for 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 and pumping power is found out for different boundary conditions. The D/d ratio is varied from 10 to 30 with an interval of 5. Nusselt number, friction factor, pumping power required and LMTD variation of inner fluid with respect to Reynolds number is found out for different D/d ratio. The maximize Reynolds number for maximum heat transfer and minimum power loss is found out by graph intersection methods. From the results complicated behaviour of fluid flow is captured for both the fluids flowing inside the tube**.

I. INTRODUCTION

Heat exchanger is a device which is used to transfer heat between two fluids which may be in direct contact or may flow separately in two tubes or channels. Recent developments in design of heat exchangers to full fill the demand of industries has led to the evolution of helical coil heat exchanger as helical coil has many advantages over a straight tube. Main aim of our project is to maximize the heat transfer rate with minimum power loss.

The helical coil-tube heat exchangers are used in industries and power sectors due to its compact structural design, larger heat transfer surface area and higher heat transfer capability. Experimental work has been done on flow pattern and heat transfer characteristic of Helical coil Heat Exchanger, see [3]. In spite of Computational and analytical studies that have been done in helical coil tube. Main aim of our project is to maximize the heat transfer rate with minimum power loss. We know that with increase in Reynolds number Nusselt number increases hence the heat transfer coefficient. But with increase in Reynolds number pumping power also increases but increase in power requirement is more compared to increases in heat transfer coefficient, more specification see [2]. So there exist a particular Reynolds number or (Dean Number) for which both the curve intersects, which is the optimum point for that particular (D/d) condition. Similarly this optimum condition can also be plotted between the variation of Nusselt number and friction factor with respect to Reynolds number or Dean Number. In my project it is plotted between (Nu) & (friction factor) with respect to (Re) Also in my project I have shown the variation of Nusselt Number, friction factor, pumping power and Log mean temperature difference with respect to Reynolds number for different D/d ratio and predict the behaviour of heat transfer for varying coil diameter. The effect of different boundary condition on thermal properties of inner fluid has also studied.

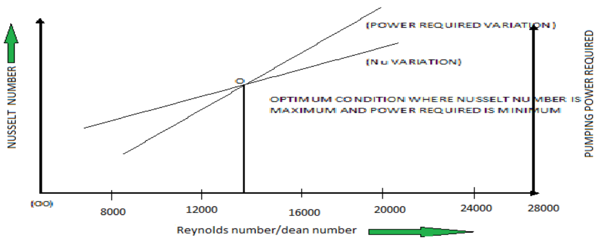

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Fig.1 (a) Optimum condition showing variation between Nu and Pumping power with Respect to Re/De;

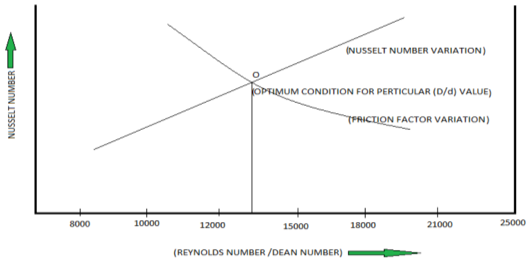
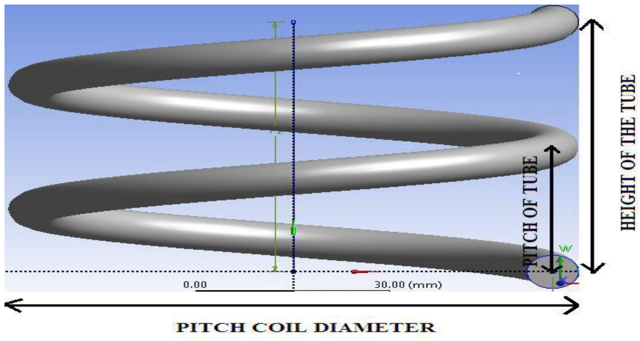


Fig.1 (b) graph between Nu and f with respect to Re/De

II. PROBLEM FORMULATION

In my work I maximize the given helical coil heat exchanger keeping in mind that it should produce maximum heat transfer rate with minimum power consummation. Because some times in the process of improving the heat transfer coefficient we consume more power without knowing the economical cost. In my study I consider [1] the double tube helical coil heat exchanger or double tube helical coil heat exchanger with two (2) numbers of turns. For simplification in Statistical analysis I consider only two turns 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. As the coil diameter increases the length of the exchanger (L) also increases. The inner tube diameter (d_1) was 8mm. the thickness (t) of the tube was taken 0.5mm. The outer tube diameter (d_2) was taken 17mm. In my study I fixed the tube diameter (both inner and outer diameter) of the heat exchanger and vary the coil diameter of the tube 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. After creating the geometry and doing the meshing in ANSYS 13 the problem was analysed in ANSYS 13 (FLUENT) for different boundary conditions as specified later. For analysis of the problem turbulent fluid flow condition was considered.



