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# Design and Development of Spiral Tube Heat Exchanger

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**Abstract –** *The Spiral Tube Heat exchanger consists of a spiral tube arranged in an array in which the liquid flows, the tubes are contained in a shell. The unique properties of a curved tube are double-flow which produces centrifugal force and thereby increasing heat transfer rate. The designed spiral tube model is for the liquid to liquid hear transfer. The spiral geometry is designed using Archimedean spiral geometry. The cold fluid flow in the curved path of the spiral tube and is heated in the spiral tube. The constant heating medium is used to heat the tube. To assess the performance the mathematical model is suitably designed and the CAD model was designed. The spiral coil is arranged in staggered array, 2 coil in set and two set connected to header tube forming the spiral tube bundle. The designed CAD model is conveniently programmed and analyzed on ANSYS Fluent. The standard k-epsilon model is used for analysis. The Nusselt correlation for the designed model was formulated by probability estimation based on the mathematical model. Simulation on a CFD model of the spiral coil bundle was conducted and the results obtained is compared to the mathematical model. The deviation of formulated correlation is in reasonable agreement with mathematical calculation. The change in Nusselt number concerning mass flow rate is selected as a comparison for the measure of performance.*

***Key Words*: Spiral tube Heat exchanger, Pancake coil, Staggered arrangement, Spiral tube bundle, Nusselt number formulation.**

## 1.INTRODUCTION

The Heat exchanger is one of the important components in the process industry. It has a broad range of application. Many industries require a certain temperature for their application and others require the temperature to run their machine efficiently. The heat is captured from by-product with help of heat exchanger and it is used again which saves energy. The Spiral heat exchanger is known for its compactness, different arrangements combination, and high efficiency. The current system for the Spiral heat exchanger is in the developing stage. Recently a Spiral tube heat exchanger was made by SENTRY-EQUIP in which the Spiral are connected to the header by welding and arranged in-line. No plugging is used. The support and baffles are not attached since the bundle forms a stable structure. Due to a variety of parallel tube configurations (diameter, number and length), efficiency is not compromised by limited shell dimension and design which increase heat performance. The maintenance is easy since the tube bundle is one part and can be easily done.

## 2. LITERATURE REVIEW

A spiral tube has a higher heat transfer coefficient than a straight tube. A spiral tube is more suitable for thermal expansion. The equation of friction factor is given by comparing various spiral coil tube[1]. The addition of fins, vortex generator, straight strip wire effect in heat transfer [4]. A theoretical model predicting the thermal performance of the spiral coil heat exchanger as a cooling and dehumidifying unit has been developed based on the assumption that the air is unmixed as it flows past each spiral coil turn. A theoretical model predicting the thermal performance of the spiral-coil heat exchanger as a cooling and dehumidifying unit has been developed [6]. Air mass flow rate and inlet-air temperature have a significant effect on the increase of outlet-air and water temperatures. The outlet-air and water temperatures decrease with increasing water mass flow rate. The enthalpy effectiveness and humidity effectiveness decrease as the air and water mass flow rates increase. The enthalpy effectiveness and humidity effectiveness increase as the inlet-air temperature increases [7]. The spirally coiled tube with three different curvature ratios of 0.02, 0.04, 0.05 under constant wall temperature is tested. The turbulent flow and heat transfer developments are simulated by using the k–e standard turbulence model. The turbulent kinetic energy, k, and turbulent kinetic energy dissipation, e, are coupled to the main governing equations via the turbulent viscosity relation. The outlet water temperature at the low curvature ratio is higher than the higher curvature ratio because the tube length for the lower curvature ratio is higher than the higher curvature ratio [8]. The induced centrifugal force in the spiral-coil tube has a significant effect on the enhancement of heat transfer. However, the pressure drop also increases. Due to the centrifugal force, the Nusselt number and pressure obtained from the spiral-coil Tube are 1.50 times higher than those from the straight tube [9]. The wire is used for increased heat transfer between coil and air. The longitudinal tube spacing still plays a major role in determining the magnitude of the Heat transfer coefficient as radial distance decrease heat transfer rate increased [10]. The use of LMTD method and equation for overall heat transfer and pressure drop in a spiral tube are formulated [11]. The spiral coil is more effective than using Nano-fluid to enhance the convection heat transfer coefficient. The addition of Nanoparticles affects the pressure drop by decreasing little [12]. The spiral geometry can significantly enhance heat transfer compared to a straight channel. Greater flow rates have shown better thermal augmentation, a greater pressure drop is expected simultaneously. The entrance effects in short spiral channels could influence the pressure drop [13].

