CFD ANALYSIS OF HEAT TRANSFER IN A HELICAL COIL HEAT EXCHANGER WITH CONSTANT WALL HEAT TRANSFER COEFFICIENT

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CFD ANALYSIS OF HEAT TRANSFER IN A HELICAL COIL HEAT EXCHANGER WITH CONSTANT WALL HEAT TRANSFER COEFFICIENT

A thesis submitted in partial fulfillment of the requirements for the degree of

Bachelor of technology

In

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By

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This is to certify that the work in this thesis entitled, "CFD ANALYSIS OF HEAT TRANSFER IN A HELICAL COIL HEAT EXCHANGER WITH CONSTANT WALL HEAT TRANSFER COEFFICIENT" submitted by LAXMIPRIYA SAHOO in partial fulfillment of the requirements for the degree of Bachelor of Technology in Mechanical Engineering, during session 2013-2014 is an authentic work carried out by her under my supervision and guidance.

To the best of my knowledge, the matter embodied in the project has not been submitted to any other University / Institute for the award of any Degree or Diploma.

Prof. Ashok Kumar Satapathy

Dept. of Mechanical Engineering

National Institute of Technology Rourkela

Date:

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ABSTRACT

There is a wide application of coiled heat exchanger in the field of cryogenics and other industrial applications for its enhanced heat transfer characteristics and compact structure. Lots of researches are going on to improve the heat transfer rate of the helical coil heat exchanger. Here, in this work, an analysis has been done for a tube-in-tube helical heat exchanger with constant heat transfer coefficient with turbulent flow. There are various factors present that may affect the heat transfer characteristics of the heat exchanger. Here, the experiment has been done by varying the curvature ratio i.e. ratio of coil diameter to inner tube diameter and inlet velocity of the hot fluid in the inner tube. The curvature ratio is varied from 8 to 25 and inlet velocity is varied from 1m/s to 2m/s step wise. The analysis has done using ANSYS 13 CFD methodology. Different parameters are calculated from the results obtained and graphs are plotted between various parameters such as Nusselt number, friction factor, pressure drop and pumping power versus Reynolds number. These graphs have been analyzed and discussed to find out the optimal result for which the heat exchanger would give the best performance.

CHAPTER 1

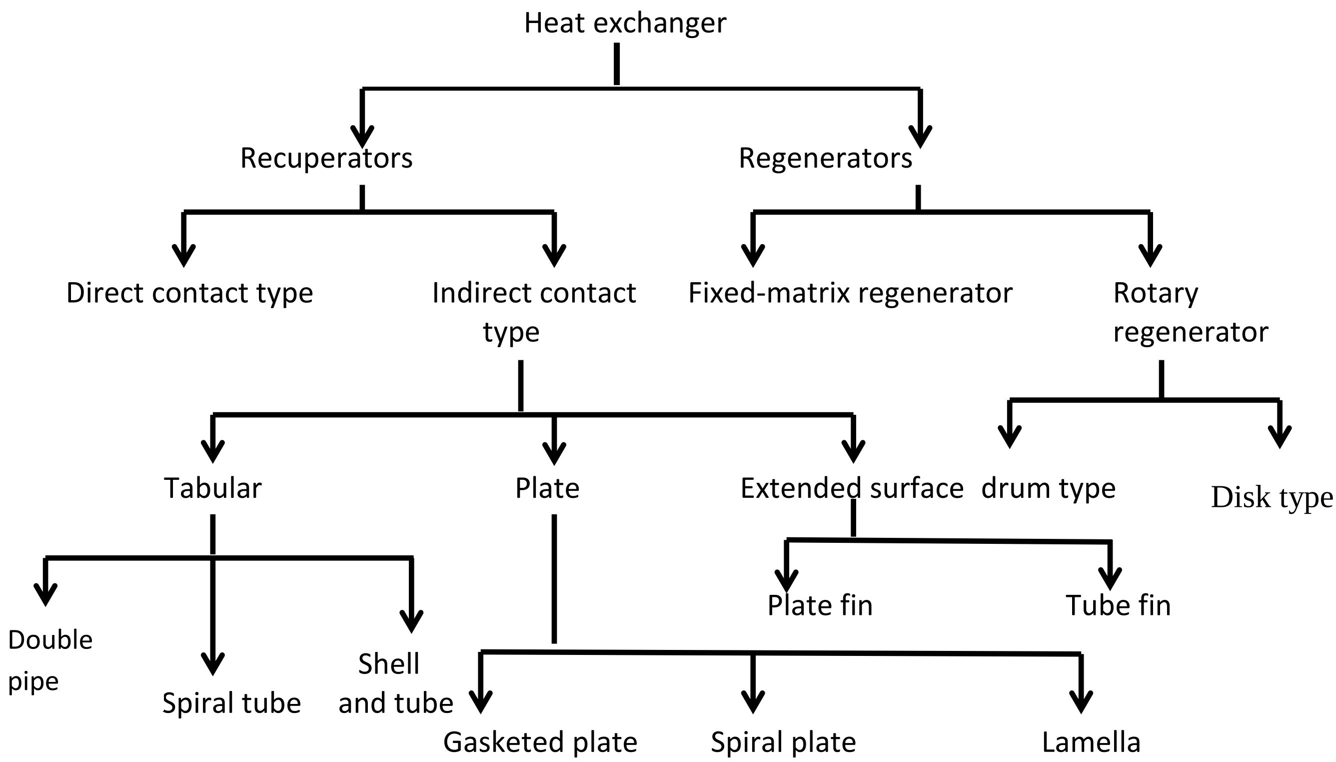
INTRODUCTION

1.1 Heat exchanger

A heat exchanger may be defined as an equipment which transfers the energy from a hot fluid to a cold fluid, with maximum rate and minimum investment and running cost. The rate of transfer of heat depends on the conductivity of the dividing wall and convective heat transfer coefficient between the wall and fluids. The heat transfer rate also varies depending on the boundary conditions such as adiabatic or insulated wall conditions.Some examples of heat exchangers are:

i. Intercoolers and pre heaters;
ii. Condensers and boilers in refrigeration units;
iii. Condensers and boilers in steam plant;
iv. Regenerators;
v. Oil coolers and heat engines;
vi. Automobile radiators etc.

1.2 Classification of heat exchangers



1.3 Tabular heat exchanger

These kinds of heat exchangers are mainly made up of circular coils whereas many different shapes are also used for different applications. They provide flexibility because the geometric parameters such as length, diameter can be modified easily. These are used for phase change such as condensation, evaporation kind of operations. Again it is classified in to three different categories i.e. double pipe heat exchanger, spiral tube heat exchanger and shell and tube heat exchanger.

1.4 Double pipe heat exchanger

These are the simplest heat exchangers used in industries. These heat exchangers are cheap for both design and maintenance, making them a good choice for small industries. In this kind of heat exchanger, two tubes or pipes having different diameters are placed concentrically, the smaller one inside the larger one. The two fluids, in between which heat transfer is required, flows in the two different tubes. The curvature of the tube gives rise to a secondary flow which makes the flow turbulent and increases the heat transfer rate.

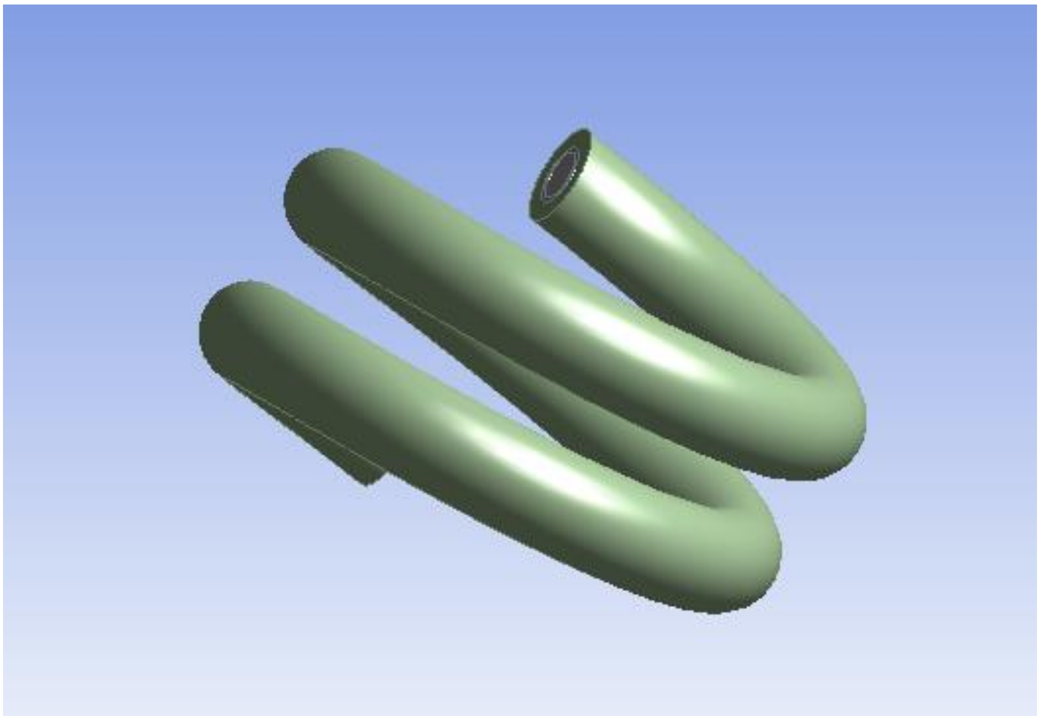


Figure 1 Model of tube-in-tube helical heat exchanger

The utilization, conversion, and recovery of energy in commercial, industrial, and domestic applications usually involve a heat transfer process such as refrigerator, air conditioner etc. Improved quality of heat exchanger above the usual practice can significantly improve the thermal efficiency as well as the economics of their design and production. It has been observed that heat transfer rate in helical coils heat exchanger are higher than that of a straight tube. They are also compact in size. For this helical coil heat exchangers are being widely used in many industrial applications such as nuclear industries, power generation, process plants, refrigeration, heat recovery systems, food industries, etc.

