Design of Helical Coil Heat Exchanger for a mini powerplant

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Abstract— Helical Coil Heat Exchanger (HCHE) is a type of heat exchanger which has a shell called annulus and inside it, there is a helical coil. It occupies less space and provides more surface area for effective heat transfer as compared to shell and tube heat exchanger. HCHEs find their use in compact nuclear reactors, e.g., Small Modular Reactors (SMR). With the help of our project work, the industry will become familiar with the benefits of HCHE. That's why, HCHE is designed by taking essential parameters like heat load, inlet and exit temperatures of both coil and the annulus and the velocity inside the coil from industry (Altern Energy Ltd). In the present work, an HCHE with a heat load of 25kW is designed and developed. Pressurized hot fluid in the annulus side at 106 Pa is used as a heating medium that transfers heat with cold fluid at 105 Pa in counterflow configuration. Both inlet and outlet temperatures have been taken from the power plant. Stainless Steel material is chosen with grade 304 (SS304) because of its good anti-corrosion characteristics and ease of availability. Water is used as a working and primary fluid. For this purpose, a helical coil of a 19 mm pitch with an average helix diameter of 305 mm is used. Temperature and flow rate of water is varied in the annulus side and helical coil, and the heat transfer characteristics of the helical coil are investigated. The design of the heat exchanger in current work provides a full range of data on helical coil heat exchanger outlet conditions as a function of input parameters. The input parameters include the amount of constant heat flux, flow rate and temperature of primary and working fluids. Results have shown the fact that there is a rise in overall heat transfer coefficient and heat transfer rate when the mass flow rate of cold water in coil or hot water in the annulus is increased. It is found that the helical coil is good in terms of heat transfer.

Keywords— Heat Exchanger, CFD, LMTD, Heat Transfer, Computer-Aided Design

1 INTRODUCTION

HEAT exchangers can be found frequently in process industries like fertilizer plants, nuclear power plants, etc. They are used to heat, cool, vaporize or condense different fluid streams in industry. There are a number of types of heat exchangers. Every type has its own benefits and drawbacks. Active and passive techniques are used to increase the heat transfer coefficient. Active techniques are those in which external forces like fluid vibration, electric field, and surface vibration are required. While passive techniques are those in which some specific surface geometries or fluid additives like various tube inserts are required. To improve the heat transfer coefficient in heat exchangers, both techniques are widely used. New types of heat exchangers have been developed by doing research on heat exchangers by following any of these techniques. Usually, in industries, shell and tube types are commonly used. Our motive is to introduce industries with some other type of heat exchanger (as HCHE) as it is better in terms of heat transfer than shell and tube heat exchanger.

Helical Coil Heat Exchanger (HCHE) is being designed and their thermal characteristics are studied for their use in next-generation nuclear power plants. Majority of new generation nuclear power plants that are under construction are expected to use helical coil heat exchangers/steam generators instead of U-tube steam generators because of their numerous merits over shell and tube type heat exchangers. Helical coil heat exchangers provide more surface area for effective heat transfer. HCHEs can be operated on high temperatures and pressures. Moreover, they have low installation and maintenance costs.

Basic objectives for this project are the determination of geometrical parameters empirically which will enable us to get the required heat transfer and then a comparison of the thermal performance of HCHE empirically and numerically. Design calculations are done using the LMTD method while, numerical analysis will be done by using commercial software.

Two types of methods can be used for the designing of heat exchangers effective NTU method & LMTD method. Owing to the requirements, LMTD method to be used. The procedure for our work according to the LMTD approach will be the process design of HCHE and then the determination of average values of temperature, density, viscosity, mass flow rate on the coil side and annulus side. Then the overall heat transfer coefficient will be found by using convection correlations. The overall surface area will also be determined. The next step will be to find the length of the coil and the pitch number of turns. Finally, a completely mechanical design based on ASME codes will be made.[1]

The essential design parameters are taken from the mini power plant installed in table 1

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TABLE 1
ESSENTIAL PARAMETERS TAKEN FROM INDUSTRY

Essential Parameters		
Capacity	25KW	
Parameter	Coil Side	Annulus Side
T_{ci}	25° C	150° C
T_{co}	85° C	70° C
Pressure	1 atm	10 MPa
ΔT_c	55 ° C	110 ° C
Velocity	3 ft/s	0.0232 ft/s
Outer diameter	0.5 in	14 in
Inner diameter	0.42 in	10 in

Tci = Inlet temperature of coil Tco= Outlet temperature coil.

2 Literature Review

The literature review related to HCHE has been divided in two sections. The first part contains research related to HCHE design and in the second part, work done on simulation software related to HCHE has been discussed. HCHE is a type of heat exchanger in which the straight coil is replaced by a helical coil, hence heat transfer coefficient is increased. They have an increased surface area for heat transfer per unit volume. Their firm structure makes them capable of bearing high temperatures and pressure. Nowadays they are in extensive use in industries because of space limitations and its efficient performance.

Patil et al. 1984[2] designed and developed a thermal-hydraulic loop for a helical coil steam generator. Pressurized hot fluid in the shell side at 106 Pa was used as a heating medium which transferred heat with cold fluid at 105 Pa in counterflow configuration. Heat transfer took place across the coil wall and the fluid flowed inside the coil and the annulus. In order to get the required heat transfer the dimensions of the two cylinders were determined by using the velocity of the fluid in the annulus. Water was used as a working and primary fluid. Temperature and flow rate of water was varied in the shell side and helical coil, and the heat transfer characteristics of the helical coil were investigated.

They designed a thermal-hydraulic loop with the input parameters which include the amount of constant heat flux, flow rate and temperature of primary and working fluids. They used the Colburn factor to find the inside coil heat transfer coefficient. They found that the Reynolds number and Colburn factor are directly related to each other.