The Nusselt number and the friction factors for a wide range of design parameters are obtained for the combined entry flows in spiral coils. With increasing Reynolds number, the heat transfer is enhanced 2-4 times over straight tubes of the same length due to secondary flow and centrifugal forces. The friction factor increases with spiral tube length [14]. As the curvature ratio increases, the heat transfer coefficient enhances. The intensity of secondary flow developed goes on increasing with an increase in curvature [15]. The spiral tube has better thermal performance (6-7%) than the tapered tube and helical tube [16]. The effects of flow configuration on the heat transfer performance of a spiral wound heat exchanger were experimentally investigated. The spiral-wound heat exchanger was made, three airflow configurations were tested, namely axial, radial, and mixed axial-radial flows. The overall heat transfer coefficient was determined from the experimental data and the shell-side convection heat transfer coefficient was calculated from the tube-side convection heat transfer coefficient and the total thermal resistance. The results showed that the mixed axial-radial flow configuration has the highest heat transfer coefficient and pressure drop followed by the axial and radial flow. The radial flow configuration has the lowest heat transfer coefficient and pressure drop [18]. An analytical model is developed for carrying out design simulations of the Pancake type heat exchanger. It is stated in the existing literature that each correlation is reasonable over a certain range of conditions, but for most engineering calculations one should not expect accuracy too much better than 20% [19]. The Nusselt number is highest for helical coil, whereas least for spiral coil, and for conical coil it reduces with increase in cone angle. The helical coil has a maximum ε whereas minimum in case of the spiral coil and conical coil, as cone angle increases ε decreases from helical to spiral [20]. The spiral tube for thermal energy storage purpose [21]. The obtained Nu and the heat transfer rate of a TSCTHE (triple spirally coiled tube heat exchanger) was greater than that of DSCTHE (double spirally coiled the tube heat exchanger) for both counter and parallel flow arrangements. Increasing the hot water inlet temperature increases the heat exchanger effectiveness while the Nu decreases. A considerable heat transfer enhancement was obtained by increasing the coil inclination angle from 45° to 90°. The thermo-hydraulic performance criteria η occurred at the minimum value and the maximum record values are 1.87 for counter flow and 1.72 for parallel flow pattern at coil inclination angles of 0°[23]. Recently a Spiral tube heat exchanger was made by SENTRY-EQUIP in which the Spiral are connected to the header by welding and arranged in-line. No plugging is used. The support and baffles are not attached since the bundle forms a stable structure. Due to a variety of parallel tube configurations (diameter, number and length), efficiency is not compromised by limited shell dimension and design which increase heat performance. The maintenance is easy since the tube bundle is one part and can be easily done. Minimum material is used than in conventional [24].

## 3. PROPOSED SYSTEM

The heat exchanger consists of CI Shell and SS Spiral tube. The flat spiral tube is arranged in stacking and in staggered order. The cold fluid passes through the spiral tube and heated sensibly by fluid in the shell, the shell fluid is heated by a heater to maintain a constant temperature. The fluid enters the first coil from outward to inward and then it enters the second coil from inward to outward both coils is connected, which forms one set likewise two sets is connected to the header where both inlet and outlet header is outside of the tube bundle

## 4. DESIGN OF EXPERIMENTAL SETUP

The problem is to design a suitable spiral heat exchange system for liquid to liquid heat transfer and use this heat for a particular process industry. The basic selection done are

1. The selection of temperatures by survey or required in application.
2. Selecting of liquid properties based on mean or selected temperature.
3. Assumptions of inner diameter of spiral coil

The spiral dimension are based on Archimedes spiral coil, using the equation of spiral ($r = a\theta$, where $a = p/2\pi$ and $p = 2d_o$) [26], the value of $R_o$ and n is found out using equation and shell diameter is calculated using equation.