The reason behind higher heat transfer rate of helical heat exchanger is that, due to the swirl flow in a coiled tube, centrifugal forces arises which gives rise to secondary flow pattern. It consists of two vertices perpendicular to the axial flow direction. As a result, the heat transfer takes place by diffusion in the radial direction and by convection. The contribution of the convective heat transfer dominates the overall process and significantly enhances the heat transfer rate per unit length of the tube, as compared to the heat transfer rate of a straight tube of equal length. Also, the coiled tube heat exchanger can provide a larger heat transfer area per unit volume having compact size.

1.4.1 Advantages

- It has larger surface area and compact volume as compared to straight tube heat exchanger.
- It eliminates the dead-zones that are common drawbacks in the shell and tube type heat exchangers because the whole surface area of the curved pipe is exposed to the moving fluid.
- They give improved heat transfer characteristics because of small wall resistance.
- Because of coil like structure it can withstand thermal shock and eliminates thermal expansion.
- More turbulence is created inside the coil tube so fouling is less.

1.4.2 Applications

- Because of compact size, it can be used in applications where space limitation is present such as marine cooling systems, cooling of lubrication oil, central cooling and industrial applications.
- In HVACs due to their compact structure and greater heat transfer rate.
- Used in chemical reactors because of high heat transfer capacity.
- In cryogenic applications for liquefaction of gases.
- Used in hydro carbon processing for the recovery of CO_2, cooling of liquid hydrocarbons, also used in polymer industries for cooling purposes.

1.5 AIM OF PRESENT WORK

The aim of this work is to determine the heat transfer characteristics for a double-pipe helical heat exchanger by varying the size of the coil diameter (i.e. varying curvature ratio D/d) and the mass flow rates (only the hot fluid) in the inner tube at constant wall heat transfer coefficient to the surrounding. Analysis has been carried out for counter flow heat exchanger using ANSYS 13 software also the optimal conditions for heat transfer has been found based on Nusselt number and pumping power required. The variation of Nusselt number, friction factor and pressure drop with Reynolds number has been plotted, and the temperature and velocity contours at the outlets are shown.

1.6 THESIS ORGANIZATION

This thesis consists of six chapters.

Chapter1 gives a brief introduction including definition and classification of heat exchangers. The description of helical heat exchanger along with its advantages and applications are described here. It also includes the aim of the present work.

Chapter2 gives a brief idea about the literature survey of the researches which are related to this work.

Chapter3 describes the detailed methodology adopted for the work which includes the steps followed in geometrical modelling and analysis of the project.

Chapter4 deals with the results obtained from the analysis and discussions related to the results obtained.

Chapter5 describes the conclusion, future scopes and the references.

CHAPTER 2

LITERATURE SURVEY

2.1 LITERATURE SURVEY

A wide range of researches are already done to study the flow characteristics and heat transfer in helical heat exchangers. The enhancement of the heat transfer in the helically coiled tubes is due to the centrifugal forces. A secondary flow field is produced due to the curvature of the tube with a circulatory motion, which causes the fluid particles to move towards the core region of the tube. The secondary flow enhances heat transfer rates by reducing the temperature gradient across the cross-section of the tube. Thus there is an additional convective heat transfer mechanism occurs, perpendicular to the main flow, which does not exist in straight tube heat exchangers.[1]

K.S. Bharuka, D.Y. Kasture studied the characteristics of heat transfer in a double pipe helical heat exchanger and found that the overall heat transfer coefficients increases with increasing inner Dean number. However, this increase is a function of the ratio of the mass flow rates.[2] Vimal Kumar, Burhanuddin Faizee, Monisha Mridha and K.D.P. Nigam conducted an experiment on tube-in-tube heat exchanger and observed that with the increase in operating pressure in the inner tube, the overall heat transfer coefficient increases and the friction factor value in the inner-coiled tube was in agreement with the literature data.[6]

N. Ghorbani , H. Taherian , M. Gorji , H. Mirgolbabaei conducted a practical experiment on a vertical helically coiled heat exchanger and found that the coil surface area was the most influential geometrical parameter on the heat transfer coefficient and effect of tube diameter is almost negligible on overall heat transfer coefficient.[9] Rahul Kharat, Nitin Bhardwaj, R.S. Jha experimented on the effect of various geometric parameters on a concentric helical coil heat exchanger. They plotted the graph between heat transfer coefficient versus tube diameter and coil gap and found that two most important design parameters are coil gap and tube diameter.[8]

J. S. Jayakumar conducted an experiment on helically coiled heat exchangers using CFD and found that the use of constant values for the heat transfer and thermal properties of the fluid resulted in inaccurate heat transfer coefficients. Based on the analysis results he developed a correlation in order to evaluate the heat transfer coefficient of the coil. In this study, analysis was done for both the constant wall temperature and constant wall heat flux boundary conditions. The Nusselt numbers that were obtained were found to be highest on the outer coil and lowest in the inner side. The coil parameters like the diameters of the pipes, the Pitch Circle Diameters have significant effect on the heat transfer and the effect of the pitch is negligible. [10]

Timothy J. Rennie studied the heat transfer characteristics for a double pipe helical heat exchanger for both counter and parallel flow with both the boundary conditions of constant heat flux and constant wall temperature. The results from the simulations were within the range of the pre-obtained results. The overall heat transfer coefficients were determined for dean numbers ranging from 38 to 350. He observed that the overall heat transfer coefficients varied directly with the inner dean number but the fluid flow conditions in the outer pipe had a major contribution on the overall heat transfer coefficient. So, he concluded that during the design of a double pipe helical heat exchanger, the design of the outer pipe should be given the highest priority in order to get a better overall heat transfer coefficient. [3]

J. S. Jayakumar, S. M. Mahajani, J. C. Mandal, Rohidas Bhoi studied the constant thermal and transport properties of the heat transfer medium and their effect on the prediction of heat transfer coefficients. Arbitrary boundary conditions were not applicable for the determination of heat transfer for a fluid-to-fluid heat exchanger. An experimental setup was made for studying the heat transfer and also CFD was used for the simulation of the heat transfer. The CFD simulation results were reasonably well within the range of the experimental results. Based on both the experimental and simulation results a correlation was established for the inner heat transfer coefficient. [7]

CHAPTER 3

METHODOLOGY

3.1 Methodology adopted

Here the analysis is done using ANSYS 14 software.

3.2 Analysis procedure

- Geometrical modelling
- Meshing
- Solution:
  - Material selection
  - Defining zones
  - Boundary conditions
  - Solution methods
  - Solution initialization
  - Iteration
- Plot results and contours
- Calculation of various parameters

3.3 Geometrical modelling

First the geometry of the model is created in ANSYS workbench. Fluid flow (fluent) module is selected from the workbench. The design modeler opens as a new window when the geometry is double clicked.

3.3.1 Sketching

Using sketching option three concentric circles are created in XY-plane(diameters of circles are 6mm, 7mm and 13mm) in three different sketches. A straight line is also created in sketch 4 along the height of the helical tube at the centre.

3.3.2 Sweep

Then the three circles were swept about the central axis with pitch 40 mm and number of turns is 1.5. At first the mean coil diameter is 50mm and then it is varied for different values of D/d. Then using boolean operation the overlapping volumes are subtracted and respective phases were chosen.

3.3.4 Merging

The model has 3 parts and 3bodies after sweep operation. So, all the 3 parts are selected using control and merged into 1 part. At the end it will have 1 part and 3 bodies. The 3 bodies are named as follows:

1. Inner fluid (fluid)
2. Thickness volume (solid)
3. Outer fluid (fluid)

3.4 Meshing

In free meshing a relatively coarser mesh is generated. It contains both tetrahedral and hexahedral cells having triangular and quadrilateral faces at the boundaries. Later, a fine mesh is generated using edge sizing. In this, the edges and regions of high pressure and temperature gradients are finely meshed.