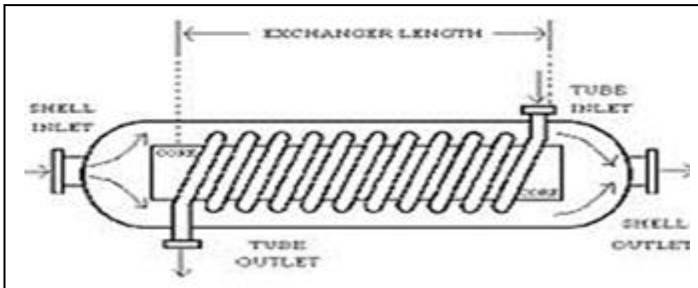


Fig. 1. Helical Coil Heat Exchanger [2]

Prabhanjan et al. 2004[3] performed the verification of the natural heat transfer coefficient from a vertical helical coil. Their results showed that for bath temperatures of 40^0 C and 50^0 C, the heat transfer coefficient for the helical coil is 1.16 and 1.43 times higher than for straight tube heat exchanger, respectively.

Dr. Elsayed [4] studied helical tubes with different fluids and for laminar and turbulent flows. He found that in laminar flow, an increase in heat transfer was much higher than turbulent flow. An increase in heat transfer of 3.25% was reported in a laminar while just 1.1% in turbulent. He found that the use of nanofluids results in a very significant increase in heat transfer and the pressure drop is almost the same for laminar and turbulent flows with a slight 5% difference for pure and nanofluids. Further, results show that the addition of nanoparticles in pure fluids also results in a greater heat transfer.

Korane et al. 2012[5] did the comparative analysis and studied about the friction factor characteristics of the helical coil heat exchanger on both laminar and turbulent flow for two geometries, one was helical circular coil and second was square coil as shown in the figure.

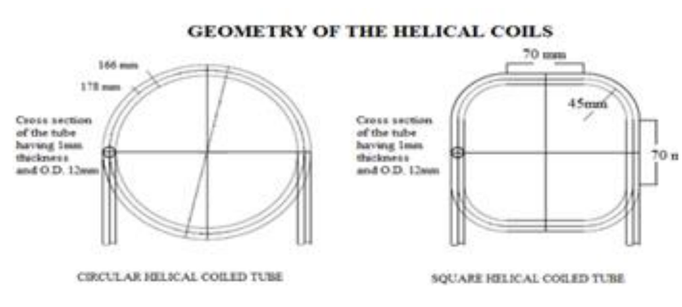


Fig. 2. Circular and square cross sections [5]

Their result shows that, in circular coil, friction factor and pressure drop is high as compared to square coil due to the curved portion of the coil. They concluded that the square helical coil heat exchanger performs well as compared to the circular coil.

Ankanna and Reddy 2014 [6] did the analysis based on the performance of the heat exchanger that affects the effectiveness of the heat exchanger by changing different parameters. They found that in the helical coil heat exchanger for parallel flow, the overall heat transfer coefficient is more as compared to the counter flow. The overall heat transfer coefficient remains the same with enhancement in the number of turns in counterflow but it decreases rapidly with the increase in a number of turns in parallel flow. The effectiveness is greater in counterflow as compared to the parallel flow.

Shirgre et al. 2014 [7] made use of a helical coil instead of a straight tube heat exchanger. Their set-up involved a helical coil made up of copper shell, heater, devices for flow measurement and cold-water source. Most of the investigations done by them on heat transfer coefficients were either for constant heat flux or constant wall temperature. First, they kept the hot water flow rate constant and then the cold-water mass flow rate. By doing this, the effect of the mass flow rate for cold and hot water on the effectiveness of heat exchanger was studied separately. Due to the heat transfer enhancement, the size of the heat exchanger became sufficiently small.

Puttewar and Andhare 2015 [8] were familiar with the inlet and outlet temperature of both hot and cold fluids, therefore, they used the LMTD method. They found the overall heat transfer coefficient and the convective heat transfer coefficient. Graphs of different parameters like average heat transfer, overall heat transfer, temperature of hot water at outlet and effectiveness against the mass flow rate of hot water were plotted and they found that the effectiveness, overall heat transfer coefficient, heat transfer rate, and hot water outlet temperature increase by increasing the mass flow rate of hot water.

Ranaware et al. 2015 [9] compared shell and tube heat exchanger with helical coil heat exchanger. It was found that the heat transfer in shell and tube heat exchanger depends on the Reynolds and Prandtl numbers for both laminar and turbulent flow. There are many factors on which the heat transfer of shell and tube heat exchanger depends, for instance, tube length and baffle number, etc. In their research, they concluded that helical coil heat exchangers are small and provide benefits. It can bear high temperatures without induced stresses and it can be operated on high pressures. The heat transfer rate of HCHE depends on the coil pitch, pitch circle diameter and the diameter of the pipe being used. HCHE counter flow has a high overall heat transfer coefficient while for straight tube parallel flow, the heat transfer coefficient is very low. Finally, they showed that the effectiveness of HCHE for counter flow is higher when compared to a straight tube parallel-flow heat exchanger.

Alimoradi and Veysi 2016 [10] investigated physical properties (CP etc.) operational parameters (i.e. the velocity v and temperature T of fluid) and geometrical parameters (pitch, diameter etc.). They investigated 42 cases and 15 test measurements were taken at the moment when there was a steady state. On both sides, water was used as a working fluid, they assumed to be dependent on temperature. Results showed that increasing pitch Nusselt number on the annulus side increases by 10% while on the coil side only 0.8%. They found that by increasing diameter and height by 50%, the Nusselt number decreases by 34.1 % and 28.3% for annulus and coil respectively. Based on results, they found two correlations to predict Nusselt and Prandtl's number on both annulus and coil side. The relations are as follow:

$$Nu_c = 0.255\,\mathrm{Re}_c^{0.685}\left(\frac{d_c}{d_{t,i}}\right)^{-0.216}\left(\frac{d_v}{d_{t,i}}\right)^{0.024}\left(\frac{d_{sh}}{d_{t,i}}\right)^{-0.012}\left(\frac{Hc}{d_{t,i}}\right)^{-0.03}\left(\frac{H_{sh}}{d_{t,i}}\right)^{-0.045}\left(\frac{f}{d_{t,i}}\right)^{0.013}\left(\frac{p}{d_{t,i}}\right)^{0.011}\mathrm{Pr}_c^{0.315} \quad (1)$$

$$Nu_{sh} = 0.247\,\mathrm{Re}_c^{0.723}\left(\frac{d_c}{d_{t,o}}\right)^{0.378}\left(\frac{d_v}{d_{t,o}}\right)^{0.556}\left(\frac{d_{sh}}{d_{t,o}}\right)^{-0.82}\left(\frac{Hc}{d_{t,o}}\right)^{0.043}\left(\frac{H_{sh}}{d_{t,o}}\right)^{-1.03}\left(\frac{f}{d_{t,o}}\right)^{0.561}\left(\frac{p}{d_{t,o}}\right)^{0.138}\mathrm{Pr}_{sh}^{0.717} \quad (2)$$