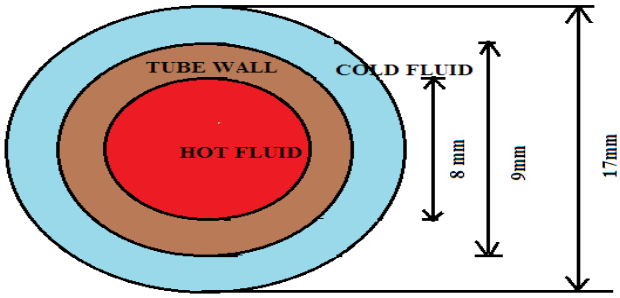


Fig.2 Front view of helical coil heat exchanger showing different fluid flowing and geometrical parameters with dimensions

In this study I considered the counter flow heat exchanger as it has better heat transfer rate compared to parallel heat exchanger. The cold fluid and the hot fluid flow in opposite directions in their respective tube. In this study for analysis, turbulent fluid flow was considered, from Aly Wael I.A [1], [2].

Boundary conditions:

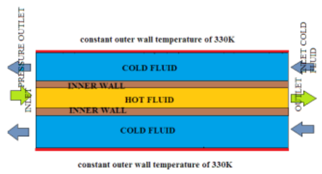
For outer wall following four conditions was taken;

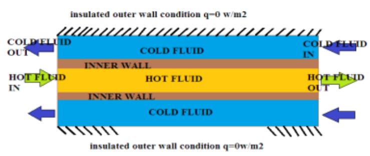
Case **1**: The outer wall of the heat exchanger has taken constant wall temperature of 330K, as shown in Fig. 3 (b). It can be expressed computationally by; at d_3=17mm; T=330K

Case 2: The outer wall of heat exchanger has taken constant heat flux of 60000 W/m^2, as shown in Fig. 3 (a). That is at d_3=17mm; q=60000W/$m^{2.}$

Case 3: Insulated outer wall was taken in the next condition, as given in the Fig. 3 (c). At d_3=17mm; q=0W/m^2.

Case 4: constant heat transfer coefficient of 4000W/m^2K was taken for outside atmosphere, as shown in Fig.3 (d). That is at d_3=17mm; h=4000W/m^2K.





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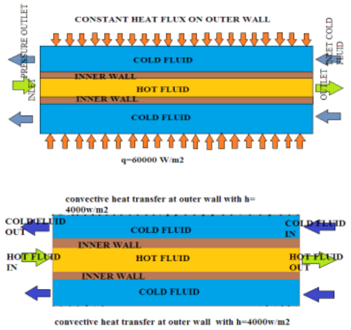


Fig.3 (a) Constant wall heat flux condition, (b) constant wall temperature boundary condition, (c) insulated outer wall condition, (d) convective heat transfer coefficient condition at outer wall.

The fluid properties of the working fluid (water) were assumed to be constant throughout the analysis with respect to temperature with the properties. The tube of the heat exchanger was made up of copper for maximize the heat transfer, because copper has good thermal conductivity. Also the properties of the copper were also remains constant throughout the analysis.

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

Log Mean Temperature Difference for counter flow heat exchanger can be taken from the figure as shown below.

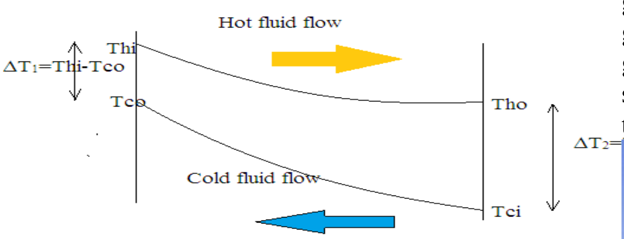


Fig4 Log mean temperature difference of hot and cold fluid

III. COMPUTATIONAL FLUID DYNAMICS (CFD)

The invention of high speed computers, combined with the accurate Computational methods for solving physical problems, has revolutionized the way we study and practice fluid dynamics and heat transfer problems, Brief about CFD [6]. This is called Computational Fluid Dynamics (CFD). CFD may thus be regarded as a zone of study combining fluid dynamics and Computational analysis.