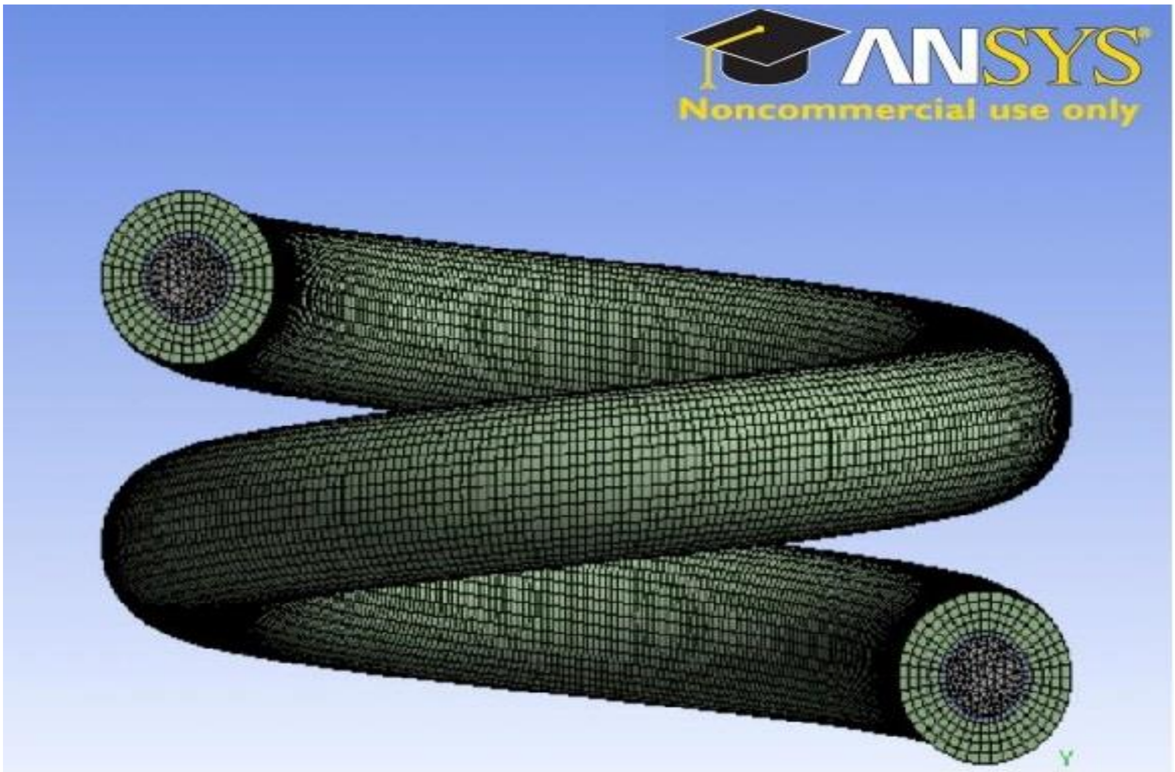


Figure 2 Geometry and mesh of the model

3.5.1 Creating named sections

Different sections are named according to their use such as cold_inlet, cold_outlet, hot_inlet, hot_outlet etc.

Now the project was updated, saved and meshing window was closed. After that the ANSYS Fluent launcher was opened double clicking on setup. The dimension was set as 3D, option as Double Precision and then OK. The Fluent window was opened.

3.6 Solution

3.6.1 Problem Setup

The mesh was checked. The analysis type was changed to Pressure Based type and the velocity formulation was changed to absolute. Time was changed to steady state.

3.6.2 Models

Energy was set ON position. And viscous model was selected as “k-ε model (2 equation).

3.6.3 Materials

Water-liquid as fluid and copper as solid was selected from the fluent database by clicking change/create.

3.6.4 Cell zone conditions

Different parts were assigned as solid or fluid accordingly.

3.6.5 Boundary conditions

Different boundary conditions were applied for different zones. Since it is a tube-in-tube heat exchanger, there are two inlets and two outlets. The inlets were defined as velocity inlets and outlets were defined as pressure outlets. The inlet velocity of the cold fluid was kept constant i.e. 2.5m/s, whereas velocity of hot fluid was varied from 1m/s to 2m/s for different experiments. The outlet pressures were kept default i.e. atmospheric pressure. The hot fluid temperature at inlet was 65^0C and cold fluid inlet temperature was kept 23^0C. The other wall conditions were defined accordingly. The surrounding air temperature was kept 27^0C and convective heat transfer coefficient between outer wall and surrounding was $2500 W/m^2K$.

3.6.6 Solution methods

The solution methods were set as follows:

1. Scheme = Simplc
2. Gradient = Least Square Cell Based
3. Pressure = linear
4. Momentum = Second Order Upwind
5. Turbulent Kinetic Energy = Second Order Upwind
6. Turbulent Dissipation Rate = Second Order Upwind
7. Energy= power law

3.6.7 Solution Control and Initialization

Under relaxation factors the parameters are:

- Pressure = 0.3 Pascal
- Density = 1 kg/m^3
- Body forces = 1 kg/m^2s^2
- Momentum = 0.7 kg-m/s
- Turbulent kinetic energy = 0.8 m^2/s^2

All the conditions were left as default. Then the “hot_inlet” was selected from the compute from drop down list and then the solution was initialized.

3.6.8 Convergence criteria

The convergence criteria were set to 10^{-5} for the three velocity components and continuity, 10^{-8} for energy and 10^{-4} for turbulent kinetic energy and dissipation energy.

3.6.9 Run calculation

The number of iterations was set to 1000 with step size 1. Then the calculation was started and it continued till the results converged.

Various contours were plotted and different parameters were calculated such as weighted average of total temperatures at out let and inner wall, total wall flux, pressure drop across the hot fluid inlet and outlet to calculate pumping power.

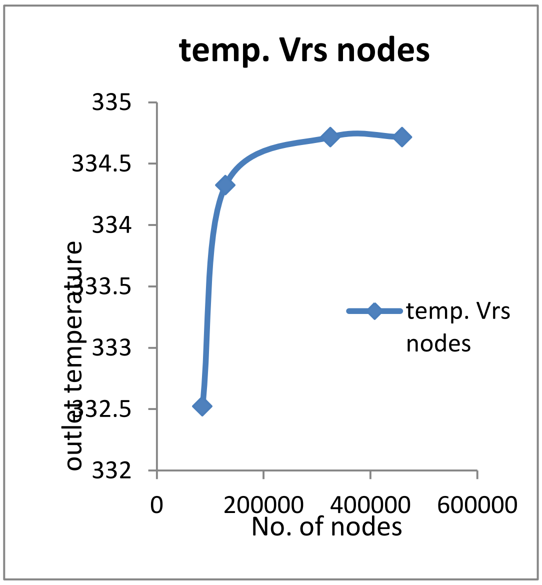
3.7 Grid independence test

This test is carried out to find out the optimum grid size of a model at which the parameters don’t change with varying grid size in mesh. In this method the same simulation is run on progressively for coarse grid, medium grid and finer grids (3 grids should be the minimum) by changing a global grid sizing parameter rather than local refinement, unless it takes too long to run/insufficient computer resources to see if the solution changes as the grid is refined. When the grid is fine enough the solution does not change, so that the solution is 'grid independent'. This gives reducing the discretization error inherent in CFD.

Grid independent test for different geometries of the heat exchangers are shown below.

Table 1 Number of divisions, nodes and outlet temperature

D/d=8		
No. of divisions	No. of nodes	outlet temp. of hot fluid
20,22,44	85623	332.524
24,26,52	128723	334.324
30,32,62	325489	334.717
40,42,84	459620	334.717
D/d=10		
No. of divisions	No. of nodes	outlet temp. of hot fluid
20,22,44	92468	332.542
24,26,52	135263	333.926
30,32,62	356923	334.252
40,42,84	489232	334.254



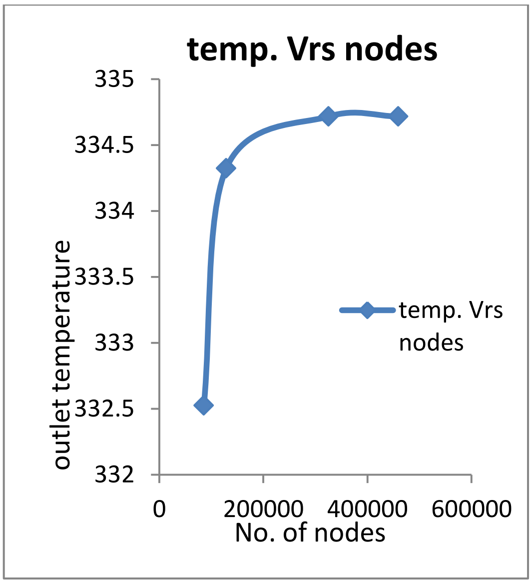


Figure 3 Graph between outlet temp. Vs no. of nodes

It is found that there is no change is outlet temperature when number of divisions is more than 30, 32, 62. Hence, it is taken as the mesh size for all the calculations. "30. 32, 62" means the no of divisions of the three diameters are 30, 32, 62 respectively.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Results and discussions

The heat transfer and flow characteristics of a helical coil pipe can be observed from the contour diagrams of temperature and pressure distribution, variation of Nusselt number, friction factor and pumping power with Reynolds number.

4.1.1 Tables

The results obtained from the CFD analysis are given in the tables below and from that Nusselt number(Nu), friction factor(f), pressure drop(△p) and pumping power(P) has been calculated.

