CFD analysis of single phase flows inside vertically and horizontally oriented helically coiled tubes

THIS THESIS IS SUBMITTED IN THE PARTIAL FULFILMENT OF THE REQUIREMENT FOR THE DEGREE OF BACHELOR OF TECHNOLOGY

IN

MECHANICAL ENGINEERING

BY

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NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA 2013

NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA

CERTIFICATE

This is to certify that the thesis entitled "CFD analysis of single phase flows inside vertically and horizontally oriented helically coiled tubes" submitted by Sandeep Sethi (109ME0419) in the partial fulfillment of the requirements for the award of BACHELOR OF TECHNOLOGY Degree in Mechanical Engineering at the National Institute of Technology, Rourkela (Deemed University) is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any degree or diploma

Date: 8th MAY, 2013

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ACKNOWLEDGEMWNT

I express my sincere gratitude to Prof. A.K. Satpathy, Department of Mechanical Engineering, NIT Rourkela for giving me an opportunity to work on this project and allowing me access to valuable facilities in the department

I avail this opportunity to express my indebtedness to my guide Prof. A.K. Satpathy Department of Mechanical Engineering, NIT Rourkela, for his valuable guidance, constant encouragement and kind help at various stages for the facilities and diddertation work.

I am also grateful to Ms. Pooja Jhunjhunwala, M. Tech Student, Department of Mechanical Engineering, National Institute of Technology Rourkela for helping me throughout the project and providing me with information and support as and when required

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ABSTRACT:-

It has been well established from previous experimental and numerical work that heat and mass transfer in a helical pipe is higher than that in a corresponding straight pipe.The detailed description of fluid flow and heat transfer inside helical coil is not available from the present literature. This paper clearly shows the variation of average Nusselt number and friction factor with the geometric variables of a vertically and horizontally oriented helical coil for constant wall temperature boundary condition .A comparison of heat transfer and head loss in helical pipe is also discussed for vertical and horizontal orientation. The effect of inlet velocity on heat transfer coefficient and average nusselt number is also described. CFD simulations are carried out for vertically and horizontally oriented helical coils by varying different geometric parameters such as (i) pitch circle diameter, (ii) helical tube pitch and (iii) pipe diameter and their influence on heat transfer and fluid flow has been analysied. After investigate the influence of these parameters, the correlations for average nusselt number and friction factor are developed.

Keywords

- Computational fluid dynamics (CFD);
- Helical coil;
- Fluid mechanics;
- Heat transfer;
- Numerical analysis;
- Mathematical modeling

CONTENTS:

LISTS OF FIGURE

CHAPTER 1 :- INTRODUCTION :-

Curved tubes, one of the heat transfer enhanced technique are adopted in industries due to compact structure, high heat and mass transfer coefficient and simplicity in manufacturing. It has been widely employed that heat transfer rates in helical coils are higher with compare to straight pipe. The predominant reason for that the formation of secondary flow which is imposed to primary flow, known as dean vortex. helical coil heat exchangers are widely used in industrial applications such as power engineering, refrigeration, food industry, thermal processing plants, heat recovery system, electrochemical plants, piping system, etc.

The paper is organized as follows: we begin with the introduction of research work followed by describing fluid flow in curved tubes and critical Reynolds number. The widely literature review of numerical, experimental and computational techniques is investigated. further, the variation of average heat transfer coefficient and darcy friction factor are plotted with pipe coil diameter, coil diameter and coil pitch for vertically and horizontally oriented coils. A brief idea of head loss in helical pipe is also given for different orientation of pipe. The correlation are developed for average nusselt number and friction factor using CFD simluations.

CHAPTER 2 :- CHARACTERISTICS OF HELICAL COIL :-

In the present simulations, we analysis helical coils which are vertically and horizontally oriented. Fig. 1 gives the schematic diagram of the helical coil in which the helical coil axis is vertically oriented. The helical pipe has an inner diameter 2r. The coil diameter of helical pipe (measured between the centres of the pipes) is represented by $2R_c$. The distance between two adjacent turns, is called pitch that is represented by H. The coil diameter is also called as pitch circle diameter (PCD). The ratio of pipe diameter to coil diameter (r/Rc) is called curvature ratio which is represented by δ. The ratio of pitch to developed length of one turn ($H/2\pi R_c$) is termed non-dimensional pitch, λ. Consider the projection of the coil on a plane passing through the axis of the coil. The angle, which projection of one turn of the coil makes with a plane perpendicular to the axis, is called the helix angle, α. For any cross section of the helical pipe, created by a plane passing through the coil axis, the nearest side of pipe wall of coil axis is termed as inner side and the farthest side is termed as outer side.
Similar to Reynolds number for fluid flow in straight pipes, Dean number is used to characterise the fluid flow in a helical pipe.

Most of the researchers have determined that a remarkably complex fluid particle flow pattern exists inside a helical pipe due to which the enhancement in heat transfer characteristics is obtained. The curvature of the coil subjects to the centrifugal force while the pitch, H (or helix angle α) governs the torsion to which the fluid flow is subjected to. The centrifugal force develops secondary flow Due to the curvature effect, the fluid stream lines in the outer side of the pipe moves faster than the fluid streams in the inner side of the pipe. The difference in velocity flow and direction in secondary flows, whose pattern changes with the Dean number of the flow.

De= Re $\sqrt{\frac{r}{Rc}}$

Where Re is the Reynolds number Re=$\frac{2ru\rho}{\mu}$

Here r is pipr radius, R_c is the pitch coil radius, u is the mean velocity, ρ is the density of the water, μ is the viscosity of the flowing fluid, as in our respective case its hot water.

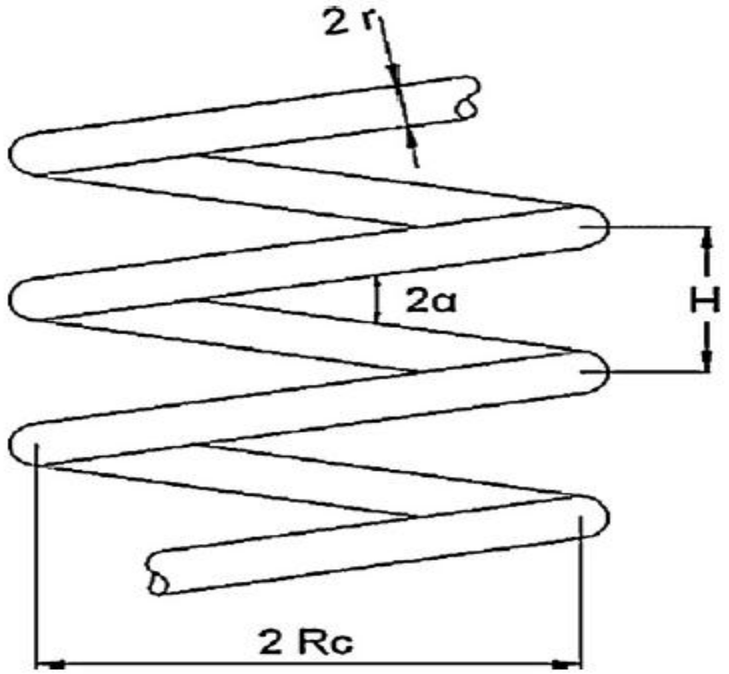


Fig. 1. Basic geometry of a helical pipe.

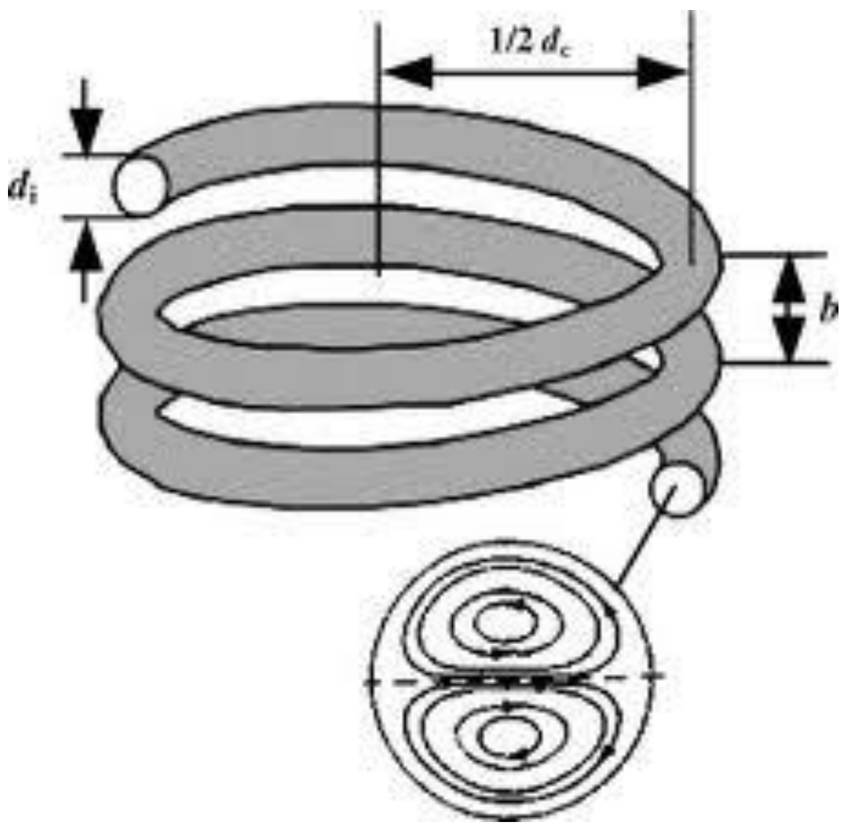


Figure 1 :- Secondary flow in helical coil.

CHAPTER 3 :- LITERATURE REVIEW:-

Berger, Talbot, Yao (1983) reviewed first heat transfer and fluid flow characteristics through a curved tube and further it is reviewed by Shah and Joshi (1987). The latest review of fluid flow, heat transfer and dimensionless heat transfer numbers is investigated by J.S. jayakumar, S.M. Mahajani, J.C. Mandal, Kannan N. Iyer and P.K. Vijayan. Most of the heat transfer and fluid flow are investigated with constant wall and constant wall flux boundary condition. The condition of constant wall temperature is suitable in heat exchangers with phase change such as condensers and cooling tower. The situation of constant wall flux boundary condition is idealized in the study of nuclear fuel elements and electrically heated tubes.

Seban and McLaughlin (1963) experimentally investigated heat transfer in helical coil both laminar and turbulent modeling for flow of water as a flowing fluid for constant wall flux boundary condition. The range of Reynolds number was maintained 6000 to 65000 and prandtl number variation was from 2.9 to 5.7. the curvature ratio of that coils were maintained .0096 to .0588. the author also stated that the assumption were considered in experiments leads to ±10% error.

Correlations were developed for the estimation of nusselt number for steady state and pulsating turbulent flow through the helical coils in the range of Reynolds number from 6000 to 18000 by Guo, Chen, Feng and Bai. The main disadvantage of the above research was that this correlation is only applicable for their set up. It is not suitable for varying curvature ratio.

Tzu-Hsiang Ko numerically investigated laminar forced convection and entropy generation in a helical coil with constant wall heat flux. This analysis is valid for a range of Reynolds no. from 1,000 to 7,500 and wall fluxes of 160,320 and 640. The development of fluid flow, secondary flow motion, distribution of temperature profile, nusselt number and friction factor were discussed. It is observed from fluid flow field analysis that there is a rapid drop of the circumferential average nusselt number and friction factor near the coil entrance, and their magnitudes develop to a constant very quickly after a mild oscillation.

Monisha Mridha Mandal and K.D.P. NigamI (2009) experimentally studied on Pressure Drop and heat transfer of turbulent flow in tube in tube helical heat exchanger at the pilot plant scale to analysis and distribution of the fluid flow and heat transfer under turbulent flow condition. The experiments were carried out with hot compressed air in the inner tube and cooling water in the outer tube in the countercurrent mode of operation. The flow rate of compressed air flowing in the inner tube was varied for Reynolds numbers from 14,000 to 86,000 and pressure of compressed air was varied from 10 to 30 kgf/cm2. On the basis of the experimental measurements, new correlations for friction factor and nusselt number in the inner tube of the heat exchanger were developed with deviation of ±4.6 to ±5%.