Acharya et al. 2001 [11] studied the enhancement of heat transfer in steady, laminar flow through a tube due to particle paths that are chaotic. They did a comparison between the regular mixing and chaotic mixing by considering two coils with regular and chaotic mixing. They found that periodic rotation of the coil axis results in a mixing which is chaotic. Lyapunov constant (it is used for the separation rate of very close trajectories) was used as mixing was chaotic. They also analyzed the velocity vectors and temperature fields. They set up the concept, in a quantitative manner, that chaotic mixing helps us to design an efficient coiled heat exchanger. They also studied the effect of changing axis coil geometry on heat transfer. They found that a 7-20% increase in heat transfer can be achieved by alternating the axis coil geometry and that increase is in terms of Nusselt number (fully developed) with a very small change in pressure drop. The range of Reynolds number was $50 \le Re \le 1200$.

Jayakumar et al. 2008 [12] studied the heat transfer between fluids in the helical coil and compared the heat transfer characteristics for various boundary conditions inside the helical coil. Computational fluid dynamics software FLUENT 6.2 was used for numerical analysis. They considered the conjugate heat transfer condition and properties that are dependent on the temperature of heat transfer media. They considered inside and outside convective heat transfer and wall conduction. They used values of thermal and transport properties of the heat transfer medium which are dependent on temperature. They found that, through the helical pipe, the flow rate of the hot fluid was low. They performed regression analysis on MATLAB to obtain relations for μ, ρ, κ, cp and the relations were later programmed by them in FLUENT. The relations which were used by them are as follow:

$$\mu(T) = 2.1897 - 11T^4 - 3.055e - 8T^3 + 1.6028e - 5T^2 - 0.003752T + 0.33158 \quad (3)$$

$$\rho(T) = -1.5629e - 5T^3 + 0.011778T^2 - 3.0726T + 1227.8 \quad (4)$$

$$k(T) = 1.5362e - 8T^3 - 2.261e - 5T^2 + 0.010879T - 1.0294 \quad (5)$$

$$Cp(T) = 1.1105e - 5T^3 - 0.0031078T^2 - 1.478T + 4631.9 \quad (6)$$

Kumar & Karanth 2013 [13] did numerical studies of helical coil heat exchanger using CFD. They applied fixed wall temperature boundary conditions to study the case of cooling hot water with the help of a heat exchanger. The CFD analysis was done using ANSYS 14.0. They performed CFD analysis on a straight tube of the same diameter and length as a coil to establish the effectiveness of the helical coil heat exchanger numerically. The three-dimensional computational domain was modeled using hex mesh for both models (straight tube and helical coil). They performed a Grid independence test to prove the validity of mesh. Parameters such as temperature drop, pressure drop, heat transfer co-efficient and Nusselt number were compared with numerical results. They used three different mass flow rates 0.005 kg/s, 0.02 kg/s, 0.05 kg/s for their analysis and their corresponding Reynolds numbers are 1068,4274 and 10685 respectively. In the coil, as the fluid particles undergo the rotational motion and due to centrifugal force movement of particles through core towards the boundary, they cause various trajectories, therefore, the particle behavior in the helical coil was observed to be in line at inlet side, but found to be in scattered state at the outlet. To validate the results, values of Nu and heat transfer coefficient from different correlations were compared with experimental values. For the Nusselt number, the average error was found up to 5% for the helical coil. The heat transfer rate was observed to be increased by 11% when the mass flow rate was increased from 0.005 kg/s to 0.05 kg/s. The experimental results and correlation had a good agreement.

For further detail and investigation on CFD analysis of HCHE, Tayde et al. 2015[14] studied different boundary conditions in order to analyze the heat transfer characteristics inside a helical coil. CFD methodology was applied by utilizing the simulation power of FLUENT 14.0. The flow field was simulated by solving governing equations. Their optimum goal was to obtain results for different boundary conditions using FLUENT and comparison of these experimental results with each other to pass some judgment. To check the properties of heat transfer medium such as thermal and transport properties, they also performed the analysis dependent on temperature as temperature change results in a change of fluid properties. By performing a comparison among above all results they found that the prediction of results should be inaccurate by using constant thermal properties and transport properties. Another reason for inaccurate prediction is the use of constant wall temperature as an arbitrary boundary condition. The temperature-dependent analysis (by keeping in view the fluid properties based on temperature) gives accurate results and has fewer errors.

Balamurugan et al. 2015 [15] studied the effect of different parameters as coil diameter, pipe diameter, coil axial pitch, number of coils, orientation between coils on the helical coil performance. The 3d simulation is initiated to understand the velocity pattern inside the helical coil, and to understand the effect of different parameters on the helical coil, the pressure drop was calculated. The simulation result of the pressure drop was compared with experimental results. They selected a suitable model on the basis of their condition. They created helical geometry on commercial CFD software COMSOL. In the mesh independent test, they analyzed both laminar and turbulent model by applying velocity boundary condition. They found through mesh independent test that pressure drop for different mesh sizes for the laminar model is almost the same while in the case of k- (turbulent) model pressure decreases by increasing mesh size. Therefore, as at higher velocities pressure drop predicted by k- is more precise so they selected k- model. Simulation results were three trends; first that pressure drop increases with respective increase in coil diameter and increases in a number of helical coils, second that pressure drop remained constant and there was no change observed by the change in helical coil pitch and third that pressure drop decreases with respective increase in pipe diameter. They found that the effect of axial pitch is almost negligible on pressure drop in the helical coil.