All the CFD software contain three basic elements

1. Pre processor
2. Main Solver
3. Post processor

This step is followed by generations of the mesh structure, which is the most important portion of the pre-processing activity. Both computation time and accuracy of solution depend on the mesh structure. The solver is the heart and mind of CFD software. It sets up the equations which are selected according to the options chosen by the analyst and grid points generated by the pre-processor, and solves them to compute the flow field. The post-processor is the final part of CFD software. It helps the user to analyse the results and get useful data and draw the conclusion [5]

CFD procedure

For computational analysis in CFD following five stages are required

1. **Geometry creation**

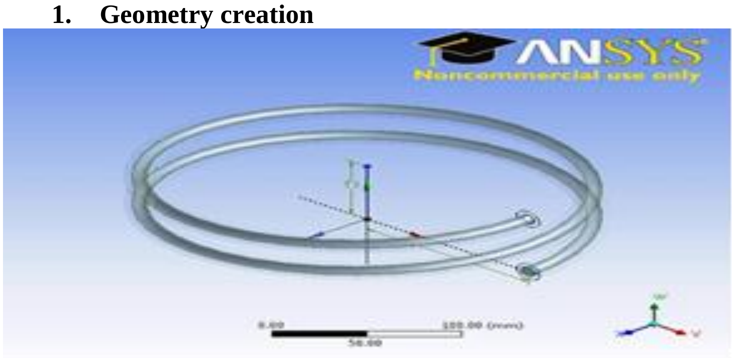


Fig 5 showing the geometry of double tube helical coil heat exchanger created in ANSYS 13 work bench

II Grid generation

In grid generation first go to the MESH option located in FLUENT tree then press the GENERATE MESH button. It will create automatic grid. Then we have to modify the grid or make the grid finer so that accurate results will come. For generating fine mesh go for the sizing option then select the EDGE for making the division. Then specify the number of divisions.

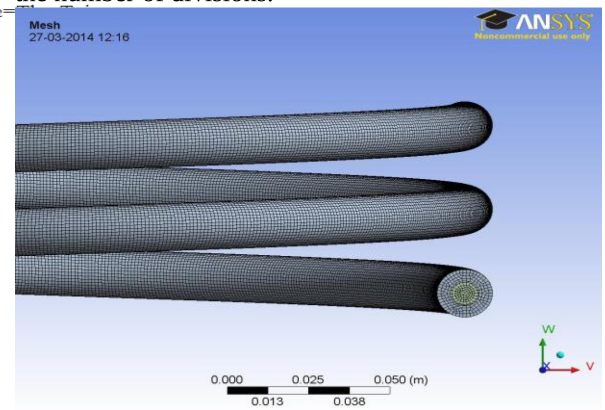


Fig 6 Grid generation for the double tube helical coil heat exchanger

For generating the uniform mesh we have to MAP the face, for that we have to select the MAPPED FACE option and select the face for which we have to do the operation. This will create uniform mesh throughout the face of the geometry. Similarly in MESHING METHOD we can specify the type of mesh we want to create for example QUAD, TRI, and QUAD/TRI etc. After that we have to name each face of the geometry. For that right click on the face and go to CREATE NAME SELECTION then name each face (for example inlet, outlet, wall etc). Then update the project.

Grid independence test

Grid independence test is the one of the most important test which should be done in computational analysis of a problem. Grid independence test was done to check the final results should be independent of the number of grids. In computational problem the results are always dependent on the number of grids generated solutions based on [10]. So if we change the number of grid the results were changed. While changing the number grids a stage may come when the results are independent of the number of grids. These minimum number of grids after which there is no change in results were observed was known as optimum grid size and the results were independent of grids. In this problem first the grid independence test were carried out for different D/d ratio. Starting from D/d=10; considering outer wall insulated condition and keeping the inlet velocity of hot fluid at 1.5072m/sec (Re=10000) and temperature at 355K the grid size was varied from 64770 to 138775.

In below two graphs outlet fluid temperatures were taken in Y axis and number of grids in X axis. The grids divisions were increased and corresponding temperature of hot fluid in Fig.7 (a) and outlet of cold fluid temperature in Fig.7 (b) were shown. After 138700 numbers of divisions the results not depend on grids hence result is grid independent.

