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"PERFORMANCE ANALYSIS OF A HELICAL COIL HEAT EXCHANGER IN THE LIGHT OF WASTE HEAT RECOVERY APPLICATIONS WITH ANSYS SOFTWARE"

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ABSTRACT

The present work has been carried out with a view to predict the performance of a helical coil heat exchanger in the light of waste heat recovery applications. The computational analysis has been performed using the CFD package. The analysis has been done using standard k-ε turbulence model. Due to the highly irregular nature of the helical tube and shell, an unstructured grid system has been employed to solve the model. The CFD simulation results have been used to estimate the overall heat transfer coefficient. A correlation has been developed to estimate the tube side heat transfer coefficient in the turbulent regime using regression analysis and the same has been used to evaluate the effectiveness of the helical coiled tube heat exchanger.

Key Words: Helical Coil, Waste Heat, Turbulence Model, Simulation, Heat Exchanger.

I. INTRODUCTION

Heat transfer is defined as the transmission of energy from one region to another as a result of temperature gradient that takes place by three modes namely Conduction, Convection and Radiation. Heat transmission, in majority of real situation, occurs as a result of these modes of heat transfer. The three modes are similar in that a temperature differential must exist and the heat exchange is in the direction of decreasing temperature. In the present work, the exhaust gas from diesel engine which comes under the category of low temperature range (66-120°C) has been used as the shell side fluid for heat transfer analysis. The exhaust gas transfer heat to the cold fluid (water) that is flowing through the helical tube.

Computational Fluid Dynamics is becoming a wide spread tool, used by a vast number of engineers. CFD provides an option, which is cheaper, obtains a complete set of results and is suitable for almost all complexity of problems. CFD is also well suited for trouble shooting and also it has a faster turnaround time than experiments.

The present work begins with the CFD analysis of helical tube heat exchanger in order to see the effect of temperature rise and pressure drop along the length of the helical tube and the shell. The exhaust gas from diesel engine has been used as the shell side fluid for heat transfer analysis. The exhaust gas transfer heat to the cold fluid (water) that is flowing through the helical tube.

CFD provides the flexibility to change design parameters without the expense of hardware changes. It therefore costs less than laboratory or field experiments, allowing engineers to try more alternative designs than would be feasible otherwise. It also reduces design cycle time and cost by optimizing through computer predictions and provides higher level of confidence in prototype or field installed performance. Moreover it investigates and understands the "why" for existing problem or new equipment. The main objective of the present study is to analyze the shell and helical tube heat exchanger both computationally and experimentally and to validate the CFD results by comparing with the present experiment.

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Several studies have indicated that helically coil tubes are superior to straight tubes when employed in heat transfer applications. In one of the work carried out by **D.G. Prabhanjan et al.** [1] on "Comparison of heat transfer rates between a straight tube heat exchanger and helically coiled heat exchanger", it was observed that the heat transfer rate was affected by the geometry of the heat exchanger and the temperature of the water bath surrounding the heat exchanger. Also the flow rate did not affect the heat transfer coefficient, most likely from the fact the flow was turbulent and increasing the flow rate does not greatly change the wall effects. Temperature rise of the fluid was found to be effected by the coil geometry and by the flow rate. In another work carried out by **M.R. Salimpour et al.** [2] on "Heat transfer coefficient of Shell and coiled tube heat exchanger", Heat exchanger with three different coil pitches were tested for counter flow configuration. From the result of the study, it was found that the shell side heat transfer coefficient of the coil with larger pitches is higher than those for smaller pitches. Finally based on the result of the study, two correlations were developed to predict the inner and outer heat transfer coefficient of the coiled tube heat exchanger.

M.R. Salimpour [3] also made an investigation to study the heat transfer coefficient of temperature dependent property engine oil flow inside shell and coiled tube heat exchanger experimentally. From the result of the study, it was observed that increasing the coil tube pitch decreases the inner nusselt number. Also, increase of coil tube pitch leads to higher value of shell side Nusselt number because in smaller coil pitches, the coolant water is confined in the space between successive coil rounds and a semi dead zone is formed, as in this region the flow is decelerated, heat transfer coefficient will be descended.

Another similar work was carried out by **W. Witchayanuwat and S. Kheawhom** [4] on "Heat transfer coefficient for particulate air flow in shell and coiled tube heat exchanger". From the result of the study it was found that variation in the pitches of coiled tube slightly affects the shell side heat transfer coefficient. Two empirical correlations were also developed to predict the inside and outside heat transfer coefficient of the coiled tube heat exchanger for the particulate air-flow water system. In another work by **H. Shokouhmand et al. [5] on** "Experimental and investigation of Shell and Coil tube Heat exchanger using Wilson Plot", **an experiment was performed for both the Parallel flow and counter flow configuration.** Overall heat transfer coefficients of the heat exchangers were calculated using Wilson plots. It was observed that the shell-side Nusselt numbers of counter-flow configuration were slightly more than the ones of parallel-flow configuration. Finally, it was observed that the overall heat transfer coefficients of counter-flow configuration are 0–40% more than those of parallel-flow configuration.

Paisarn Naphon et al. [6] made a detailed survey on Single phase and double phase flow and Heat transfer characteristic in helically coiled tubes, spirally coiled tubes and other coiled tubes. In one of his paper "Thermal performance and pressure drop of the Helical coil Heat Exchanger with or without helically crimped fins", [7] he studied the thermal performance and pressure drop of Heat Exchanger in which the heat exchanger consists of thirteen turns concentric helical coil tubes with coil tubes consisting of two different coil diameters. He concluded that outlet cold water temperature increases with increasing hot water mass flow rate. Inlet hot and cold water mass flow rates and inlet hot water temperature also have a significant effect on the heat exchanger effectiveness. Paisarn Naphon et al. [8] also made a study on effect of curvature ratio on the heat transfer and flow development in the horizontal spirally coiled tubes. It was observed that because of centrifugal force, the heat transfer and pressure drop are more in spirally coil tube compared to that of straight tube.

Andrea cioncolini et al. [9] made a study on laminar to turbulent flow transition in helically coiled tubes. The influence of curvature on the laminar to turbulent flow transition in helically coiled pipes was analyzed from direct inspection of the experimental friction factor profiles obtained for twelve coils. The coils studied had ratios of coil diameter to tube diameter ranging from 6.9 to 369 while the coil pitches were small enough to neglect the effect of torsion on the flow.