3 Methodology

3.1 Design

In simple shell and tube heat exchangers, heat is transferred from one fluid to the other working fluid without appreciably interacting with the separating walls while in helical coil heat exchangers, there occurs turbulence due to helix of the coil. This turbulence of the fluid causes vibrations in the helical coil. But, these-vibration are not sufficient at low velocity (mass flow rate).

Mechanical design of helical coil heat exchanger consists of the design calculations which relate the material strength, stability, robustness etc. with the operating conditions in which the system is to be installed. In the helical coil heat exchanger, the strength required of an element in a system is an important factor. Therefore, strength is a design consideration for the determination of geometry and the dimensions of the element. Hence, there should be modifications in the mechanical design of the helical coil to reduce the effects of these limitation design conditions.

When an enquiry of the heat exchanger is received, the first step is to analyze the application of its use. The design engineer must define the type of heat exchanger that can meet the requirements of the application of its use. The design temperature and pressure and maximum allowable pressure drop must be defined for product and service fluid. Inside the annulus, the pressure is 106 Pa while inside the coil, it is 105 Pa.

The minimum required thickness of the cylindrical shell under internal pressure shall not be less and Maximum operating pressure shall not be greater than that computed by the following formulas, given

E = 0.85 from Table UW-12 ASME Section VIII – Division I.
Maximum $P = 1.2\ MPa$
$R = 190.5mm$

$t = 12.7mm$
Maximum T= 150^{o}C
Maximum Allowable Pressure Drop=5Psi

The next step is to analyze the product and service fluids involved in the application. The better the fluid properties are understand, the better will be the design of the heat exchanger. Any mistake in the physical properties can lead to the wrong design of the heat exchanger.

After defining physical properties, the next step is to apply energy balance. Normally the product flow rate and desired entry and exit temperature of the product fluid are defined. Type of service fluid should also be defined and two of these three parameters: flow rate, service entry or exit temperature should be defined. With any of these two knowns, applying the energy balance gives the third parameter. After this step, flow rates and entry and exit temperatures of product and service fluid are fixed.

geometry of heat exchanger using an empirical approach is defined by the design engineer. Tube diameter, shell diameter in which tubes are to be placed, number of tubes, wall thickness and length of the inner tube are defined. The choice of materials is also made at this step. Material SS-304 is selected and its properties are used in evaluating the minimum thickness and load calculations of the pressure vessel.

From ASME Code Section II Part D Table 1A

Nominal Composition: 18Cr – 8Ni
Type: 304
Minimum Tensile Strength: 515 MPa
Minimum Yield Strength: 205 MPa
Maximum Allowable Strength @ 150^{o}C = S = 130 MPa

Thermal design of heat exchanger includes the consideration of many interactive design parameters which are summarized as follows:

- Process fluid assignments to shell and tube side
- Stream temperature specification
- Shell and tube side pressure drop design limits
- Shell and tube side velocity limits
- Shell and tube side heat transfer coefficient and fouling factor coefficient

At start, desired outlet temperature of secondary or product fluid for given inlet temperature is defined. So,

Inlet temperature = Tci = 25^{o}C
Outlet temperature = Tco = 85^{o}C

In heat transfer, the bulk temperature or average bulk fluid temperature is considered to be the convenient reference point for the evaluation of properties related to convective heat transfer. Like density, viscosity, thermal conductivity and specific heat vary with temperature so the question arises what temperature should be used for evaluation of these properties. For natural convection, the fluid properties are evaluated at film temperature which is the average of wall temperature and bulk temperature. But for forced convection as in our case, the fluid properties are evaluated at a mean bulk temperature of the helical coil which is the average of bulk inlet and bulk outlet temperature.

$$\Delta Tc = (T_{ci} + T_{co})/2 = 55^oC \qquad (7)$$

As stated earlier that thermal properties of water like thermal conductivity, density and specific heat are found to vary with temperature so using the same values of properties at all temperatures leads to the failure of design.

Density, specific heat, thermal conductivity, and viscosity for secondary fluid i.e. water flowing through helical coil are evaluated at 55^{o}C.

TABLE 2
PROPERTIES OF H2O AT ΔTC = 55^{O}C

ρ	985.2
C_p	4183
μ	0.504×10^{-3}
k	0.649
Pr	3.25

Density = ρ (kg/m3), Specific heat = Cp (J/kg. K), Dynamic Viscosity = μ (kg/m. s), Thermal Conductivity = k (W/m. K), Prandtl number = Pr, Tci = Inlet temperature of coil, Tco= Outlet temperature coil.

Similarly, the same procedure is used for the primary fluid coming from the pressurizer which exchanges its heat with the secondary fluid flowing through the helical coil.

Inlet temperature = Thi = 150 oC
Outlet temperature = Tho = 70 oC

$$\Delta Th = (T_{hi} + T_{ho})/2 = 110^oC \qquad (8)$$

Density, specific heat, thermal conductivity, and viscosity for secondary fluid i.e. water flowing through helical coil are evaluated at 110o C.

TABLE 3
PROPERTIES OF H2O AT ΔTC = 110^{O}C

ρ	950.6
C_p	4229
μ	0.255×10^{-3}
k	0.682
Pr	1.58

Density = ρ (kg/m3), Specific heat = Cp (J/kg. K), Dynamic Viscosity = μ (kg/m. s), Thermal Conductivity = k (W/m. K), Prandtl number = Pr, Tci = Inlet temperature of coil, Tco= Outlet temperature coil.

Electric heat source of capacity 25 kW is used which will increase the temperature of the primary fluid to 150^{o}C kept under pressure to avoid boiling. Then primary fluid at increased temperature level will be allowed to exchange heat with secondary fluid flowing through the helical coil.

