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# NUMERICAL ANALYSIS OF HELICALLY COILED HEAT EXCHANGER USING CFD TECHNIQUE

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**ABSTRACT**

In the present study, 3 dimensional numerical analysis of helically coiled tube is carried out using commercial CFD tool ANSYS CFX 12.1. It is very difficult and time consuming if these analyses could be carried out experimentally and hence in the present work, efforts are taken to optimize the helically coiled tube with respect to heat transfer and flow parameters for various coil pitch. Analysis of heat exchanger is done using conjugate heat transfer. The 3-dimensional flow through the helically coiled tube is considered which would overcome the anisotropic flow properties that would arise due to complex turbulence phenomenon and flow deviations. The flow field through the helically coiled tube is simulated by solving the appropriate governing equations: conservation of mass, momentum and energy. The turbulence is taken care by Shear Stress Transport (SST) k-$\varepsilon$ model of closure. SST $k$-$\varepsilon$ has a blending function which acts as standard $k$-$\varepsilon$ in the main stream flow and as Standard $k$-ω near the boundary layer where the gradient is much steeper. The numerically predicted results are compared with the experimental data available in the literature and a very good agreement exists between the experimental and numerically predicted data.

**Keywords:** helical coil heat exchanger, turbulence, effectiveness, CFD.

## INTRODUCTION

Based on the various literatures, the enhancement of heat transfer rate in helical coil is high as compared to straight tube. Helical coil heat exchangers are used in nuclear industry, refrigeration, heat recovery systems etc. due to its solid structure and enhanced heat transfer coefficient. Manna *et al*., 1996 had conducted studies on the helical coils used in residual heat removal systems. Jayakumar *et al*. investigated the performance of residual heat removal system which uses a helical coil for distinct process parameters in 1997. For the above studies, to obtain the rate of heat transfer and pressure drop an empirical correlation was used for helical coil. In the present work, it is expected to obtain a correlation through numerical studies after validating the predicted results with the experimental data available in the literature. Numerical predictions of fluid flow and heat transfer through various flow devices and heat exchangers are analyzed by Thundil *et al*., using CFD codes more reliably.

Investigations carried out on heat transfer coefficients are done by considering the boundary conditions like constant wall temperature or constant heat flux. In the present work, it is suggested to develop a correlation for inner heat transfer coefficient while considering fluid-wall heat exchange in helically coiled heat exchanger. Temperature dependent values of thermal and transport characteristics of the heat transfer medium are taken in to consideration in these analyses. CFD has been used efficiently for the analysis of various parameters for heat exchangers. An early study was done by Rustum and Soliman in 1990. Then reliable results were obtained by Shah in 2000. Design optimization for heat exchangers was done using CFD tool by Grijspeerdt *et al*., in 2003. Heat exchanging process in tube in tube heat exchanger was done in STAR-CD code by Van der vyver *et al*., in 2003 and the same code was applied for performance evaluation of Fractal heat exchangers. Heat transfer coefficient for plate type heat exchanger in series and parallel configurations was experimentally and numerically determined by Flavio *et al*., in 2006 using Fluent 6.1for numerical analysis. Numerical analysis of double piped heat exchangers were also carried out, such type of analysis was done by Rennie and Raghavan in 2005. Hydrodynamics and heat transfer characteristics of double pipe was carried out in Fluent by Kumar *et al*., in 2006. The paper is organized as follows: begins with the introduction of helical coil system, the experimental setup fabricated and then the methodology of experimentation. It is followed by the numerical results for heat transfer properties. Afterwards the analysis of actual helical coil tube for fluid-wall heat transfer is carried out and then the comparisons of computational and experimental results are done. Finally, the results that were justified are converted into a correlation for obtaining heat transfer inside a helical coil.