Rahul Kharat, Nitin Bhardwaj*, R.S. Jha have developed a correlation of heat transfer coefficient for helical coil heat exchanger to take into account of experimental and CFD results of different functional dependent variables such as gap between the concentric coil, tube diameter and coil diameter which strongly effects the heat treansfer within error band of 3-4%. The heat transfer coefficient is validated for a wide range of Reynolds number from 20,000 and 1,50,000 and specific ratio is from 0.55 to 2.25 that covers the most engineering helical coil heat exchanger applications.

S S Pawar, Vivek K Sunnapawar and B A Mujawar(2011) critically reviewed experimental and computational fluid dynamics work of heat transfer through coils of circular cross section in terms of dimensionless number, their validity, and effect of geometry, friction factor, different coil curvature ratios, fluid types, laminar and turbulent flow on heat transfer rates. This review indicates that there is a need of analyzing dynamic similarities amongst the geometrical similarities on large scale models covering industrial applications. Further research is required to be conducted at large scale on considerable range of curvature ratio, low range of prendtl number and reylnold number, temperature etc. to consider these parameter and geometry in order to address scalability issues, applicable to industries.

M. M. ABO ELAZM1, A. M. RAGHEB1,*, A. F. ELSAFTY2, M. A. TEAMAH numerically investigated and studied the heat transfer enhancement in helical cone coils over ordinary helical coils using mathematical modeling and computational fluid dynamics simulation. The simulation done in this paper indicates that nusselt number greatly influenced by taper angle of coil, curvature ratio and dean number.

Heat transfer in helical coils has been both experimentally and computational fluid dynamically investigated in helical coil heat exchanger by S.D.SANCHETI DR.P.R.SURESH for flow of water for constant wall temperature and constant wall flux boundary condition. Variation of physical properties with temperature changes were taken into account in their research work. The variation of inner nusselt number, heat removed and overall heat transfer coefficient with dean number are remarkably observed.

S.Naga Saradaa, A.V.Sita Rama Rajua , K.Kalyani Radha carried out experiments to study the enhanced heat transfer in a horizontal circular tube using mesh inserts in turbulent region at Reynolds number range of 7,000 to 14,000 and porosity range of 99.73 to 99.98. CFD techniques were also employed to perform optimization analysis of the mesh inserts. they found the remarkably variations of temperatures, heat transfer coefficients, Nusselt number in the horizontal tube fitted with various mesh inserts.

Paisarn Naphon carried out CFD analysis to investigate the enhancement of heat transfer and flow characteristics in a spiral-coil tube. The spiral-coil tube was fabricated by bending a 8.00 mm diameter straight copper tube into a spiral-coil of five turns. The innermost and outermost diameters of the spiral-coil were 270.00 mm and 406.00 mm, respectively. Hot and cold water were used as working fluids in this research work. The k–ε standard two-equation turbulence model was subjected to simulate the turbulent flow and heat transfer characteristics in spiral coil. The main governing equations were solved by a finite volume method with an unstructured nonuniform grid system. Three-dimensional turbulent convective heat transfer in the spiral-coil tube being subjected to constant wall temperature has been studied numerically with control volume method. The results obtained from the numerical study are validated and analysied by comparing with the measured data.

B.S.V.S.R Krishna (2012) conducted experiments to study the pressure drop in helical coil with single phase flow of non-Newtonian fluid of Carboxy Methyl Cellulose (CMC). Single helical coil with five different helix angles were used in this study to identify the effect of helix angle on pressure drop. Modified Correlations were developed for predicting the frictional pressure drop in laminar and turbulent regions from the results obtained from experiments.

Recently, To estimate the heat transfer coefficient correlations have developed for single phase flow through helically coiled heat exchanger by Jayakumar, Mahajani, Mandal, Vijayan and Bhoi. There correlation were validated against the experiments which were suitable for a specific experiment configuration. The local variation of nusselt number and heat transfer coefficient were not described.

From the above literature review it is clearly shown that the effect of various geometric and kinematic parameters on heat transfer are not studied for all the operating range of dean number and curvature ratio. Mostly authors considered the fluid properties such as density, viscosity are constant with varying temperature that leads to error in numerical and CFD analysis. In the my current analysis a large range of dean number and curvature ratio are simulated with near wall treatment turbulent modeling.

CHAPTER 4 :- NATURE OF TURBULENT FLOW AND HEAT TRANSFER IN HELICAL COIL :-

Analysis and simulation of Heat transfer to water which is flowing in helical coil is carried out using ANSYS 13.0. as a representative case, coil of pipe coil diameter 200mm and coil pitch of 75mm is taken for our discussion. For this particular case the coil diameter of pipe is kept as 30mm. the solid geometry and meshing is generated using ANSYS13.0.the heat transfer analysis is employed the CFD package of Fluent associated with ANSYS 13.0. in meshing the smoothing and transition are kept as medium and slow.in order to taking consideration of near wall treatment inflation layer option is used to make Y PLUS value in specific range correspond to particular turbulent flow model. The number of inflation layers are taken as 2 using pre inflation algorithm. The number of nodes and elements for the current case are 33372 and 26531 which are generated by ANSYS13.0 mesh launcher. Mesh grid is chosen such as it follows the near wall treatment condition for proper heat transfer results.

The meshed grid is subjected to following fluent launcher condition. The processing option is chosen as serially with double precision. The fluent solver is pressure based and velocity formulation is taken as absolute with steady time condition. To analysis the heat transfer phenomena in helical coil the standard k-epsilon model is used and for near wall treatment, standard wall functions are used. The flowing fluid is chosen as water in our respective analysis. The wall material is kept copper which is widely used in mostly engineering applications. For our entire analysis of helical coil the turbulent kinetic energy and turbulent dissipation rate is subjected to unity.

Pressure velocity coupling is chosen as simple scheme. Second order upwind equations are used for momentum and energy equation. This equation is also applied for turbulent kinetic energy and turbulent dissipation rate. In our all respective cases to investigate the heat transfer phenomena standard initialization method is used for in initialization the computational simulation. Reference frame is selected relative to the cell zone.

The fluid properties are taken as constant with respect to the varying temperature. But in the real world fluid properties such as density, viscosity etc. varies with the temperature. So this leads to error in the calculation of heat transfer and nusselt number.

The standard k-epsilon model is selected to analyze the heat transfer and contour presentation. This model does not perform very well in the case of large adverse pressure gradients. It is basically two equation model. The first variable is turbulent kinetic energy k. it describes the energy phenomena in the case of turbulence. The second transported variable is the turbulent dissipation ε which determines the scale concentration in turbulence model. This model is well suitable for flows including boundary layers with strong adverse pressure gradient, rotation, separation and recirculation.

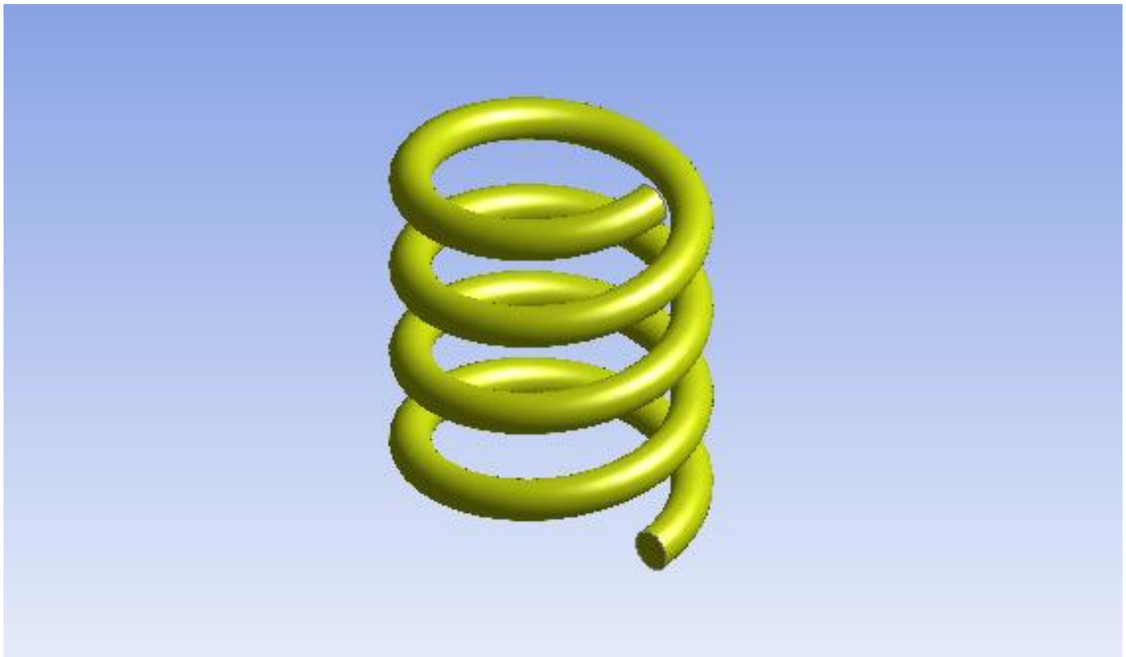
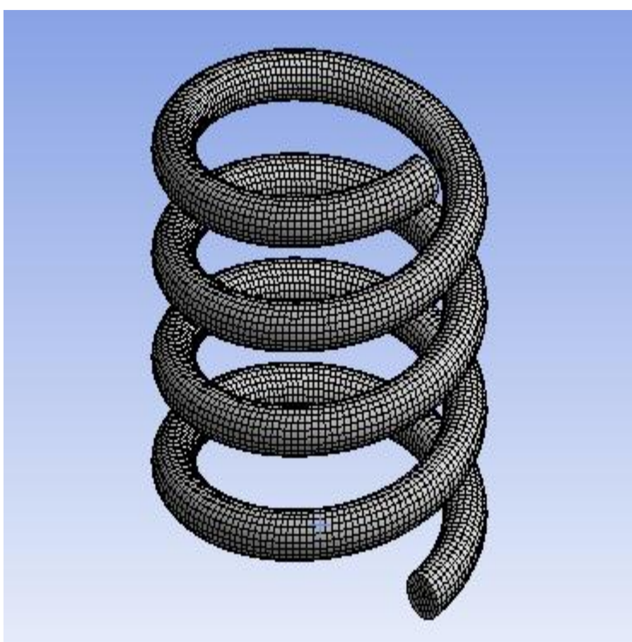


Figure 2 :- Solid model of helical coil generated by ANSYS13.0 Geometry launcher.



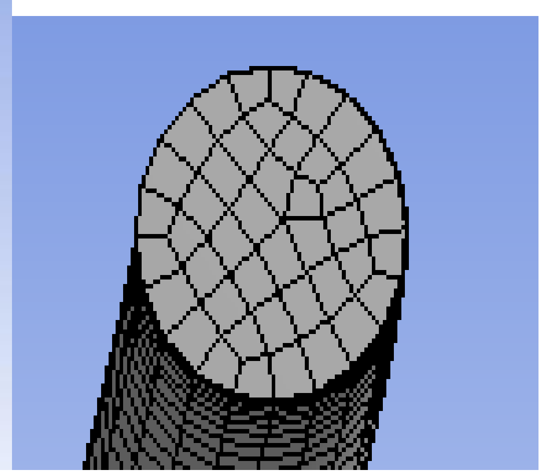


Figure 3 Grid mesh of helical coil

figure 4 cross section of mesh at any plane.

CHAPTER 5 :- Analysis and formulation:-

To analyze heat transfer and CFD simulations different heat and mass transfer formulas are used. In all simulations, heat transfer coefficient and nusselt no. are studied to predict the heat transfer rate behavior of helical coil. The heat transfer coefficient and nusselt number for a single phase fluid flow is given as below:-

$$h = \frac{q}{Tw - Tf}$$

$$Nu = \frac{hd}{k}$$

Where q=heat flux per unit area, Tw=wall temperature, Tf is the average fluid temperature during heat simulation, h is the heat transfer coefficient, Nu is dimensionless nusselt number, d is the inner pipe diameter of helical coil in meter, k is thermal conductivity of fluid as in our case it is considered as water.

The length of the helical coil is given as

$L=n[p^2+(2\pi R)^2]^{1/2}$

Here n is the number of turns in a helical coil, R is the pitch coil radius, p is the pitch and L is the length of the coil.