Heating load = Q = 25 kW

TABLE 4
0.5 IN DIAMETER BWG GAUGE 18 TUBE DIMENSIONS

BWG Gauge	18
D (in)	0.402
d_o (in)	0.5
t (in)	0.049
v (ft/s)	4

Inner Diameter of Coil = D (in), Outer Diameter of Coil = do (in), Thickness = t (in), Velocity flow rate inside the Coil = v (ft/s)

The required outer diameter of the tube is 0.5 in and the required gauge is BWG gauge is 18. The dimensions of the tube are:

TABLE 5
NPS 10 AND 14 SCHEDULE 40 PIPE DIMENSIONS

Pipe Size (inch)	Pipe size (meters)
Inside diameter of cylinder (B) = 10 in	0.254 m
Outside diameter of cylinder (C) = 14 in	0.3556 m

The required NPS for the shell is NPS 14 – Schedule 40 and for helical coil support, pipe of NPS 10– Schedule 40 is used. The dimensions of the pipe are:

In our case for helical coil, the tubes are made of stainless steel. The reason for using stainless steel is due to its increased thermal conductivity in addition to stability and firmness provided to the structure. It is also resistant to corrosion, chemical damage and heat damage. Thermal conductivity of stainless steel from literature is given as:

Thermal conductivity = k = 17 W/m2K = 14.627 kcal/h.m.K

The next step is to find out the velocity of the secondary fluid through the helical coil. The velocity is calculated by first computing the mass flow rate in the coil (m_c) by energy balance which is given as:

$$Q = m_c\, c_p\, (T_{co} - T_{ci}) \qquad (9)$$

$$m_c = 25000 / [4183\ (85 - 25)] = 0.0996 \text{ kg/s}$$

Now with a known mass flow rate, the equation of continuity is used to find out the velocity.

$$m_a = \rho\, v_c\, A_f \qquad (10)$$

$$A_f = \pi D^2/4$$

$$A_f = 3.1415 \times (0.0102)^2/4$$

$$A_f = 8.171\times10^{-5} \text{ m}^2$$

Substituting the values of Area A_f, density p, and mass flow rate m_a in eq. (10), the velocity is given as:

$$v_c = m_c / \rho A_f = 0.0996 / (985.2\times8.171\times10^{-5})$$

$$v_c = 1.22 \text{ m/s}$$

The following figure gives us the true picture of the helical coil and some parameters which are required to be calculated.

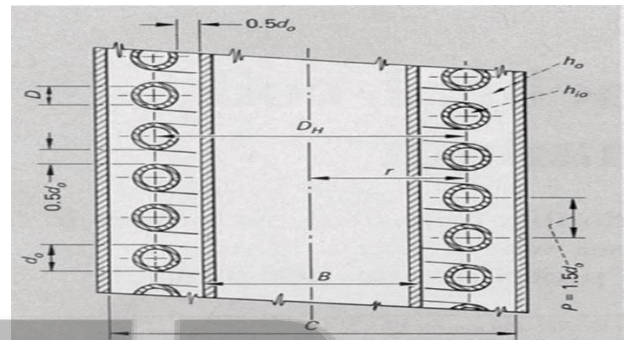


Fig. 3. Schematic cutaway view of an HCHE [2]

From the geometry as shown in figure.3, the average diameter of helix is given by:

Average diameter of helix = $D_h = (B+C)/2 = 0.3048$ m

Average radius of helix is calculated as:

Average radius of helix = $r = D_h/2 = 0.1524$ m

Inside Diameter of helix:

$$D_{h1} = D_h - d_o = 0.2921 \text{ m}$$

Outside Diameter of helix:

$$D_{h2} = D_h + d_o = 0.3175 \text{ m}$$

Pitch of the coil = $p = 1.5\times d_o = 0.019$

Length of coil for one turn can be easily calculated by the formula given by:

$$L = p + \sqrt{(2\pi r)^2} \qquad (11)$$

$$L = 0.019 + \sqrt{(2\times3.1415\times0.1524)^2}$$

$$L = 0.9577 \text{ m}$$

Volume occupied by one turn of coil= $V_c = \frac{\pi}{4}\times d_o^2\times L$ (12)

Substituting the values,

$V_c = 0.000121315\ m^3$

Volume of annulus (1 turn coil) = $V_a = \frac{\pi}{4}(C^2 - B^2) \times p$ (13)

$V_a = \frac{\pi}{4}(0.3556^2 - 0.254^2) \times 0.019 = 0.0009242\ m^3$

Volume in annulus = $V_f = V_a - V_c$ (14)

$V_f = 0.0008029\ m^3$

$D_E = 4V_F / \pi D_O L = 0.084021$ (15)

Clearance = $\left(\frac{C-B}{2} - d_o\right) / 2 = 0.01905m$ (16)

LMTD depends on the hot and cold fluid temperature differences at the inlet and exit of the heat exchanger.

$$\Delta T_m = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln\left(\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right)}$$

$$\Delta T_m = \frac{(150 - 85) - (70 - 25)}{\ln\left(\frac{150 - 85}{70 - 25}\right)}$$

$$\Delta T_m = 54.392\ K$$

- Prandtl number = $Pr = \frac{C_p \mu}{K}$ (18)

The thermal conductivity of water at the mean bulk temperature of secondary fluid i.e. 55° C, flowing through the tube is given as:

Thermal Conductivity = k = 0.649 W/m.K

Substituting the values of specific heat, viscosity and thermal conductivity in eq.18 Prandtl number come out to be:

$Pr = (4183 \times 0.504 \times 10^{-3}) / 0.649 = 3.25$

On shell Side:

$Pr = (4229 \times 0.255 \times 10^{-3}) / 0.682 = 1.58$

Reynolds number for the fluid flowing through the tube is given as:

$Re = (\varrho \times v_c \times D) / \mu$ (19)

$= (985.2 \times 1.22 \times 0.0102) / (0.504 \times 10^{-3})$

$Re = 24334.843 \approx 24335$

The mass flow rate of hot fluid in the annulus is given by energy balance eq.19 for annulus as:

$Q = m_a\, c_p\, (T_{hi} - T_{ho})$ (20)

Mass flow rate in the annulus = $m_a = Q / c_p (T_{hi} - T_{ho})$

$m_a = 25000 / 4229\ (150 - 70) = 0.07389\ kg/s$

Area of flow in annulus = $A_a = \frac{\pi}{4}[(C^2 - B^2) - (D_{h2}^2 - D_{h1}^2)$

$= \frac{\pi}{4}[(0.3556^2 - 0.254^2) - (0.3175^2 - 0.2921^2)] = 0.0365\ m^2$

Velocity at the annulus side (v_a) can be found from eq. as:

$m_a = \varrho \times v_a \times A_a$ (21)

$v_a = m_a / \varrho A = 0.07389 / (950.6 \times 0.365) = 2.1295 \times 10^{-3}\ m/s$

The Reynolds number now is given by eq.20 :

$Re = (\varrho \times v_a \times D_e) / \mu = (950.6 \times 2.1295 \times 10^{-3} \times 0.0840521 / (0.255 \times 10^{-3})$

$Re = 669.465 \approx 670$

Nusselt number = $Nu = \frac{hd}{K}$ (22)

Heat transfer coefficient on Coil Side:

As Re > 10000 so, the following co-relation is used for turbulent flow:

$h_i d_o / K = 0.023\ Re^{0.8} Pr^{0.4}$ (23)

$h_i d_o / K = 0.023 \times 24335^{0.8} \times 3.25^{0.4}$

$h_i = 118.97 \times (K / d_o) = 118.97 \times (0.609 / 0.0127)$

$h_i = 6078.85\ W/m^2K$

Heat transfer coefficient on Shell Side:

$h_o d_o / K = 0.6\ Re^{0.5} Pr^{0.31}$ (24)

$h_o = 144.79\ W/m^2K^o$

Fouling factor of shell side (For distilled water) = R_a
$= 0.0005\ h.m^2.K^o/kcal = 4.3 \times 10^{-4}\ m^2K^o/W$
Fouling factor of shell side (For tap water) = R_t
$= 0.002\ h.m^2.K^o/kcal = 1.72 \times 10^{-3}\ m^2K^o/W$

Overall Heat Transfer Coefficient:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + \frac{x}{k_c} + R_a + R_t \quad (25)$$

$$= U = 107.58\ W / m^2K^o$$

Area of Heat Transfer:

$A = \frac{Q}{U \times \Delta T_m} = 4.27\ m^2$ (26)

Turns:

$N = \frac{A}{\pi d_o \times L} = 113$ (27)

Height:

$H = (N \times p) + d_o = 2.14m$ (28)

Length:

L_t = N×L (length for one turn) (29)
= 113×0.9577
L_t = 107.1 m

3.2 Simulation

TABLE 6
SCALED-DOWN DESIGN PARAMETERS

Modeling Parameters	Original (mm)	Original/6 (mm)
Coil Diameter	12.7	2.116
Shell Diameter	355	59.16
Effective Shell length	2178	363
No. of coil Revolutions	113turns	19 turns
Helix dia	306	51

TABLE 7
SCALED-DOWN OTHER PARAMETERS

Parameters	Original (mm)	Scaled down (mm)
Coil Side		
Inlet Temperature	25^0 C	4.167 ^{0} C
Outlet Temperature	85^0 C	14.167 ^{0} C
Mass flow rate	0.0736 kg/s	0.012267 kg/s
Pressure	1atm	1atm
Shell Side		
Inlet Temperature	150^0 C	25^0 C
Outlet Temperature	70^0 C	11.66^0 C
Mass flow rate	0.073kg/s	0.012166kg/s
Pressure	10Mpa	1.67Mpa

CFD analysis of heat by following phases Pre-Processing, Solution Phase & Post Processing. First of all, the dimensions are utilized to develop the model. The model is scaled-down to save computational time.

Following are the assumptions made for the fluid flow in the heat exchanger.

1. Working fluid used is water
2. Fluid (water) is incompressible
3. Fluid properties assumed to be constant for the given temperature conditions
4. Zero-gauge pressure is assigned to the outlet

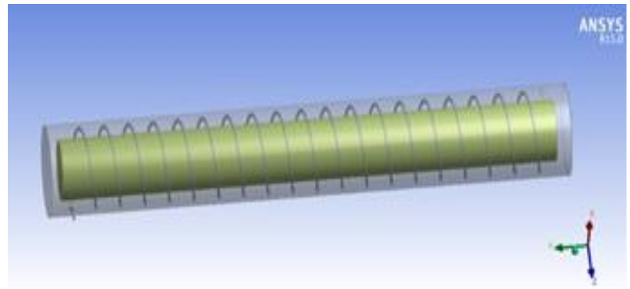


Fig. 4. Scaled Down model

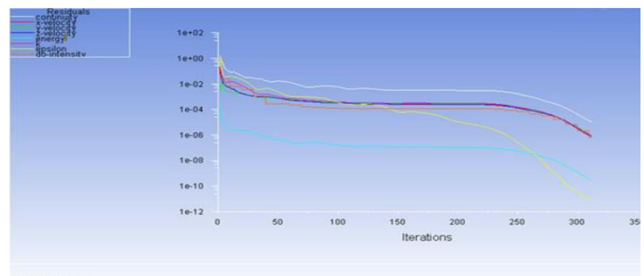
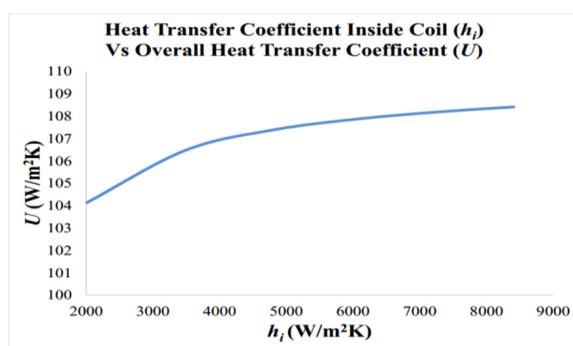