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**Nomenclature**

A area of heat transfer (m2)
De Dean number (dimensionless)
H tube pitch (m)
L length of the pipe (m)
Nu Nusselt number (dimensionless)
Pr Prandtl number (dimensionless)
Q heat removed (W)
R resistance the flow of thermal energy (W−1 m2 K)
Rc pitch circle radius of the pipe (m)
Re Reynolds number (dimensionless)
U velocity (m s−1)
V volume (m3)
h heat transfer coefficient (Wm−2 K−1)
k thermal conductivity (Wm−1 K−1)
r inner radius of the tube (m)

Greek letters
$\alpha$ helix angle (rad)
$\delta$ curvature ratio (dimensionless)
$\Delta$(temperature) difference (K)
$\mu$ viscosity (kgm−1 s−1)
$\rho$density (kgm−3)

Subscripts
Av average
$f_i$ internal fouling
$f_o$ external fouling
i internal
LM log mean
o external
ov overall
w wall

**Characteristics of helical coil**

The schematic of helical coil is given in Figure-1. Inner diameter of pipe is 2r and the coil diameter is $2R_c$ (measured between the centers of pipes) and the distance between two adjacent turns is pitch (H). The pipe diameter to coil diameter ratio ($r/R_c$) is called curvature ratio, $\delta$ . The non-dimensional pitch $\lambda$ , is the ratio of pitch to develop length of one turn. The angle made by the projection of a turn of the coil making with a plane perpendicular to the axis of the coil is called the helix angle, $\alpha$ . Dean number is used to characterize the flow in helical pipe as same to Reynolds number for flowing pipes. Dean number (De):

$$De = Re\sqrt{\frac{r}{R_c}},$$

Where Re is the Reynolds number.

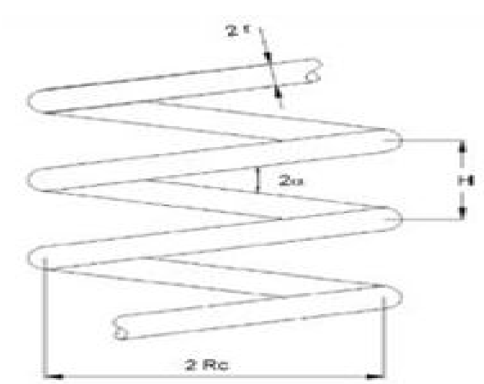


**Figure-1.** Basic geometry of helical pipe.

The enhancement in heat transfer in helical pipe is due to the complex flow pattern existing inside the pipe. The helix angle and the pitch of the coil results in the torsion of the fluid and the curvature of the coil determines the centrifugal force. The centrifugal force develops a secondary flow inside the helical tube as per the research done by Darvid *et al*., in 1971. 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 velocity setting up a secondary flow which changes correspondingly with the dean number of the flow.

## 2. EXPERIMENTAL SETUP

In order to understand the difficulty and instrumentation involved with the experimental setup of helical coiled tube the experimental setup of J.S. Jayakumar *et al*., heat exchanger is presented here in this study. The heat transfer coefficient and Nusselt number of a helically coiled heat exchanger is determined in their studies. The heat exchanger consists of a helical coiled tube inside and a hollow cylindrical shell as shown in the drawing of Figure-2. The helical pipe carries hot water though the shell inner and the shell carries the cold water. The pipe is made of SS 316 (annealed stainless steel). The pipe has an inner diameter of 10mm and outer diameter of 12.7mm. The (PCD) Pitch circle diameter of the coil is 300mm and the pitch denoted by H is 30mm. All the remaining parts of the heat exchanger are made of SS 304. The schematic of the heat exchanger used in the experimental setup is shown in the Figure-2.

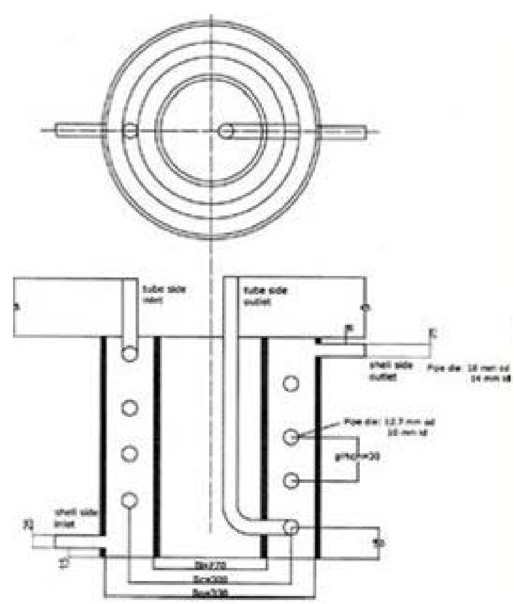

**Figure-2.** Schematic line diagram of the test section. [Jayakumar *et al*.,].

The heat exchanger is in counter flow mode. Cold water enters the shell from the bottom and exits through the outlet in the top. The coils and baffle plates in the

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assembly are assembled in such a way that they can be replaced easily.

