DESIGN AND ANALYSIS OF OVAL-TUBE HEAT EXCHANGER BY USING ANSYS V15.0 AND COMPARING WITH CIRCULAR-TUBE

[1]BODDETI JAGADISH, [2]P.NAVEEN, [3]M.SANDEEP, [4]K.APPALARAJU, [5]G.AZAD
[1]Assitant Professor & [2,3,4,5]UG Students
Department Of Mechanical Engineering
Godavari Institute of Engineering and Technology, Rajahmundry, India

Abstract: Heat exchanger is a device which is used for efficient heat transfer from one medium to another in order to carry and process energy. In this device, one medium is cooled while the other is heated. Generally all the heat exchangers have circular tubes with segmental baffles. Circular tube has less surface area contact than oval tube according to the fluid flow on it. Circular tube does not have aerodynamic configuration and increases total drag forces of fluid. In this project, design and analysis are done for oval tube heat exchanger with different orientation of oval tube to overcome these problems. So, we would like to modify the tube cross-section with elliptical profile for better heat exchange. It also increases the overall efficiency, heat transfer rate and smooth flow in the heat exchanger. The design of elliptical-tube is based upon the circular tube dimensions at equal fluid flow rate. These can be used in power plants as oil coolers with better performance. Modeling is done by using Catia-v5 or Solid-works-v18 and fluid flow and thermal analysis are carried out in Ansys-v15.0 software.

***Keywords* – ELLIPTICAL TUBE, SHELL AND TUBE HEAT EXCHANGER & ANSYS.**

1.INTRODUCTION

A heat exchanger is a device used to transfer heat between two or more fluids. The fluids can be single or two phase and, depending on the exchanger type, may be separated or in direct contact. Devices involving energy sources such as nuclear fuel pins or fired heaters are not normally regarded as heat exchangers although many of the principles involved in their design are the same. In order to discuss heat exchangers it is necessary to provide some form of categorization.

Shell and Tube Exchangers:

A Shell and Tube Exchanger consists of a number of tubes mounted inside a cylindrical shell. Figure 8 illustrates a typical unit that may be found in a petrochemical plant. Two fluids can exchange heat, one fluid flows over the outside of the tubes while the second fluid flows through the tubes. The fluids can be single or two phase and can flow in a parallel or a cross/counter flow arrangement. The shell and tube exchanger consists of four major parts:

1. Front end–this is where the fluid enters the tube-side of the exchanger.
2. Rear end–this is where the tube-side fluid leaves the exchanger or where it is returned to the front header in exchangers with multiple tube-side passes.
3. Tube bundle–this comprises of the tubes, tube sheets, baffles and tie rods etc. to hold the bundle together.
4. Shell—this contains the tube bundle.

The popularity of shell and tube exchangers has resulted in a standard being developed for their designation and use by the Tubular Exchanger Manufactures Association (TEMA) Standards [1]. In general shell and tube exchangers are made of metal but for specialist applications (e.g., involving strong acids of pharmaceuticals) other materials such as graphite, plastic and glass may be used. It is also normal for the tubes to be straight but in some cryogenic applications helical or Hampson coils are used.

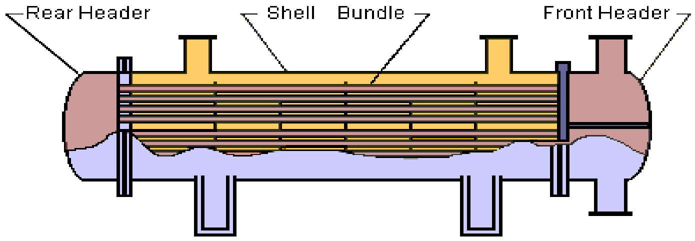


Fig. 1. Shell and tube exchanger.

Several thermal designs are to be considered before designing the heat exchangers. It is seen that the ends of each tube are connected to water boxes through holes in tube sheets. The tubes may be of different types depending upon the requirement. The design is based on the $\varepsilon = NTU$ method and kern method.

2. LITERATURE SURVEY

Piyush Gupta et.al [2017] studied the way for energy saving which is most important parameter in Industrial & Chemical Sector. For industrial growth it is necessary to enhance heat transfer in shell and tube Heat Exchanger. In the present study, the conventional cross section of tube is modified from circular to elliptical. The code developed for circular tube is giving the same result as available in existing literature. Then the code is modified for elliptical tube and heat transfer coefficient is determined for the various ratio of major axis to minor axis. It is found that heat transfer coefficients is better when this ratio is less than one as compared to the ratio more than one. It is also reported that when this ratio is less than 0.5, then pressure drop in shell become almost double.

Fadi A. Ghaith et.al [2015] studied to enhance the thermal performance of an industrial shell-and-tube heat exchanger utilized for the purpose of cooling raw natural gas by means of mixture of Sales gas. The main objective of this work is to provide an optimum and reliable thermal design of a single-shelled finned tubes heat exchanger to replace the existing two- shell and tube heat exchanger due to the space limitations in the plant. A comprehensive thermal model was developed using the effectiveness-NTU method. The shell-side and tube-side overall heat transfer coefficient were determined using Bell-Delaware method and Dittus-Boelter correlation, respectively. The obtained results showed that the required area to provide a thermal duty of 1.4 MW is about 1132 m^2 with tube-side and shell-side heat transfer coefficients of 950 W/m^2K and 495 W/m^2K, respectively. In order to verify the obtained results generated from the mathematical model, a numerical study was carried out using HTRI software which showed a good match in terms of the heat transfer area and the tube-side heat transfer coefficient.