Table 2 Nu, f,△ p and P for different values of Re at D/d=8

Re	Nu	△p	f	P
12937.79	129.6194	1191.57	0.06534	157.8453
15525.34	149.5578	1653.1	0.06295	262.7802
18112.9	168.8872	2182.83	0.06107	404.8183
20700.46	187.7711	2748.04	0.058863	582.4454
23288.02	208.8607	3442.76	0.058267	820.9023
25875.57	224.2856	4170.54	0.057173	1104.929

Table 3 Nu, f, △p and P for different values of Re at D/d=10

Re	Nu	△p	f	P
12937.79	125.6922	1251.53	0.057493	165.7881
15525.34	145.1506	1730.68	0.055211	275.1124
18112.9	164.0135	2278.97	0.053414	422.648
20700.46	182.4424	2894.788	0.051946	613.5485
23288.02	200.4091	3535.32	0.050125	842.9726
25875.57	218.0824	4324.312	0.049663	1145.669

Table 4 Nu, f, △p and P for different values of Re at D/d=15

Re	Nu	△p	f	P
12937.79	146.2556	1626.759	0.046057	215.4941
15525.34	166.4906	1653.09	0.032501	262.7786
18112.9	187.5804	2182.826	0.031531	404.8176
20700.46	208.1515	2780.175	0.030747	589.2564
23288.02	228.2546	3442.77	0.030084	820.9047
25875.57	247.932	4170.551	0.029519	1104.932

Table 5 Nu, f, △p and P for different values of Re at D/d=20

Re	Nu	△p	f	P
12937.79	149.6472	1903.089	0.043789	252.0991
15525.34	171.9615	2619.015	0.041849	416.324
18112.9	193.5491	3433.737	0.040311	636.8061
20700.46	214.5315	4344.977	0.039053	920.9153
23288.02	234.9751	5350.214	0.037996	1275.722
25875.57	254.9566	6446.76	0.037084	1707.984

Table 6 Nu, f, △p and P for different values of Re at D/d=25

Re	Nu	△p	f	P
12937.79	157.4227	2186.524	0.040277	289.6452
15525.34	180.7256	3002.343	0.038406	477.2586
18112.9	203.2441	3929.159	0.036928	728.6851
20700.46	225.1019	4962.713	0.03571	1051.844
25875.57	267.2465	7340.522	0.033804	1944.774

Table 7 Velocity, Nu, △p, f for different D/d values

D/d	v	Nu	△p	f
10	1	125.6922	1251.53	0.057493
15	1	146.2556	1626.759	0.046057
20	1	149.6472	1903.089	0.043789
25	1	157.4227	2186.524	0.040277
10	1.2	145.1506	1730.68	0.055211
15	1.2	166.4906	1653.09	0.032501
20	1.2	171.9615	2619.015	0.041849
25	1.2	180.7256	3002.343	0.038406
10	1.4	164.0135	2278.97	0.053414
15	1.4	187.5804	2182.826	0.031531
20	1.4	193.5491	3433.737	0.040311
25	1.4	203.2441	3929.159	0.036928
10	1.6	182.4424	2894.788	0.051946
15	1.6	208.1515	2780.175	0.030747
20	1.6	214.5315	4344.977	0.039053
25	1.6	225.1019	4962.713	0.03571
10	1.8	200.4091	3535.32	0.050125
15	1.8	228.2546	3442.77	0.030084
20	1.8	234.9751	5350.214	0.037996
`	2	218.0824	4324.312	0.049663
15	2	247.932	4170.551	0.029519
20	2	254.9566	6446.76	0.037084
25	2	267.2465	7340.522	0.033804

4.1.2 Graphs

Using the values obtained from CFD analysis as given in the above tables, graphs are plotted between various parameters from which the fluid flow characteristics and heat transfer can be easily visualized.