$L_{sp} = 1/2a \times (R_o^2 - R_i^2)$

$n = (R_o - R_i) / p$

$D_i = (R_o + d_{h,o}) \times 2 + \text{clearance}$

This shell diameter is modified as per the nearest value available in TEMA standards. [5], [19]

$A_{sh} = \pi/4\, D_i^2 - [(L \times d_o) + (2 \times \pi/4 \times d_{h,o}^2) + (\pi/4 \times d_i^2)]$

$Re = (\rho \times V_{max} \times d_i)/ \mu.$

$V = \dot{m} / (\rho \times A_s)$

$V_{max} = S_T / (S_T - d_o) \times V$ --- inline

$V_{max} = S_T / (2 \times (S_D - d_o) \times V$ --- staggered

Several correlations, all based on experimental data, have been proposed for the average Nusselt number for cross flow over tube banks. More recently, Zukauskas (1987) has proposed correlations whose general form is [3]

$Nu_D = C \times Re^m \times D \times Pr^n \times (Pr/Pr_s)^{0.25}$

*Nomenclature :*

*A* surface area [$m^2$]
*$C_p$* specific heat [J/kg K]
c capacity ratio [-]
*D* coil diameter [m]
*d* tube diameter [m]
*f* friction factor [-]
*g* gravitational acceleration [$m/s^2$]
*h* film heat transfer coefficient [$W/m^2$ K]
*k* thermal conductivity [W/m K]
*L* length of the tube [m]
*ṁ* mass flow rate [kg/s]
*N* number of pancake
*Nu* Nusselt number [-]
*$Nu_D$* Corrected Nusselt number
*NTU* number of transfer units
*n* number of turns
*P* pressure [$N/m^2$]
*Pr* Prandtl number [-]
*p* coil pitch [m]
*Q* discharge [lit/s]
*q* heat transfer rate [W]
*R* radius of coil [m]
*Re* Reynolds number [-]
*Rf* fouling factor
t thickness
*T* temperature [C°]
*U* overall heat transfer coefficient [$W/m^2$ K]
*V* fluid velocity [m/s]
*(r, θ)* polar coordinates

*Greek symbols :*

*Δ* change in
*ε* roughness factor [mm]
*η* effectiveness [-]
*μ* dynamic viscosity [N-s/$m^2$]
*ρ* density [$kg/m^3$]

*Subscripts :*

*avg* average
*e* exit
*h* header
*lm* logarithmic mean difference temperature (LMTD)
*i* inner, inlet
*o* outer, outlet
*s* staggered
*sh* shell
*sen* sensible
*sp* spiral
*st* straight
*t* tube
*w* *water*

$h = (Nu_D \times k) / d_o$
The water side heat transfer coefficient is calculated,
Assuming n pancakes Mounted on one header.
For heating circulation fluid velocity is 1.3 – 3.0 m/s [25].
$Re = (\rho \times V \times d_i)/ \mu$.
$A_i = \pi/4 \times d_i^2$
$Q= \dot{m}/\rho$
$\dot{m} = A_i \times V \times \rho$

Using Gnielinski's Correlation for Nusselt Number ; from pg 498, 497 & Table 8-3, [3].

$$Nu_{st} = \frac{\left\{\left(\frac{f}{8}\right)\times(Re-1000)\times Pr\right\}}{\left\{1+12.7\times\left(\frac{f}{8}\right)^{1/2}\times\left((Pr)^{2/3}-1\right)\right\}} \quad 0.5 \le Pr \le 2000 \text{ and}$$

$3 \times 10^3 < Re < 5 \times 10^6$

Where *f* Colebrook equation :

$$\frac{1}{\sqrt{f}} = -2.0 \times \log\left\{\left(\frac{\left(\frac{\varepsilon}{d_i}\right)}{3.7}\right) + (2.51)/\left(\frac{Re}{\sqrt{f}}\right)\right\}$$

But Nusselt number for spiral coil is estimated using Mikheev correlation [1] :

$$\frac{Nu_{sp}}{Nu_{st}} = \left\{1 + 3.54 \times \frac{a}{R_{avg}}\right\} \times \left(\frac{Pr_{avg}}{Pr}\right)^{0.25} \quad \text{when } R_{avg}/d_i > 6$$

$R_{avg} = (R_i + R_o)/2$

The water side heat transfer coefficient can be calculated by

$$Nu_{sp} = \frac{h_{spi} \times d_{spi}}{k_w}$$

Calculation of overall heat transfer coefficient , Resistance(R) is given by [3],

$$R = \frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o}$$

$$R = \frac{1}{hspi\ Ai} + \frac{Rf_i}{A_i} + \frac{\ln(\frac{do}{d_i})}{2\pi kL} + \frac{Rf_o}{A_o} + \frac{1}{hoAo}$$

$A_i = \pi d_i L$
$U_o = 1/RA_o$ ; $A_o = \pi d_o L$

The surface temperature of tube is constant the exit temp from tube is given by
$T_e = T_s - (T_s - T_i) \exp(-U_oA_s / \dot{m}\, C_p)$
$T_e = T_{t,o}$ ; $T_i = T_{t,i}$ ; $A_s = A_o$
Total number of pancake required for final temperature,

