Post Experimental Analysis of Tube-In-Tube Helical Coil Heat Exchanger

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Abstract

Tube-in-tube helical coil heat exchangers consist of long, small diameter concentric tubes bent in the form of helix. Helical coil heat exchangers can be better choice in some cases such as, where space is limited, so that not enough straight pipe can be laid. Also, under conditions low flow rates where shell and tube heat exchanger would become uneconomical due to low heat transfer coefficients, the tube-in-tube helical coil heat exchanger can be proved economical. In this work only post experimental analysis is considered, which includes the variations of parameters such as curvature ratio, pitch of coil and its effects on heat exchanger and heat transfer coefficient. For experimentation water was used as working fluid. In this analysis, all the possible flow arrangements are considered such as parallel flow and counter flow.

Keywords

Curvature ratio, secondary flow development, post experimental analysis, pre experimental analysis, Nusselt number, Reynolds number, heat transfer coefficient.

Nomenclature

d_i = Inner tube diameter, mm

d_o = Outer tube diameter, mm

D = Coil diameter, mm

δ = Curvature Ratio

Re = Reynolds Number

Nu = Nusselt Number

h = Heat transfer coefficient, W/m^2K

b = Pitch of coil, mm

γ = Dimensionless pitch, $b/\pi D$

Pr = Prandtl Number

De =Dean Number

t = Tube thickness,mm

r = Tube radius, mm

R = Coil radius,mm

d_h = Hydraulic diameter, mm

Introduction

Along with improvisation of performance of heat exchangers, heat transfer augmentation techniques enable the decrease in the size of heat exchanger. Currently, single tube helical coil or helical coil and shell type heat exchangers are being widely used in industrial applications. For any heat exchanger to be used in industry proper experimentation and evaluation of it is must.

Pre experimental analysis consist the checking of the construction of heat exchanger computer aided analysis of all the possibilities of failure of heat exchanger and numerical analysis. This includes the CFD (Computational Fluid Dynamics) analysis of using ANSYS Fluent or any other similar package. This gives the manufacturer and designer the overview of approximate performance of heat exchanger without actually putting the heat exchanger in service. T. Elango[9] used CFD tool ANSYS CFX 12.2 for finding out the temperature profile at various sections of single tube helical coil heat exchangers. He found that the variation in the temperature profile goes on increasing with increase in pitch of the coil. Resultantly, thermal loading on coil goes on increasing with increase in pitch. Also, he plotted pressure contour along the wall at fluid inlet velocity of 1 m/s. after doing such analysis he concluded that the helical coil with 60mm pitch is better as compared to coil with 30mm pitch with limitation in space and pressure drop. In this way, without actual conducting the experiment, the most economical heat exchanger can be designed and redesigned. J.S. Jaykumar[4] did the CFD analysis at constant wall temperature boundary condition and plotted the velocity contours at various planes along the length of the coil. He used Scientific Visualization package 'Anu Vi' developed by Computer Division, BARK, India. Soumya Mohanty[10] did the CFD analysis with ANSYS Fluent 13.0 and plotted the contours of turbulent dissipation rate, dynamic pressure, turbulent kinetic energy, effective thermal conductivity, total surface heat flux, effective Prandtl number. He did numerical analysis of heat transfer characteristics of helical coil and the results then compared with counter flow arrangement. The CFD results when compared with experimental results from different studies, they were well within error limits. So, the pre experimental analysis is helpful to find out solutions for most of the problems that might arise after experimentation. Not only thermal analysis, but dynamic analysis such as vibrations in coil due to velocity of fluids, velocity of fluid at different sections such as inlet and outlet, pressure acting at joints can be found out and compared with theoretical calculations of designer of heat exchanger. Pre experimental analysis also gives the idea about material which can be used for manufacturing and economical performance of heat exchanger.