Unlike the study made by **Andrea cioncolini et al.** [9] on laminar to turbulent flow transition in helically coiled tubes, **R.A. Seban et al.** [10] have done an investigation on laminar flow of oil and turbulent flow of water in coiled tubes having ratio of coil to tube diameter of 17 and 104. The friction factor for laminar and turbulent flow corresponding with the results of Ito and are predictable by his equations when for non-isothermal flow the properties are evaluated at the mean film temperature. **B.V.S.S.S. Prasad et al.** [11] also conducted experiments on helical tube heat exchanger and developed a correlation for pressure drop and heat transfer coefficient for the tube and shell side. In the tube side, the laminar friction factor and Nusselt numbers are represented as functions of $Re\sqrt{(d/D)}$, whereas in turbulent flow the results are correlated with $Re(d/D)^2$. The pressure drop and heat transfer

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values for the shell side are found to follow the classical Blasius and Dittus-Boelter type relations, while a strong dependence on the coil to tube diameter ratio is detected. The performance of the exchanger has been tested not only as simulated experimental exchanger but also as a waste heat recovery device for a 60 HP gas turbine. In one of the paper carried out by **J.S. Jayakumar et al.** [12] on “Experimental and CFD estimation of Heat Transfer in helically coiled Heat Exchanger”, they made an attempt to find out the boundary condition for proper modeling considering different boundary conditions. They found that constant temperature or constant heat flux boundary conditions does not yield proper modeling. Hence, the heat exchanger was analyzed considering conjugate heat transfer. The CFD analysis was made using FLUENT . The experimental and CFD results were compared and based on the experimental results, a correlation was developed to calculate the inner tube heat transfer coefficient of the helical coil.

J.S. Jayakumar et al. [13] also made an investigation on “CFD analysis of single phase flows through helical coil”. Here, they made an attempt to see the outcome by varying the coil pitch, pipe diameter and pitch circle diameter using the CFD package FLUENT. It was observed that when the coil pitch is zero, local Nusselt number at the top and bottom points on the periphery of a cross section are almost the same. For this case, only centrifugal but no torsonal force is acting on the fluid. As we increase the pitch, torsonal or rotational forces comes into effect. When the pipe diameter is small, the secondary flows are weaker and hence mixing is lesser. This produces nearly the same heat transfer in the upper half cross section in a given plane. When the pitch coil diameter is more, the effect of coil curvature on flow decreases and hence centrifugal force plays a lesser role in flow characteristic.

In another paper on “Development of Heat transfer coefficient correlation for concentric helical coil heat exchanger”, by **Rahul Kharat, Nitin Bhardwaj and R.S. Jha**, [14] improved heat transfer coefficient correlation was developed for the flue gas side of heat exchanger from experimental and CFD data. Also the effect of different functional dependent variable such as gap between the concentric coil, tube diameter and coil diameter which affects the heat transfer were analyzed.

Based on the above mentioned comprehensive literature review, it can be concluded that the geometry of a helical tube is the main concern in order to obtain increasing heat load which is the first priority in the modern day heat exchanger. The parameters that affect the heat transfer coefficient are coil to tube diameter ratio, pitch of the coil and coil diameter. So, while doing an analysis, these parameters need to be taken into account with the aim of achieving higher heat transfer coefficient.

II. GOVERNING EQUATIONS OF CFD

Each CFD software package has to produce a prediction of the way in which a fluid will flow for a given situation. To do this the package must calculate numerical solutions to the equations that govern the flow of fluids. For the analyst, therefore, it is important to have an understanding of both the basic flow features that can occur, and so must be modeled, and the equations that govern fluid flow. The physical aspects of any fluid flow and heat transfer are governed by three fundamental principles [12].

a) Continuity equation
b) Momentum equation and
c) Energy equation.

a) Continuity Equation:

The continuity equation is essentially the equation for the conservation of mass. It is derived by the mass balance on the fluid entering and leaving a volume element taken in the flow field. The equation for the conservation of mass for two dimensional steady flows may be stated as

$$\begin{bmatrix}\text{Net rate of mass flow entering}\\ \text{volume element in x direction}\end{bmatrix} + \begin{bmatrix}\text{Net rate of mass flow entering}\\ \text{volume element in y direction}\end{bmatrix} = 0 \qquad (3.1)$$

For an incompressible fluid, the continuity equation for a steady two dimensional flow can be written as

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad (3.2)$$

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b) Momentum Equation:

The momentum equation are derived from Newton's second law of motion, which states that mass times the acceleration in a given direction is equal to the external force acting on the body in the same direction. The external force acting on the volume element in a flow field is considered to consist of the body forces and the surface forces.

$$[\text{Mass}]\begin{bmatrix}\text{Acceleration in} \\ \text{i direction}\end{bmatrix} = \begin{bmatrix}\text{Body forces acting in} \\ \text{i direction}\end{bmatrix} + \begin{bmatrix}\text{Surface force acting} \\ \text{in i direction}\end{bmatrix} \qquad (3.3)$$

x Momentum:
$$\rho\left[u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right] = F_x - \frac{\partial p}{\partial x} + \mu\left[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right] \qquad (3.4)$$

y Momentum:
$$\rho\left[u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right] = F_y - \frac{\partial p}{\partial y} + \mu\left[\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right] \qquad (3.5)$$

where F_x and F_y are the body forces per unit volume acting in the x and y direction respectively.

The physical significance of the various terms in equation (3.4) is as follows: The terms on the left hand side represent the inertia forces, the first term on the right hand side is the body forces, the second term is the pressure forces, and the last term in the parentheses is the viscous force on the fluid element.

c) Energy Equation:

The temperature distribution in the flow field is governed by the energy equation, which can be derived by writing an energy balance according to first law of thermodynamics for a differential volume element in the flow field. If radiation is absent and there are no distributed energy sources in the fluid, the energy balance on a differential volume element may be stated as

$$\begin{bmatrix}\text{rate of energy} \\ \text{input due to} \\ \text{conduction}\end{bmatrix} + \begin{bmatrix}\text{rate of energy} \\ \text{input due to} \\ \text{work done by} \\ \text{conduction}\end{bmatrix} + \begin{bmatrix}\text{rate of energy} \\ \text{input due to} \\ \text{work done by} \\ \text{surface stress}\end{bmatrix} = \begin{bmatrix}\text{rate of increase} \\ \text{of energy in} \\ \text{element}\end{bmatrix} \qquad (3.6)$$

The energy equation for two dimensional flow of an incompressible, constant property, Newtonian fluid is determined as

$$\rho c_p\left[u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right] = k\left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right] + \mu\phi \qquad (3.7)$$

Where the viscosity-energy-dissipation function ϕ is defined as

$$\phi = 2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2\right] + \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y}\right)^2 \qquad (3.8)$$

The basic procedural steps for solving a problem in FLUENT include:

- Define the modeling goals.
- Create the model geometry and grid.
- Set up the solver and physical models.
- Compute and monitor the solution.
- Examine and save the results.