LMTD for constant wall temperature is given by
$\Delta T_{lm} = (\Delta T_e - \Delta T_i)/\ln(\Delta T_e/\Delta T_i)$ where $\Delta T_i = T_s - T_i$ & $\Delta T_e = T_s - T_e$

The number of pancakes required for sensible heating is [19],

$$N = \frac{q_{sensible}}{(U_o) \times \pi \times (d_o) \times L \times (\Delta T_{lm})}$$

$q_{sensible} = \dot{m}_w \times Cp_w \times \Delta T_w$ ; $\Delta T_w = T_{t,i} - T_{t,o}$

The NTU is given by [19],

$$NTU = \frac{U_o \times A_s}{\dot{m}_w \times Cp_w}$$

For constant surface temperature the effectiveness is given by :
$\varepsilon_{effectiveness} = 1 - \exp(-NTU)$
Pressure drop in spiral coil is given by,
$\Delta P = f \times \frac{L}{D_e} \times \frac{\rho V^2}{2} \times 4$ where $D_e = d_{t,i}$ ; [17], [19]

Pump work [3], $W_{pump} = (\dot{m}_{@Q=1lit/s}\, \Delta P)/\rho_w$

The pancakes are arranged in staggered array, the clearance between two pancake is 1.25 × $d_{t,o.}$ Since the staggered array header is added on both side i.e. 1inch pipe and 5mm clearance.

The Dean number [27] :
$De = Re \times \sqrt{\frac{di}{R_c}}$ Where [26], $R_c = \frac{(a^2+r^2)^{\frac{3}{2}}}{2a^2+r^2}$
r = aθ, where a = p/2π , No. of turns, n = 5.5,since for 1 turn, θ is 2π
r is the length of the radius from the centre, or beginning, of the spiral.
θ is the angular position (amount of rotation) of the radius.

## 4.1 Problem solution

The model was designed for the particular process industry, for design purpose, the milk pasteurization is considered where the required water temperature is 73°C, the water is heated by spiral tube bundle at constant wall temperature at 80°C, [2]. The available water is at room temperature at 30°C. The assumption taken is Steady operating conditions exist. The heat exchanger is well insulated so that heat loss to the surroundings is negligible. Changes in the kinetic and potential energies of fluid streams are negligible. Fluid properties are constant. There is no fouling. The fluid in the shell is heated by a heater and constant fluid temperature is maintained. Tube material of SS304, BWG 22 [5] of length 6m. The spiral pancake of core diameter of 204mm. The calculated shell diameter is 22inch. The number of pancakes required for heating is 1 with an effectiveness of 0.9.

**Table -1:** Dimension of the spiral coil.

| Description | Value |
|---|---|
| Tube material | SS304 |
| Dimensions: | ½ inch, BWG 22 |
| Tube outer diameter | 12.7 mm |
| Tube inner diameter | 11.28 mm |
| Thermal Conductivity | 17 W/mK |
| Roughness height (ε) | 0.002 mm |
| Pitch (p) | 25.4 mm |
| No. of turns, (n) | 5.54 |
| Length of pipe | 6 m |
| Archimedes constant, (a) | 4.0425 mm |
| Length if radius from center (r) | 139.699 mm |
| Core diameter of pancake | 204 mm |
| Outer diameter of pancake | 485.44 mm |
| Angular position (θ) | 11 π |
| Radius of curvature (Rc) | 139.5 mm |
| Shell dimensions | 22 inch |