Post experimental analysis, on the other hand deals with the actual results and impacts of experiment. It advises, the manufacturer about the plus points along with flaws if any in the heat exchanger. These are direct observable changes such as, by changing the inlet velocity of fluid one can detect the vibrations in the heat exchanger assembly, actual temperature at various sections can be directly measured by fitting sensors at different locations. Also, the leakages in the actual setup, supports at various points and the actual fluid pressure drop can be detected after experimental run only. Post experimental analysis consists of effects of change in curvature ratio, pitch of the coil, helix angle of the coil, tube diameter, coil diameter on overall performance of the heat exchangers. This analysis can be done by direct recording the observations in specially designed software, algorithms and by hand calculations using different correlations for Nusselt number and heat transfer coefficient. J.S. Jaykumar[4] did experimental and theoretical analysis of a helical coiled heat exchanger, in which heat transfer is between fluid-fluid. Experimental setup was fabricated to get the output in estimation of heat transfer characteristics. He found out influence of secondary flows on heat transfer coefficient. Pramod S. Purandare[7] carried out analysis of single tube helical coil heat exchanger. He performed the calculations as per the data reduction procedure for helical coil configuration and are tabulated the results for heat transfer analysis. His analysis concluded as, increase in tube diameter with constant coil diameter increases the curvature ratio and hence the secondary flows. This increased intensity of secondary flows increases Nusselt number and hence the heat transfer coefficient. V.K. Matawala[3] carried out similar analysis for single tube spiral coil heat exchanger and collected design procedure from different literature. His analysis gives idea about the changes needed to be incorporated for reducing the shell side pressure drop and improvisation of heat transfer.

Pre and post experimental analysis improves the quality of the heat exchangers delivered to the customer. By following this analysis manufacturer can supply flawless or heat exchangers with minimum flaws along with having optimum performance characteristics.

Curvature Ratio

The ratio of tube radius to coil radius is the curvature ratio (δ). It is one of the most important factors to be considered while evaluating heat transfer in tube-in-tube helical coil heat exchanger. Dean number (De) which is used to characterize flow in helical coil is calculated from curvature ratio. The curvature of coil determines the centrifugal force acting on fluid particles. The enhancement in heat transfer in helical coil is mainly due to the complex flow pattern existing inside the tube. The curvature effect makes the fluid in the outer side of the pipe to move faster than that present inside, which gives a difference in the velocity setting up a secondary flow which changes correspondingly with the dean number of the flow.

$$\delta = \frac{r}{R}$$

$$De = Re\left(\frac{d}{D}\right)^{0.5}$$

For flow through Inner tube, d = d_i.
For flow through Annular space, d = d_h.

In this experimental work curvature ratio is changed in three ways:

a. Change in inner tube diameter while keeping outer tube diameter and coil diameter constant.
b. Change in outer tube diameter while keeping inner tube diameter and coil diameter constant.
c. Change in coil diameter while keeping inner and outer tube diameter constant.

Secondary Flow Development

In helically coiled tube-in-tube arrangement flow is modified due to curvature effect. When fluid flows through straight tube the velocity of the fluid at centre of tube is maximum, zero at tube wall and symmetrically distributed about tube axis. In helically coiled tube arrangement, due to curvature effect, velocity is highest at centre. The fluid at centre is subjected to maximum centrifugal action and pushes the fluid towards the outer wall. The fluid at the outer wall moves inwards along the tube wall to replace the fluid ejected outwards. Resultantly, two vortices symmetrical about horizontal plane though the centre of tube, are formed. Thus, Secondary flow is characterized by centrifugal action and acts in plane perpendicular to primary flow. Secondary flow development plays an important role in heat transfer in helical coil tube-in-tube heat exchanger.

Critical Reynolds Number (Re_{Cr})

Reynolds number gives the ratio of inertia forces to viscous forces acting on fluid particles in flow. It is the factor on which, whether the flow is turbulent or laminar is decided. The Reynolds number value which separates laminar region from turbulent is the critical value of Reynolds number, which is 2100 for flow through straight tube.

$$Re_{Cr} = 2100\ (1 + 12\sqrt{\frac{d}{D}})$$

For flow through Inner tube, $d = d_i$.
For flow through Annular space, $d = d_h$.