Ala Ali Hasan [2015] investigated the features of oval tubes appear clearer in a dry heat transfer process. Five shapes of dry oval tubes are experimentally investigated in a cross-flow of air. The measurement results for the oval tubes are compared with those for an equivalent circular tube. It is found that the Nusselt numbers Nu_D for the studied tubes are close for Reynolds numbers $Re_D < 4000$. While for higher Re_D, the Nu_D decreases with the increase of the 2 oval tube axis ratio. The drag measurements indicate lower drag coefficients C_d avg for the oval tubes. It is revealed that the investigated oval tubes have favourable combined thermal-hydraulic performance, which is expressed in terms of (Nu_D /C_d avg). The ratio of (Nu_D /C_d avg) for the oval tubes to that for the circular tube is from 1.3 to 2.5.

James E. O'Brien et.al [2014] presents the results of an experimental study of forced convection heat transfer in a narrow rectangular duct fitted with an elliptical tube and one or two delta-winglet pairs. The duct was designed to simulate a single passage in a fin-tube heat exchanger. Heat transfer measurements were obtained using a transient technique in which a heated airflow is suddenly introduced to the test section. High-resolution local fin-surface temperature distributions were obtained at several times after initiation of the transient using an imaging infrared camera. Comparisons of heat transfer and pressure-drop results for the elliptical tube versus a circular tube with and without winglets are provided. Mean heat transfer results indicated that the addition of the single winglet pair to the oval-tube geometry yielded significant heat transfer enhancement, averaging 38% higher than the oval-tube, no-winglet case. The corresponding increase in friction factor associated with the addition of the single winglet pair to the oval-tube geometry was very modest, less than 10% at Re_{Dh} 5500 and less than 5% at Re_{Dh} 55000.

S. Yamani Douzi Sorkhabi [2014] studied heat transfer and flow resistance of alternating elliptical axis tubes is investigated both experimentally and numerically. The working fluid is heat transfer oil, and the flow's Reynolds number ranges from 300 to 2000. The grid and numerical models are generated using Gambit 2.4.6 and Fluent 6.3, which are verified by the experimental results. The numerical results show that decreasing the aspect ratio and pitch length, increases heat transfer and flow resistance. However, in order to compare the heat transfer and flow resistance simultaneously, the non-dimensional heat transfer enhancement ratio is defined. The comparison of this ratio shows that alternating elliptical axis tubes perform better than the flattened or circular ones. It is observed that this ratio increases with the increase in Reynolds number.

Muhammad Mahmood Aslam Bhutta et.al [2011] focused on the applications of Computational Fluid Dynamics in the field of heat exchangers. It has been found that CFD has been employed for the following areas of study in various types of heat exchangers: fluid flow mal-distribution, fouling, pressure drop and thermal analysis in the design and optimization phase. Different turbulence models available in general purpose commercial CFD tools i.e. standard, realizable and RNG k - ε RSM, and SST k - ε in conjunction with velocity-pressure coupling schemes such as SIMPLE, PISO and etc. have been adopted to carry out the simulations. The quality of the solutions obtained from these simulations are largely within the acceptable range proving that CFD is an effective tool for predicting the behavior and performance of a wide variety of heat exchangers.

Ting D S-K et.al [2003] experimentally investigated the flows of hot air at 41.5 ± 1.5 °C across an array of elliptical tubes carrying cold water were experimentally in the range of waterside and airside Reynolds numbers of $1.0\times10^3<Re_w<3.7\times10^3$ and $1.0\times10^4<Re_a<3.3\times10^4$respectively. The waterside Reynolds number was based on the inner hydraulic diameter and that of the airside, on the major axis length of the tube. Cold water at 6.5 ± 1.0 °C entered and jig jagged through a single passage tube array, and exited at a temperature dictated by the total heat transferred. The array consisting of 18 elliptical tubes each 30 cm long, with minor to major outside axis ratio of 0.30, and equally spaced by 0.61 cm gap, was oriented at zero angle of attack in a 30 cm by 30 cm test section with the major axis parallel to the air flow. The results showed that the heat transfer rate (q) increased with the increase of both water and air flows. The pressure coefficient (C_p), across a single tube in the array was found to remain at approximately 0.16 for $Re_a>2\times10^4$. Both q and Nusselt number (Nu) were found to vary in power law relationship with Re. The Nu–Re correlation obtained from the idealized test bench was found to be in the form $Nu_a=0.26Re_a^{0.66}$.

Jang JY et.al [1998] investigated the fluid flow and heat transfer over a 4-row. Elliptic, finned-tube heat exchanger having an axis ratio of 2.83:1 are studied experimentally and numerically. Three types of finned-tube configurations have been investigated under dry and wet conditions for different values of inlet frontal velocity ranging from 2 to 7 m/s: two elliptic finned tubes with staggered and in-line arrangements and one circular finned tube with staggered arrangement. The experimental results indicate that the average heat transfer coefficient of an elliptic finned lube is 35–50% of the corresponding circular finned tube having the same tube perimeter, while the pressure drop for an elliptic finned-tube bank is only 25–30% of the circular finned-tube bank configuration. Three-dimensional numerical results of a laminar model for dry coils are also presented and are compared with the experimental data.