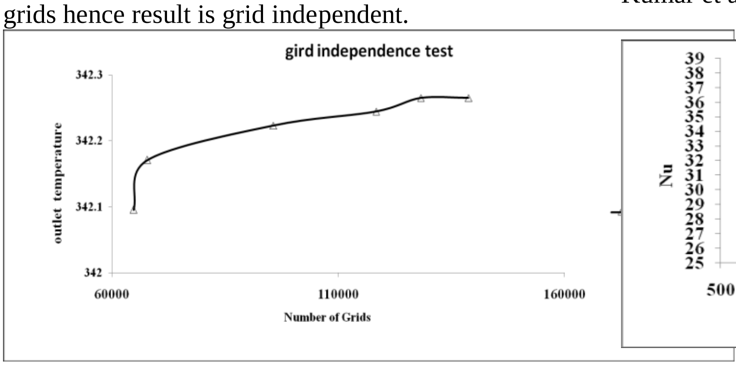


Fig 7 (a)Grid independence test based on the outlet of hot fluid temperature

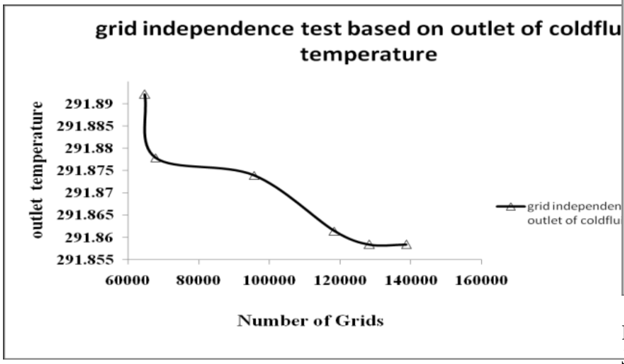


Fig 7 (b) Grid independence test based on outlet of cold fluid temperature

Similarly for other D/d ratio the grid independence test has been done on the basis of outlet temperature of hot fluid and cold fluid and optimum grids were chosen for further COMPUTATIONAL analysis.

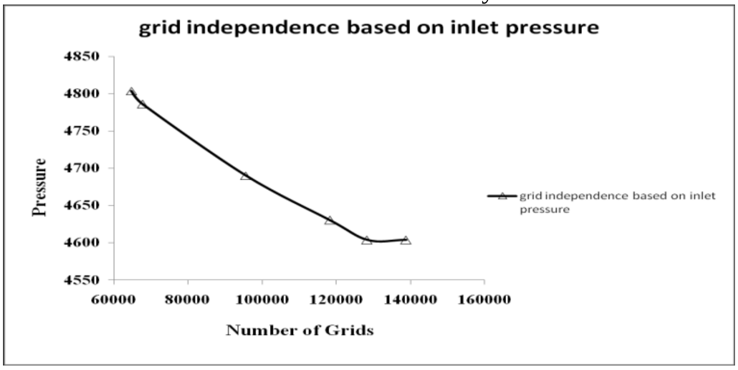


Fig 7 (c) Grid independence test based on the pressure of hot fluid at inlet

IV RESULTS

The inner Nusselt number was calculated and compared with the Nusselt number predicted by Kumar et al. (2006) and it is found that the calculated results match with the predicted results with reasonable accuracy.

Here in the fig.8 (a) Nusselt number is taken in Y axis and Dean number which is a function of Reynolds number and curvature ratio is considered on X axis. With increase in Dean Number the Nusselt number increases. In both the cases of calculated results matches fair accurately with the results given by Kumar et al. (2006).

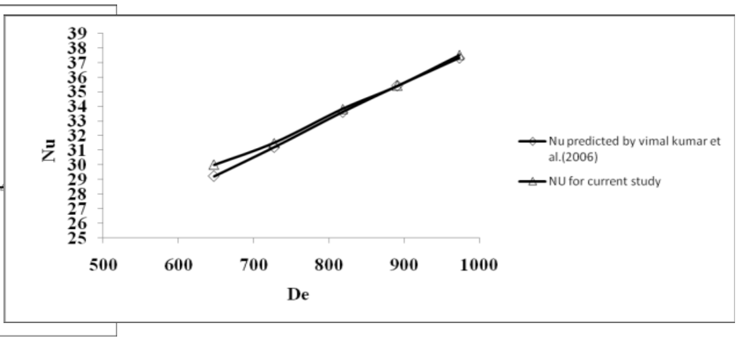


Fig 8(a) Variation of Nu with respect to De predicted by Vimal Kumar et al. (2006)

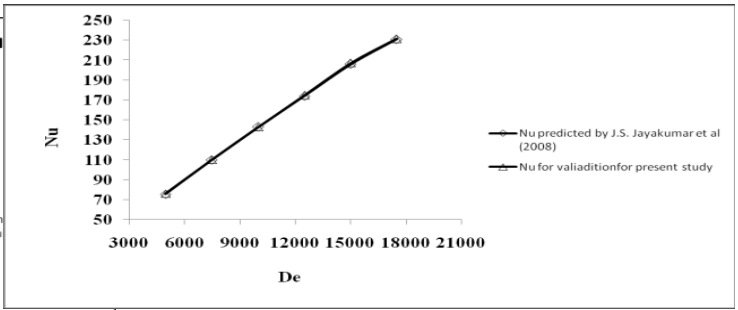
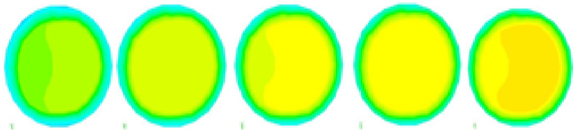


Fig 8 (b) Variation of Nu with respect to De predicted by J.S. Jayakumar et al. (2008).