The water through the helical coil is heated using an electric heater. Three heaters were used with a total power of 5000 W. The temperature is set constant at the end of the inlet using a controller provided in the setup. The hot water passing through the helical coils are pumped using a ½ HP centrifugal pump. The flow rate is measured using a rotameter. RTDs are used to measure inlet and outlet temperatures and displayed using digital displays. Cooling water at constant temperature is fed from a tank to the shell of the heat exchanger. The inlet, outlet and flow rate parameters are measured. The cooling water flow rate is kept constant. The cold water flow rate is set in such a way that the rise in temperature of the cooling water is approximately in range of 3 to $5^0$C.The following data was taken to model the helical coil:

Pitch circle diameter = 300mm
Inner diameter of the coil = 10mm
Outer diameter of the coil = 12.7mm
Pitch of the coil = 30mm

## 3. GOVERNING EQUATIONS AND BOUNDARY CONDITIONS

Governing equations like conservation of mass, momentum and energy were solved for simulation of 3-dimensional heat flow in helical coiled heat exchanger using ANSYS CFX code. The simulations were carried out under steady state conditions in a 3-dimensional geometry. To calculate the turbulence flow, the Standard $k$-$\varepsilon$ model is taken into consideration. The boundary conditions considered areconstant wall temperature of 300K. Hot water at a temperature of 360K is entering the coil at top (velocity inlet boundary condition) and leaves through the bottom (pressure outlet boundary condition). The analysis has been done for various inlet velocities for the helical coil. The properties of water were kept at constant values corresponding to atmospheric conditions (360K temperature and 1atm pressure).

Conservation of mass:

$$\nabla.(\rho V) = 0 - (1) -$$

Conservation of x-momentum:

$$\nabla.(\rho u\vec{V}) = -\frac{\partial \rho}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + \rho g - (2) -$$

Conservation of y-momentum:

$$\nabla.(\rho u\vec{V}) = -\frac{\partial \rho}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{yz}}{\partial z} + \rho g - (3) -$$

Conservation of z-momentum:

$$\nabla.(\rho u\vec{V}) = -\frac{\partial \rho}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + \rho g - (4) -$$

Conservation of energy:

$$Energy : \nabla(\rho e\vec{V}) = -p\nabla\vec{V} + \nabla(\mathrm{k}\,\nabla\,\mathrm{T}) + \mathrm{q} + \phi - (5) -$$

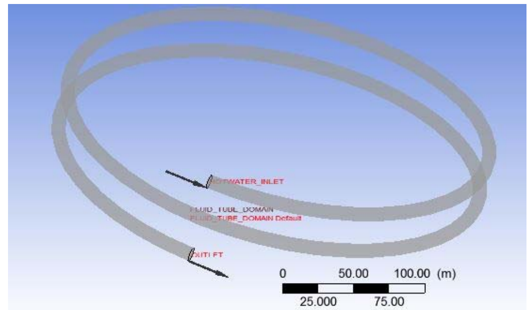


**Figure-3.** 3D modeling of 30mm coil pitch case.

## 4. CFD MODELLING OF HEAT TRANSFER IN A HELICALLY COILED TUBE

The 3d Modeling of the helical pipe was carried out using CATIA V6.0. A hollow coil of inner diameter 10 mm, pitch circle diameter 300 mm and coil pitch of 30 mm was created using the sweep tool; this part indicates the solid wall for the tube and the inner hollow portion denotes the fluid region. Another hollow helical pipe having pitch circle diameter of 300 mm, coil pitch of 60mm and inner diameter of 10mm was created for analysis for different pitch.