In this experiment, Hydraulic diameter (d_h) is given by,

$$d_h = d_o - (d_i + 2t)$$

In earlier literature, it has been widely observed and studied that flow of fluid inside the coiled tube remains in viscous regime up to much higher Reynolds number than that for straight tubes. The curvature induced vortices i.e. Dean vortices suppress the onset of turbulence and delay the transition. Therefore, critical Reynolds number for flow through coiled tube is more as compared to flow through straight tubes.

Results and Post experimental analysis

For this experiment all the possible fluid flow arrangements are taken into consideration such as:
Series 1 = parallel flow with hot water flowing through inner tube.
Series 2 = counter flow with hot water flowing through inner tube.
Series 3 = parallel flow with cold water flowing through inner tube.
Series 4 = counter flow with cold water flowing through inner tube.

1] Effect of Curvature Ratio

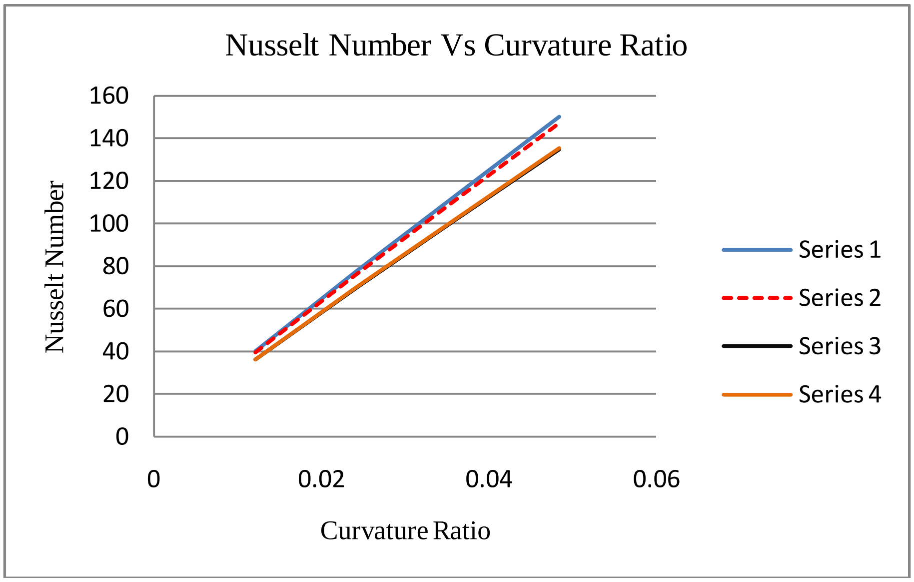
Case 1:

d_i = Varying	D = Constant = 165.2mm	d_o = Constant = 12 mm

Here, the flow conditions of the fluid flowing through inner tube will be changed. So, average temperature of fluid flowing through inner tube only is considered.

Series 1: T_{avg} = 43.5˚C
Series 2: T_{avg} = 40˚C
Series 3: T_{avg} = 30.5˚C
Series 4: T_{avg} = 31˚C

From Graph 1, as inner tube diameter is increased, curvature ratio also increases linearly. Also, the Nusselt number increases steadily with increase in curvature ratio. For this analysis the Nusselt number is calculated using the correlation given by Roger et al[7], because this correlation considers the effect of curvature directly without imposing any restrictions of dean number. Also, the Reynolds number depends on fluid viscosity which in turn depends on fluid temperature. As the average temperature for Series 3 and Series 4 are nearly equal, their graphs overlap. From this graph it can also be concluded that, higher average temperature in addition to higher curvature ratio yields higher values of Nusselt number.