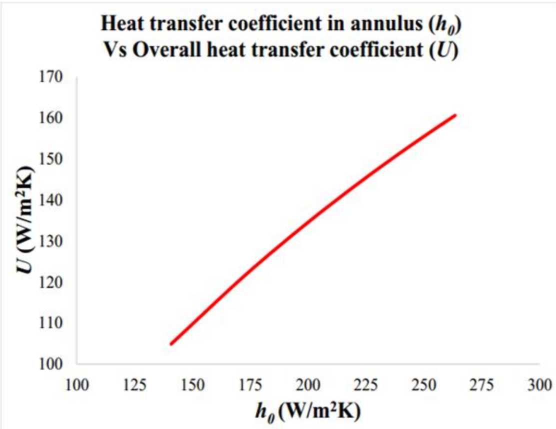
Fig. 5. Scaled Residual

4 RESULTS AND DISCUSSION

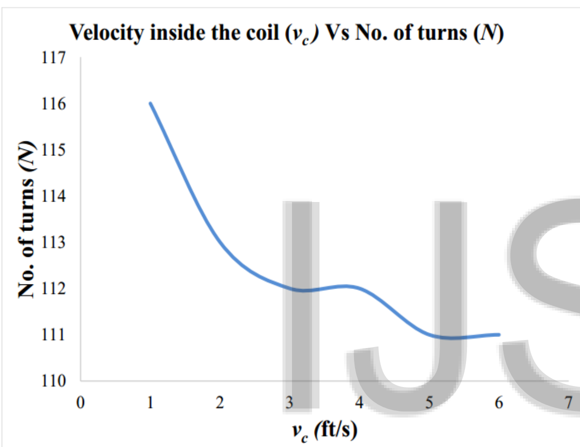
By calculations of our results, It is found that the overall heat transfer coefficient U is dependent on some factors like heat transfer coefficient inside the coil hi and the heat transfer coefficient in the annulus side ho.
It can be seen from the graph that the variation of U with hi is of less importance as compared to that with ho. There is a linear relationship between ho and U. As U increases, ho also increases and their trend is very close to an ideal trend i.e. a straight line. Overall heat transfer mainly depends on the heat transfer coefficient in the annulus (h_o).



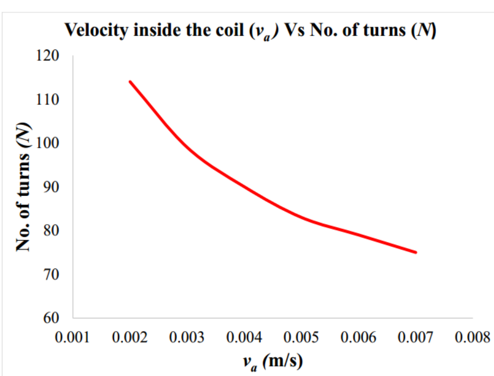
Plot 1. U vs h_i



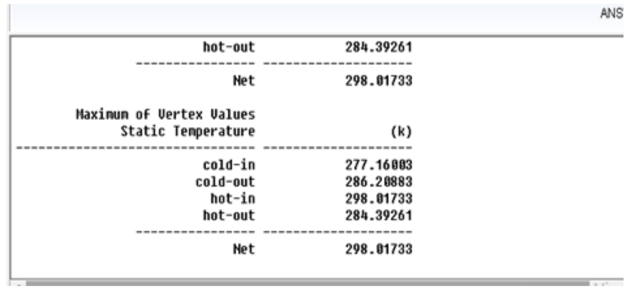
Plot 2. U vs h_o



Plot 3. N vs v_c



Plot 4. N vs v_a


```
                                                   ANS
                hot-out          284.39261
       ---------------- --------------------
                    Net          298.01733

  Maximum of Vertex Values
     Static Temperature                (k)
 ----------------------- --------------------
                cold-in          277.16003
               cold-out          286.20883
                 hot-in          298.01733
                hot-out          284.39261
       ---------------- --------------------
                    Net          298.01733
```

Fig 6. Temperature values

Similarly, the number of turns N is also dependent on velocity in the coil (v_c) and velocity in the annulus (v_a). As the number of turns is a major part of designing HCHE, hence its behavior with both velocities need to be studied. Graphs between these parameters clearly depict the dependence of N on v_c and v_a. When velocity in the coil increases, the number of turns initially decreases, then the number of turns remains constant for some values of increasing velocity inside the coil then again, the number of turns starts decreasing but this has fewer decreasing values. On the other hand, by increasing the velocity in the annulus, the number of turns decreases constantly. Its behavior is ideally inverse. Number of turns changes (decreases) abruptly with the velocity in the annulus and not with the velocity in the coil.

TABLE 8
RESULTS FOR THE DIFFERENT MASS FLOW RATE FOR SHELL FLUID

Sr. No.	Mass flow rate (m) kg/s	Temperature	
		inlet	outlet
1	0.012166	25° C	12° C
2	0.012222	25° C	11° C

TABLE 9
RESULTS FOR THE DIFFERENT MASS FLOW RATE FOR COIL FLUID

Sr. No.	Mass flow rate (m) kg/s	Temperature	
		inlet	outlet
1	0.012267	4° C	12° C
2	0.012333	4° C	13° C

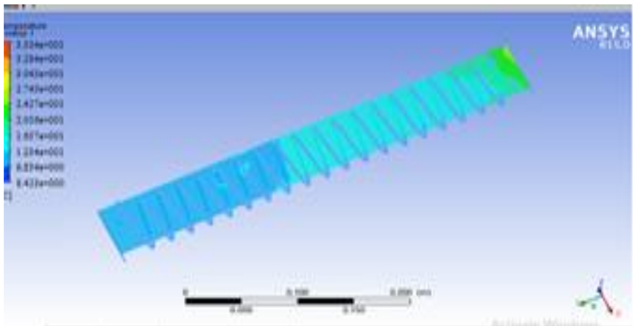


Fig 7. Temperature contours, shell (HCHE)

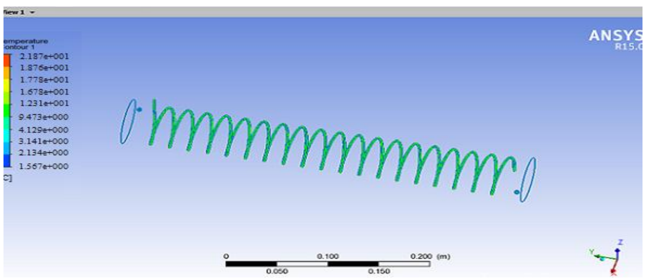


Fig 8. Temperature contours, coil (HCHE)

It is investigated that an increase in the mass flow rate increases the heat transfer rate. As the fluid is moving through the shell so the temperature is varying.