**Table -2:** Thermal Properties at varying velocity.

| Parameters | Solutions | | | |
|---|---|---|---|---|
| Velocity (V), m/s | 1.5 | 2 | 2.5 | 3 |
| Reynolds No. (Re) ($\times 10^3$) | 30.93 | 41.24 | 51.55 | 61.86 |
| Discharge (Q), $m^3/s$ | 0.15 | 0.20 | 0.25 | 0.30 |
| Mass flow rate ($\dot{m}$), kg/s | 0.15 | 0.2 | 0.25 | 0.3 |
| friction factor (f) | 0.023 | 0.022 | 0.021 | 0.020 |
| Kinematic Viscosity (μ), N-s/$m^2$ ($\times 10^{-3}$) | 0.547 | 0.547 | 0.547 | 0.547 |
| Nusselt No. ($Nu_{st}$) | 164.7 | 224.7 | 285.3 | 346.3 |
| Nusselt No. ($Nu_{sp}$) | 207.1 | 282.5 | 358.8 | 435.5 |
| Film Heat transfer coefficient ($h_{sp}$), $W/m^2K$ | 11825 | 16133 | 20485 | 24868 |
| Resistance (R) ($\times 10^{-3}$) | 0.58 | 0.47 | 0.41 | 0.37 |
| Overall heat transfer coefficient ($U_o$) ($\times 10^3$) | 7.168 | 8.766 | 1.007 | 1.116 |
| Outlet temperature ($T_e$), °C | 76.77 | 75.94 | 75.06 | 74.09 |
| LMTD ($\Delta T_{LMTD}$), °C | 17.07 | 18.29 | 19.47 | 20.64 |
| q sensible (W) | 29286 | 38367 | 46937 | 55234 |
| No. of Coils (N) | 1 | 1 | 1 | 1 |
| NTU | 2.74 | 2.54 | 2.31 | 2.13 |
| Dean's Number (De) | 8797 | 11730 | 14660 | 17594 |
| Pressure drop for one coil (Δp), bar | 0.57 | 0.95 | 1.42 | 1.97 |

## 4.2 Formulation of Nusselt number

By analyzing Mikheev correlation
$Nu_{sp} = C \times Re \times R_c \times Pr$ where C = constant
The correlation for spiral tube can be given as
$Nu = C \times De^n \times Pr^m$ , where dean's number is product of Reynolds number and radius of curvature

The constant C, m, n are found by probability analysis. By varying constant from 0 to 0.99 and the best fit is produced by using numerical python. The above problem is calculated for mass flow rate 0.15, 0.20, 0.25, 0.30 keeping all other parameters same. The probable the range of C, m, n are found at initial by plotting the Formulated correlation with constant from 0.1 to 0.9 with 0.1 interval the range for the actual value of Nusselt number is found between 0.5 to 0.65, taking C =0.5.

Selecting interval 0.5 to 0.6 and choosing min value and close to 207, the value of m=0.53. Constant m value is plotted for interval 0.56 to 0.65 and minimum close value is selected.

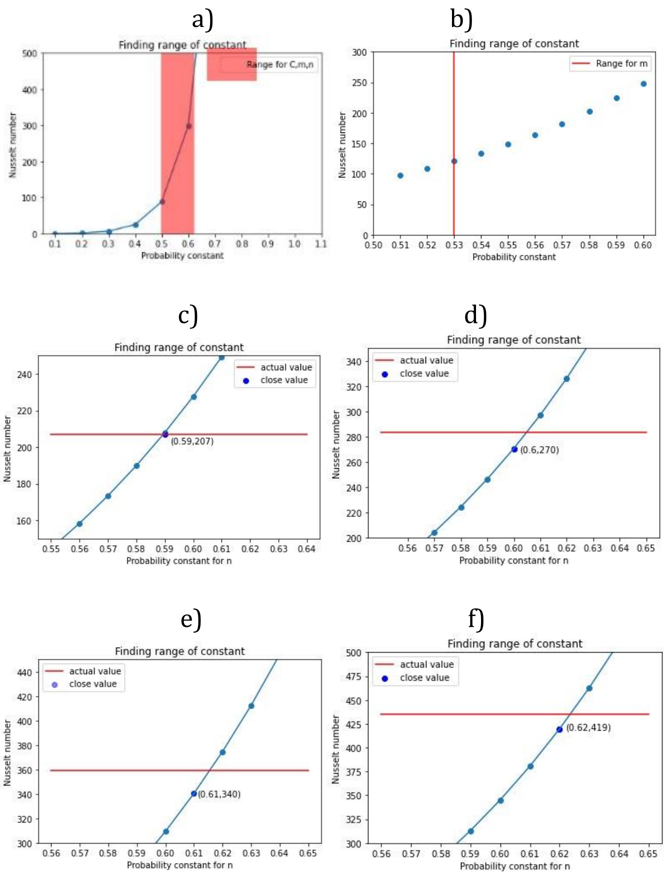


**Chart -1**: a) Probability Analysis to find constant C.
b) Probability Analysis to find constant m.
c)Probability Analysis to find constant n1
d) Probability Analysis to find constant n2.
e) Probability Analysis to find constant n3.
f) Probability Analysis to find constant n4