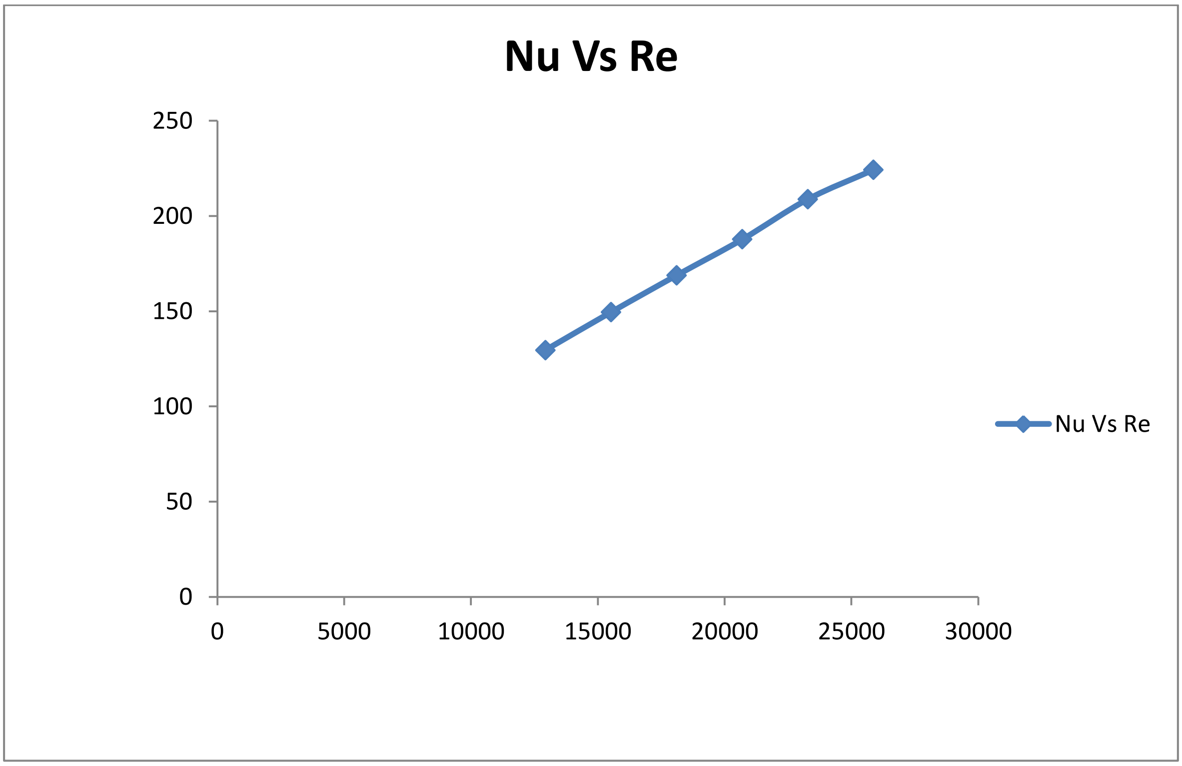


Figure 4 Graph between Nusselt number Vs Reynold number for D/d=8

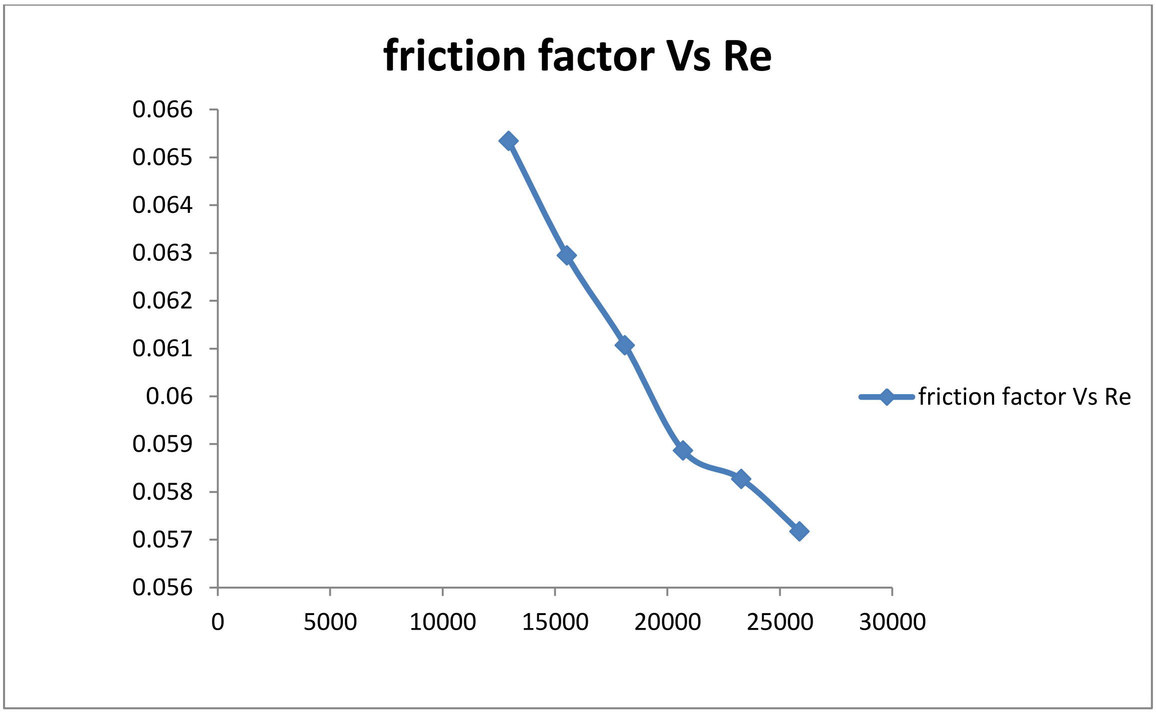


Figure 5 Graph between Reynold number Vs friction factor for D/d=8

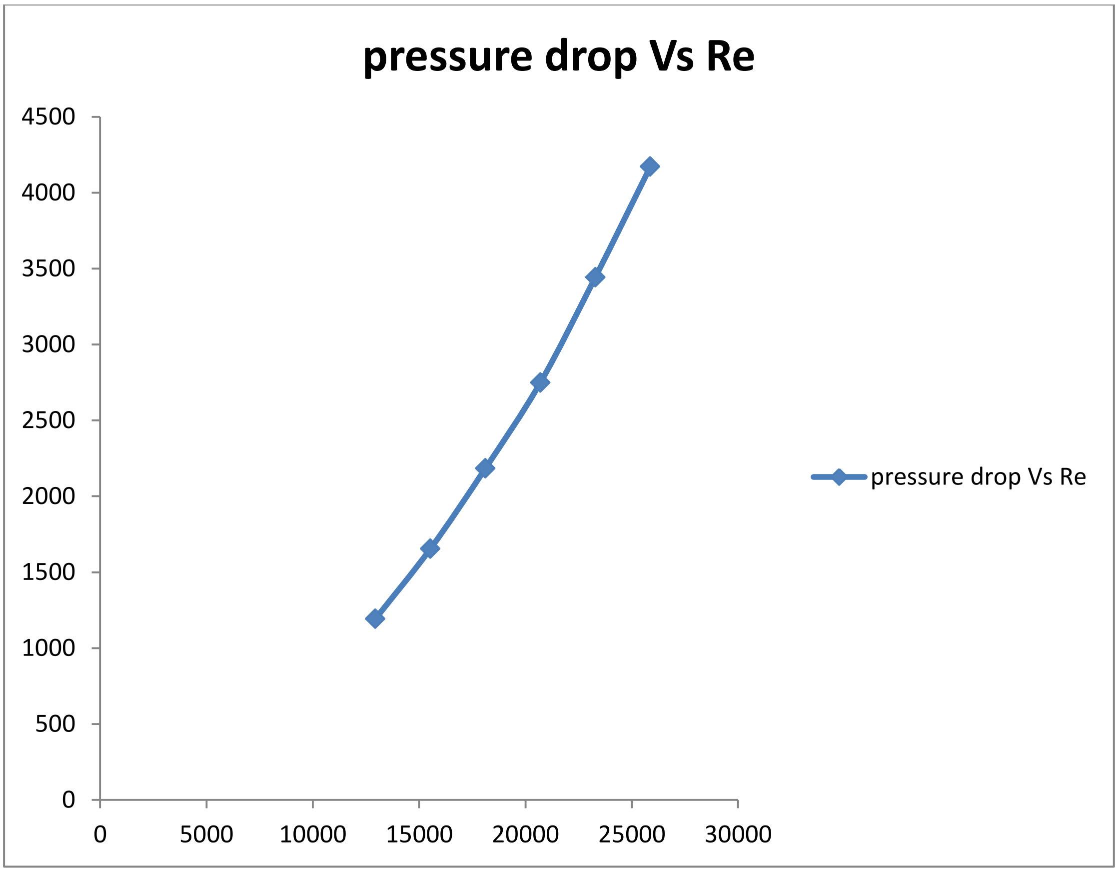


Figure 6 Graph of Reynold number Vs pressure drop for D/d=8

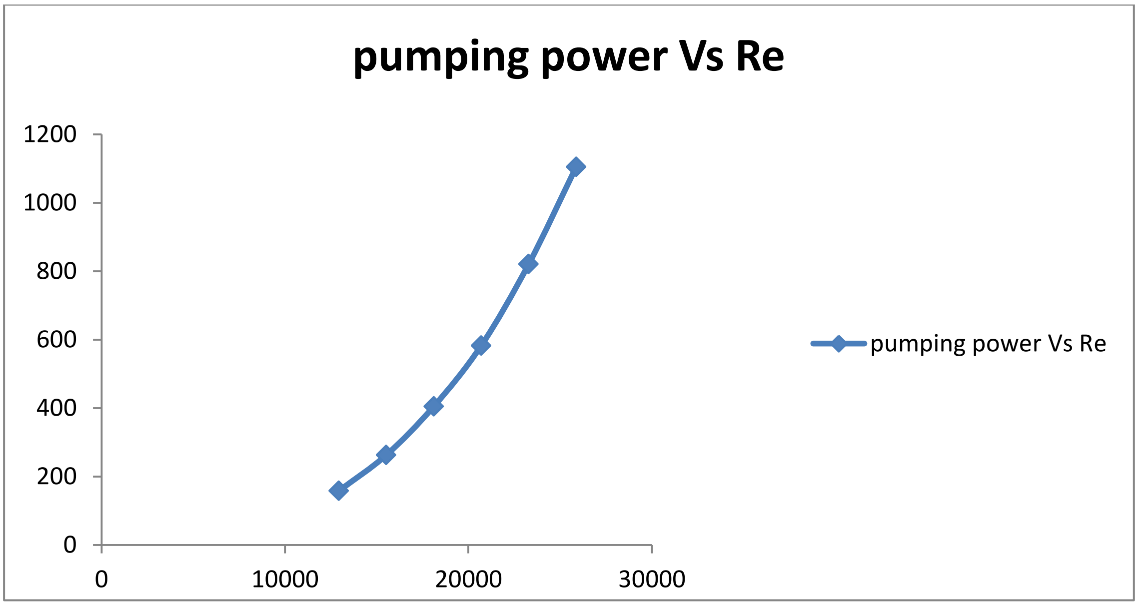


Figure 7 Graph of Reynold number Vs pumping power for D/d=8

From the above graphs (Figure 4, 5, 6, 7) it is observed that the Nusselt number, pressure drop and pumping power increases with increasing Reynolds number whereas friction factor decreases with increasing Re. For the optimal condition of heat exchanger, pumping power should be minimum with maximum Nusselt number, i.e. the power consumption should be less with more heat transfer rate.