To analyze the head loss in helical coil fiction factor is calculated. Normally in all the industries it is designed on the basis of experiments. So the fanning friction factor is described as

$$f = \frac{(Pout - Pin)d}{2l\rho v * v}$$

$$h = \frac{f\ l\ V * V}{d\ 2g}$$

Here Pout and Pin are the pressure at the outlet and inlet of the coil. d is the pipe diameter in SI units. L is the length of the coil. ρ is the density of the water which is kept is constant in our all respective. v is the mean velocity of the water.

To simulate the heat transfer phenomena the k epsilon model is taken into consideration.it is the most common turbulence model which is widely used in power and thermal industries. The parameter k, turbulent kinetic energy shows the energy in the turbulence. The second parameter ε, turbulence dissipation that describes the degree of turbulence in fluid flow. Transport equation for standard k-epsilon model.

For turbulent kinetic energy k

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+P_k+P_b-\rho\epsilon-Y_M+S_k$$

For dissipation ϵ

$$\frac{\partial}{\partial t}(\rho\epsilon)+\frac{\partial}{\partial x_i}(\rho\epsilon u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x_j}\right]+C_{1\epsilon}\frac{\epsilon}{k}\left(P_k+C_{3\epsilon}P_b\right)-C_{2\epsilon}\rho\frac{\epsilon^2}{k}+S_\epsilon$$

Navier stokes equations describes the motion of the fluid control volume. These equation are derived from the newton’s second law of fluid motion. A simplification of the resulting flow of a Newtonian fluid is employed with considering incompressible fluid flow. But this have very less practical interest as the industrial point of view because the density of the fluid varies with the temperature and other thermodynamic terms. The simplified navier stokes equation for incompressible fluid flow is described as

Navier – Stockes (momentum) equations:

$$\rho\left(\frac{\partial u_r}{\partial t}+u_r\frac{\partial u_r}{\partial r}+\frac{u_\theta}{r}\frac{\partial u_r}{\partial \theta}+u_z\frac{\partial u_r}{\partial z}-\frac{u_\theta^2}{r}\right)=-\frac{\partial p}{\partial r}+\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_r}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 u_r}{\partial \theta^2}+\frac{\partial^2 u_r}{\partial z^2}-\frac{u_r}{r^2}-\frac{2}{r^2}\frac{\partial u_\theta}{\partial \theta}\right)+\rho g_r$$

$$\rho\left(\frac{\partial u_\theta}{\partial t}+u_r\frac{\partial u_r}{\partial r}+\frac{u_\theta}{r}\frac{\partial u_\theta}{\partial \theta}+u_z\frac{\partial u_\theta}{\partial z}-\frac{u_r u_\theta}{r}\right)=-\frac{1}{r}\frac{\partial p}{\partial \theta}+\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_\theta}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 u_\theta}{\partial \theta^2}+\frac{\partial^2 u_\theta}{\partial z^2}-\frac{u_\theta}{r^2}-\frac{2}{r^2}\frac{\partial u_\theta}{\partial \theta}\right)+\rho g_\theta$$

$$\rho\left(\frac{\partial u_z}{\partial t}+u_r\frac{\partial u_z}{\partial r}+\frac{u_\theta}{r}\frac{\partial u_z}{\partial \theta}+u_z\frac{\partial u_z}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_z}{\partial r}\right)+\frac{1}{r^2}\frac{\partial^2 u_z}{\partial \theta^2}+\frac{\partial^2 u_z}{\partial z^2}\right)+\rho g_z$$

The continuity equation:

$$\frac{\partial \rho}{\partial t}+\frac{1}{r}\frac{\partial}{\partial r}(\rho r u_r)+\frac{1}{r}\frac{\partial(\rho u_\theta)}{\partial \theta}+\frac{\partial(\rho u_z)}{\partial z}=0$$

CHAPTER 6 :- ANALYSIS OF THERMAL BEHAVIOR OF CONTOURS:-

In the present case hot water at 340k temperature and 1m/sec. velocity is entering into the helical coil at the top. The wall of the helical coil is maintained at 300k temperature to carried out the heat transfer phenomena between hot fluid and wall. In the respective analysis the inlet boundary condition as specified as the form of inlet velocity. The fluid is cooled as it flows along the tube. In all the simulations fluid properties such as velocity, density etc. are considered as constant with temperature.

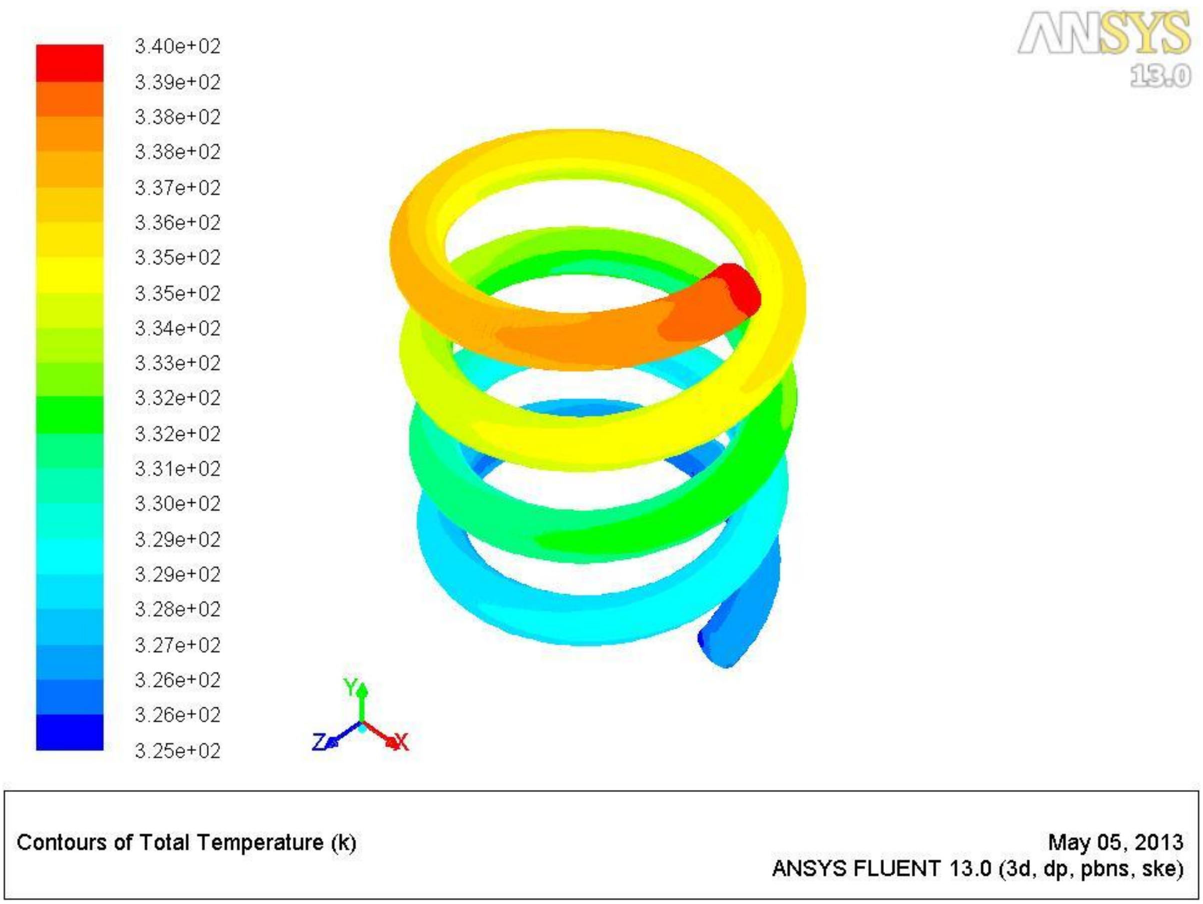


Figure number 5

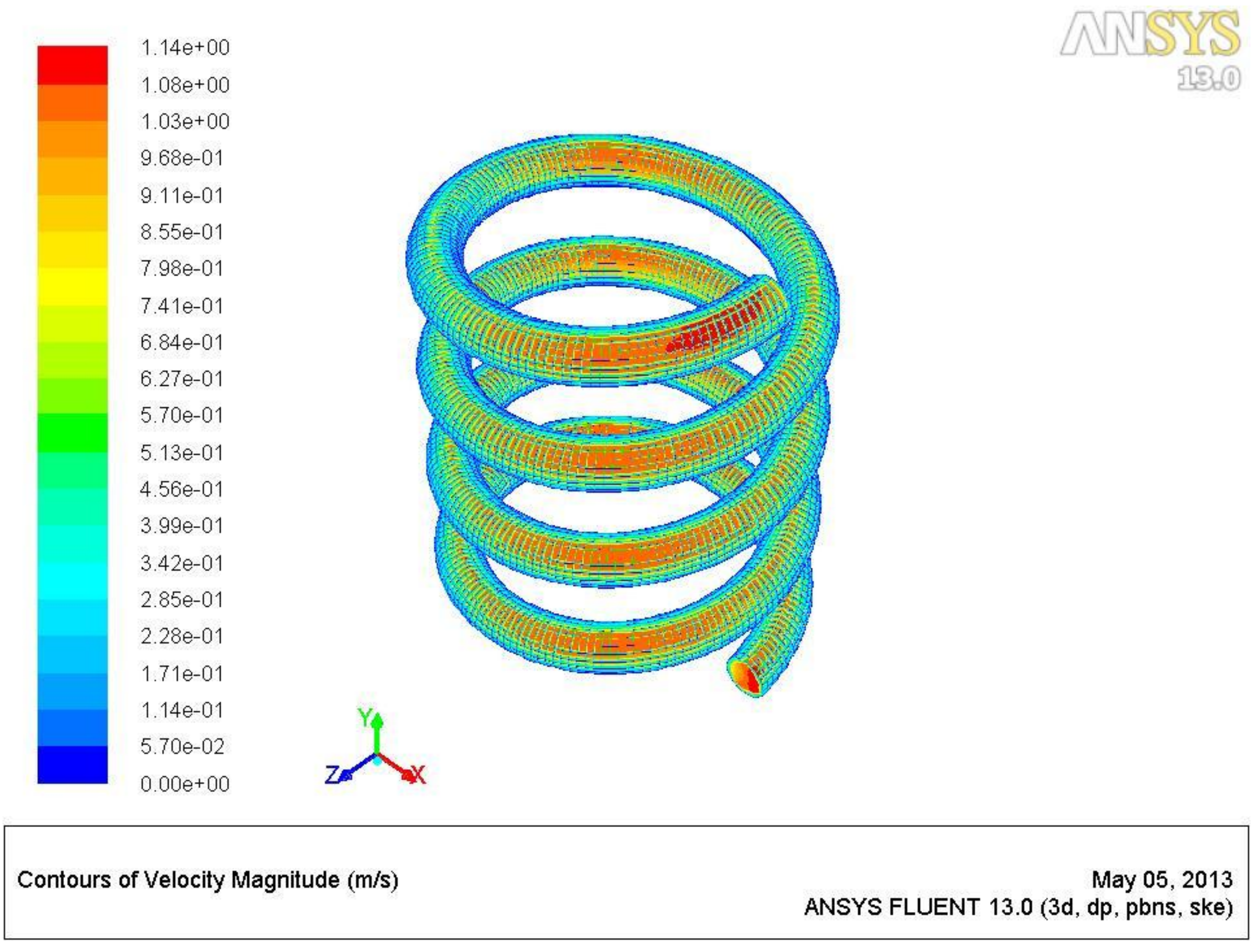


Figure number 6

Figure number 4 shows the temperature distribution throughout the helical coil. The pressure at the outer layer is more than the inner side so, secondary flow comes into picture. The temperature at the outer side is more than the inner side it can be justified at the first coil of helical tube. Temperature falls down from 335k to 329 k during the 2nd and 3rd turn of the tube. From the contour it is observed that the heat transfer rate is found in the first turn of the tube. The figure number 5 shows the average velocity distribution throughout the fluid cell zone. Due to the adverse pressure gradient the velocity near the wall is much higher than the internal side of the tube. It can be easily understood from the contour representation that the mean velocity remains constant throughout the fluid flow.