G. P. Merker et.al [1986] studied heat transfer and pressure drop of the cross-flow on the shell-side of staggered tube-banks, having different transversal and longitudinal pitches in the range of $1.97 \leq t_q \leq 3.16$ and $0.67 \leq t_1 \leq 1$ .O, respectively. The heat transfer was determined with the aid of the analogy between heat and mass transfer from measurements of the mass diffusion of naphthalene. The results show that exchangers with oval-shaped tubes have considerably smaller front areas on the shell-side compared to those with circular tubes.

Taoka Y et.al [1984] studied heat transfer characteristics and flow behaviors have been made clear for an elliptic cylinder of axis ratio 1:3. The testing fluid was air and the Reynolds number ranged from about 8000 to 79000. The angle of attack α was varied from 0° to 90°. The local and overall heat transfer features are clarified in relation to the flow behaviors around the cylinder. The critical Reynolds number is detected, where the heat transfer and flow characteristics change drastically. It is found that the mean heat transfer coefficient is at its highest at α = 60°–90° over the whole Reynolds number range studied and also that even the lowest value of the mean heat transfer rate is still higher than that for a circular cylinder. Effects of the axis ratio of the elliptic cylinder are also discussed in comparison with previous works.

3. METHODOLOGY

Basically CFD analysis involves three major steps say Pre-Processing, Processing and Post Processing.

3.1 Pre-Processing: This is preliminary step of CFD simulation0process which assists in explaining geometry in good suitable manner. The flow domain of interest is divided into equal number of smaller parts known as elements. There are different Pre-Processing software available are Gridgen, CFD-GEOM, ANSYS Meshing, ANSYS ICEM CFD, T Grid etc. Pre-processing this includes defining the problem, creating the 3D model, fetching the model to Ansys workbench, meshing, and applying physical operating condition called boundary conditions.

3.2 Solving or Processing: Once the issue material science has been recognized, liquid properties, stream physical science model, limit circumstances are located to tackle utilizing PC. There are renowned business programming accessible for this including: CFD++, Open ANSYS CFX, Star CCM, ANSYS FLUENT and so on. All the above given programming have their individual abilities. Utilizing this product it is conceivable to fathom the administering conditions identified with stream material science issue. Handling includes unraveling of numerical or scientific conditions of liquid stream until the point when joining in result is accomplished. Typically it requires the PC to understand a huge number of conditions and may take couple of hours to few days

3.3 Post processing: The last stride in the wake of getting the outcomes from the solver is to examine the outcomes with various techniques like weight and speed shape plots, vector plot, streamlines, temperature contour and so forth At the point when the model has been comprehended, the outcomes can be broke down both numerically and graphically. Post preparing is about perception either in basic 2-D to 3-D portrayals.

4. DESIGN CALCULATIONS

In this design calculation, the thermos-physical properties of hot oil and water are required.

Properties	Water	Hot oil
Specific heat (c_p)	4198 $\frac{j}{kg\ K}$	2000$\frac{j}{kg\ K}$
Density (ρ)	1000$\frac{Kg}{m^3}$	884$\frac{Kg}{m^3}$
Viscosity (μ)	1.452*10^{-3} Pa.s	1.2 Pa.s
Thermal conductivity (k)	0.68 $\frac{W}{mk}$	1.41 $\frac{W}{mk}$
Mass flow $(\frac{m}{s})$	0.9 $^{kg}/_{s}$	0.3 $^{kg}/_{s}$
Initial temperature (°C)	30 °C	120 °C

Table 1 Properties of fluids

Material of the body parts of shell and tube heat exchanger is copper which has thermal conductivity (k),$k_{Cu} = 400\frac{W}{mk}$.

4.1 THEORITICAL DESIGN CALCULATIONS:

4.1.1 Calculations of outlet temperatures: In spite of defining the ΔT_{LM} method, we will not use this it, because in order to use it the temperatures of inlet and outlet should be known. For that reason the ε-NTU method will be used.

$$Q_{max} = m * c_{p_{min}} * (T_{h_1} - T_{c_1})$$

$$m_c * c_{pc} \rightarrow 0.9 * 4186 = 3764.4\ {}^{W}/_{K} \rightarrow C_{max} \quad ; \quad m_h * c_{ph} \rightarrow 0.3 * 2000 = 600\ {}^{W}/_{K} \rightarrow C_{min}$$

In our case, $m_h * c_{ph} \leq m_c * c_{pc}$ so, $\Delta T_{max} = \Delta T_h$

$$Q_{max} = m * c_{p_{min}} * (T_{h_1} - T_{c_1}) = 54000\ W$$

Then, number of temperature units (NTU) [14] [15] is obtained by the formula and assuming the overall heat transfer coefficient (U), $U = 350 \frac{W}{m^2}$.