Constant outer wall temperature

For constant outer wall condition the temperature contour of hot fluid outlet is shown in below fig.9. The inner Nusselt number, friction factor, pumping power are calculated subsequently with respect to Reynolds number.



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

Fig 9 Temperature contour for D/d=10 to 30 at constant wall temperature of 330K

In the above Fig 9 temperature contour of outlet of hot fluid for different Reynolds number for constant outer wall temperature of **330K** is shown. In above Fig 9 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, avail in[6]. 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. we 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. It is shown in the Fig. When the D/d ratio increases from 25 to 30, there is a large decrease in outlet temperature which can be conformed from above fig.[6],[10].

Fig 10 (a) shows the variation of Nusselt number with respect to D/d ratio (inverse of curvature ratio) for different Reynolds number. As in Fig.10 (a) it is explained that with increases in D/d ratio the Nusslet number will decreases. From the graph it is clear that Nusselt number will decreases faster rate when D/d changes from 10 to 15, then between D/d ratios 15 to 20 the change is less. For Re=12000 the percentage change in Nu from D/d ratio 10 to 25 is 7.58% while that for Re=25000, it is 7.8%.

Fig 10 (b) shows the variation of pumping power with respect to D/d ratio (inverse of curvature ratio) for different Reynolds number. It can easily understood that with increase in D/d ratio pumping power increase and pumping power is maximum for Reynolds number=25000.

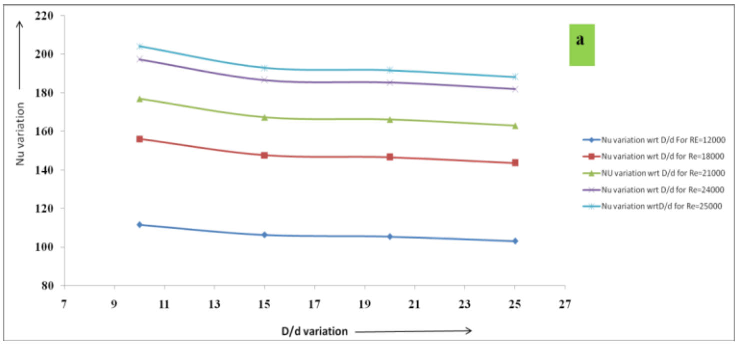


Fig 10 (a) variation of Nu with respect to D/d

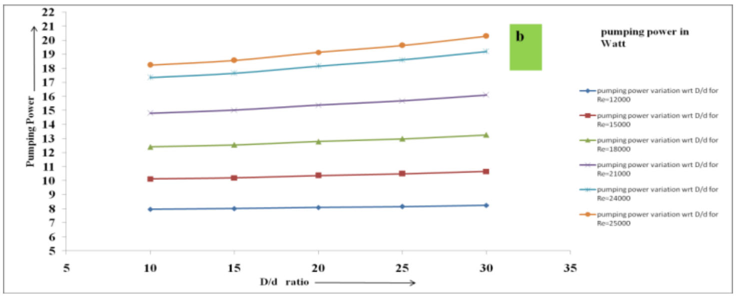


Fig 10 (b) variation of pumping power with respect to D/d

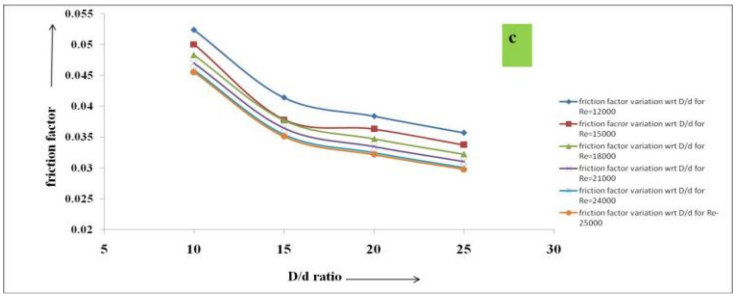


Fig 10 (c) variation of friction factor[f] with respect to Re for different Reynolds number.

With increases in D/d ratio the length of pipe increases and the pressure loss increase which will increase the pumping power requirement. Fig10 (c) shows the variation between Darcy friction factor with respect to D/d ratio for different Reynolds number. For a particular value of Reynolds number friction factor decreases with decreases in curvature ratio (d/D). For Re=12000 friction factor is maximum and decrease with increase in Re. The change in friction factor is insignificant after Re=24000 for particular D/d ratio which can be understood from the fig 10 (c). The D/d ratio varying from 10 to 25 percentage decrease in friction factor is 31.84 % for Re=12000.