5 Conclusions

From the study and calculations being carried out, it is concluded that:

(1) Mean bulk temperature of annulus side, i.e. 110^0 C must be higher than the mean bulk temperature inside the coil i.e. 55^0 C (almost double)

(2) Temperature drop and pressure drop are greatly affected by changing the diameter of the tube

(3) We must use a coil of small diameter and tube of large diameter. This will enable us to have desirable pressure and temperature drop because the temperature drop is inversely proportional to mass flow rate while pressure drop is directly proportional to the flow rate. Increase in pressure drop increases cost larger pumps are more expensive it also increases operational cost

(4) Area of the cross-section of coil A_f which is dependent on the inner diameter of coil D is responsible for any change in the velocity inside the coil v_c

(5) Results clearly show that that variation of U with hi is of less importance as compared to that with ho. There is a linear relationship between h_o and U.

(6) The number of turns N is inversely proportional to the velocity of the annulus v_a and also to the velocity inside the coil v_c. The significant change in U occurs with a change in velocity of the annulus v_a as compared to the velocity inside the coil v_c

(7) At low Reynolds number, the efficiency of the helical coil is much better

(8) It was investigated that when there is an increase in mass flow rate of cold water in coil or hot water in annulus, then there is a rise in overall heat transfer coefficient and heat transfer rate. So, when the mass flow rate increases, the heat transfer rate also increases

6 Acknowledgment

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Nomenclature

T_{ci}	Inlet temperature of coil
T_{co}	Outlet temperature coil
T_{hi}	Inlet temperature of annulus
T_{ho}	Outlet temperature
d_o	Outer diameter of coil
D	Inner diameter of coil
C_p	Specific heat
k	Thermal conductivity
Pr	Prandtl no
Re	Reynolds no
C	Outside diameter of cylinder
B	Inside Diameter of cylinder
p	Pitch of the coil
D_h	Average diameter of helix
r	Average radius of helix
D_{h1}	Inside diameter of helix
D_{h2}	Outside diameter of helix
L	Length of coil for one turn
A_f	Area of cross section of coil
A_a	Area for fluid flow in annulus
V_c	Volume occupied by one turn of coil
V_a	Volume of annulus for one turn of coil
V_f	Volume available for flow of fluid in annulus
D_e	Shell side equivalent diameter of coil tube
A_a	Area for fluid flow in annulus
m_a	Mass flow rate in the annulus
m_c	Mass flow rate in the coil
x	Thickness of the coil wall
R_a	Fouling factor of shell side (distilled water)
R_t	Fouling factor of shell side (For tap water)
h_i	Heat transfer coefficient inside the coil
h_o	Heat transfer coefficient in the annulus
U	Overall heat transfer coefficient
A	Area for heat transfer
N	Number of turns
H	Height of coil
L_t	Total length of the coil
NPS	Nominal Pipe Size
μ	Dynamic viscosity
ρ	Density
ci	Cold inlet
co	Cold outlet
hi	Hot inlet
ho	Hot outlet
o	Outer diameter

h	Helix
h1	Inside helix
h2	Outside helix
a	Annulus
c	Coil
f	Flow
e	Equivalent diameter
i	Inside the coil
o	Outsie the coil (annulus)
t	Total

REFERENCES

[1] J.P.Holman, Heat Tranfer,10th Edition. .

[2] R. K. Patil, B. W. Shende, and P. K. Ghosh, "Designing a helical-coil heat exchanger." pp. 1–8, 1984.

[3] D. G. Prabhanjan, T. J. Rennie, and G. S. V. Raghavan, "Natural convection heat transfer from helical coiled tubes," Int. J. Therm. Sci. 43, 2004.

[4] A. M. Elsayed, "Heat Transfer in Helically Coiled Small Diameter Tubes for Miniature Cooling Systems," no. September, 2011.

[5] A. B. Korane, P. S. Purandare, and K. V. Mali, "Pressure drop analysis of Helical Coil Heat Exchanger (HCHE) for circular and square-coiled pattern," Int. J. Eng. Sci. Res., no. 5, pp. 361–369, 2012.

[6] B. C. Ankanna and B. S. Reddy, "Performance Analysis of Fabricated Helical Coil Heat Exchanger," Int. J. Eng. Res., vol. 5013, no. 3, pp. 33–39, 2014.

[7] [N. D. Shirgire, A. Thakur, and S. Singh, "Comparative Study and Analysis between Helical Coil and Straight Tube Heat Exchanger," Int. J. Eng. Res. Appl., vol. 4, no. 8, pp. 130–133, 2014.

[8] A. S. Puttewar and A. M. Andhare, "Design and thermal evaluation of Shell and Helical Coil Heat Exchanger," Int. J. Res. Eng. Technol., pp. 416–423, 2015.

[9] N. D. Ranaware, P. G. Student, M. E. H. Power, K. N. Molawade, and B. T. Student, "A Review on Comparison between Shell and Tube Heat Exchanger And Helical Coil Heat Exchanger," Int. J. Innov. Eng. Res. Technol., vol. 2, no. 2, pp. 1–9, 2015.

[10] A. Alimoradi and F. Veysi, "Prediction of heat transfer coefficients of shell and coiled tube heat exchangers using numerical method and experimental ...," Int. J. Therm. Sci., 2016.

[11] N. Acharya, M. Sen, and H. Chang, "Analysis of heat transfer enhancement in coiled-tube heat exchangers," vol. 44, pp. 3189–3199, 2001.

[12] J. S. Jayakumar, S. M. Mahajania, J. C. Mandala, P. K. Vijayanb, and R. Bhoia, "Experimental CFD estimation of heat transfer in helically coiled heat exchanger Experimental and CFD estimation of heat transfer in helically coiled heat exchangers," Chem. Eng. Res. Des., no. March 2008, 2008.

[13] S. Kumar and K. V. Karanth, "Numerical Analysis of a Helical Coiled Heat Exchanger using CFD," pp. 126–130, 2013.

[14] M. M. Tayde, J. Wankhade, and P. S. Channapattana, "Helically Coiled Heat Exchanger Heat Transfer Analysis using Fluent," vol. 2, no. 11, pp. 50–53, 2015.

[15] M. Balamurugan et al. , "CFD simulation on helical coil to understand the effects of different design parameters by ...," no. December, 2015.