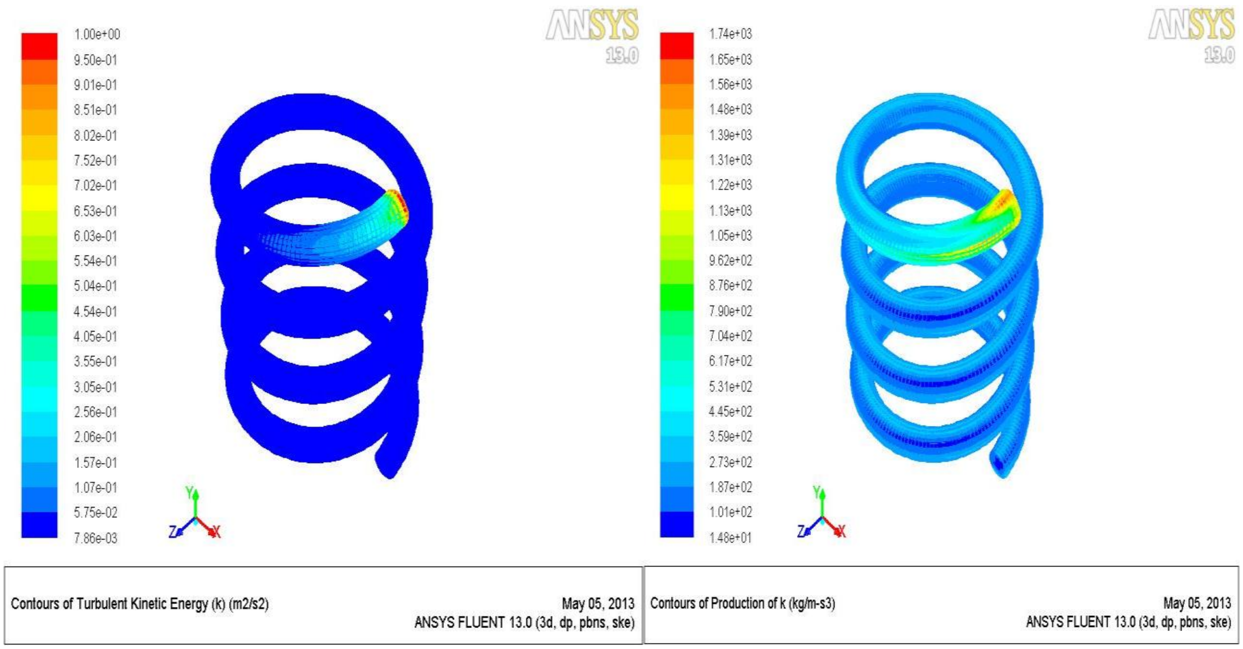


Figure 7

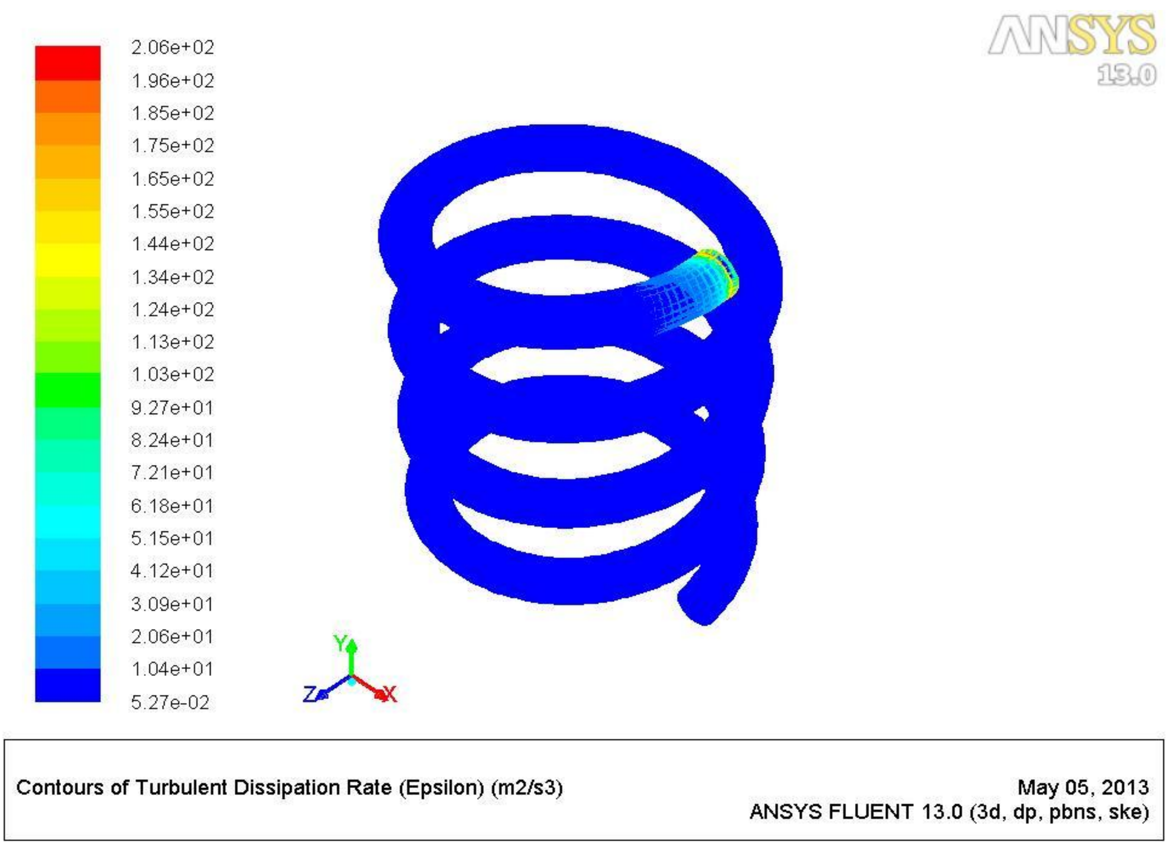


Figure 8

All the simulations are carried out using standard k-epsilon turbulence model. The parameter k, turbulence kinetic energy represents the mean kinetic energy per unit mass associated with eddies in turbulent fluid flow. Figure 6 represents the contours of the turbulence kinetic energy and production of k. it is observed that near the entering region the turbulence kinetic energy is more with compare to rest of the tube. The rate of production of turbulence kinetic energy is also much more at the entering region after that the production rate of mean turbulence kinetic energy is observed as constant due to fully developed fluid flow.