The outer diameter was created using the sweep tool using thickness option with a thickness of 1.35mm. The hollow coil represents the actual solid part of the tube having the hot water inside and constant wall temperature of 300K outside. Study was carried out to determine the effect of boundary condition of the heat transfer characteristics of the coil. The heat transfer characteristics for flow inside a helical pipe were analyzed for constant wall temperature of 300K as the boundary condition. The results obtained from the specified boundary conditions were studied. In all the cases, the inlet conditions were kept the same and the analysis were carried out for various flow parameters inside the helical coil. The flow inlet velocity was changed from 1 to 3 m/s with an increment of 0.5 m/s in each numerical study. It is found out that heat transfer characteristics are dependent on the flow and geometrical dimensions of the helical coil. Hence fluid-wall heat transfer can be taken as a relevant design for helically coiled heat exchangers. Flow velocity through the helical pipe was considered to be in the range of 1 to 3 m/s. The SST $k$-ω turbulence with standard wall function was used for this analysis.

Pressure-velocity coupling was resolved using the SIMPLE algorithm having skewness correction factor as 1. For pressure and momentum linear discretization was done. Power law was used for turbulence kinetic energy and turbulence dissipation rate. Second order unwinding was used for energy equation. The convergence values were given as 1.0e-0.8 for energy equation and 1.0e-06 for turbulence, mass and momentum.

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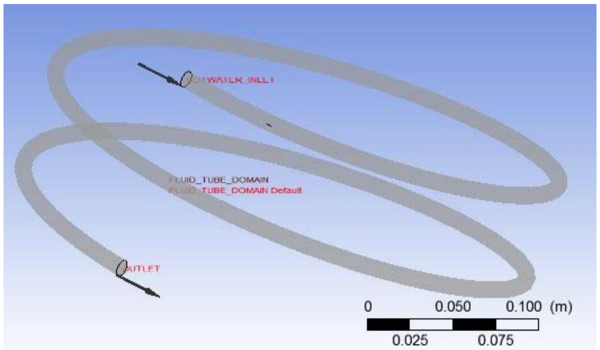


**Figure-4.** 3D modeling of 60 mm coil pitch case.

The geometry created in UniGraphics software is converted into parasolid kernel so that data transfer from UG to ICEMCFD is achieved. The Figures 3 and 4 shows the geometry of the helical coil with 30 mm pitch and 60 mm pitch respectively after importing from ICEM CFD to ANSYS CFX Pre for defining the boundary conditions. Figure-5 represents the meshing done on the 30mm helical coil using ICEM CFD tool with all hexahedral elements. It can be observed that the first cell height is located at a distance of 0.001 mm from the wall in order to capture the thermal and hydraulic boundary layers. The y+ value obtained is less than 1 and hence SST k-ω model can be applied with confidence to capture the real heat transfer characteristics happening between the fluid wall interfaces. The entire domain is discretized into more number of finite volumes by hexahedral elements for capturing this physics. The meshing achieved on the inlet and the outlet of the helical pipe of 30 mm coil pitch with bunching is shown in Figures 6 and 7, respectively.