Insulated outer wall condition:

This is the general case of heat exchanger where the main aim is to transfer the heat between two fluids which are maintained at different temperature. We find applications of this type of heat exchangers in air conditioning plant, food storage, auto mobile industries and research lab.

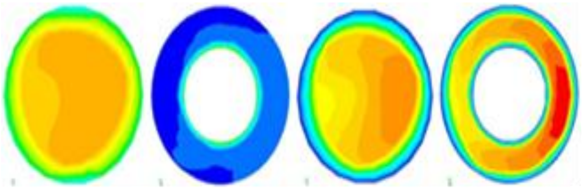


Fig 11 (a) outlet temperature contour of hot fluid (b) outlet temperature contour of cold fluid (c) velocity contour of hot fluid outlet (d) velocity contour of cold fluid outlet

In the above fig11 shows the temperature and velocity contour of hot fluid outlet and cold fluid outlet for D/d=15 at Reynolds number 24000. The Fig.11 (a) shows the temperature contour of hot fluid at outlet. The temperature decreases from center towards wall of inner tube. For outer tube (fig.11(b)), it is maximum at the center of annulus and decreases toward the outer wall. Fig.11(c-d) shows the velocity contour of hot and cold fluid and the velocity is minimum at the wall and maximum at the centre.

Fig. 12 (a) shows the variation of Nusselt number with respect to Reynolds number for different D/d ratio. In this Fig. 13 (a), Nu is taken in Y-axis and Re in X axis keeping D/d ratio constant. Then the D/d ratio varied from 10 to 30 with an interval of 5 and Nusselt number variation with respect to Reynolds number was drawn in the figure. With increases in Reynolds number the Nusselt number increases because with increases in flow rates the turbulence between the fluid elements increases. Due to flow past the helical tube there is strong a centrifugal force acting on fluid element which will enhance the heat transfer rate.[9],[10].

Fig. 12 (b) shows the variation of pumping power (Watt) with respect to Reynolds number for different D/d ratio (inverse of curvature ratio). Pumping Power is taken in Y-axis and Reynolds number in X-axis for a constant D/d ratio. Then the D/d ratio varies to get series of lines showing the variation of power requirement. The change in pumping power is due to the pressure drop and the change in the pressure drop with Reynolds number is due to the increasing contraction and expansion pressure losses at the inlet and outlet portion of the helical coil heat exchanger from the reference in [7].