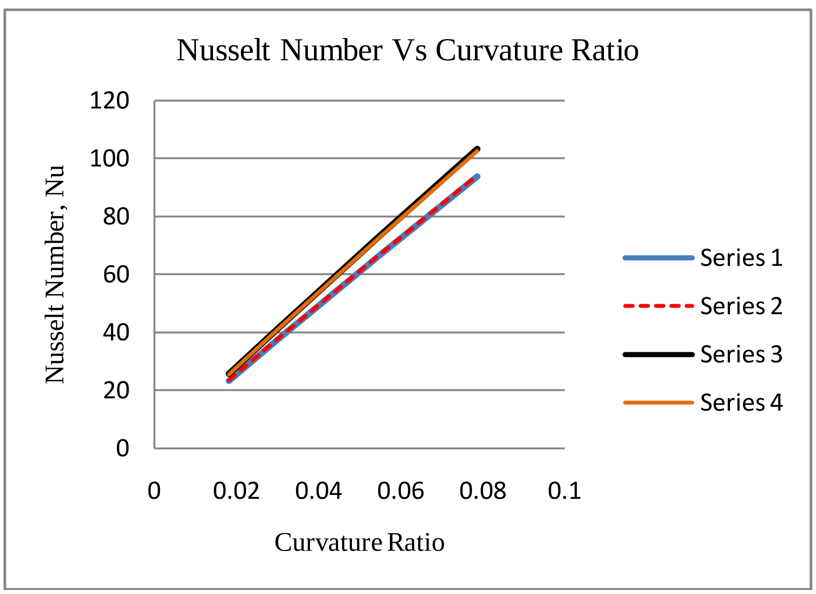
Graph 1: Nusselt Number Vs Curvature Ratio with d_i varying

Case 2 :

d_o = Varying	D = Constant = 165.2mm	d_i = Constant = 6 mm

Here, the flow conditions of the fluid flowing through annular space will be changed. So, average temperature of fluid flowing through annular space only is considered.

Series 1: T_{avg} = 30.5˚C
Series 2: T_{avg} = 31˚C
Series 3: T_{avg} = 41.5˚C
Series 4: T_{avg} = 40.5˚C



Graph 2: Nusselt Number Vs Curvature Ratio with d_o varying

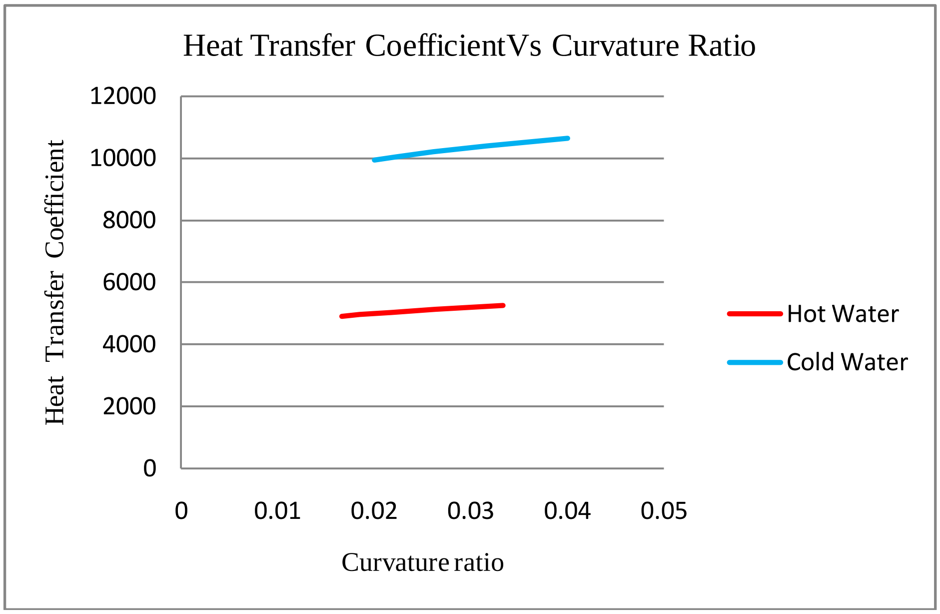
In Graph 2, the similar trend as that of Graph 1 is observed. Comparing Graph 1 and Graph 2, even though curvature ratio in Graph 1 is lower (0.04 to 0.06) the Nusselt number whoops to 150, where as in Graph 2, Nusselt number remains around 100 even at higher curvature ratio of (0.08). This is due to the Reynolds number variation. The Reynolds number of the fluid flowing through annular space remains well below of the Critical Reynolds number and hence the flow is laminar. And hence, the heat transfer will be lower as compared to Case 1.