Consider revisions to the numerical or physical model

III. CFD MODELING OF A HELICAL TUBE HEAT EXCHANGER

The helical tube consists of twenty two numbers of turns. The analysis has been done for cross-counter flow configuration by varying the tube side fluid velocity from 0.5 m/s to 3 m/s while the shell side fluid velocity was kept constant. The investigations were carried out using the CFD package ANSYS FLUENT.

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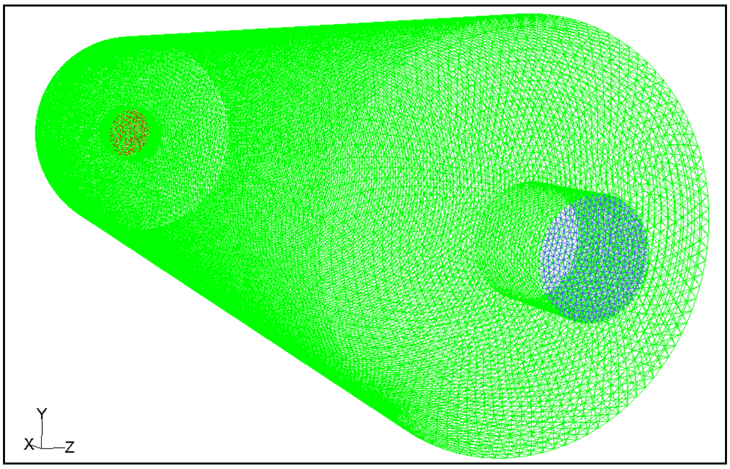

Fig.1 Grid for shell

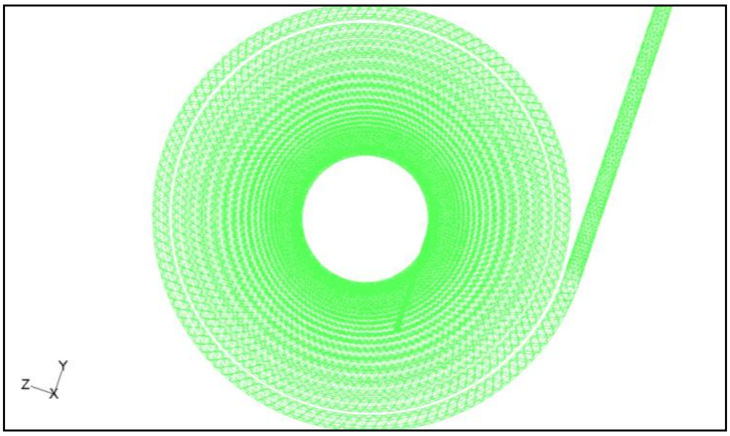

Fig. 2 Enlarged view of grid for helical coil

3.1 Properties of fluid and material

Table 1 Properties of Tube side fluid (water) at 26°C

Sl.no.	Property	Unit	Value
1.	Density	Kg/m^3	998.5
2.	Viscosity	Kg/m-s	899.9x10^{-6}
3.	Specific heat	KJ/kgK	4178.0
4.	Thermal conductivity	W/mK	6068.6x10^{-4}

Table 2 Properties of Shell side fluid (Exhaust gas) at 120°C

Sl.no.	Property	Unit	Value
1.	Density	Kg/m^3	897.7x10^{-3}
2.	Viscosity	Kg/m-s	1011.0
3.	Specific heat	KJ/kgK	323.5x10^{-4}
4.	Thermal conductivity	W/mK	226.4x10^{-7}

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Table 3 Properties of Tube material (Copper)

Sl.no.	Property	Unit	Value
1.	Density	Kg/m^3	8030.00
2.	Specific heat	J/kgK	502.48
3.	Thermal conductivity	W/mK	16.27

Table 4 Properties of Shell material (Steel)

Sl.no.	Property	Unit	Value
1.	Density	Kg/m^3	8978.0
2.	Specific heat	J/kgK	381.0
3.	Thermal conductivity	W/mK	387.6

3.2 Grid Independence Test

To find out the most independent grid for CFD analysis of a helical tube heat exchanger, grid independency of the solution was established. The resolution of the grid has a great quantitative impact over the result obtained. There exists a level of refining of a computational domain beyond which there is no significant changes in the results achieved. Based on the different grids, analysis have been made and it was observed that after refining the grid from nodes 209686, results are not varying significantly. So, nodes 209686 have been used for further analysis.

Table 5 Overview of all the grids used for Grid Independence Study

Sl.no.	No. of nodes	Nusselt number
1.	157848	4.7
2.	179010	7.1
3.	190038	6.9
4.	209686	8.2
5.	215050	8.2

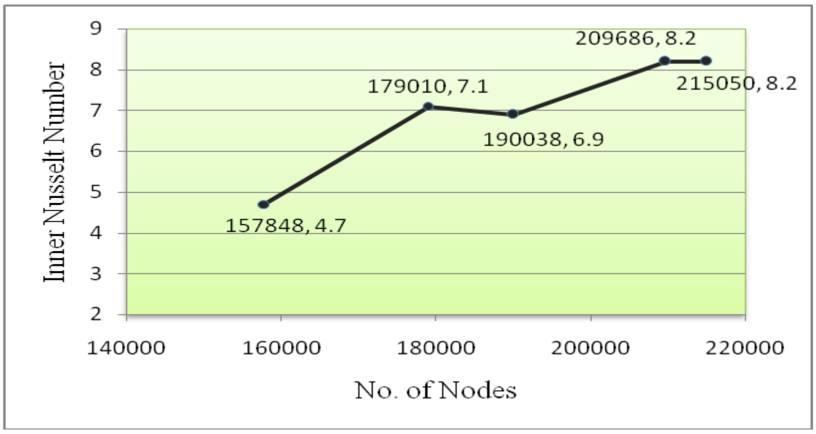

Fig.3 Grid Independence Test

Using the above mentioned nodes, the analysis has been carried out for seven cases i.e by varying the tube side fluid velocity from 0.5 m/s to 3.0 m/s.