$$NTU = {}^{UA}/_{C_{min}} = 0.374$$ Then the ration (R) is,

$$R = \frac{C_{min}}{C_{max}} = \frac{600}{3764.4} = 0.16$$

From the heat transfer text book, the effectiveness of the heat exchanger for counter flow is

$$\varepsilon = \frac{1 - \exp[-NTU(1-R)]}{1 - R * \exp[-NTU(1-R)]} = 0.37$$

Here $m_h * c_{ph} = C_{min}$ so, according to the NTU method, the effectiveness of heat exchanger is

$$\varepsilon = \frac{m_h C_h}{C_{min}} [\frac{T_{h1} - T_{h2}}{T_{h1} - t_{c1}}]$$

$$T_{h2} = 355.7\ k$$

$$\boldsymbol{T_{h2} = 82}\ ℃$$

From the energy equation,

$$Q = m_h * c_p * (t_{hi} - t_{ho}) = m_c * c_p * (t_{c2} - t_{ci})$$

$$\boldsymbol{t_{c2} = 44.8 \cong 45}℃$$

4.1.2 Calculation of area for heat exchanger: From the log mean temperature difference (LMTD) method, the energy equation is [2]

$$Q = U * A * \Delta T_{LM} = \frac{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})}{\ln[\frac{(T_{h_1} - T_{c_2})}{(T_{h_2} - T_{c_1})}]}$$

From these equations, the area of heat exchanger can be obtained:

$$A = \frac{Q}{U * \Delta T_{LM}}$$

$$\Delta T_{LM} = \frac{(393 - 318) - (355 - 303)}{\ln[\frac{(393-318)}{(355-303)}]} = 62.7\ K$$

And total energy, $Q = m_h * c_p * (t_{hi} - t_{ho})$

$$Q = 0.3 * 2000 * (120 - 82) = 22800\ W$$

So by using the values of Q, U and ΔT_{LM}, area can be calculated by using formula is:

$$\therefore A = \frac{Q}{U * \Delta T_{LM}} \qquad A = \frac{22800}{350 * 62.7} = \boldsymbol{1.04\ m^2}$$

4.2 Number of tubes:

Area of tube is obtained by, $A_t = \frac{m_h}{\rho v}$ $A_t = \frac{0.3}{884 * 0.05}$ $A_t = 0.0054\ m^2$

No. of tubes obtained by formula is:

$$A_t = n\pi \frac{d^2}{4} \qquad \therefore n_t = \frac{A_t * 4}{d^2 \pi} = \frac{0.0054 * 4}{(0.019^2)\pi} = \textbf{19 Tubes.}$$

4.3 length of the heat exchanger:

Length of heat exchanger is same as the length of shell and the length of tubes because design is based on the fixed support single pass shell and tube heat exchanger. Length can be obtained by

$$A = n\pi dL \qquad \therefore L = \frac{A}{n\pi d} = \frac{0.0054}{19 * 0.019 * \pi} = \boldsymbol{0.92 \cong 1\ m}$$

4.4 Diameter of heat exchanger: Diameter of the shell can be obtained by the formula,

$$D_0 = \frac{A}{L * \pi} \qquad \therefore\ D_0 = \frac{1.04}{1 * \pi} = \mathbf{0.21 \cong 0.2\ m}$$

4.5 Convective Heat Transfer Coefficient:

$$h_o = \frac{k_c . Nu_o}{D_H} = 285.62\ \frac{W}{m^2 K}$$

$$h_i = \frac{k_h . Nu_i}{D_i} = 1989.5\ \frac{W}{m^2 K}$$

Overall Heat Transfer coefficient can be based on inner area of a pipe as well as the outer area.

$$U_o = \frac{1}{\frac{A_O}{A_i} * \frac{1}{h_i} + \frac{A_O * \ln\left(\frac{r_O}{r_i}\right)}{2\pi K_{cu} L} + \frac{1}{h_O}} = 233.33\ \frac{W}{m^2 K}$$

5. MODELING OF SHELL AND TUBE HEAT EXCHNAGER USING ANSYS:

5.1 GEOMETRY: A software model is accomplished by using proportions of shell, tubes and baffles in ANSYS 15.0. This heat exchanger is counter flow type and the tube side consists of one inlet and one outlet representing of 19 complete tubes