**Table -3:** The probable value of formulated Nusselt No.

| Nusselt number | | Constant | | |
|---|---|---|---|---|
| Actual value | Formulated value | C | m | n |
| 207.12 | 207.84 | 0.5 | 0.53 | 0.59 |
| 282.58 | 271.5 | 0.5 | 0.53 | 0.6 |
| 358.8 | 340.35 | 0.5 | 0.53 | 0.61 |
| 435.58 | 419.5 | 0.5 | 0.53 | 0.62 |

Formulated Nusselt number correlation for designed model.

$$Nu_{sp} = 0.5 \times De^{n} \times (Pr)^{0.53}$$

Where n = 0.59 to 0.60 for $6\times10^3 \le De \le 9\times10^3$
n = 0.60 to 0.61 for $9\times10^3 \le De \le 12\times10^3$
n = 0.61 to 0.62 for $12\times10^3 \le De \le 15\times10^3$
n = 0.62 to 0.63 for $15\times10^3 \le De \le 18\times10^3$

## 5. DESIGN

The arrangements of tubes are done in staggered array

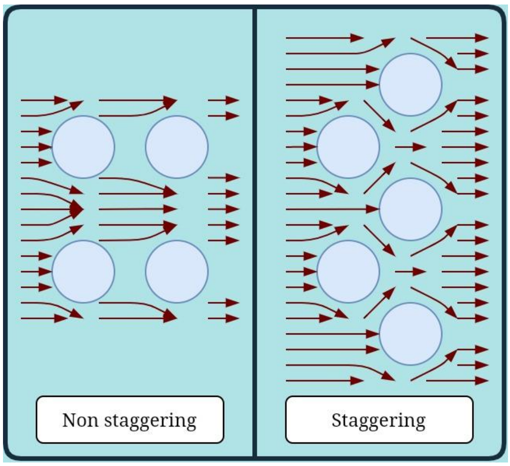


**Fig -1**: Comparison of in-line and staggered Arrangements

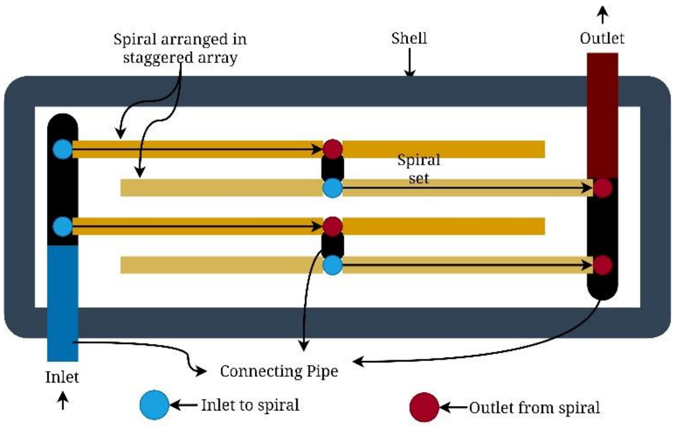


**Fig -2**: Flow Diagram

### 5.1 CAD Model

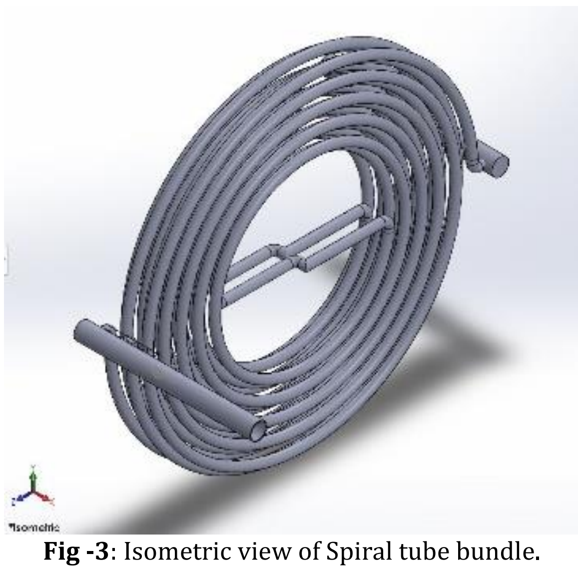

**Fig -3**: Isometric view of Spiral tube bundle.


**Fig -4**: Top view of Spiral tube bundle.

## 5.2 ANSYS Fluent Analysis

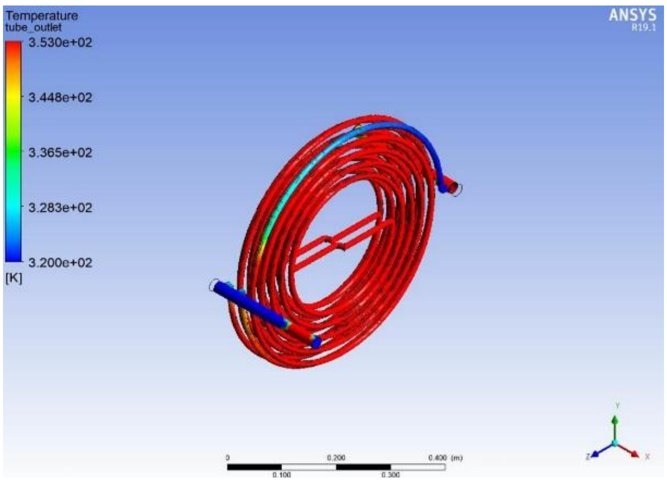


**Fig -4**: The temperature profile of the spiral tube bundle.

The CAD model was imported to the workbench and it was segregated into parts and meshed. The Meshing for the inner domain is tetrahedral and for the tube the octahedral shape element.