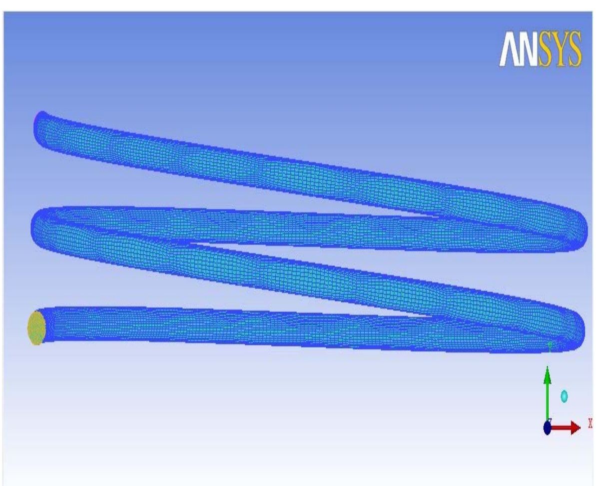


**Figure-5.** Mesh done on the helical coil surface for 30 mm coil pitch.

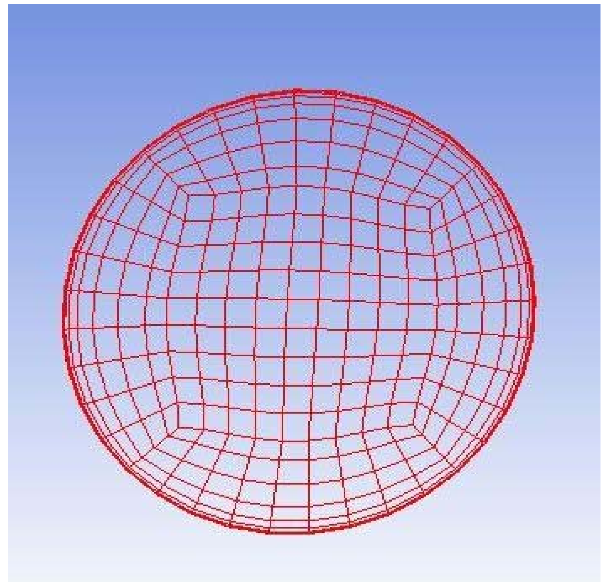

**Figure-6.** Meshing done on the inlet of the helical coil.

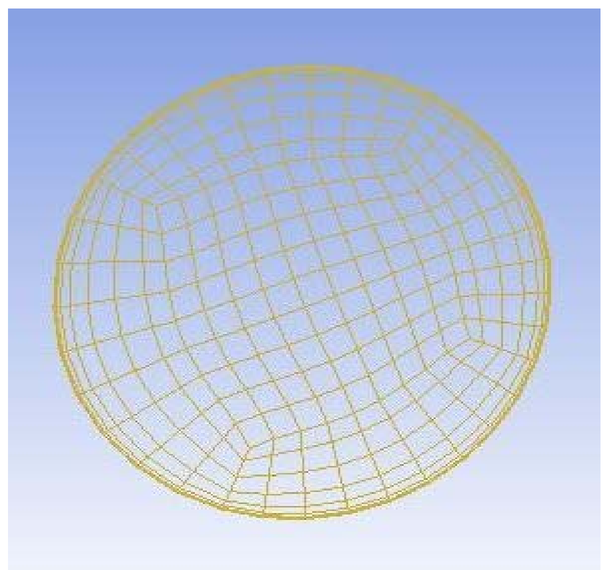

**Figure-7.** Meshing done on the outlet of the helical coil.

## 5. GRID INDEPENDENCE STUDY

The helical coil with pitch of 30 mm is meshed with hexahedral elements with three different nodes of 278512, 398729 and 509345 nodes. It was found that the numerical results predicted do not vary beyond 398729 nodal points (i.e.) the solution is grid independent beyond 398729 nodes. Similar grid independence studies were carried out for all the simulations carried out in this simulation.

## 6. VALIDATION

The helical coil with 30 mm pitch is modeled as per the literature of Jayakumar *et al*. Figure-8 shows the variation of inner Nusselt number to than of Dean number

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for the present numerical study and that of the literature of Jayakumar *et al*. Figure-8 shows the comparison and it was seen that the predicted values are in good agreement with that of the experiments available in the literature. The maximum deviation between the numerical results and experimental data occurs at higher Dean Number compared to lower Dean number and this deviation is less that 6.5 % even at higher Dean numbers. Thus the validity of the CFD code is carried out and hence the CFD code can be further extended to undergo parametric study for helical coil with 60 mm pitch.

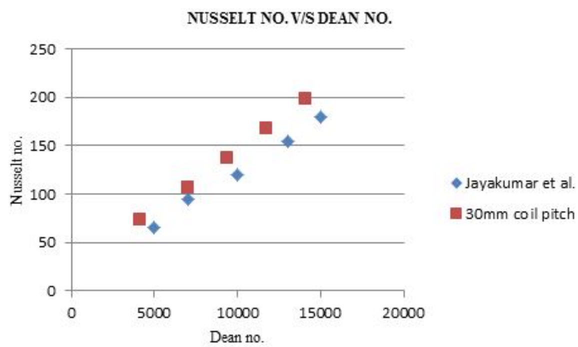


**Figure-8.** Comparison between present and studies of Jayakumar *et al*.

## 7. RESULTS AND DISCUSSIONS

Parametric studies are done for helical coil heat exchanger with 30 mm and 60 mm coil pitch by varying the inlet velocity from 1 m/s to 3 m/s with increment of 0.5 m/s. The different heat transfer characteristics of the helical coil heat exchangers are plotted for both 30 mm coil and 60 mm coil pitch. The heat removed from helical coil heat exchangers is plotted against pressure drop as shown in Figure-9. The blue color readings indicate for 30 mm coil pitch and red color are those of 60 mm coil pitch. The heat removed increases with increases in velocity from 1.0 m/s to 3.0 m/s which may be due to the increase in the heat transfer coefficient and thereby resulting in increasing the Nusselt Number as shown in Figure-10. For both 30 mm and 60 mm coil pitch. The maximum heat removed from the working fluid in case of 60 mm coil pitch is nearly 32 kJ whereas it is only 18 kJ for 30 mm coil pitch for the same maximum velocity of 3 m/s. The corresponding pressure drops are 22000 Pascal and 8000 Pascal respectively for 60 mm and 30 mm coil pitch respectively. The maximum Nusselt number obtained are 410 and 200 for 60 mm coil and 30 mm coil pitch respectively.