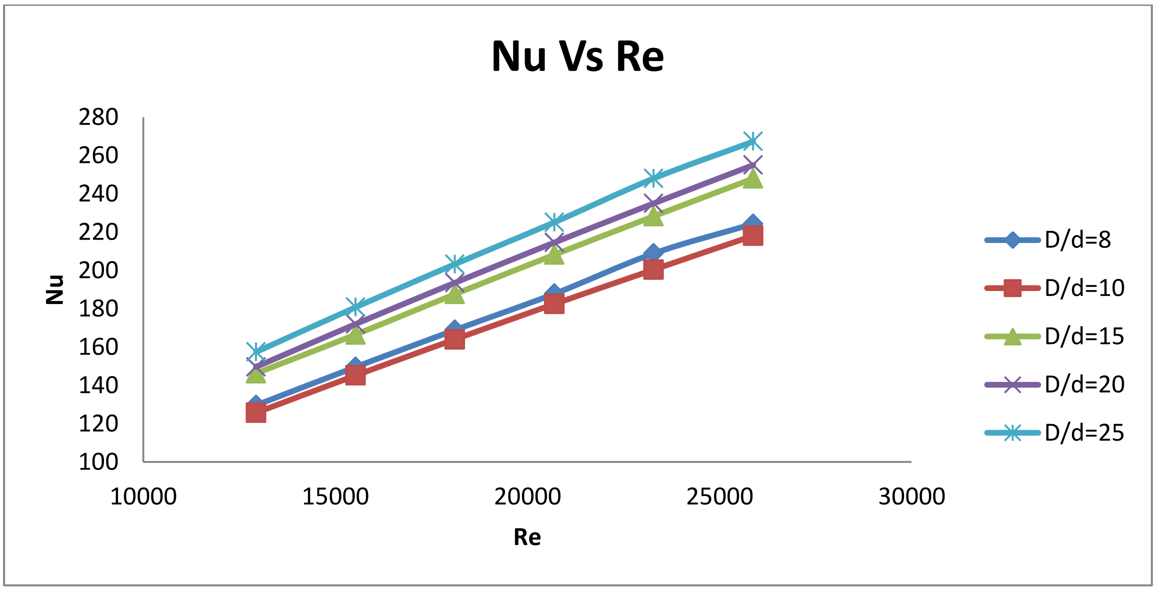


Figure 8 Graph between Nu & Re for different D/d

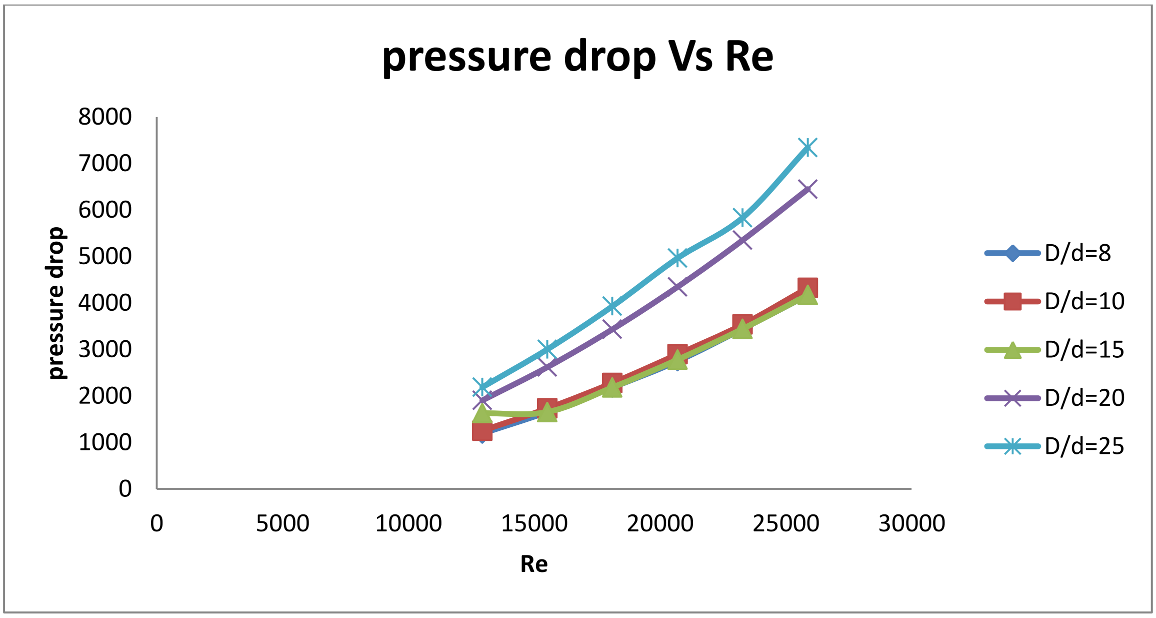


Figure 9 Graph between pressure drop & Re for different D/d

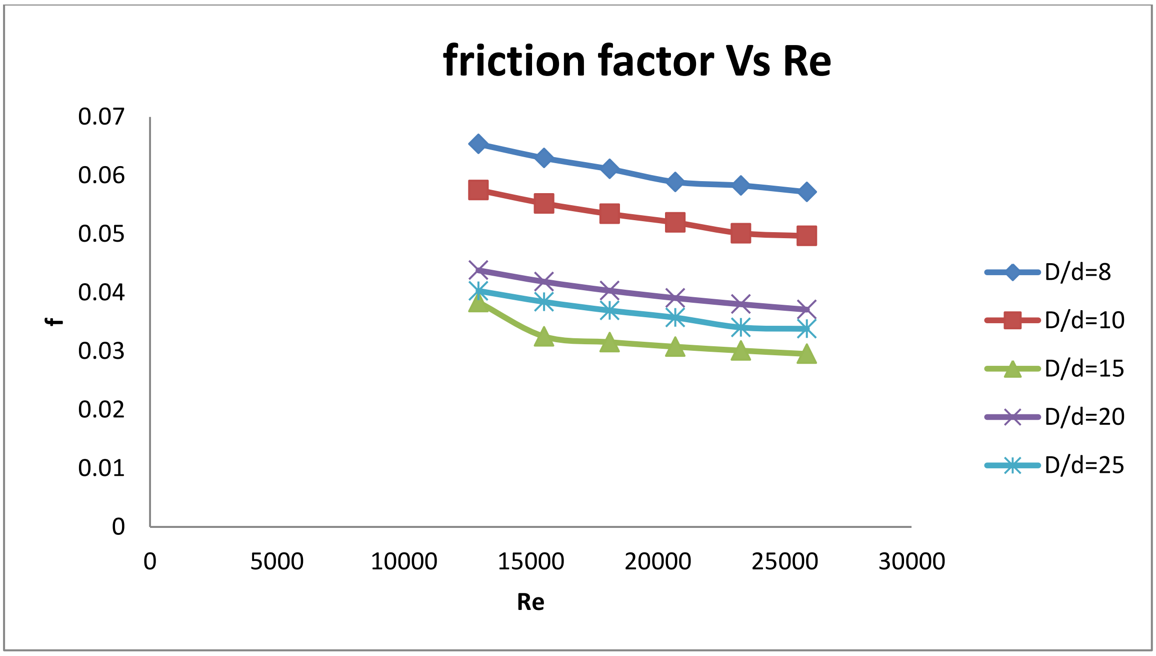


Figure 10 Graph between friction factor & Re for different D/d

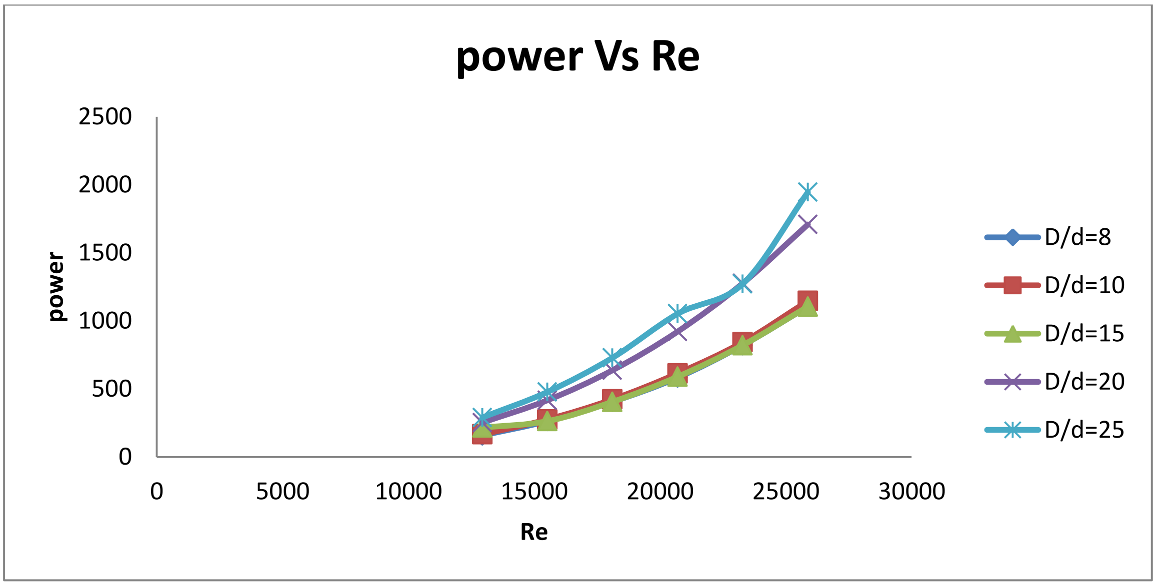


Figure 11 Graph between power & Re for different D/d

In the figures 8, 9, 10, 11 it can be seen that with increasing Reynolds number, Nusselt number, pressure drop and pumping power increasing whereas friction factor decreases.

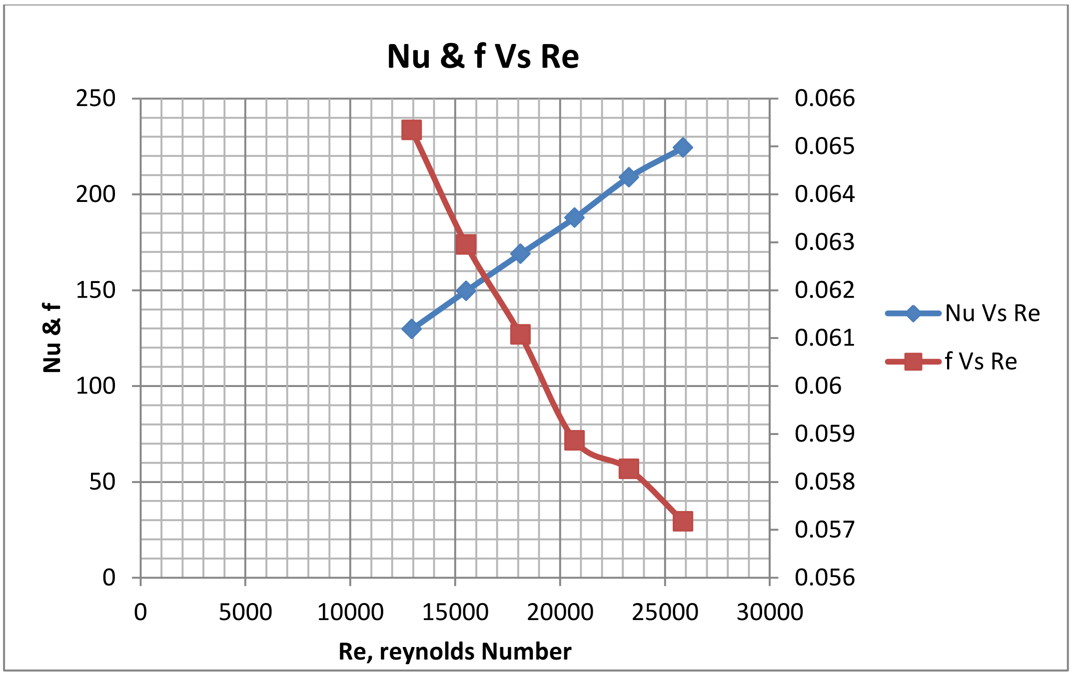


Figure 12 variation of Nusselt number and friction factor with Reynolds number for D/d=8

From the above graph it can be seen if Re is less than 16500, friction factor increases but Nusselt number decreases and if Re is more than 16500, Nu increases but f decreases. So. here Re=16500 is the optimal condition for the heat exchanger with D/d=8.