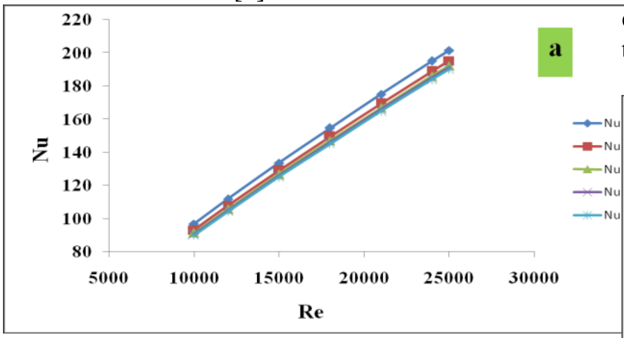


Fig 12 (a) variation of Nu with respect to Re

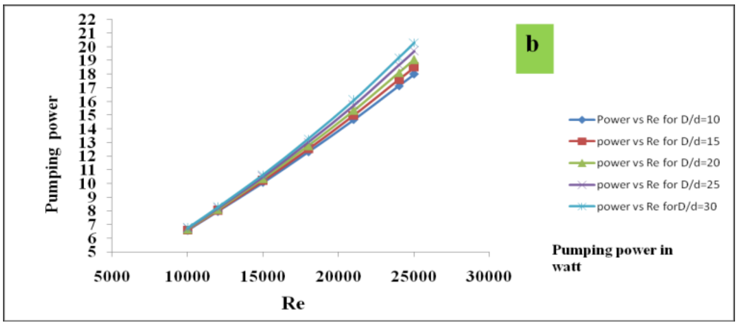


Fig 12 (b) variation of pumping power with respect to Re

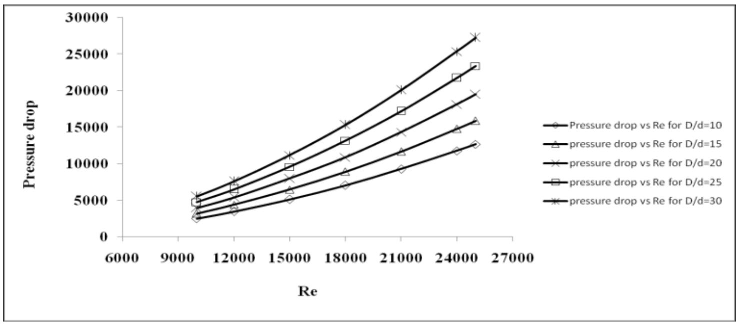


Fig 13 Pressure drop variation with respect to Re for insulated condition

In above Fig. 13 pressure drop variation with respect to Reynolds number is shown for outer wall insulated condition. The pressure drop variation is in Pascal unit. With increases in Reynolds number the pressure drop increases. When the flow is more turbulent more amount of energy is lost due to conversion pressure energy to kinetic energy. With increases in D/d ratio the pressure loss increases, because with increase in D/d ratio the length of the pipe increases and with increase in length of pipe the loss is more.

Fig.14 (a) shows the variation of Nusselt number with respect to D/d ratio for different Reynolds number. As in Fig.14 (a) it is explained that with increases in D/d ratio the Nusslet number will decreases. From the graph it is clear that Nusselt number will decreases when D/d changes from 10 to 15, then for 15 to 20 the change is less and it gradually decreases with further increase in D/d ratio because turbulence due to curvature effect disappear and flow through coiled tube changed to flow through a straight tube.[8]

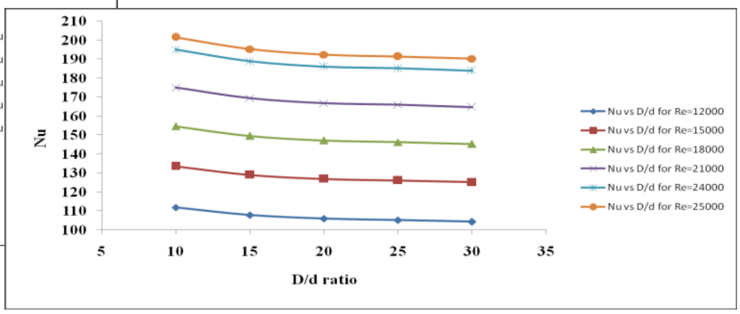


Fig 14 (a) variation of Nu with respect to D/d ratio for different Re

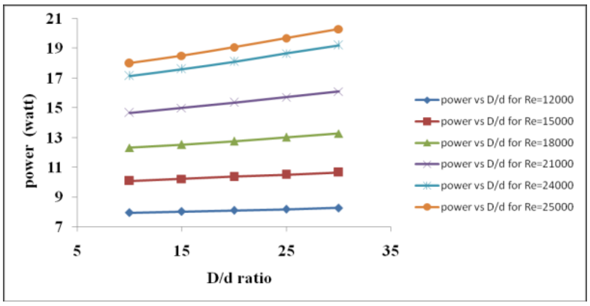


Fig 14 b) variation of pumping power with respect to D/d ratio for different Re

Fig. 14 (b) shows the variation of pumping power with respect to D/d ratio for different Reynolds number. It can easily understood that with increase in D/d ratio pumping power increase and pumping power is maximum for Reynolds number=25000. With increases in D/d ratio the length of pipe increases and the pressure loss increase which will increase the pumping power requirement. Also with increase in Reynolds number the turbulent flow will increase which will increases the pressure loss and ultimately the power requirement. Fig 14 (c) shows the variation between Darcy friction factor with respect to D/d ratio for different Reynolds number at insulated outer wall condition. For a particular value of Reynolds number friction factor decreases with increase in D/d ratio. For Re=12000 friction factor is maximum and decrease with increase in Re. The decrease in friction factor is more for D/d ratio varying from 10 to 15 and it decreases with further increase in D/d ratio.[6].

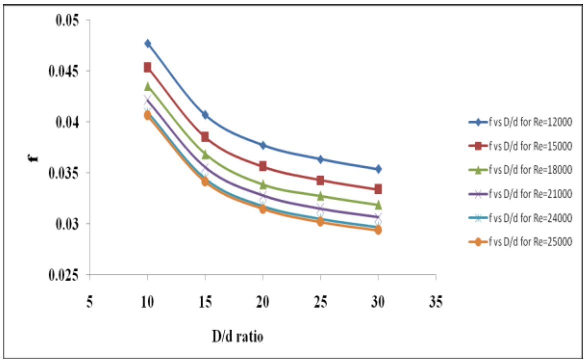


Fig 14 c) variation of friction factor[f] with respect to D/d ratio for different Re

Fig 14 (d) shows the variation between LMTD with respect to D/d ratio for different Reynolds number at insulated outer wall condition. For a particular value of Reynolds number LMTD decreases with increase in D/d ratio. For Re=25000 LMTD is maximum and decrease with decrease in Re. For D/d ratio varying from 10 to 30 the percentage decrease in LMTD is 18.003% for Re=12000. For Re=25000 the percentage decrease in LMTD is 14.3%.