As expected, the heat removed is directly proportional to the Nusselt number, (i.e.) as the Nusselt number increases the heat removed from the working fluid also increases as shown in Figure-11. For the same Dean number the Nusselt number variation for a 60 mm coil pitch is nearly more than twice compared to a 30 mm coil pitch helical heat exchanger as shown in Figure-12. From the analysis made it was found that 60 mm coil pitch is better compared to a 30 mm coil pitch at all velocities from 1 m/s to 3 m/s.

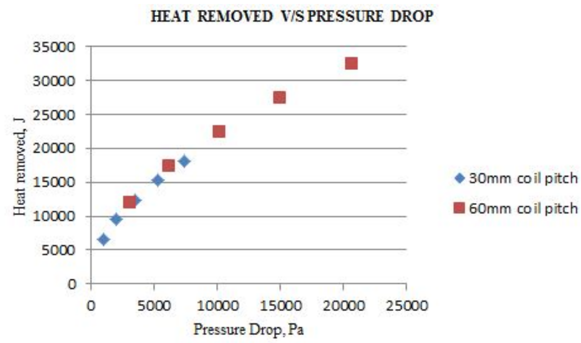


**Figure-9.** Heat removed v/s pressure drop plot.

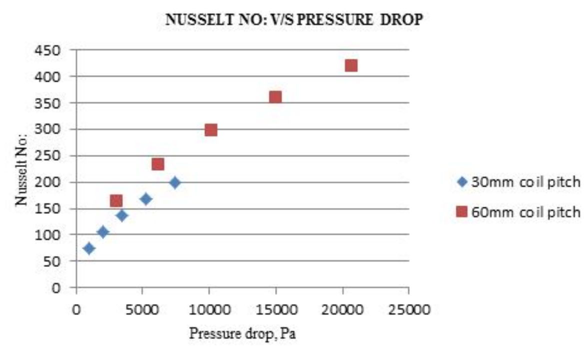


**Figure-10.** Nusselt number v/s pressure drop plot.

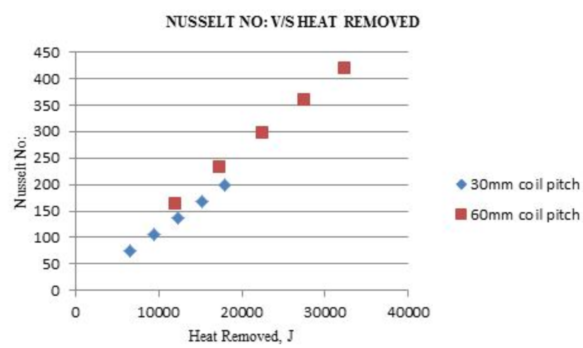


**Figure-11.** Nusselt number v/s heat removed plot.

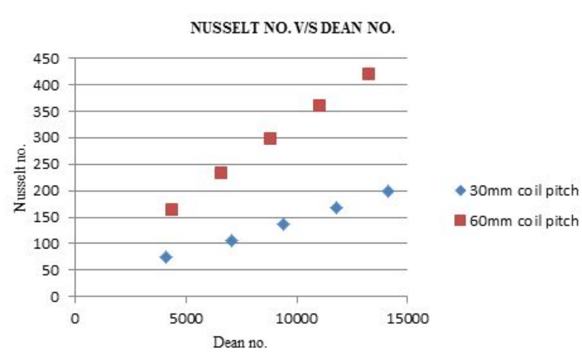


**Figure-12.** Nusselt number v/s Dean Number plot.

The above argument of 60 mm coil pitch is better than 30 mm coil pitch with respect to heat transfer characteristics can be further justified with the help of Figures from Figures 13 to 18. The temperature contour at the outlet of the 30mm coil pitch for 1 m/s velocity is shown in Figure-13. The temperature drops from 360 K to 342 K along one side and on the other side the temperature is 327.3 K. The difference in the temperature in the outlet plane is due to the difference in the velocity of the working fluid due to the curvature of the helical coil heat exchanger. This variation would be minimal for a smooth circular tube without curvature. As expected this temperature variation is higher for 60 mm coil pitch compared to a 30 mm coil pitch as shown in Figure-14 where the temperature variation at the outlet plane is 21 K whereas it is 15 K for the corresponding 30 mm coil pitched helical heat exchanger. Thus from these contours it is identified that the thermal loading on a 60 mm coil pitch would be more compared to a 30 mm coil pitch helical heat exchanger. The temperature contours along the wall for a 30 mm and 60 mm coil pitch for an inlet velocity of 1 m/s is shown in Figures 15 and 16, respectively and the temperature drops along the wall is more for 60 mm coil pitch. It should also be noted that more temperature drop is obtained with a limitation of increasing coil length of 60 mm coil pitch corresponding to a 30 mm coil pitch. Hence the space occupied by a 60 mm coil pitch is more than a 30 mm coil pitch.