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IV. RESULTS AND DISCUSSION

The analysis has been carried out for seven cases by varying the tube side fluid velocity from 0.5 m/s to 3.0 m/s. The shell side fluid velocity was kept constant at 0.3 m/s. The purpose of CFD analysis was to see the effect of temperature rise and pressure drop in the helical tube with the increase in mass flow rate of the tube side fluid. And also to obtain the Overall heat transfer coefficient and the Effectiveness of the helical tube heat exchanger for cross counter flow configuration.

Case I: Tube side fluid velocity: 0.5 m/s, Shell side fluid velocity: 0.3 m/s

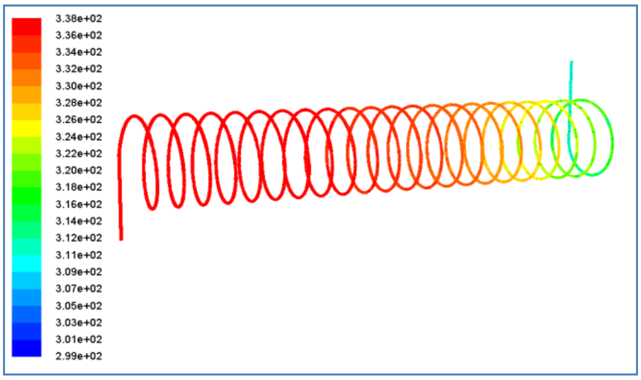

Fig.4 Temperature contour of helical tube

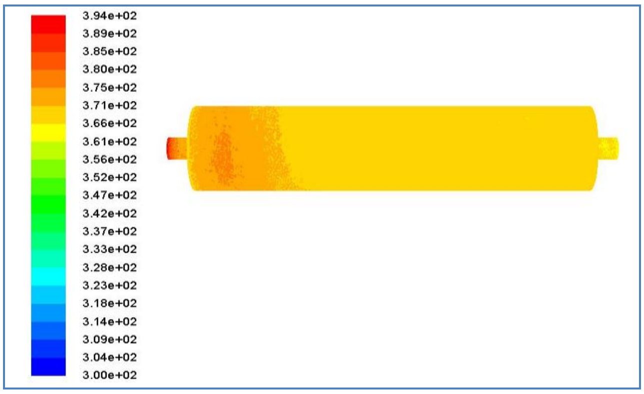

Fig.5 Temperature contour of Shell

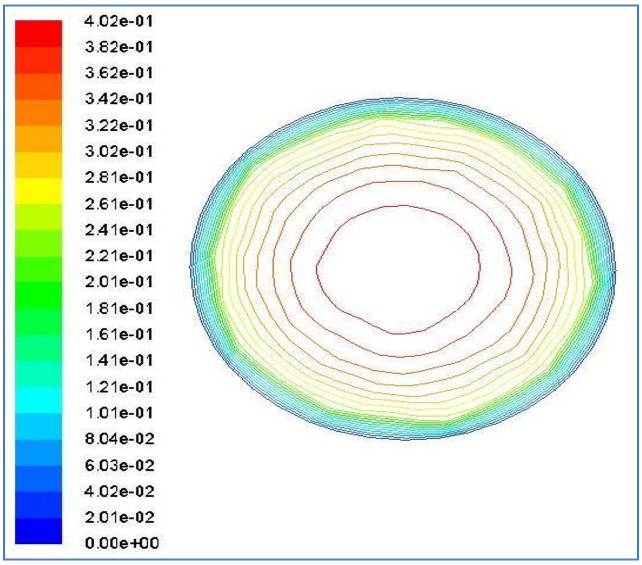

Fig.6. Velocity profile at the exit of the shell for the case of inlet velocity 0.3 m/s.