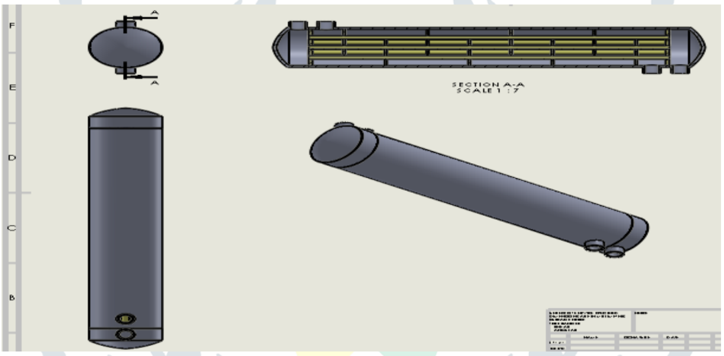


Fig. 2 Shell and tube heat exchanger assembly

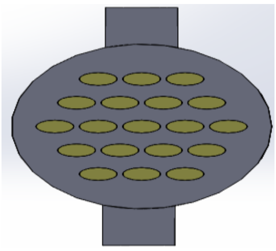


Fig. 3 Oval Tube at 0^0

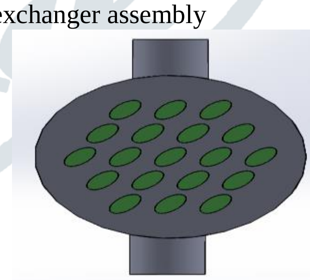


Fig. 4 Oval Tube at 45^0

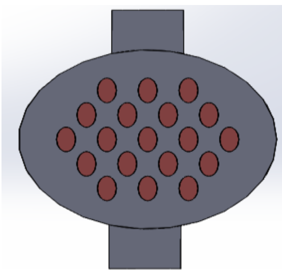


Fig 5 Oval Tube at 90^0

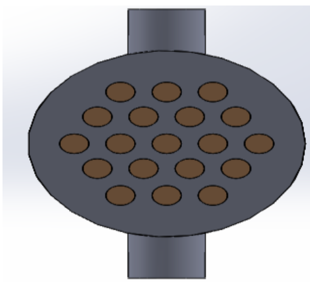


Fig 6 Circular Tube

5.2 MESH GENERATION: In this module, meshing of the heat exchanger can be done. Meshing is the process of the discretization of the heat exchanger model into small nodes or elements for better solutions. Initially a relatively coarser mesh is generated with 2858273 nodes and 8211991 elements This mesh contains mixed cells (Tetra and Hexahedral cells) having both triangular and quadrilateral faces at the boundaries. Care is taken to use structured cells (Hexahedral) as much as possible, for this reason the geometry is divided into several parts for using automatic methods available in the ANSYS meshing client. It is meant to reduce numerical diffusion as much as possible by structuring the mesh in a well manner, particularly near the wall region

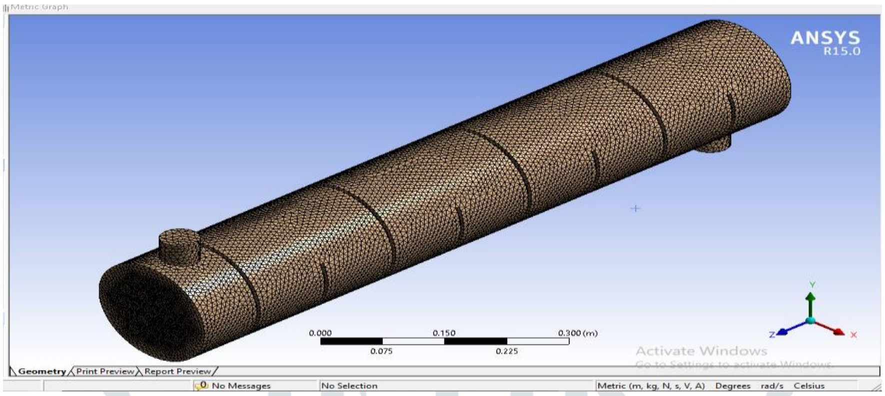


Fig. 7 Meshing

5.3 Problem setup: Simulation was carried out in ANSYS® FLUENT® v15.0. In the Fluent solver Pressure Based type was selected , absolute velocity formation and steady time was selected for the simulation . In the model option energy calculation was on and the viscous was set as standard k-e, standard wall function (k-epsilon 2 eqn.). In cell zone fluid water-liquid was selected. Water-liquid and copper was selected as materials for simulation. Boundary condition was selected for inlet, outlet. Water and hot oil taken as fluids, the properties of fluids taken from Table 1.

5.4 Solution Initialization: Pressure Velocity coupling selected as SIMPLE. Skewness correction was set at zero. In Spatial Discretization zone Gradient was set as least square cell based, Pressure was standard, Momentum was First order Upwind, Turbulent Kinetic energy was set as First order Upwind, and Energy was also set as First order upwind. In Solution control, Pressure was 0.7, Density 1 , Body force 1, Momentum 0.2 , turbulent kinetic and turbulent dissipation rate was set at 1, energy and turbulent Viscosity was 1. Solution initialization was standard method

6. RESULTS AND DISCUSSION:

To investigate the fluid flow and heat transfer of shell and tube heat exchanger, a three-dimensional model is designed. Using CFD software (ANSYS FLUENT 15.0), numerical calculations have been carried out.

6.1 CIRCULAR TUBE HEAT EXCHANGER:

In this circular tube exchanger, temperature variation of every tube is at 88.6 ℃ as outlet temperature but theoretical got 82℃ of outlet temperature. Fig 8 & 9 gives the temperature variation of fluid in both water and oil. Due to cross flow, the heat exchange is more and at the 500 mm to 1000 mm water temperature range is raised to 360 k and also hot oil releases the energy to water. It decrease to 372 k. The water has theoretical calculated temperature of 48.7 ℃ but ansys has the temperature of 60 ℃