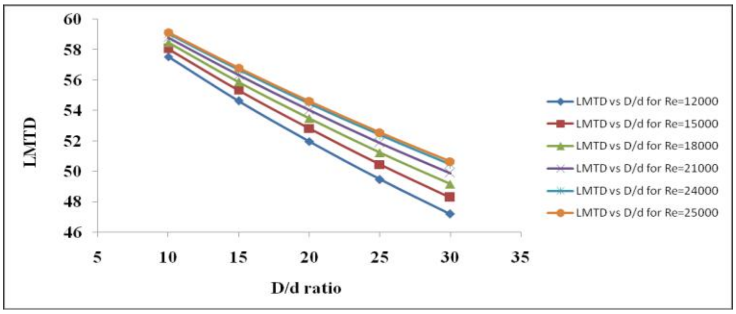


Fig 14 (d) variation of LMTD with respect to D/d ratio for different Re

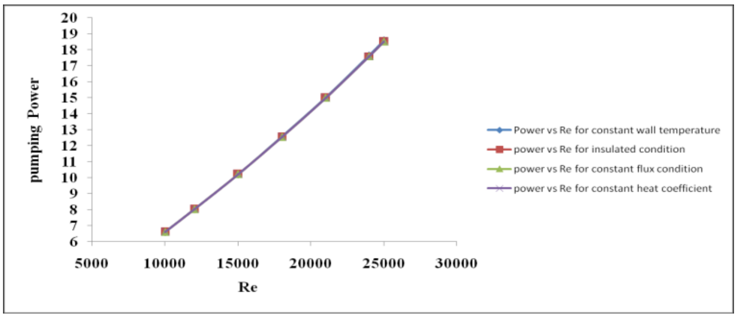


Fig 15 Variation of Pumping power with respect to Re for different boundary conditions

In Fig.15 shows the variation of pumping power of hot fluid with respect to Reynolds number for different boundary conditions.[4],[10].

Shows the optimum Reynolds number variation with respect to D/d ratio in fig 16. As the Nusselt number decreases with D/d ratio and power required increases with D/d ratio, the optimum Reynolds number decreases with increases in D/d ratio.

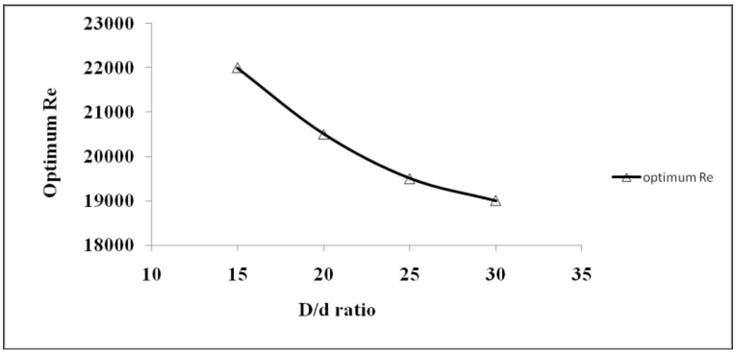


Fig 16 Optimum Reynolds number with respect to D/d ratio

V CONCLUSIONS

Statistical simulation has been carried out for Double tube helical coil heat exchanger subjected to different boundary conditions. Nusselt number, Darcy friction factor, pumping power required, Log mean temperature difference, pressure drop 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. Heat transfer behaviours for different boundary conditions are predicted and maximize condition for maximum Nusselt number (Nu) and minimum friction factor (f) was plotted against Reynolds number. Following are the outcome of above Statistical study;

1. 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.
2. Which will enhance the mixing of the fluid and ultimately the Nusselt number or the heat transfer rate increases.
3. 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.
4. The outer wall boundary condition does not have any significant effect on the inner Nusselt number, which can be confirmed from the results.
5. Friction factor decreases with increase in Reynolds number due to relative roughness of surface, and velocity of flowing fluid Pumping power increases with increase in Reynolds number for all the condition of D/d ratio and for all the boundary conditions. This is due to increase in pressure loss caused. by more turbulent flow. Pumping power is independent of the outer wall boundary conditions.
6. Log mean temperature difference increases at a steady rate with increase in Reynolds number.
7. From the optimization graph shown between Nu and f with respect to Re; the intersection point shifts toward the lower Reynolds number with increase in D/d ratio. It indicates that lower Reynolds number is required to get maximize condition for higher D/d ratio because the power requirement is more as we move towards higher D/d ratio.
8. The optimum Reynolds number decreases with increase in D/d ratio.

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