Case II: Tube side fluid velocity: 0.75 m/s, Shell side fluid velocity: 0.3 m/s

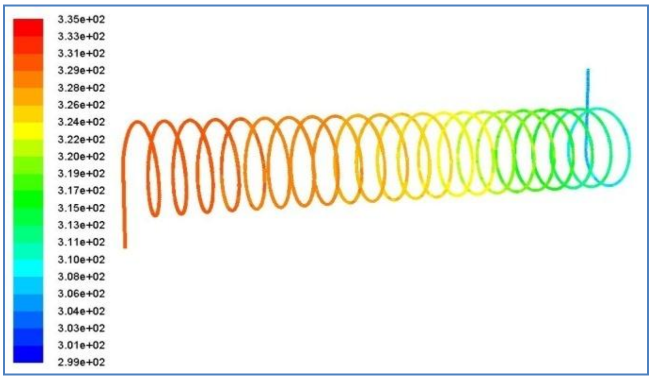

Fig.7 Temperature contour of helical tube

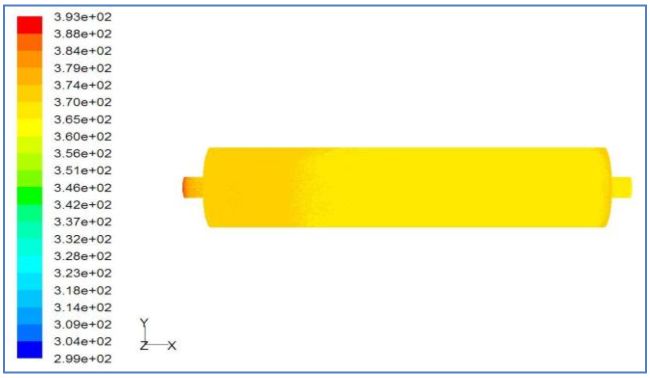

Fig.8 Temperature contour of shell

Case III: Tube side fluid velocity: 1.0 m/s, Shell side fluid velocity: 0.3 m/s

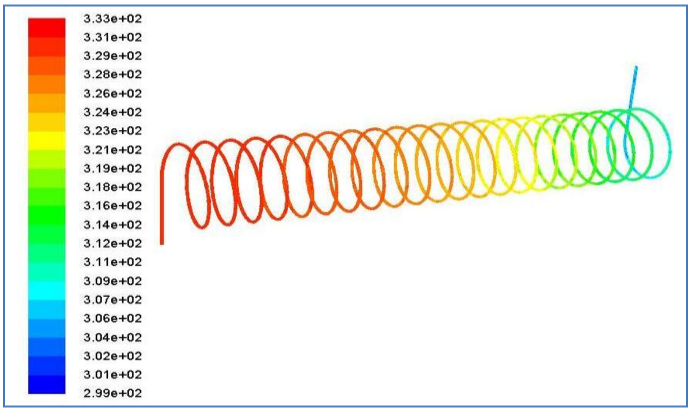

Fig.9 Temperature contour of helical tube

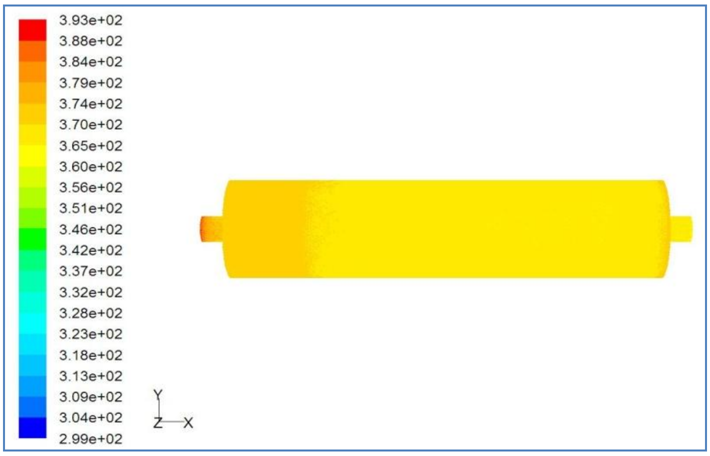

Fig.10 Temperature contour of shell

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Case IV: Tube side fluid velocity: 1.75 m/s, Shell side fluid velocity: 0.3 m/s

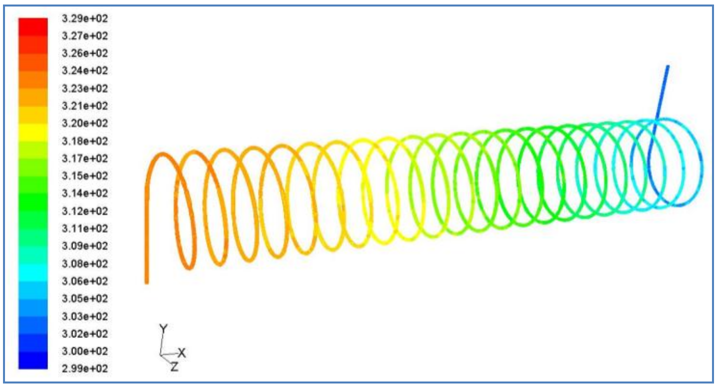

Fig.11 Temperature contour of helical tube

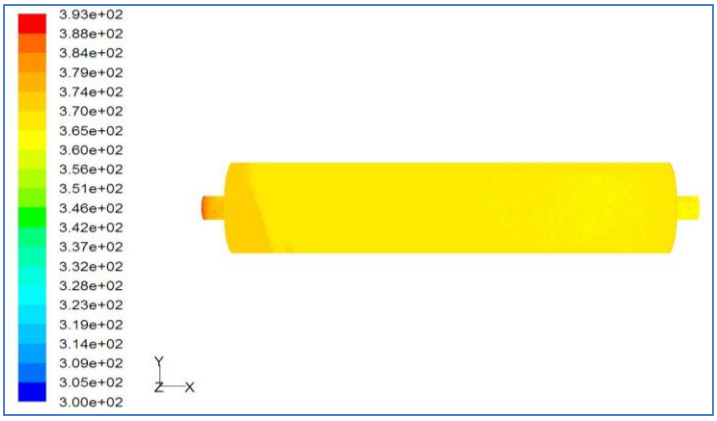

Fig.12 Temperature contour of shell

Case V: Tube side fluid velocity: 2.0 m/s, Shell side fluid velocity: 0.3 m/s

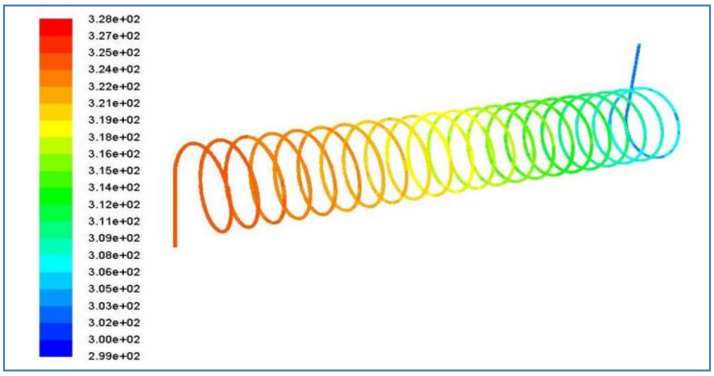

Fig.13 Temperature contour of helical tube

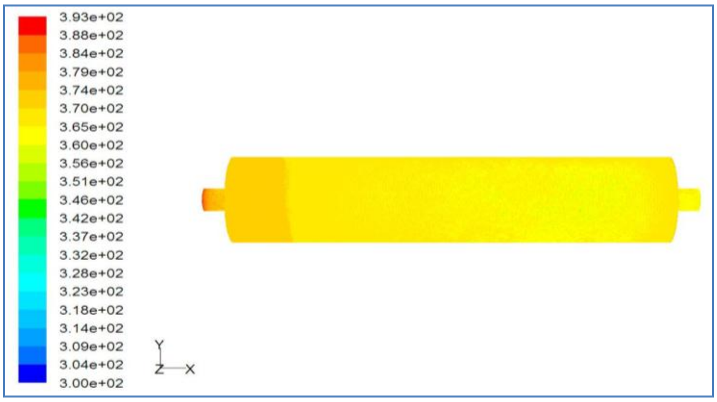

Fig. 14 Temperature contour of shell

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The temperature contour of the helical tube and shell are shown in fig.3 to 14. The effect of cross flow in the shell region is visible from the temperature contour. It is also visible from the temperature contour of the shell that the temperature of the shell side fluid decreases after exchanging its heat with the fluid that is flowing through the helical tube. Since the shell side fluid become mixed along the path of flow, therefore the exit temperature for the shell side fluid become nearly uniform. Pressure drop is quite high in helical tube as observed from the fig.5 and 8, thereby making the necessity of large pumping power. Reducing the length may solve the problem. The pressure drop of exhaust side is very low as observed from fig.4. This means that there is no back pressure and hence, there is no effect on working in the engine.