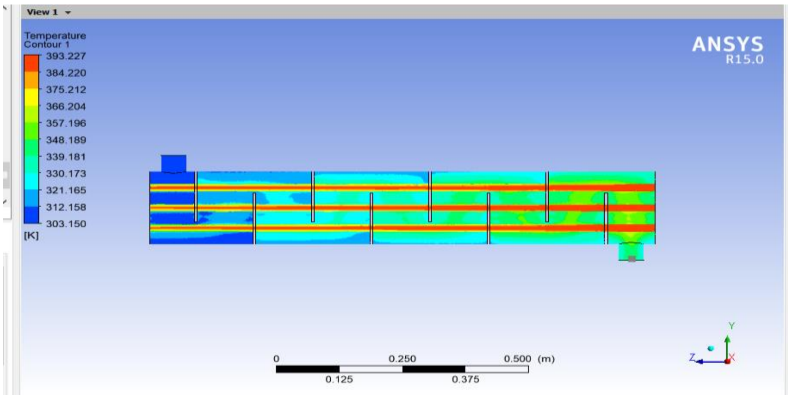


Fig. 8 Temperature Variation

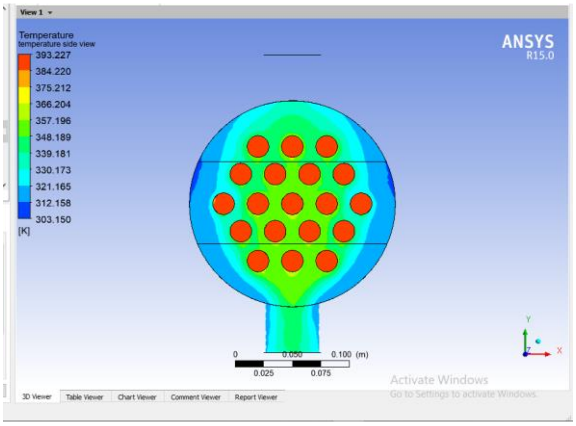


Fig. 9 Temperature Variation Side View

The temperature stream lines as shown in Fig 10 and it is observed that most of the stream lines are passed through tube bundle and shell but not in between the tubes. If the tubes are increased, the gaps size will be less and heat transfer will be more. Fig 11 explains about the velocity. Initial velocity of water is 0.02 m/s and decreased to 0.011 m/s when it strikes the tube. At the baffle cut, the velocity is raised to 0.022 m/s and again decreased at tubes and the outlet of water has velocity of 0.027 m/s. The oil has 0.02 m/s of initial velocity and gradually increased to 0.038 m/s.

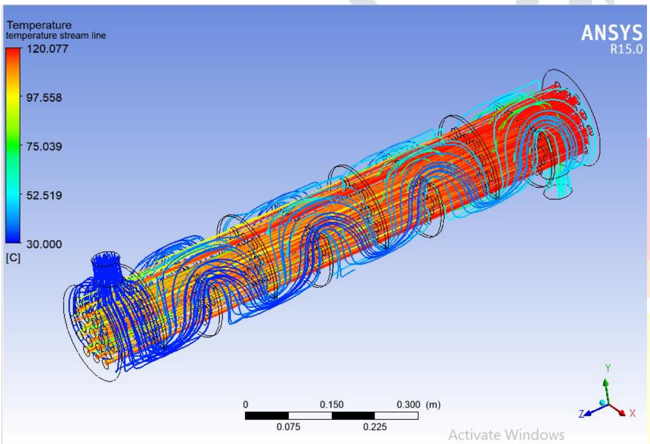


Fig. 10 Temperature Stream Line

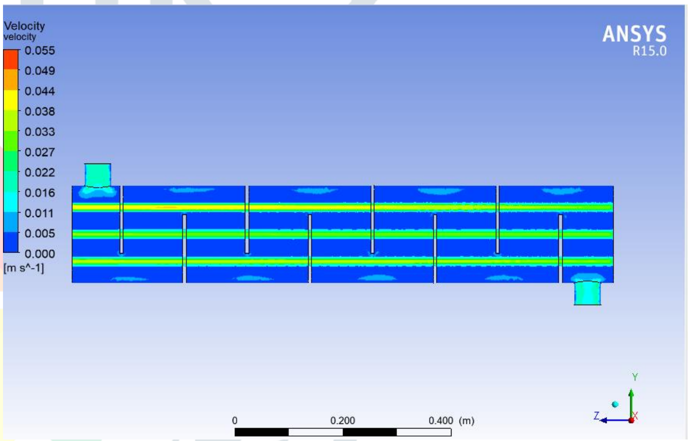


Fig.11 Velocity of circle tube

Fluid	Temperature [°C]		velocity $[m/s]$		Pressure $[Pa]$	
	Inlet	Outlet	inlet	Outlet	Inlet	Outlet
Water	30	60	0.1	0.057	125.67	0
Hot oil	120	88.6	0.02	0.015	1980	800

Table 2 Circular tube heat exchanger results

6.2 OVAL TUBE HEAT EXCHANGER:

Oval tube heat exchanger at 45^0 orientation has temperature variation as shown in Fig 12 & 13. The temperature of oil has initially at 120°C due to the heat exchange it decreases to 100 °C up to mid length 500 mm. Due to cold water interaction at oil temperature gradually decreased to 72°C. The stream lines are shown in Fig 15 and it represented the variation of temperature.