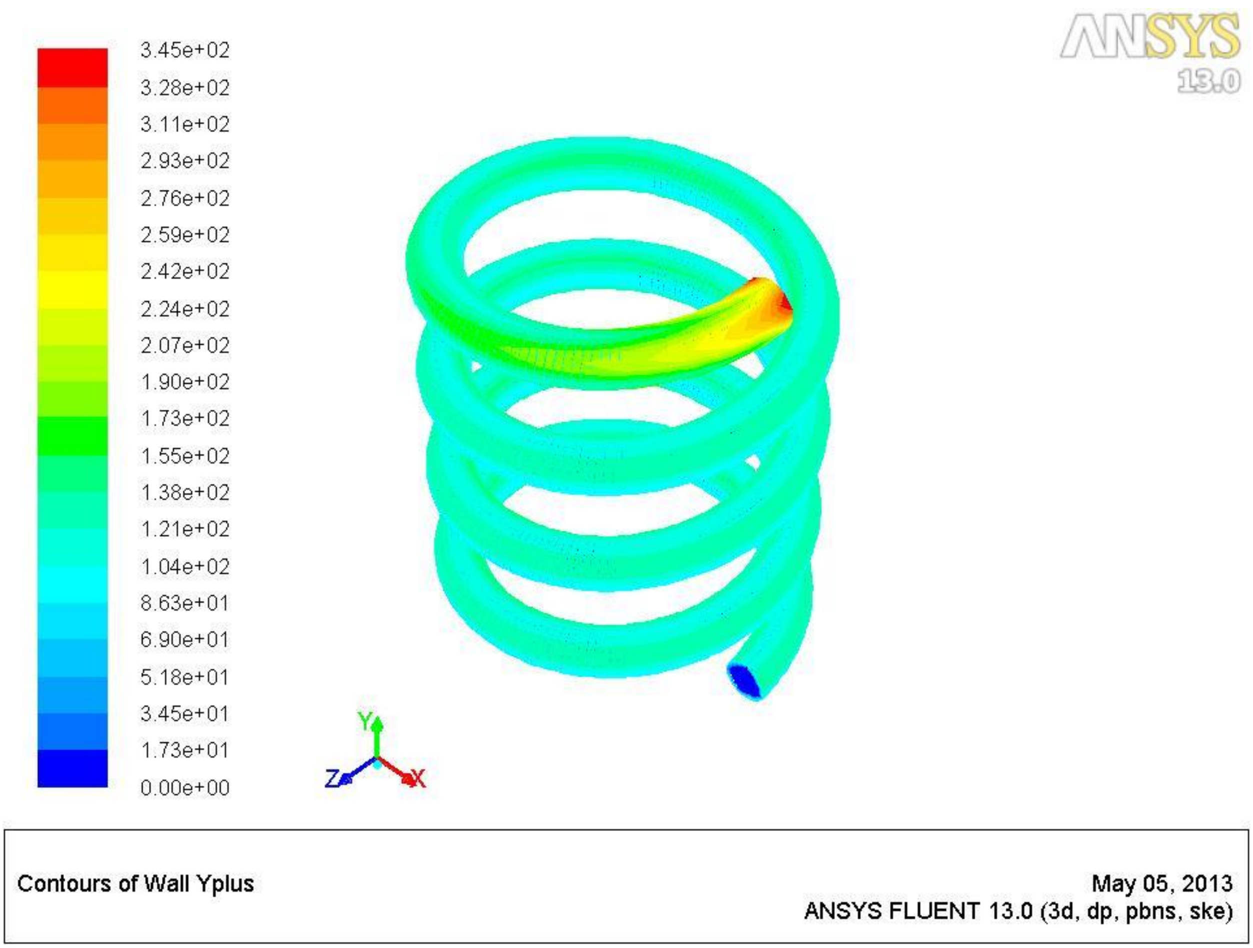


Figure 9

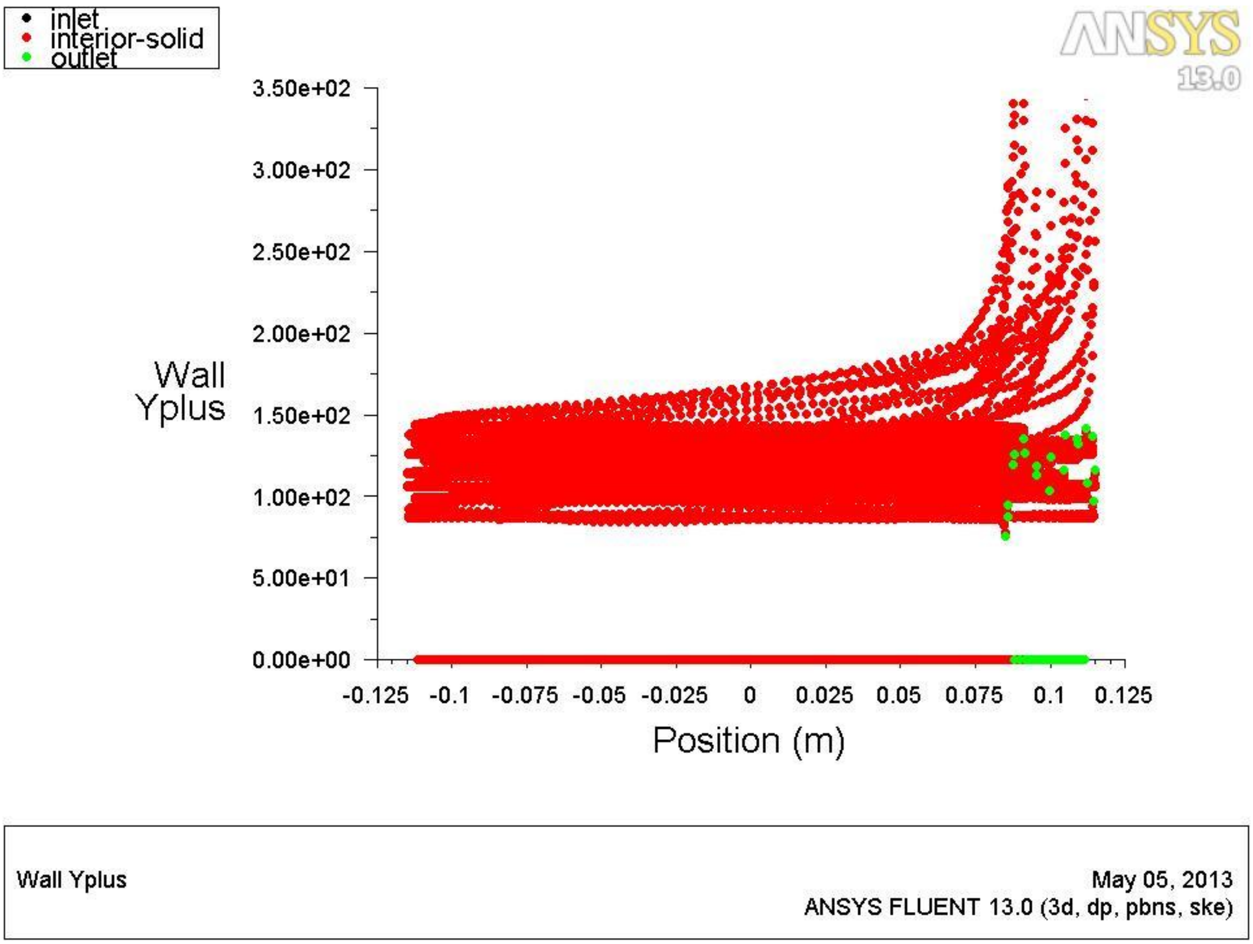


Figure 10

Y PLUS is a non dimensionless distance. It is mostly used to describe how coarse or fine mesh of a particular fluid flow. It is very important to determine the Y PLUS values near the wall to satisfy the turbulence behavior of the fluid flow. Figure 8 represents the contour phenomena of the Y PLUS value and figure 9 shows the X-Y plot of the Y PLUS value with position of fluid particle. It is observed from the above defining diagram that the value of non dimensionless Y PLUS varies from 100 to 200. This condition satisfies the near wall treatment condition of turbulent modeling.

CHAPTER 7 :- <u>INFLUENEC OF KIMEMATIC AND GEOMATRIC PARAMETERS ON HEAT TRANSFER :-</u>

In our present work, CFD simulations are carried out by varying the different coil parameters and inlet boundary conditions such as (i) pitch coil diameter (ii) pipe diameter (iii) coil pitch (iv)inlet velocity of the entering fluid. various graphs of heat transfer coefficient and average nusselt number are plotted against curvature ratio and dean number. In all CFD calculations the wall temperature of helical coil is taken as constant. Subsequently the results of cfd simulations are discussed and correlations are predicated for average nusselt number and darcy friction factor. The total analysis is carried out using ANSYS 13.0 with fluent 6.3(3D, double precision). The input boundary conditions are declared with reference to cell zone. The standard k-epsilon model with standard wall function is used to compute the different field parameters such as wall flux, total temperature of cell zone, inlet and outlet pressure etc. wall material is employed as copper from fluent database with roughness height 0 meter. All the readings are taken by windows 7 processor intel core i3 with 3 GB Ram.

<u>Analysis with constant wall temperature boundary condition:-</u>

This is the most idealized condition which is widely used in condensers and boilers in power generation and refrigeration. The details of discretization scheme and other fluent parameter are already discussed.

7.1 Influence of pitch circle diameter for vertically oriented helical coil:-

In vertically oriented helical coil the hot water flows form top to bottom at 340 K. the coils with PCDs 25mm, 30mm,40mm, 50mm, 100mm, 200mm, 300mm 400mm, 500mm, 600mm were analysed for inlet velocity boundary condition 1m/sec. and 1.2 m/sec. in all the simulations the pipe diameter and coil pitch were kept as 15mm and 75mm.

The effect of PCDs directly influences the heat transfer coefficient. The variation of PCDs strongly effects the centrifugal force subjected to moving fluid element in helical coil. As the value of PCDs rises up, the centrifugal force acted on fluid element plays a lesser role. this will in turn effects the secondary flow and coil curvature effects comes down. The degree of straightness also increases with rising PCDs.

A comparison of the average nusselt nember and heat transfer coefficient is shown in table 1 for different PCDs. The analysis is done for two differ velocity value as 1 m/sec. and 1.2 m/sec. the nusselt number, which is employed here is the average nuesslt number for fully developed region. From table no. 1 as the value of PCDs increases, the data of average nusselt number and heat transfer coefficient corresponding to that comes down. All the simulation are verified against dimensionless number Y PLUS value subjected to near wall treatment.

Average values of nusselt number for different PCDs

D	Δ	h for v=1m/sec.	Nu for v=1m/sec.	Y PLUS for v=1m/sec.	h for v=1.2m/sec.	Nu for v=1.2m/sec.	Y PLUS for v=1.2m/sec.
25	.6	10213.37	255.33	68	116672.215	291.68	74
30	.5	9100.57	227.15	65	10550.69	227.15	76
40	.375	7869.98	196.74	64	9080.51	196.74	75
50	.3	6880.47	172	66	8026.337	200	73
100	.15	5743.51	143.58	58	6588.7365	164.71	67
200	.075	5276.58	131.968	65	6123.4728	153.08	77
300	.05	5129.8	128.23	65	5950.7648	148.7961	76
400	.0375	5045.8415	126.146	64	5857.3735	146.4343	75
500	.03	5045.735	126.143	64	5825.8076	145.6451	72
600	.025	4928.88	123.220	61	5738.088	143.45221	72

Table number 1

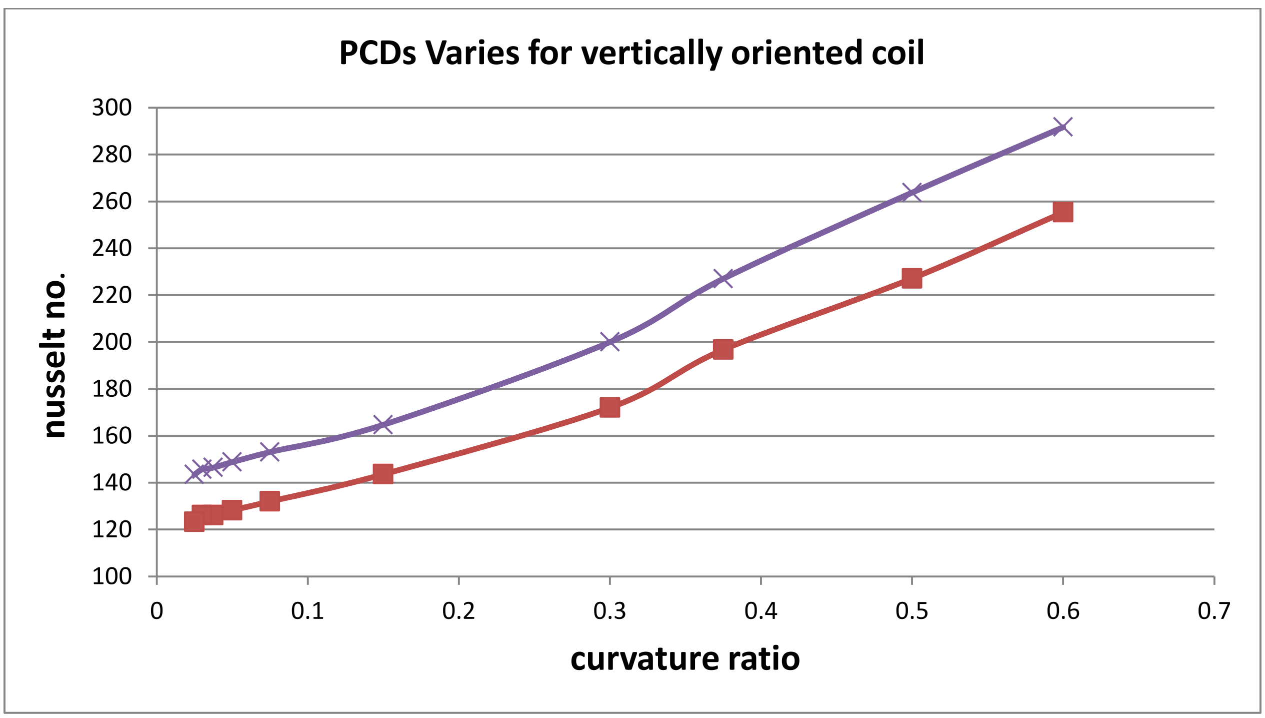


Figure no. 11

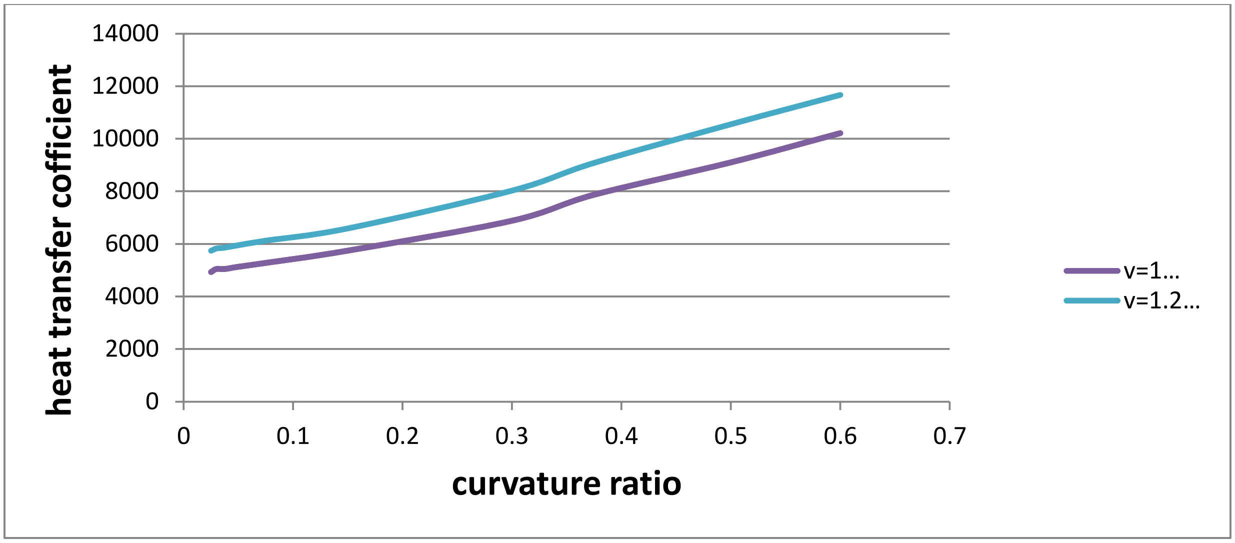


Figure no. 12

From the figure no. 10 and 11 it is clearly shown that heat transfer coefficient and average nusselt number rises with curvature ratio. And the slope of the above graphs does not vary with varying velocity. All the Y PLUS value lies between30 to 300, which verifies the near wall treatment for standard k-epsilon model.

To determine the correlation of average nusselt number with curvature ratio with varying PCDs is used. The general form of the correlation is

$Nu = c\,(\delta)^{x}$

Figure 12 shows a plot of ln(δ) and ln(Nu). The equation is drawn as a continue line. Regression analysis is made to fit a straight line using Microsoft excel 2012. From the regression analysis using excel average nusselt number is correlated to curvature ratio as

$Nu = 190.3377(\delta)^{.1256}$

This can be observed that as the PCDs increases the nusselt number approaches correspond to straight tube.

7.2 Influence of coil pitch for vertically oriented coil:-

In this analysis the helical coil with PCD of 200mm and inner pipe diameter 20mm is considered. The CFD simulations are carried out for two different velocity conditions such as 1.2 m/sec. and 1.4 m/sec. the cases are analysed with pitch 20mm, 40mm, 60mm, 80mm. the direction of flowing water is considered from top to downward.

As we increases the coil pitch, the torsional or rotational behavior of fluid stream rises up so, the turbulent intensity of fluid flow also comes up with increasing turbulent behavior of fluid streams. The rate of forced convection also increases with rising coil pitch. This will bring the result of high heat transfer coefficient and high average nusselt number.