4.1 Data from FLUENT analysis

Table 6 Tube side CFD data from FLUENT analysis

Sl. no.	Velocity (m/s)	Inlet temperature (K)	Outlet temperature (K)	Mean temperature (K)	Inner tube heat transfer coefficient (h_i) (W/m^2K)	Outer tube heat transfer coefficient (h_o) (W/m^2K)	Overall heat transfer coefficient (U) (W/m^2K)
1.	0.50	299	337	318.0	832	37.7	36.6
2.	0.75	299	334.4	316.7	1059	38.8	37.4
3.	1.00	299	332.5	315.8	1106	39.3	38.0
4.	1.75	299	328.8	313.9	1412	42.9	41.6
5.	2.00	299	328	313.5	2187	51.5	50.3
6.	2.50	299	326.2	312.6	2353	53.2	52.0
7.	3.00	299	324.8	311.9	2943	60.0	58.8

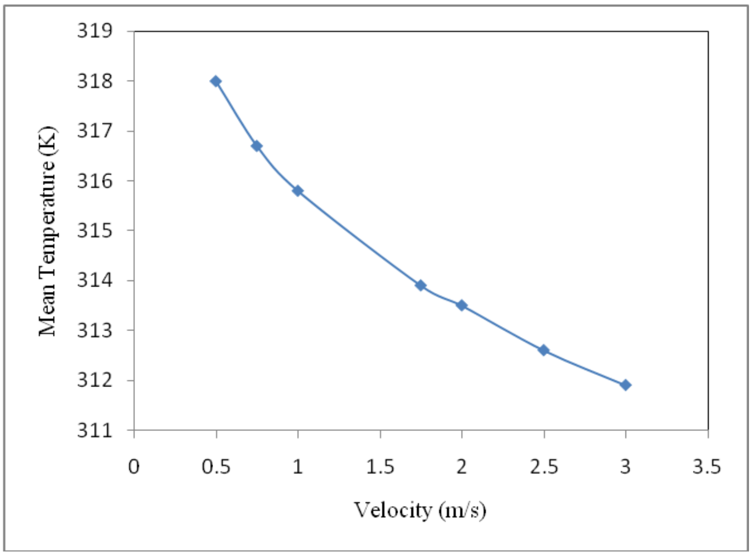

Fig.15 Mean Temperature (K) V/S Velocity (m/s)

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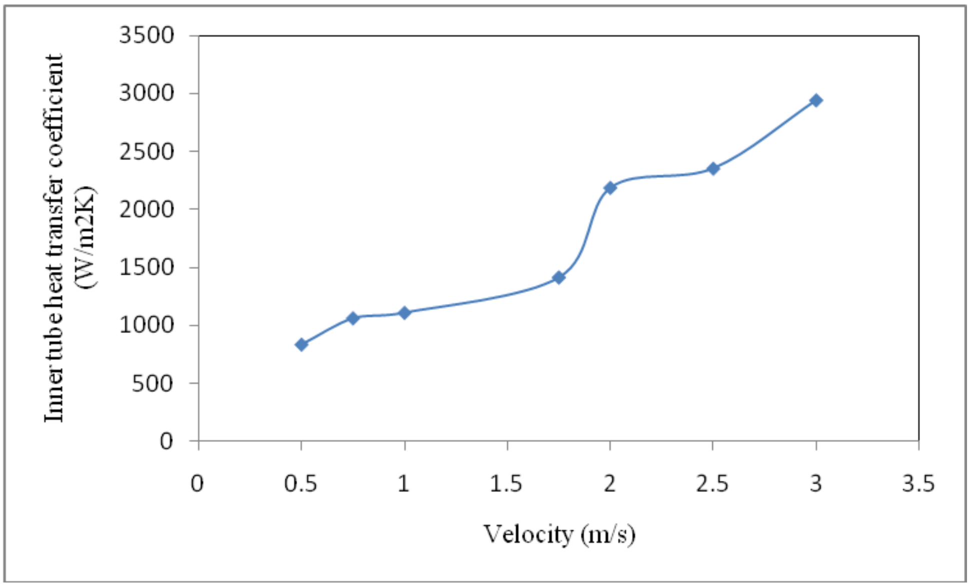

Fig.16 Inner tube heat transfer coefficient (W/m^2K) V/S Velocity (m/s)

Table 6 gives the inlet and outlet temperature of the tube side fluid and the inner and outer tube heat transfer coefficient that was obtained from CFD analysis. It is observed from fig.15 and fig. 16 that with the increase in velocity of the fluid, there is a decrease in temperature rise but increase in heat transfer coefficient. The decrease in temperature rise might be due to the decreased residence time of the fluid. And the increasing heat transfer coefficient is most likely from the fact that the flow was turbulent and increasing the flow rate does not greatly change the wall effects. The increased heat transfer coefficients are a consequence of the curvature of the coil, which induces centrifugal force to act on the moving fluid, resulting in the development of secondary flow. Fluid from the inside of the tube is thrown through the centre of the tube towards the outer wall and then returns to the inner wall through the wall region. The secondary flow enhances heat transfer and temperature uncertainty due to increased mixing.