Total elements = 1,572,574.
Total nodes = 468,969.
The average skewness of element is 0.88.
Average mesh quality is maintained up to 0.63.
The model is calculated using the k-epsilon method.
Limitations: Due to low processing power, the model is calculated for 3 iteration.

| Parameters | Solutions | | | |
|---|---|---|---|---|
| Mass flow rate ($\dot{m}$), kg/s | 0.15 | 0.2 | 0.25 | 0.3 |
| Outlet temperature ($T_e$), °C | 75.93 | 75.43 | 74.04 | 74.49 |
| Overall heat transfer coefficient ($U_o$) ($\times 10^3$) | 6.573 | 8.360 | 9.286 | 11.555 |
| Resistance (R)($\times 10^{-3}$) | 0.63 | 0.49 | 0.44 | 0.37 |
| Film Heat transfer coefficient ($h_{sp}$), $W/m^2K(\times 10^3)$ | 10.44 | 14.94 | 17.76 | 26.64 |
| Nusselt No. (by CFD) | 182.9 | 261.8 | 311.2 | 466.7 |

## 3. RESULTS AND DISCUSSIONS

The analytical model outlet temperature is close to 75°C. There is a chance of error in the analysis since it is calculated for 3 iteration and mesh quality is maintained low. The average deviation between the Mikheev correlation and Formulated correlation is 3%. Due to double-flow, the heat transfer rate is also increased. The Nusselt number for the Spiral coil is 1.5-time Nusselt number of straight tube.
The comparison is shown below :