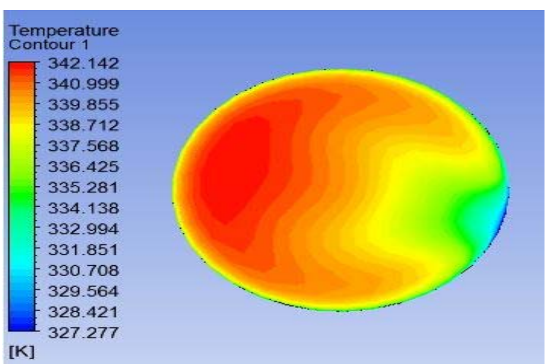


**Figure-13.** Temperature contour plot for outlet with inlet velocity 1m/s and coil pitch 30 mm.

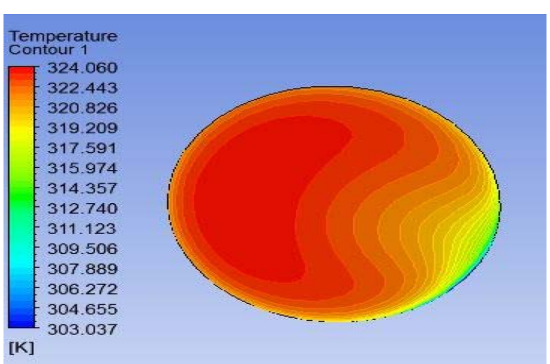


**Figure-14.** Temperature contour plot for outlet with inlet velocity 1 m/s and coil pitch 60 mm.

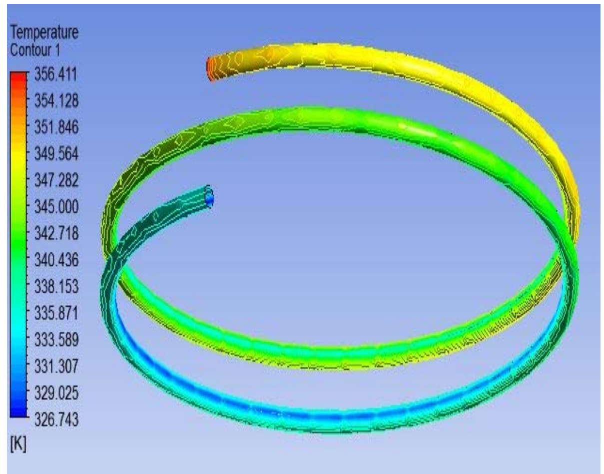


**Figure-15.** Temperature contour plot along wall with inlet velocity 1m/s and coil pitch 30 mm.

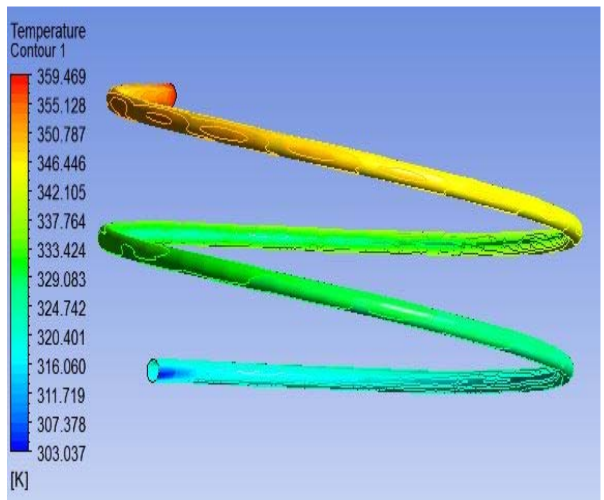


**Figure-16.** Temperature contour plot along wall with inlet velocity 1 m/s and coil pitch 60 mm.

The pressure contours along the wall of the 30 mm and 60 mm coil pitch of a helical coil heat exchanger is shown in Figures 17 and 18, respectively for an inlet velocity of 1 m/s. The pressure drop variation is around 1150 Pascal for 30 mm pitch coil whereas it is 3275 Pascal for 60 mm pitch coil for the same inlet velocity of 1 m/s. Thus it is inferred that the heat transfer characteristics of 60 mm coil pitch are achieved with an associated more pressure drop compared to its corresponding 30 mm coil pitch counterpart.

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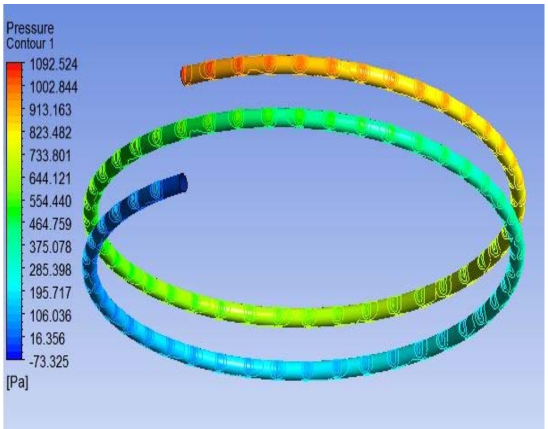


**Figure-17.** Pressure contour plot along wall with inlet velocity of 1 m/s and coil pitch 30 mm. velocity of 1 m/s and coil pitch 30 mm.

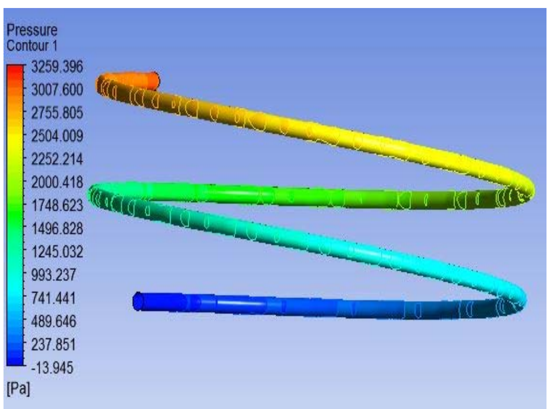