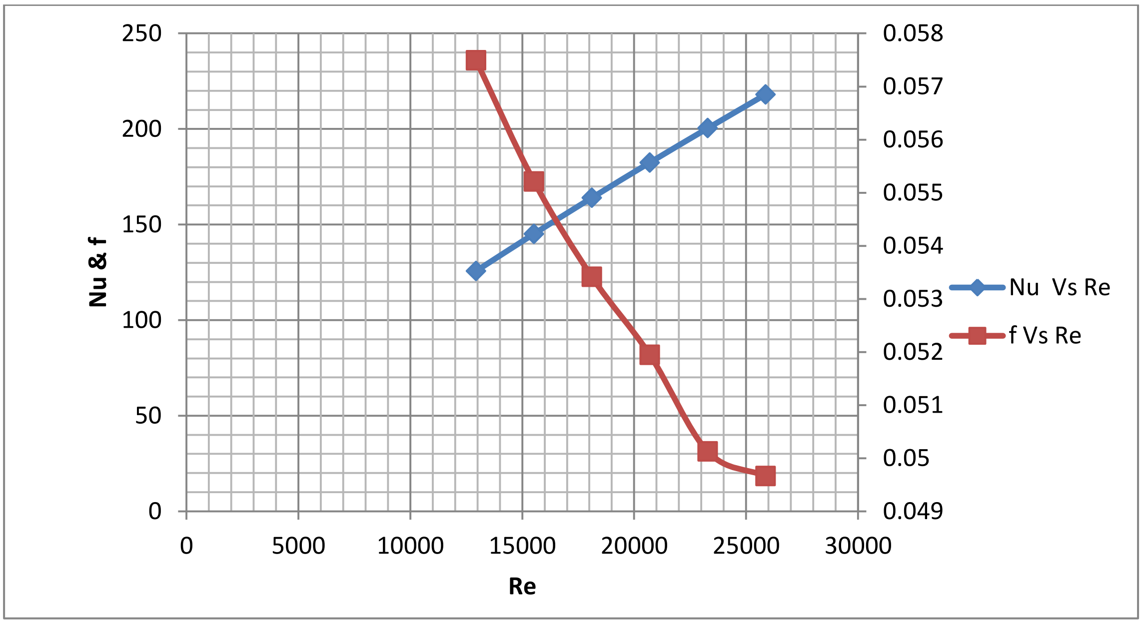


Figure 13 Variation of Nusselt number and friction factor with Reynolds number (D/d=10)

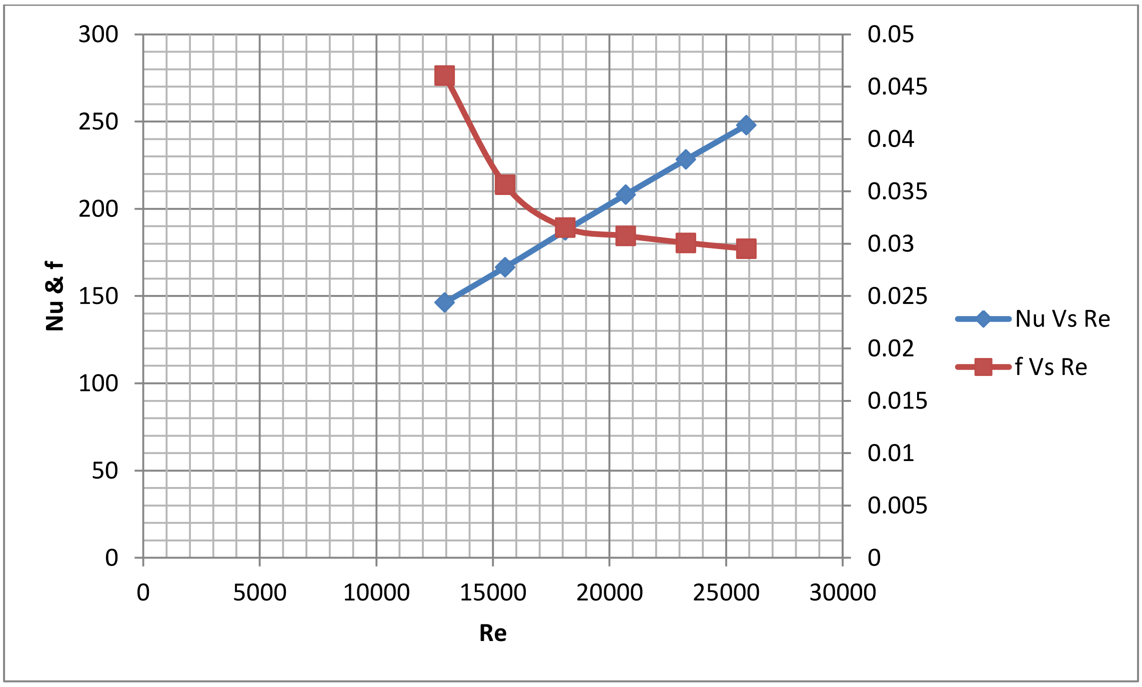


Figure 14 Variation of Nusselt number and friction factor with Reynolds number(D/d=15)

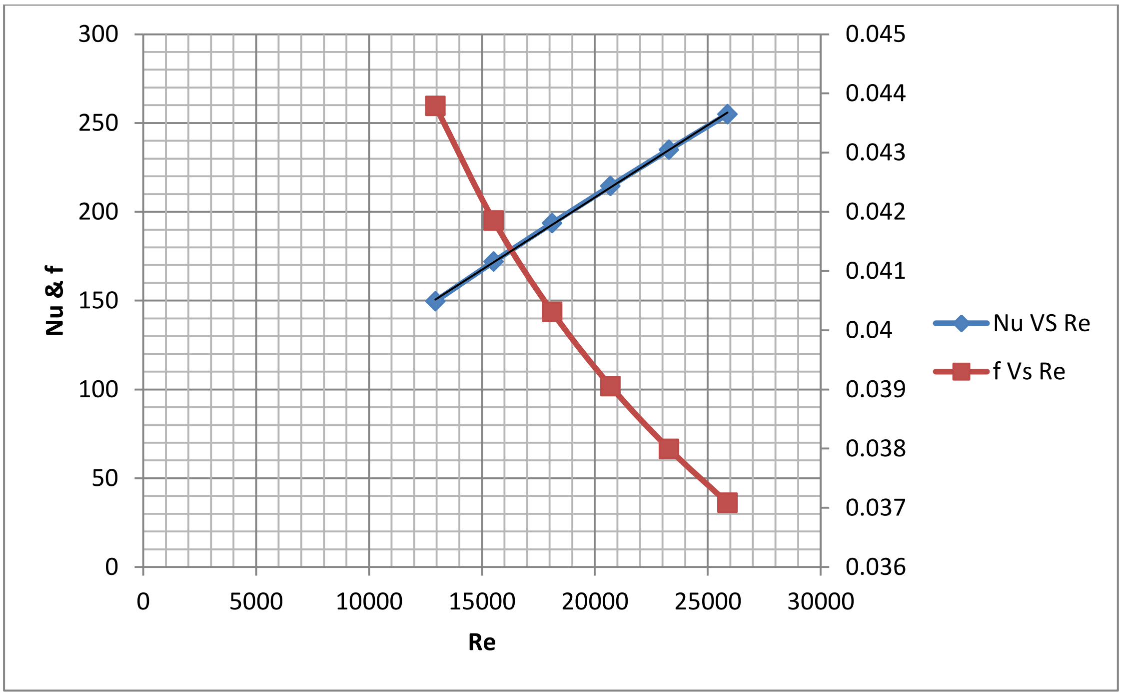


Figure 15 Variation of Nusselt number and friction factor with Reynolds number(D/d=20)

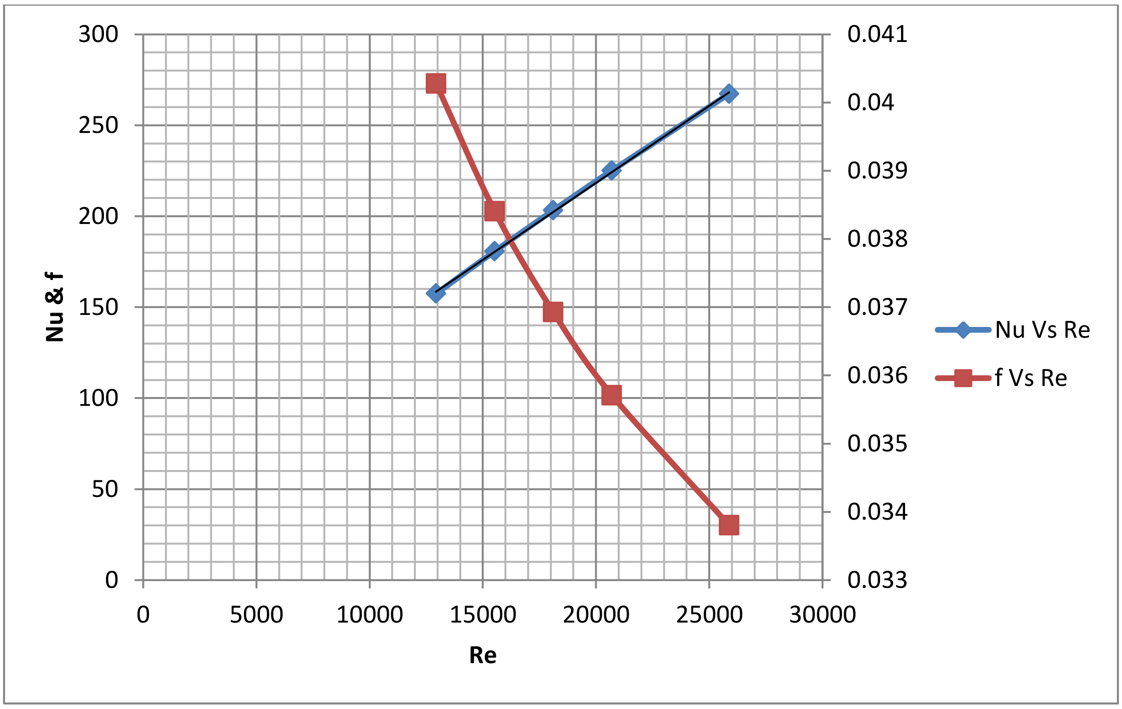


Figure 16 Variation of Nusselt number and friction factor with Reynolds number(D/d=25)

The optimal values of Reynold number for the heat exchanger for different values of D/d can be taken as the point of intersection of both the curves as shown in figures 13, 14, 15 and 16.