The different simulations are shown into table 2 with different coil pitch. Heat transfer coefficient and average nusselt number are calculated for varying inlet velocity values. The variation of heat transfer coefficient and average nusselt number with coil pitch are shown into table 2. All the simulations are verified against dimensionless number Y PLUS, which clearly shows the validity of near wall treatment of vertically oriented helical coil.

Coil pitch	h for v=1.2 m/sec.	Nu for v=1.2 m/sec.	Y PLUS for v =1.2 m/sec.	h for v=1.4 m/sec.	Nu for v=1.4 m/sec.	Y PLUS for v=1.4 m/sec.
20	5852.9	195	96	6473.686	215.7895	110
40	5956.34	198.54	90	6742.07	224.7359	103
60	5984.92	199.49	99	6778.9162	225.9638	113
80	6018.07	200.602	105	6826.4757	227.5491	120

Table no. 2

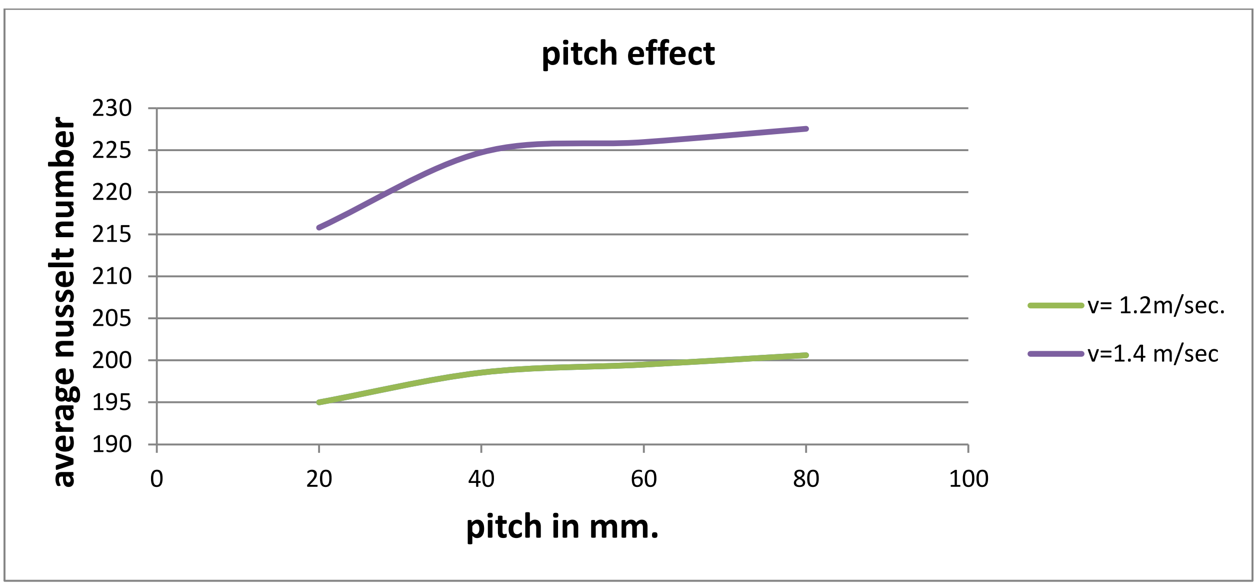


Figure no. 13

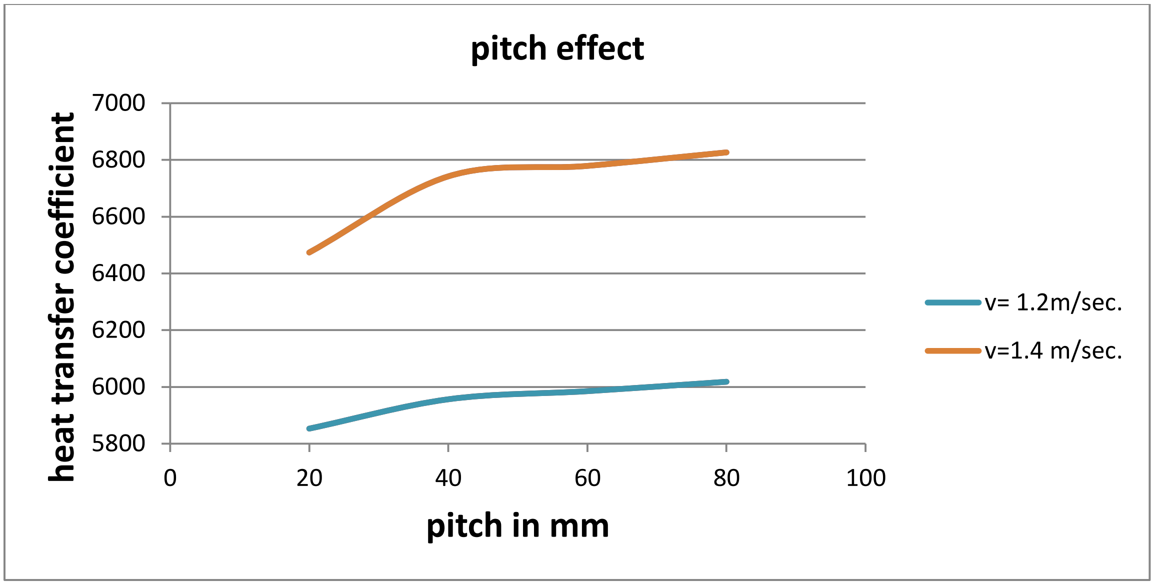


Figure no. 14

Form the figure 12 and figure 13. It is clearly investigated that average nusselt number and heat transfer coefficient marginally increases with rising coil pitch. When the coil pitch varies from 20mm to 40mm the change in nusselt number and heat transfer coefficient is significant after that when the coil pitch increases from 40mm to 80mm. the nature of graph is almost horizontal straight line. It mentions that the change in heat transfer rate at higher pitch is negligible. There is also one more observation is considered that the slope of above graphs does not vary with varying velocity. In mostly engineering applications the coil pitch is kept higher than the pipe diameter. In that range the change in $NU_{avg.}$ with pitch coil is negligible. The effect of pitch on heat transfer rate of helical coil is not considered for pratical applications of engineering problems. So there is no need to bother about pitch variation subjected to change in heat transfer rate.

7.3 Influence of pipe diameter:-

In this current analysis the effect of inner pipe diameter on heat transfer of a vertically oriented helical coil is considered. The direction of flowing hot water is kept from top to bottom side. The inlet temperature of flowing fluid is considered as 340 K. for the present analysis the pitch circle diameter and pitch are taken as 200mm and 75mm. the constant wall temperature is kept as 300 Kelwin. The cases that are analysed with inner pipe diameter are 8mm, 10mm, 15mm, 20mm, 25mm, 30mm.

As the inner pipe diameter of vertically oriented helical coil is low, at that situation the influence of secondary flow on heat transfer is low. This causes the low heat and mass transfer rate. As the inner pipe diameter increases the contact surface area of flowing fluid to wall surface comes up and the effect of secondary fluid flow also becomes predominant. In result the heat transfer rate rises up.

The different simulations are carried out with ansys 13.0 for varying inner pipe diameter in table 3. The analysis is carried out with varying velocity magnitude such as 1 m/sec, 1.4 m/sec and 1.8m/sec. table shows that increasing value of heat transfer rate and nusselt no. with inner pipe diameter. All the readings are verified against the dimensionless parameter Y PLUS. all the Y PLUS values come under 30 to 300 which validates the near wall treatment.

d	δ	h for v=1m/sec.	Nu for v=1m/sec.	Y PLUS for v=1m/sec.	h for v=1.4m/sec.	Nu for v=1.4m/sec.	Y PLUS for v=1.4m/sec.	h for v=1.8m/sec.	Nu for v=1.8m/sec.	Y PLUS for v=1.8m/sec.
8	.04	5619.28	74.92	37	7375.08	98.3344	50	9056.99	120	62
10	.05	5496.37	91.6	43	7177.8	119.63	57	8853.182	147.55	71
15	.075	5276.76	131.919	65.84	6938.144	173.4536	88	8521.98	213.04	110
20	.1	5152.271	171.742	83.92	6776.808	225.8936	112	8316.99	277.23	140
25	.125	5191.41	216.30	104.97	6807.87	283.66	140	8400.06	350	175
30	.15	5172.65	258.63	122	6793.719	339.68	164	8290.533	483	66

Table no. 3

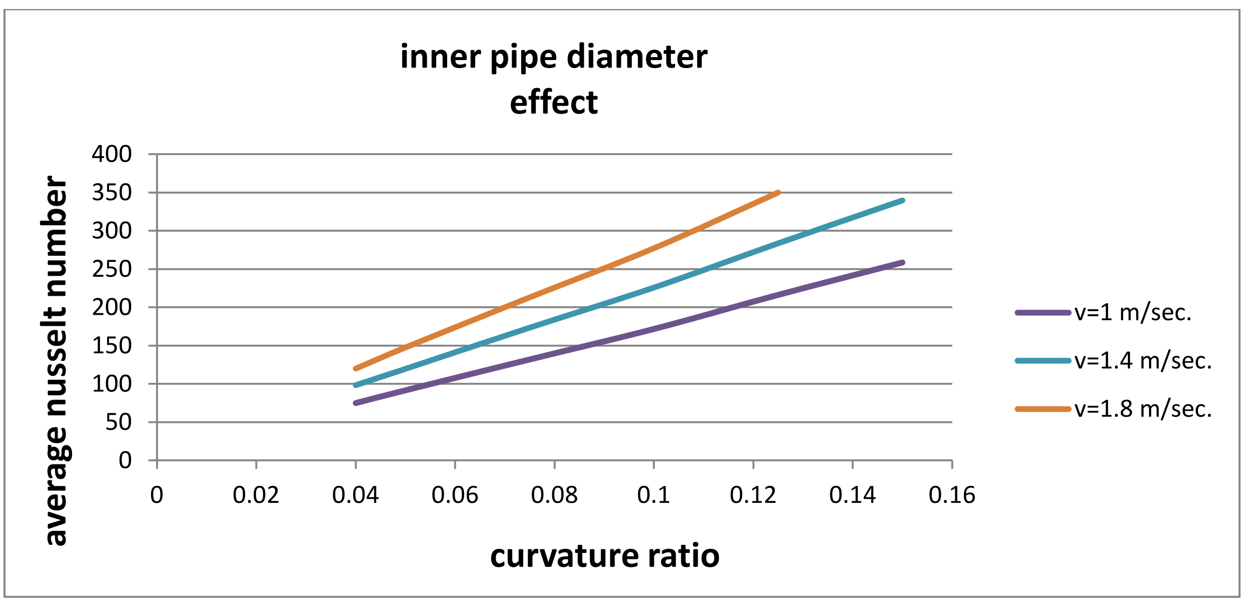


Figure no. 15

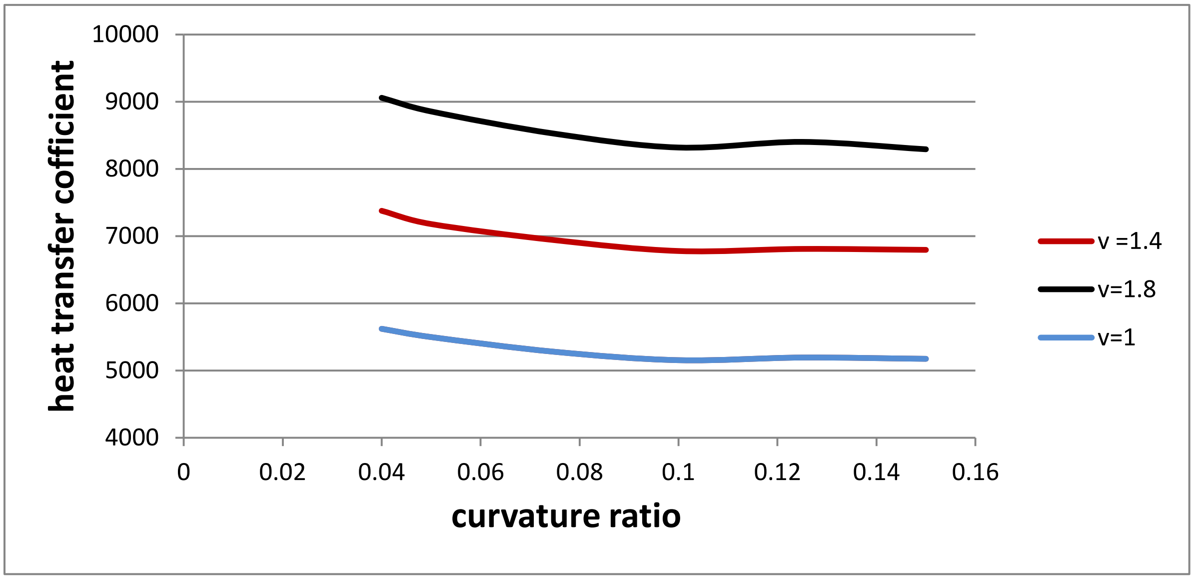


Figure no. 16

From the figure no. 14 and 15 it shows that average nusselt number and heat transfer coefficient rise with varying curvature ratio marginally. The effect of pipe diameter is strongly observed with compare to other geometric parameter. In figure no. 5 the slope of lines for different velocities are not parallel so, the slope of graph rises with inlet velocity of hot water.it is employed that to increase pipe diameter is better option to increase heat and mass transfer rate for vertically oriented helical coil.