Case 3:

D = Varying	d_o = Constant = 12mm	d_i = Constant = 6 mm

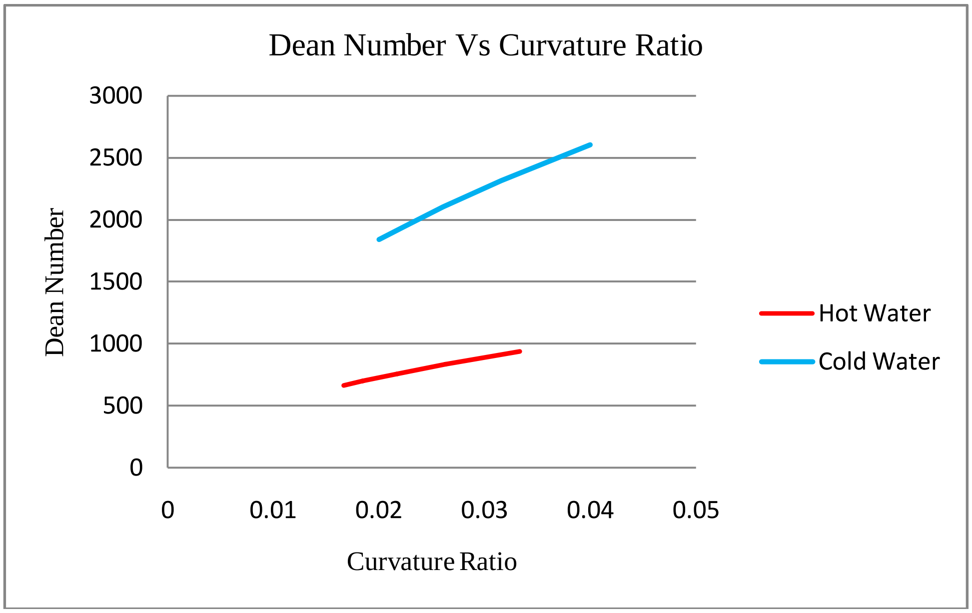
Here, due to variation in coil diameter, different curvature ratios for flow through inner tube and annular space are obtained. The experimentation for this case is performed only for Series 4 arrangement. For this average temperature of fluid flowing through inner tube and annular space is considered.