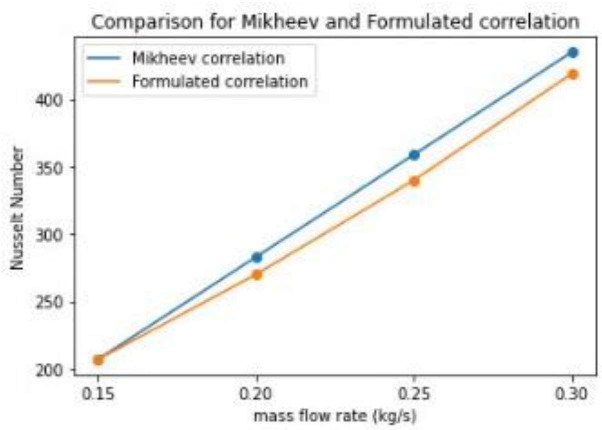


**Chart -2**: Comparison of Mikheev and formulated correlation with mass flow rate.

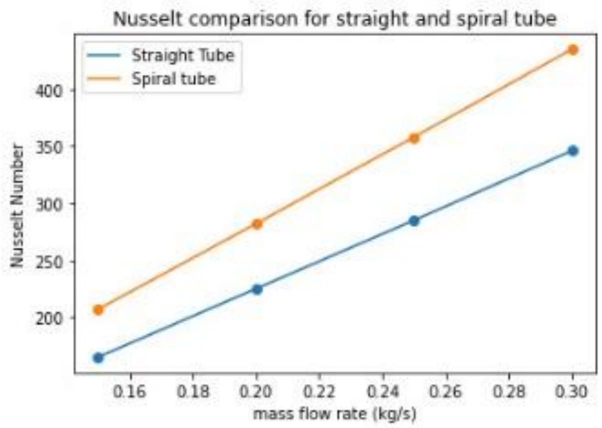


**Chart -3**: Comparison of straight and spiral tube with mass flow rate.

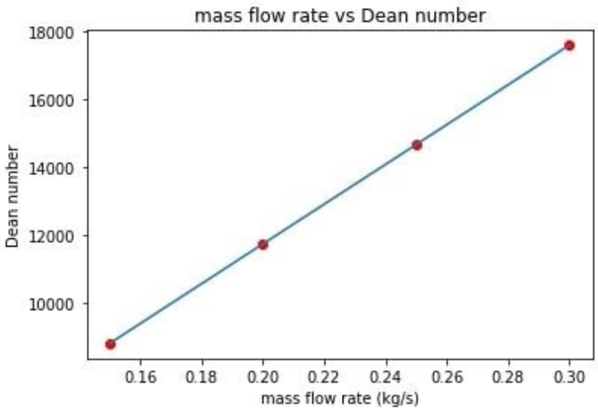


**Chart -4**: Mass flow rate vs Dean No.

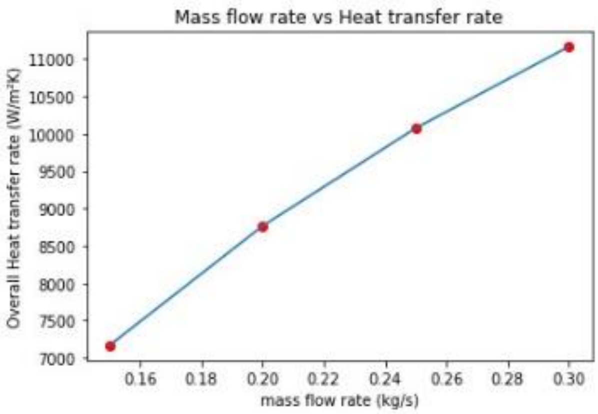


**Chart -5**: Mass flow rate vs overall heat transfer coefficient.

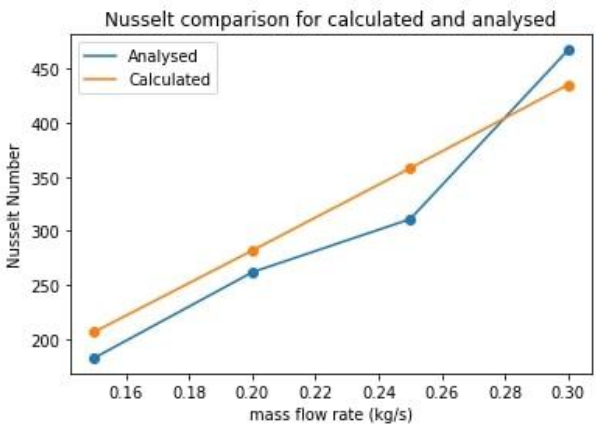


**Chart -4**: Comparison of Analyzed and calculated Nusselt number with mass flow rate.

## 6. CONCLUSIONS

The spiral tube heat exchanger is analyzed for the heat transfer to water through shell and water through tubes. The deviation between the mathematical results and analytical values obtained are within 10%. The pressure drop estimated is also compared with actual values observed during analysis, which is found in an acceptable range. The deviation between the mathematical model and formulated correlation for this model is obtained within 3%. Due to secondary flow and high Dean number, fully turbulent flow is observed which can decrease the scaling. There is uneven heating fluid due to change in Dean number as the diameter of spiral varies. The temperature profile observed is between is 74°C to 76°C. The deviation of Nusselt number between the mathematical model and Analyzed model is obtained within 12%. The Spiral heat exchanger works best for pure fluids. The staggering of spiral tubes reduces the size but increases complexity in connection. It is stated in the existing literature that each correlation is reasonable over a certain range of conditions, but for most engineering calculations one should not expect accuracy too much better than 20%.

## 7. FUTURE SCOPE

The standard design for the spiral tube heat exchanger can be developed. The experiment and analysis can be done by changing the material, curvature of the coil, the inner diameter of the pipe and coil. The spiral tube coil can be arranged in staggered by combining more than two. Using easy connection of coil so it can easily be removed if the coil is damaged. Internal core support can be given if the inner diameter of the coil is more. The spiral tube heat exchanger can be used where space is less also in a harsh environment like geo hot well. This heat exchanger removes the complex design of the shell since heat performance doesn't depend on the shell. The pressure drop needs to be reduced. And more research to fully develop the industrial application model.

## APPENDIX

Properties of saturated water [3].
The Moody chart for friction factor [3].
Nusselt no. correlations for crossflow over tube banks [3].
Correction factor for tube banks [3].
Chart for Friction factor and Corrections factor [3].

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