7.4 Influence of inlet velocity of fluid on average nusselt number:-

In vertically oriented helical coil hot water at 340k flows from top to bottom. The pitch circle diameter and pipe diameter are taken as 300mm and 20mm. the present simulation is carried out with 75mm pitch by varying inlet velocity of the fluid such as .6m/sec, .8m/sec, 1.2m/sec, 1.4m/sec, 1.6m/sec, 1.8m/sec, 2.0m/sec, 2.2m/sec, 2.4m/sec. the wall of the helical coil is maintained at 300k.

The velocity is the prime phenomena to describe the motion of fluid particle whether its laminar or turbulent region. If in our present cases, the inlet velocity of the fluid rises up so, the average mass momentum flux and degree of turbulence increases. This leads up to mixing of fluid stream lines and turbulence intensity. Finally heat transfer rate and nusselt number in a helical coil rises by employing secondary flow predominantly.

In the figure 16 it is clearly represented that heat transfer rate and average nusselt number rises strongly with rising curvature ratio. The nusselt number value 338 is simulated at 2.4 m/sec. inlet velocity of fluid. This shows the predominant effect of velocity on heat transfer coefficient and heat transfer for helical coil.

Inlet velocity of fluid	Average nusslt number
.6	109.8214
.8	138.2209
1.0	165.5435
1.2	191.9162
1.4	217.5055
1.6	242.7034
1.8	267.1079
2.0	291.4893
2.2	315.34
2.4	338.5242

Table no. 4

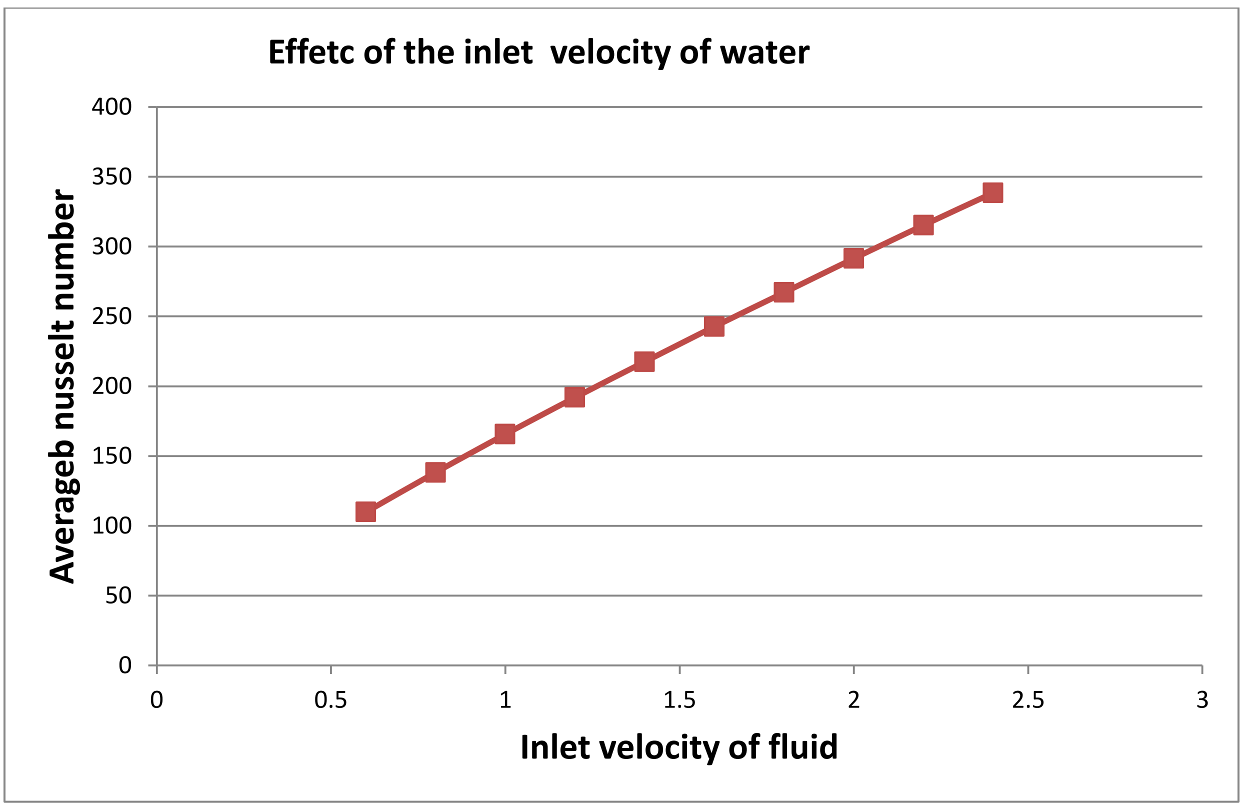


Figure 17

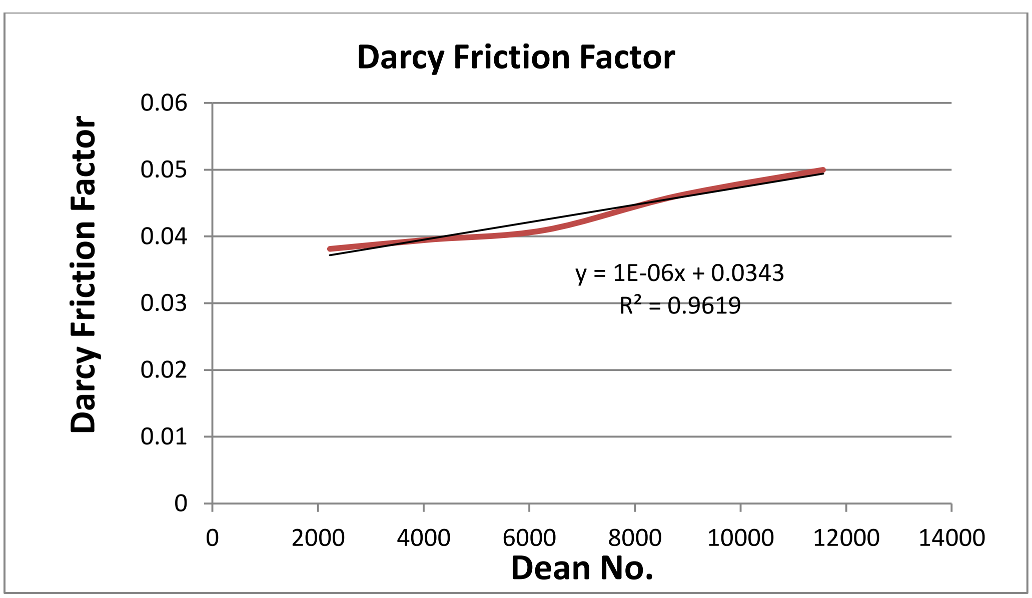
CHAPTER 8 :- DARCY FRICTION FACTOR AND HEAD LOSS IN HELICAL PIPE :-

The hot water at 340k inlet temperature and 1m/sec. inlet velocity is entering into helical oil from top face. In the present case the PCD and wall temperature were considered 200mm and 300k. the simulations are carried out with varying pipe diameter as 10mm, 15mm, 20mm, 25mm, 30mm. the number of turns of the present helical pipe are subjected to 4.

To analysis the hydrodynamic and thermal behavior of tubes it is necessary to calculate the friction factor and loss of head in the pipe. So we can estimate the power require for the pipe fluid flow. Darcy friction factor is calculated by CFD analysis with employing pressure difference phenomena. Further, loss of head in helical pipe is investigated using darcy friction factor by darcy weisbach equation.

d in mm	De	f_d	h_f in meter
10	2225.2600	0.0381411	0.4920250
15	4088.26	0.0395122	0.3398076
20	6294.2882	0.0409590	0.2641876
25	8796.5352	0.0460653	0.2376988
30	11563.3458	0.0499809	0.2149199

Table no. 5



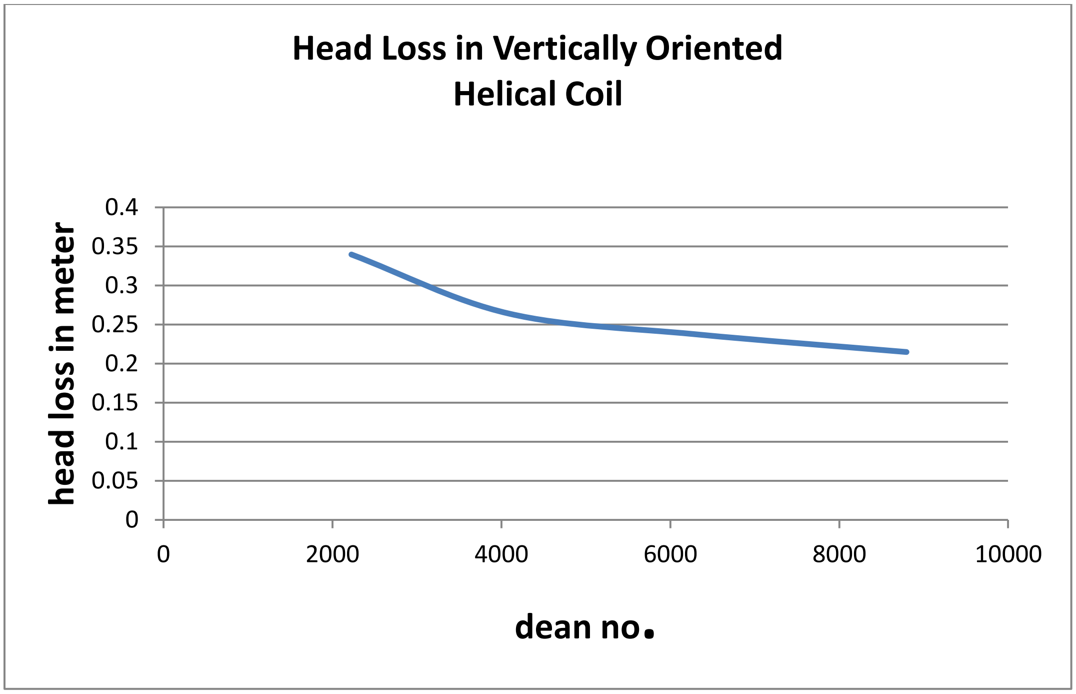


Figure 18

in figure 17 dracy friction factor and loss of head in pipe are plotted with dean number. It is observed that darcy friction factor varies with dean number linearly. When dean number rises from 2000 to 4000, head loss decreases rapidly. When it varies from the range of 4000 to 9000. Then almost linear behavior is observed. the linear variation of darcy friction factor with dean number is given below as:-

$$f_d = 10^{-6} D_e+0.0343$$

CHAPTER 9 :- COMPARSION OF AVERAGE NUSSELT NUMBER FOR VERTICALLY AND HORIZONTALLY ORIENTED HELICAL COIL TUBES :-

In this analysis the effect of different orientation on average nusselt number is observed. The pipe diameter and pitch of the helical coil are considered as 15mm and 75mm. the boundary condition of hot water is given as 340k temperature and 1m/sec. inlet velocity. The present analysis is carried out with varying pitch circle diameter as 25mm, 30mm, 40mm, 50mm. the wall of helical coil is kept at 300 k as constant temperature boundary condition.