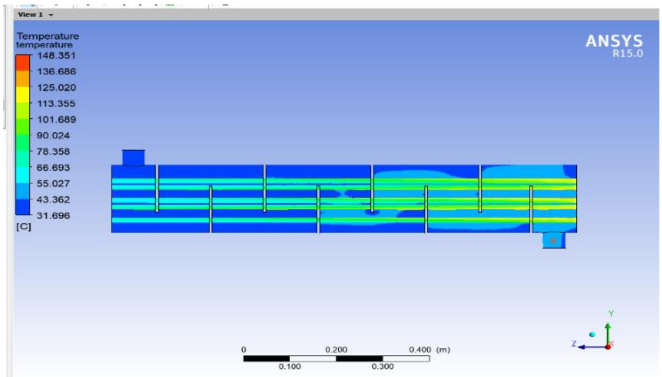


Fig. 12 Temperature Variation at 45^0

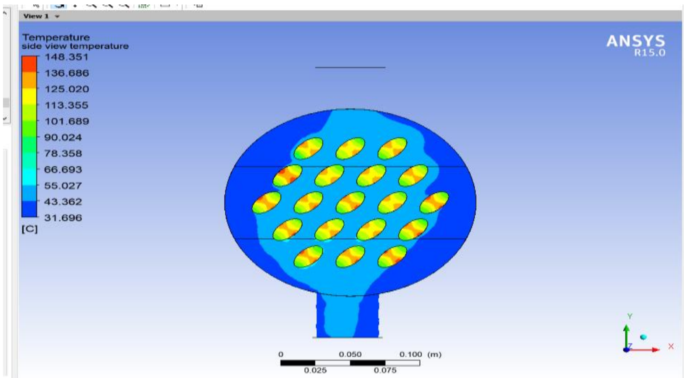


Fig. 13 Temperature Side View at 45^0

The velocity variation of oval tube at 45^0 and oil has initially at 0.02 m/s and it is gradually increases to 0.04 m/s. The water has initial of 0.1 m/s and decreased to 0.05 m/s when it strikes the oval tube and it has velocity of 0.1 m/s at the baffle cut due edge of baffles. Fig 14 shows the velocity variation as below:

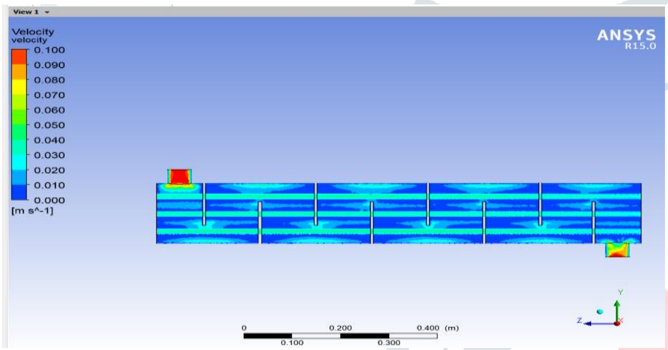


Fig. 14 Velocity at 45^0

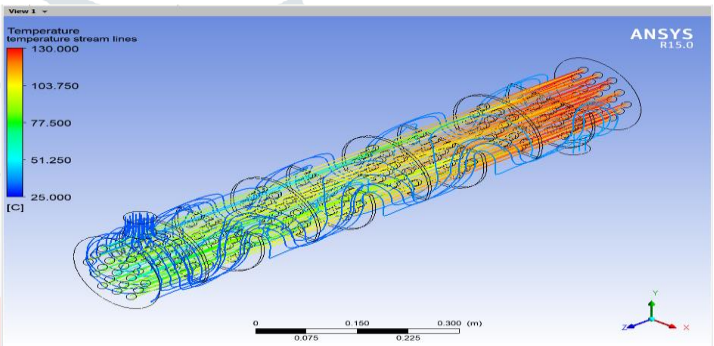


Fig. 15 Temperature Stream Lines at 45^0

Fluid	Temperature [℃]		velocity $[m/s]$		Pressure $[Pa]$	
	Inlet	Outlet	inlet	Outlet	Inlet	Outlet
Water	30	48.7	0.1	0.022	80.5	0
Hot oil	120	70.4	0.02	0.0346	6125.8	628.6

Table 3 oval tube heat exchanger at 45^0

The temperature variation of oval tube heat exchanger at 90^0 is as shown in Fig.16 & 17. Initial oil at 120 ℃ is passed and reached 98 ℃ in the mid span of heat exchanger and finally has temperature 85.6 ℃.The temperature is high in the central axis of oval tube and at surface temperature is low due to water interaction.

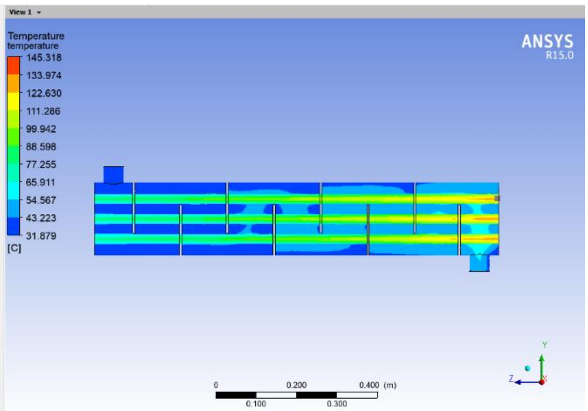


Fig. 16 Temperature Variation at 90^0

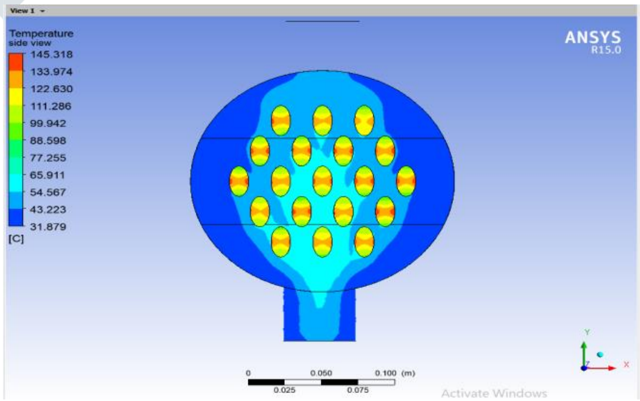


Fig. 17 Temperature Side View at 90^0

The velocity variation of oval tube at 90^0 is as shown in Fig 18. Initially water has 0.1 m/s and it gradually reduced and increased at the baffle edge. Finally, it got final velocity as 0.041 m/s. The temperature stream lines as shown in Fig 18.