$T_{avg(inner\ tube)} = 31^{\circ}C$ $T_{avg(annular\ space)} = 40.5^{\circ}C$



Graph 3: Heat Transfer Coefficient Vs Curvature Ratio with D varying

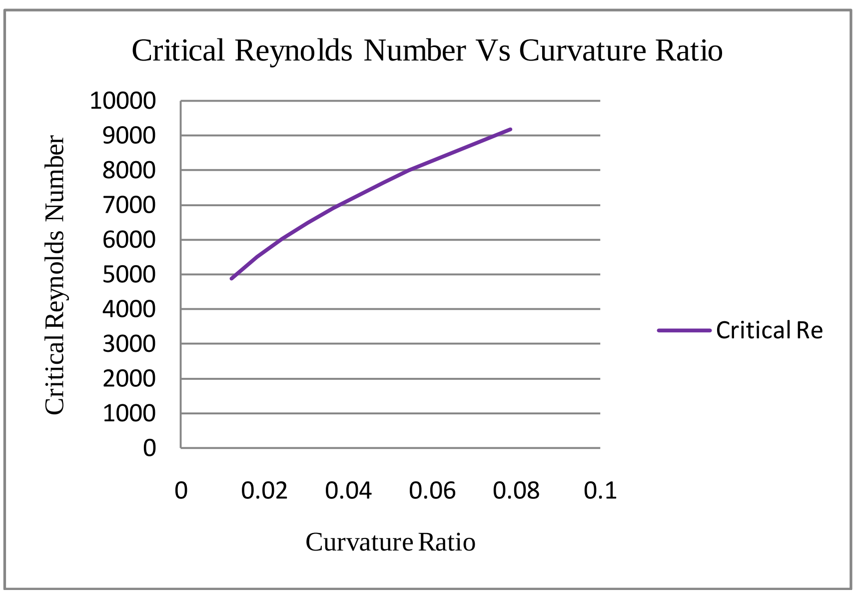
Graph 3 shows that, the heat transfer is directly proportional to curvature ratio. As we go down the line with increasing coil diameter from 150mm, the Reynolds number of hot fluid (annular space) remains well below the corrousponding Critical Reynolds number and hence the flow remains laminar. Also, due to low intensity of secondary flows, the heat transfer coefficient is adversly affected and the same is reflected from graph.



Graph 4: Dean Number Vs Curvature Ratio for D varying

As seen from Graph 4, the plot of Dean number Vs Curvature ratio for Cold fluid (inner tube) is steeper than hot fluid. The dean number is varied directly with square root of curvature ratio. The Dean number for cold fluid is 3 to 4 times that of hot fluid(annular space). So the dean vortices are more intense for cold fluid which is directly reflected in Graph 3 in terms of heat transfer coefficient.

2] Effect on Critical Reynolds Number



Graph 5: Critical Reynolds Number Vs Curvature Ratio

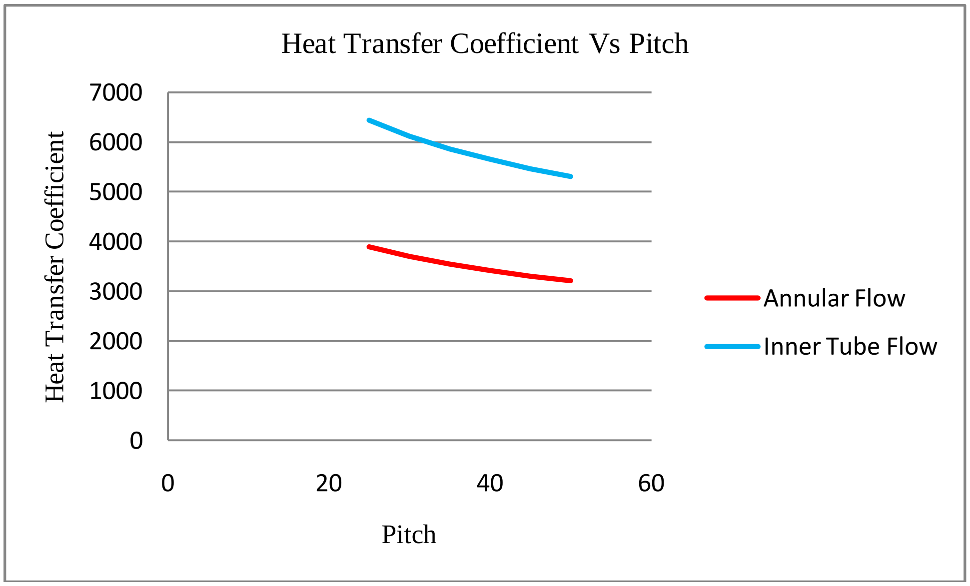
The Graph 5 advocates the Critical Reynolds number's correlation given by Srinivasan et al.
As curvature ratio increases from 0.01 to 0.08, the value of critical Reynolds number gets doubled over this span.