D	δ	Horizontal orientation	Vertical orientation
		Average nusselt number.	Average nusselt number.
25	0.6	246.32	255.33
30	0.5	226.41	227.15
40	0.375	196	196.74
50	0.3	173.78	172

Table no. 6

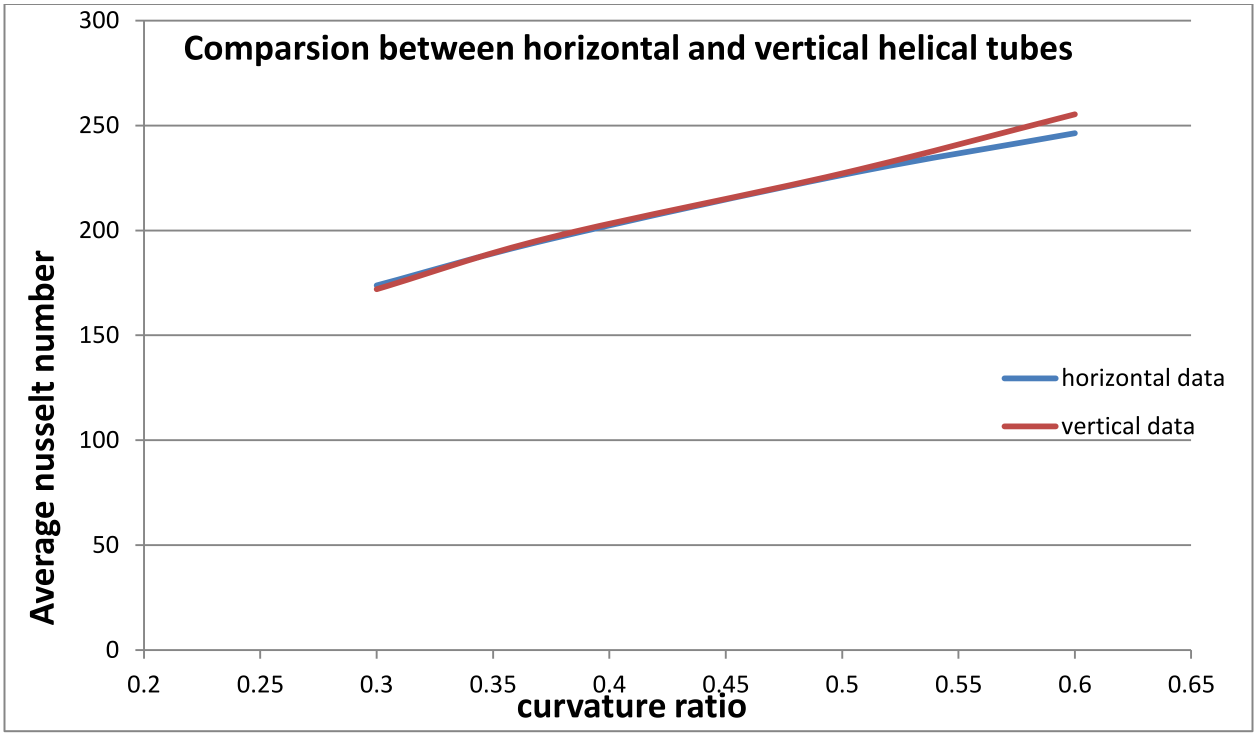


Figure 19

Direction of flow	Average Nusselt Number
Top to Bottom	131.9704
Left to Right	130.8743
Bottom to Top	131.7258

Table no. 7

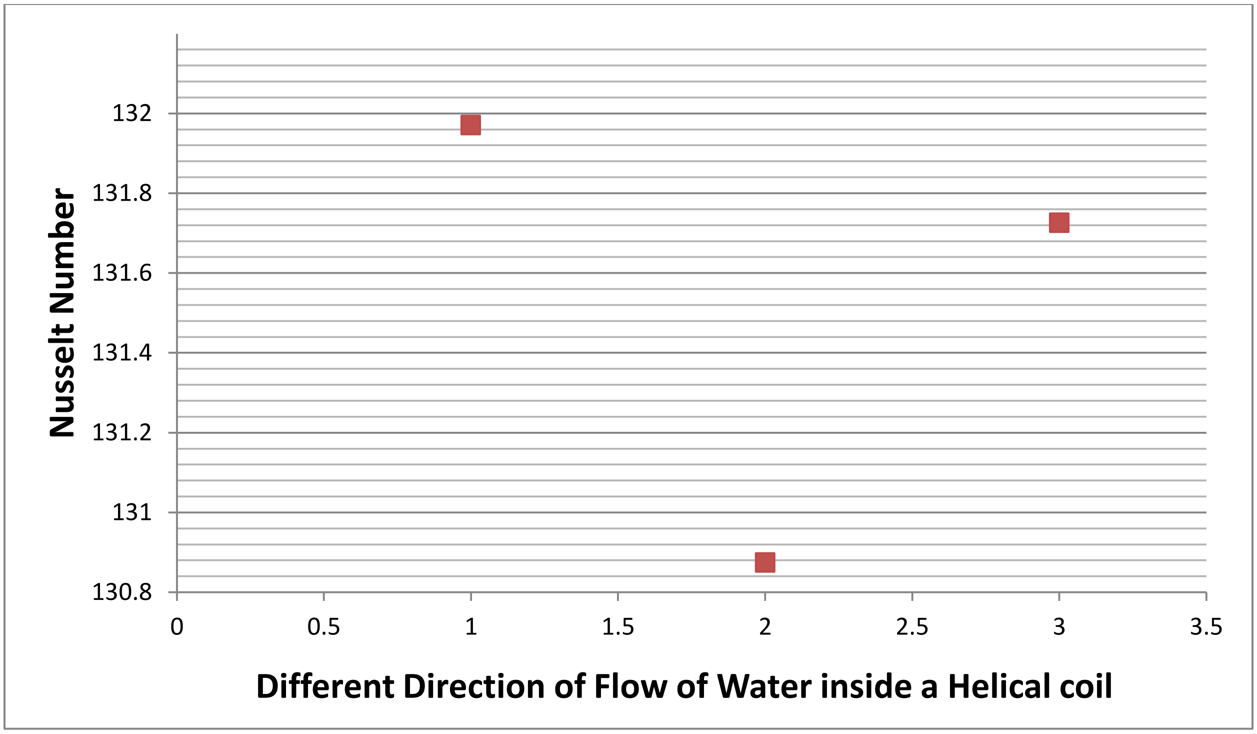


Figure 20

In figure 18 the effect of flow of direction is plotted against average nusselt number. It is observed that when curvature ratio increases from 0.3 to 0.5. the value of nusselt number for both the orientation is same. When curvature ratio rises from the range of .5 to .65. the heat transfer rate is higher in vertical orientation with compare to horizontal orientation. In the vertical orientation the effect of gravitation on fluid particle is more predominant with compare to horizontal orientation.

CHAPTER 10 :- DEVELOPMENT OF CORRELATION FOR AVERAGE NUSSELT NUMEBR:-

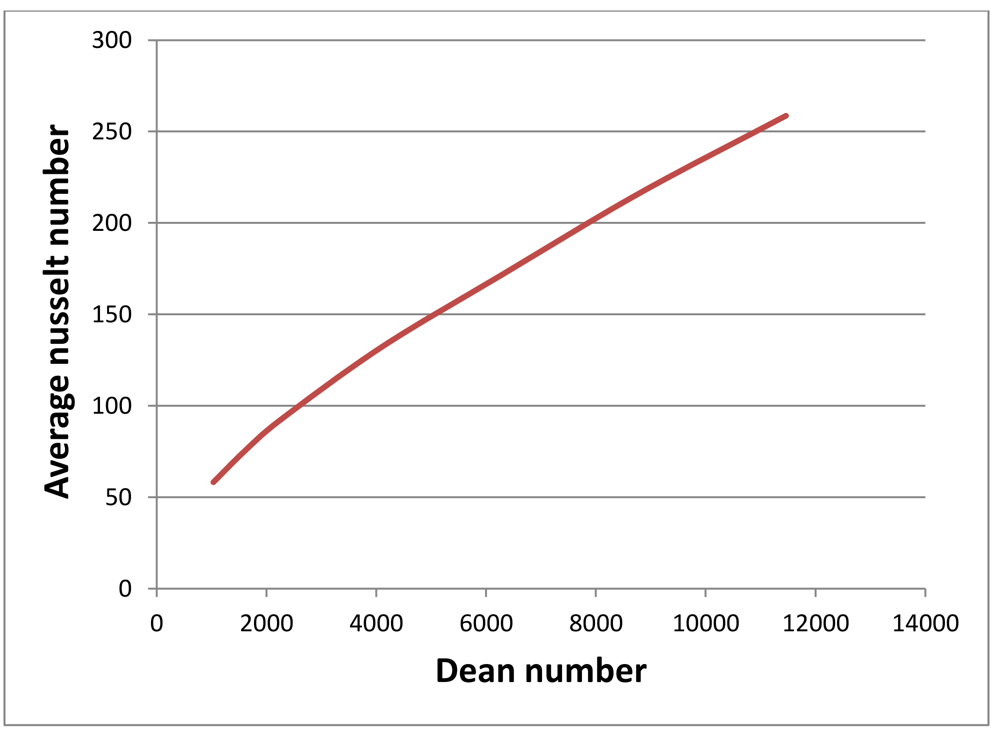
The correlation for nusselt number already consists of pipe diameter in trems of Reynolds number and curvature ratio. Hence the correlation in the form described by ditties bolter equation,

$$Nu = C\, Re^{n}\, Pr^{0.4}\, \delta^{m}$$

Where C, n and m can be evaluated. If we use dean number in the formulation, the curvature ratio term needs to be included twice. Hence Reynolds number is chosen as the general form tio estimate the average nusselt number to predict the correlation. In table no. 3. A large range of Reynolds number, dean number and curvature ratio have been analyzed. The corresponding date are clearly shown in the table no. 3

Multiple regression analysis based on the above generated data has been carried out using MATLAB, MINITAB and MICROSOFT EXCEL. The developed correlation is given as below, the range of these parameter are (1) $7{,}000<Re<54{,}000$ (2) $.04<\delta<.15$ (3) pr = 6 (4) $2000<De<14000$

$$Nu = 0.023925\, Pr^{0.4}\, \delta^{0.1078}\, Re^{0.8515}$$



CHAPTER 10 :- CONCLUSION:-

1. Characterisrics of heat transfer under turbulent flow modeling of single phase water through vertically and horizontally oriented helical axis are presented in this article. The development work also employs various MINITAB and MATLAB codes.
2. To describe the thermal and wall treatment behavior of fluid flow inside the helical coil Different contours are represented in this thesis. It is observed that Y PLUS value varies between the range of 30 to 300 for standard k-epsilon model for the large range of curvature ratio and dean number.
3. The effect of various kinematic and geo matric parameters on average nusselt number is plotted. It is observed that coil pitch does not effect the average nusselt number. The rising value of pitch circle diameter greatly increases heat transfer and average nusselt number. But the slope of this graph is not effected by varying inlet velocity of fluid.
4. The effect of pipe diameter on heat transfer and average nusselt number is plotted. It is found that the heat transfer rate and average nusselt number predominantly increases with pipe diameter of helical coil. The slope of this graph also rises with increasing inlet velocity of fluid flow.
5. The influence of inlet velocity of fluid on heat transfer is also observed. It is investigated that heat transfer rate and average nusselt number strongly rises with inlet velocity of water.
6. The effect of dean number on darcy friction factor and head loss in coil is also plotted. It is observed that head loss in the helical pipe continue decreases with dean number. Darcy friction factor rises almost linearly with dean number.
7. The heat transfer phenomena is also investigated for different orientation of helical coil. the main conclusion is drawn that heat transfer rate in vertically oriented coil is higher than the horizontally oriented helical coil
8. After establishing the parametric influence, a correlation has been developed for estimation of average nusselt number. This correlation is compared with those available in the literature and deviation are within limits. It is also observed this correlation is applicable for a large range of curvature ratio and dean number for constant wall temperature boundary condition

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