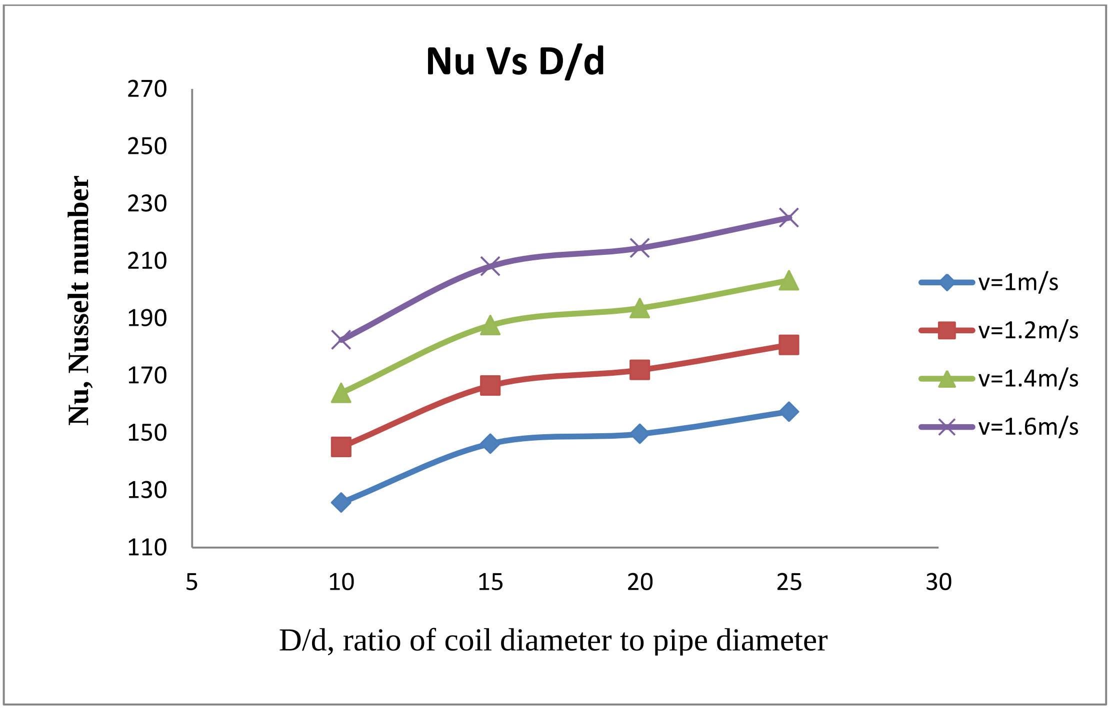


Figure 17 Nusselt Number Vs D/d for various inlet velocities

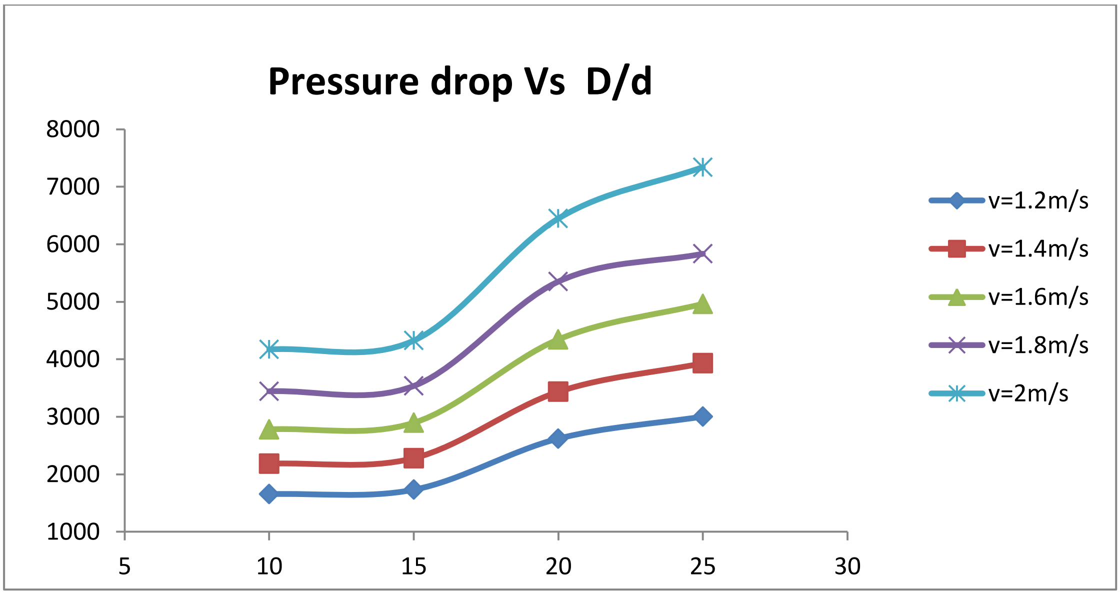


Figure 18 Pressure drop Vs D/d for various inlet velocities

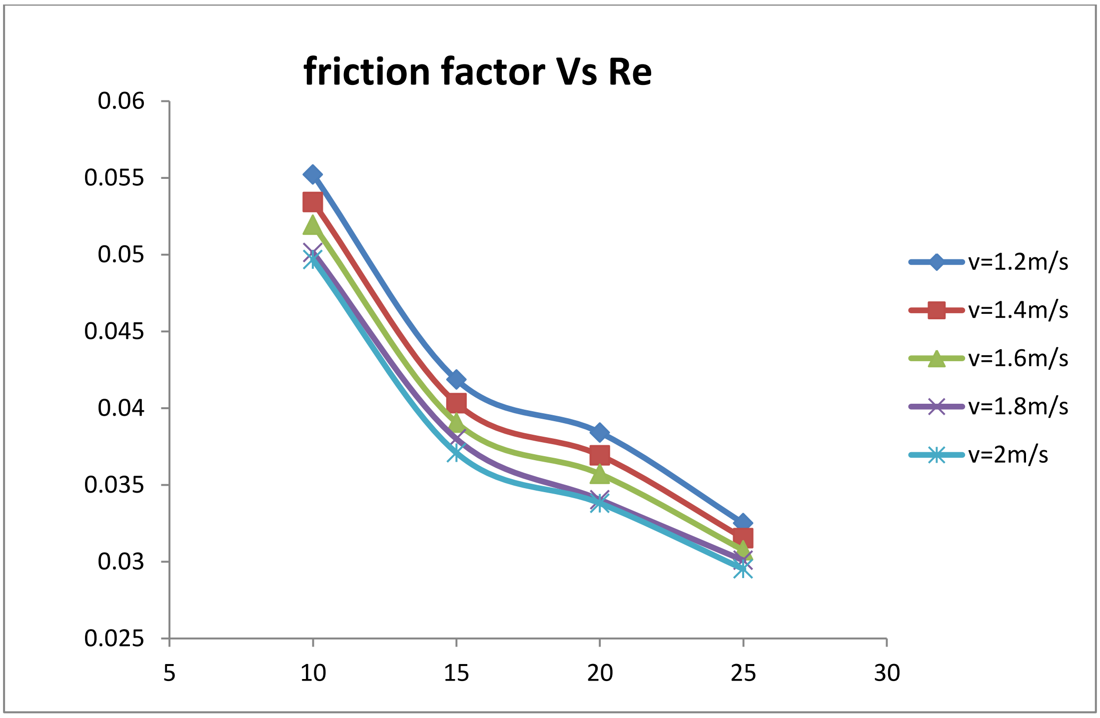


Figure 19 Friction factor Vs D/d for various inlet velocities

From the above graphs (figure 17, 18, 19) we found that, Nusselt number and pressure drop increases with increasing curvature ratio, D/d, whereas friction factor decreases with increasing curvature ratio for different values of inlet velocities in the inner tube. So, the optimal dimension of a heat exchanger can be calculated by considering both the parameters.

CHAPTER 5

CONCLUSION & FUTURE SCOPES

5.1 Conclusion

This work investigates the heat transfer and flow characteristics of a tube-in-tube helical heat exchanger for counter flow using CFD methodology. The effect of mass flow rate in the inner tube and curvature ratio are studied and the various conclusions drawn are:

- Nusselt depends on the curvature ratio, i.e. the ratio of coil diameter to inner tube diameter. Nu increases with increasing curvature ratio. Also it was found that Nu increases with increasing mass flow rate or Reynolds number.
- For turbulent flow in the pipe, friction factor decreases with increasing Reynolds number(Re) whereas heat transfer rate increases with Re. So, there exists an optimal value of mass flow rate and curvature ratio for which the heat exchanger would give the best performance.
- For more heat transfer rate, higher curvature ratio should be preferred irrespective of the power low.
- From the velocity and temperature contours it can be observed that the velocity is higher towards the outer side of the coil whereas temperature is higher towards inner side of coil.
- The heat transfer performance of a helical tube heat exchanger is more than that of a straight tube heat exchanger.

5.2 Future scope

Future works required to be carried out for further improvement of helical heat exchangers are:

- CFD analysis and optimization of the curvature ratio using Dean number and Colburn factor for boundary conditions of constant wall temperature and constant wall heat flux for both laminar and turbulent flow.
- To analyze the results and optimize the heat transfer rate with varying the pitch of the helical coil.

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