**Figure-18.** Pressure contour plot along wall with inlet velocity of 1 m/s and coil pitch 60 mm.

## CONCLUSIONS

The analysis on the 30 mm and 60 mm coil pitch for the helical coil heat exchanger has put forward the below observations. This observation can been seen very helpful for the enhancement in product life cycle of the heat exchanger especially in design and service sectors. The heat removed increases for 60 mm coil pitch compared to that of 30 mm but with an evident increase in the pressure drop. Velocity and the least for 60 mm coil pitch with 3 m/s inlet velocity. As the velocity increases the ratio of pressure drop to heat removed is comparatively less for 60 mm coil compared to the change in 30 mm coil pitch. The Nusselt number increases in direct proportion to the pressure drop at lower Dean Number and the profile is parabolic and the steepness decreases for higher Dean Numbers.

The Nusselt number to heat removed shows that the values are directly linear to each other. The ratios for different velocities are nearly the same. So no much variation has been observed when comparing flow and dimensional parameters in this case. The Nusselt number to dean number shows that it is same for 60 mm case, but it shows that the ratio is more in 60 mm coil pitch than that for 30 mm coil pitch.

Thus the heat transfer characteristics of a 60 mm coil pitch are better compared to a 30 mm coil pitch at higher Dean Number with limitation in space and more loss in pressure drop.

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