3] Effect of Pitch

The ratio of pitch of helical coil to the developed length of one turn is called dimensionless pitch. This dimensionless pitch is considered while calculating Nusselt number and for this the following correlation given by M.S. Salimpour[6] is used.

$$Nu = 0.152 De^{0.431} Pr^{1.06} \gamma^{-0.277}$$

The calculations were performed and graph is plotted for Series 4 only.



Graph 6: Heat Transfer Coefficient Vs Pitch

It can be seen from the Graph 6 that, the flow of fluid through inner tube(here, cold fluid) is highly affected by variation of pitch. The heat transfer coefficient of fluid flowing through inner tube decreases rapidly as compared to fluid flowing through annular space.

Conclusion

1. As curvature ratio increases, the transition of fluid flow from laminar to turbulent regime gets delayed. This results in higher values of Critical Reynolds Number for helically coiled tube-in-tube heat exchanger.
2. With tube-in-tube helical coil heat exchangers it is possible to accommodate larger heat transfer surface area within small space when compared with straight tube arrangement.
3. As curvature ratio increases, the heat transfer coefficient also increases. This increase in heat transfer coefficient for fluid flowing through inner tube remains higher than that of fluid flowing through annular space.
4. Increase in coil pitch weakens the secondary flows with the same Reynolds number and ultimately approaches to straight tube characteristics. Therefore, when coil pitch is increased, heat transfer coefficient of the fluid decreases.

References

1. Abo Elazm M.M. , Ragheb A.M. , Elsafty A.F. , Teamah M.A., “Experimental and Numerical Comparison between the performance of Helical cone coil and ordinary helical coils used as dehumidifier for humidification dehumidification in desalination units”, International Journal of Applied Engineering Research, Vol. 2, No. 1, 2011.

2. Bibin Prasad, Sujith V, Mohammed Shaban K, Saju Haneef, Sandeep N, Vishnu Raj, “ Comparison of Heat Transfer Between a Helical and Straight tube Heat exchanger”, International Journal of Engneering Research and Technology, Volume 6, Number 1, 2013.

3. Jay J. Bhavsar, V.K. Matawala, S. Dixit, “Desighn ans Experimental Analysis of Spiral Tube Heat Exchanger”, International Journal of Mechanical and Production Engineering, Volume 1, Issue 1, July 2013.
4. J. S. Jayakumar, Helically Coiled Heat Exchanger, Heat Exchanger – Basics Design Applications, Dr. Jovan Mitrovic(Ed.), March 2012.
5. M. Necati Özisik“Heat Transfer – a Basic Approach”, McGraw-Hill, New York, 1985
6. M.R. Salimpour, “ Heat Transfer Coefficient of Shell and Coiled Tube Heat Exchangers”, Experimental Thermal and Fluid Science, Jan. 2009
7. Pramod S. Purandare, Mandar M. Lele, Rajkumar Gupta, “ Parametric Analysis of Helical Coil Heat Exchanger”, International Journal of Engineering Research and Technology, Volume 1, Issue 8, October 2012
8. Ramesh K. Shah, Dušan P. Sekulić,“Fundamentals of Heat Exchangers Design”, John Wiley and Sons Inc.,2003.
9. R. Thundil Karuppa Raj, Manoj Kumar S., Aby Mathew C. and T. Elango,“ Numerical Analysis of Helically Coiled Heat Exchanger Using CFD Technique”, ARPN Journal of Engineering and Applied Sciences, Volume 9, Number 3, March 2014.
10. Souyma Ranjan Mohanty, “CFD Analysis of Heat Exchanger Using Fluent”, National Institute of Technology, 2013. (M.Tech Report)
11. Timothy J. Rennie “ Numerical Experimental Studies of Double Pipe Helical Heat Exchanger “ , Department of Bioresearch Engineering , McGill University , Montreal , August 2004