Table 7 Shell side CFD data from FLUENT analysis

Sl.no.	Velocity (m/s)	Inlet temperature (K)	Outlet temperature (K)	Mean temperature (K)
1.	0.3	393	368	380.5
2.	0.3	393	367.6	380.3
3.	0.3	393	367.6	380.3
4.	0.3	393	366.3	379.7
5.	0.3	393	365.9	379.5
6.	0.3	393	365.4	379.2
7.	0.3	393	365	379.0

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Table 8 Tube and Shell side Reynold number

Sl.no.	Tube side				Shell side		
	Reynold no.	Dean no.	Nusselt no.	Prandtl no.	Reynold no.	Nusselt no.	Prandtl no.
1.	5104	993	8.2	4.01	3770	7.6	0.7096
2.	7513	1462	10.5	4.09	3774	7.8	0.7097
3.	9513	1851	11.0	4.16	3774	7.9	0.7097
4.	16854	3280	14.0	4.28	3784	8.7	0.7098
5.	19156	3728	21.7	4.31	3788	10.4	0.7099
6.	23532	4580	23.4	4.39	3793	10.8	0.7099
7.	29984	5836	29.4	4.49	3796	12.1	0.7100

4.2 Development of correlation for inside tube heat transfer coefficient

Based on the nature of correlation available in the literature [12], Nusselt number for inside tube heat transfer can be represented in the form,

$$Nu_i = CDn_i^m Pr_i^n$$

Where, C and m are unknown which are to be determined and index of the Prandtl number,

n = 0.3 for cooling i.e wall temperature less than mean temperature

n = 0.4 for heating i.e wall temperature more than mean temperature

Using Regression analysis, the following correlation was developed for estimating the Inside tube Heat transfer coefficient.

$$Nu_i = 0.038 Dn_i^{0.686} Pr_i^{0.4}$$
$$993 \leq Dn \leq 5836$$

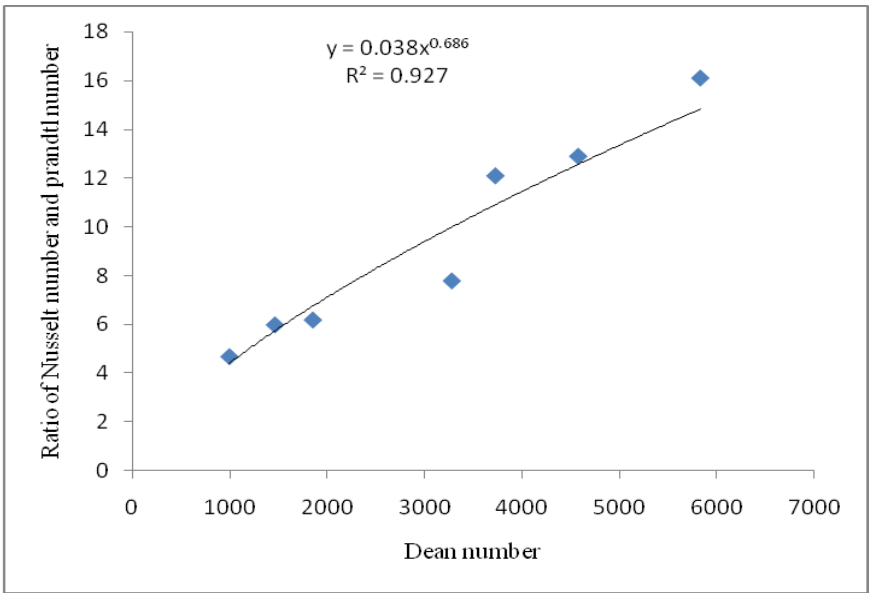

Fig.17 Plot of data set with line of best fit

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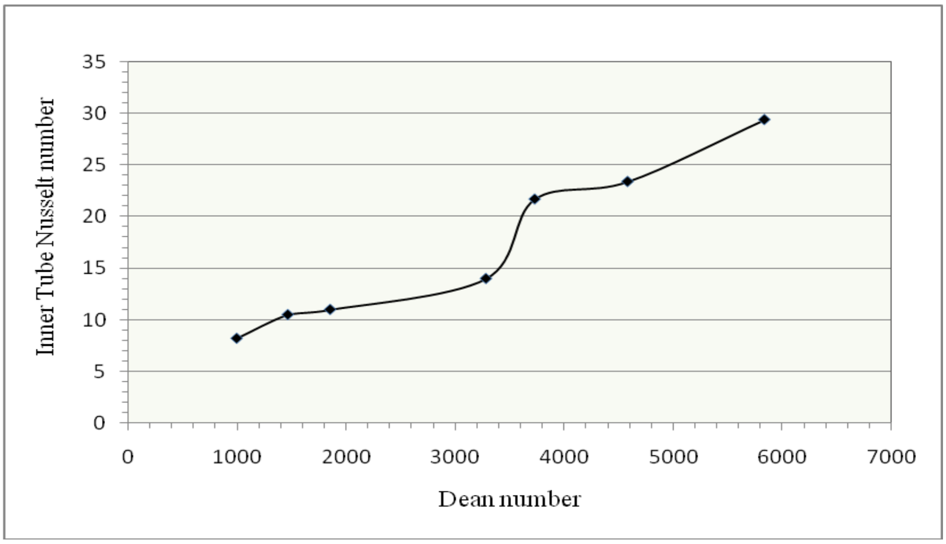

Fig.18 Inner Tube Nusselt number V/S Dean Number

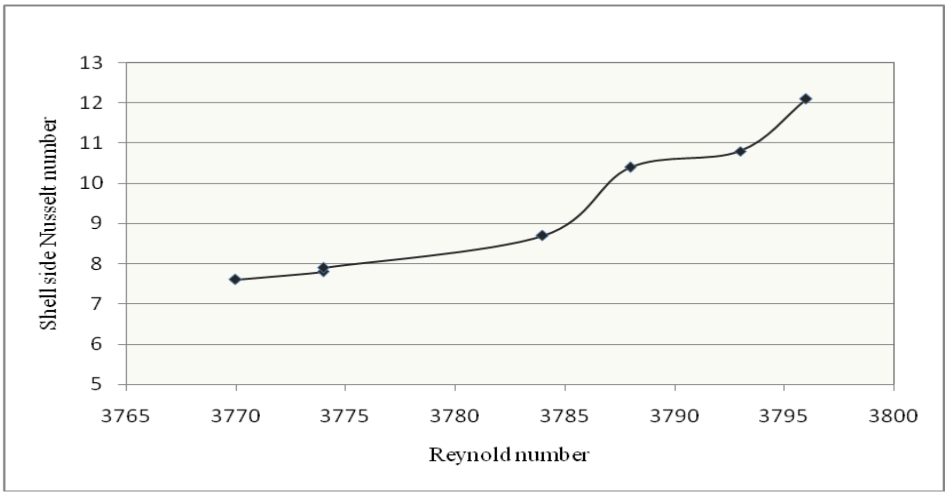

Fig.19 Shell side Nusselt number V/S Reynold number

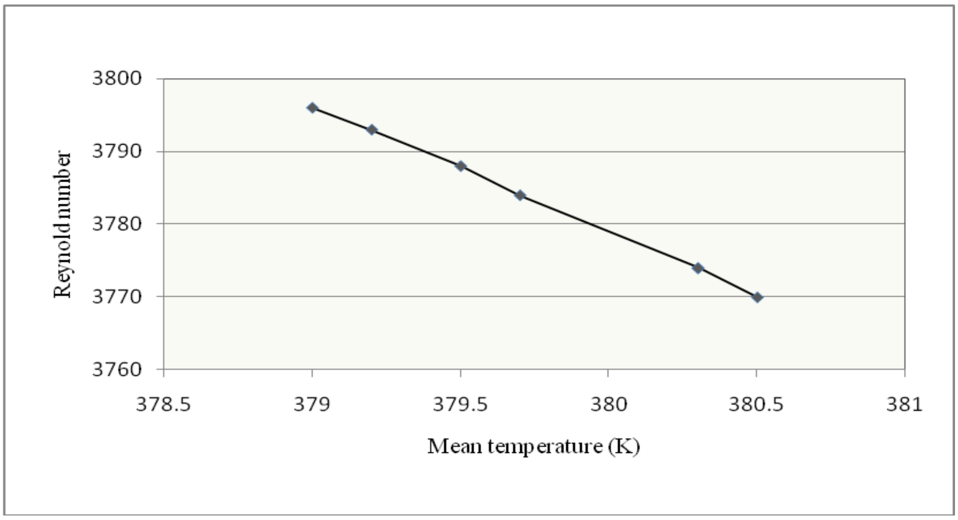

Fig.20 Shell side Reynold number V/S Mean temperature

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It is observed from the fig.17, that the change in temperature along the length of the tube is low. This leads to the Nusselt number to be closer at low Dean number. Fig.19 represents the shell side Nusselt number with shell side Reynold number based on results obtained from CFD analysis. It is observed that shell side Nusselt number increases with Reynold number. Also, from the fig.20, it is observed that the shell side Reynold number goes on decreasing with the increase in mean temperature of the fluid. Since the shell side fluid is a gas and the viscosity of a gas increases with the increase in temperature, therefore, the shell side Reynold number decreases with the increase in mean temperature.

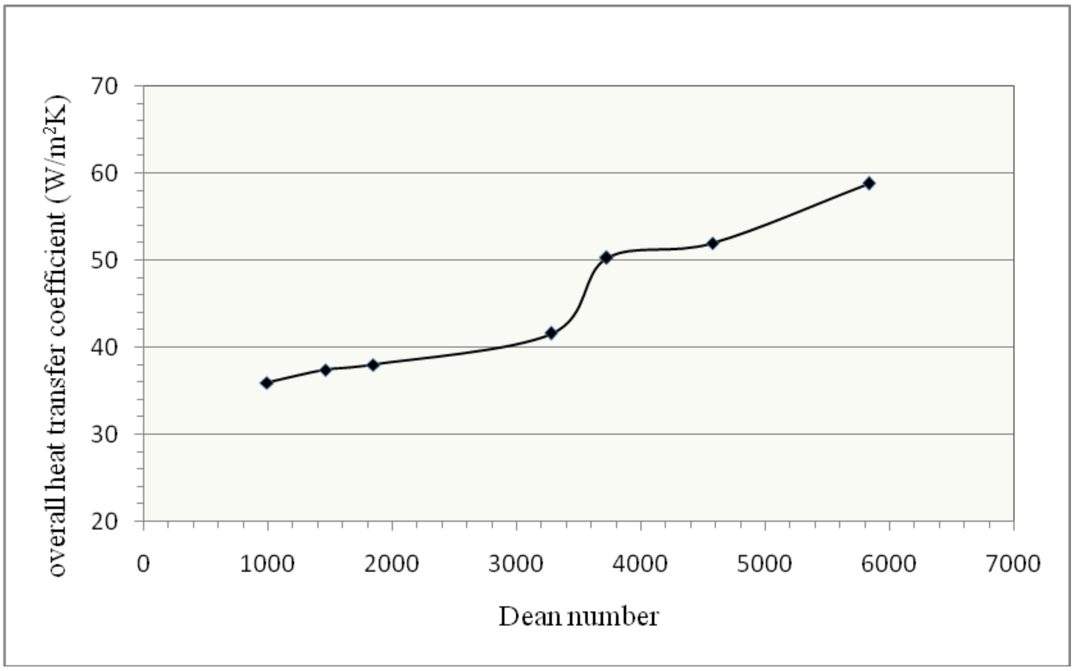

Fig.21 Overall Heat transfer Coefficient V/S Dean Number

Table 9 Heat capacity

Sl.no	Cp of cold flow (J/kgK)	Cp of hot flow (J/kgK)	Heat capacity of cold flow (Cc) (W/K)	Heat capacity of hot flow (Ch) (W/K)	Capacity ratio (C)
1.	4179.25	1009.75	63.60	19.85	0.312
2.	4178.93	1009.73	95.40	19.87	0.208
3.	4178.70	1009.73	127.30	19.87	0.156
4.	4178.23	1009.67	223.00	19.90	0.089
5.	4178.13	1009.65	254.90	19.91	0.078
6.	4178.00	1009.62	318.70	19.92	0.063
7.	4178.00	1009.60	382.50	19.93	0.052

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Table 10 Effectiveness

Sl.no.	Overall heat transfer coefficient (W/m^{2}K)	NTU	Effectiveness (%)
1.	36.6	0.419	32.5
2.	37.4	0.427	33.5
3.	38.0	0.434	34.3
4.	41.6	0.475	37.2
5.	50.3	0.573	42.9
6.	52.0	0.593	44.1
7.	58.8	0.670	48.2

V. Conclusion

An investigation was carried out to study the Shell and helical tube heat exchanger computationally. The analysis was done for seven cases and the effect on temperature rise and the pressure drop in the helical tube and shell was observed. A correlation was developed to predict the inner tube heat transfer coefficient based on CFD data.

a) The results from analysis appear to be in good agreement and as such the correlation so developed for helical tube heat exchanger and the turbulence model used therein can be said to be applicable for helical configurations.
b) There is an augmentation of heat transfer coefficient on helical coil. This happens due to secondary flow thereby causing greater amount of turbulence in the coil.
c) Pressure drop in the shell being very low, there exists no back pressure and hence there is no effect on working in the engine.
d) The effectiveness of the helical tube heat exchanger is quite comparable with other conventional heat exchanger design.

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