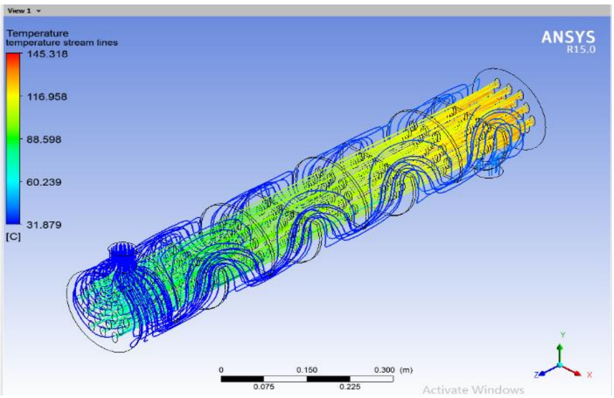


Fig. 18 Temperature Stream Lines At 90^0

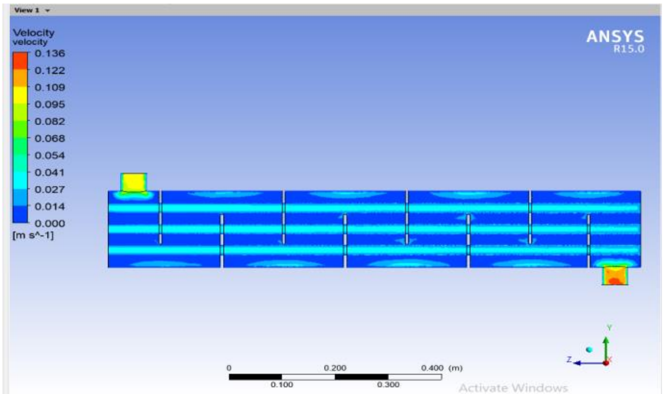


Fig. 19 Velocity At 90^0

Fluid	Temperature [°C]		velocity $[m/s]$		Pressure $[Pa]$	
	Inlet	Outlet	inlet	Outlet	Inlet	Outlet
Water	30	47.9	0.1	0.14	110	0
Hot oil	120	85.6	0.02	0.041	2033.3	585.7

Table 4 Oval tube heat exchanger at 90^0 orientation

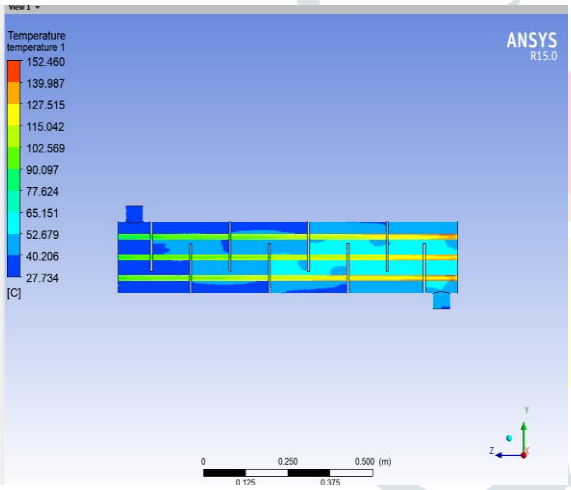


Fig. 20 Temperature of oval at 0^0

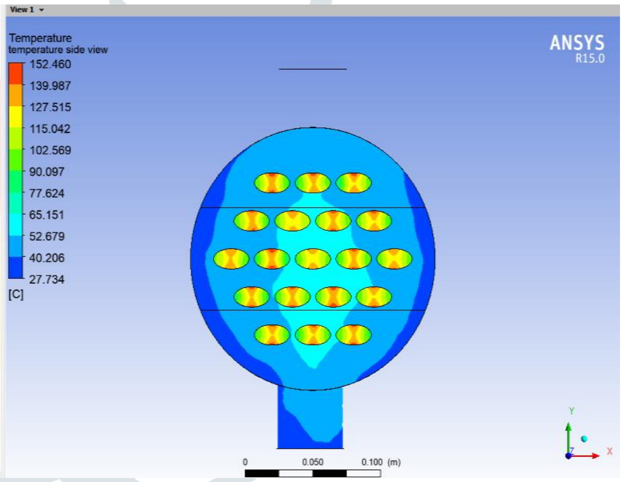


Fig. 21 Temperature side view

The temperature variation of oval tube heat exchanger at 0^0 orientation is as shown in Fig 20 & 21. It has oil temperature of 120 °C as outlet and at the mid span 102°C. Finally, it transfer heat to water and has 79 °C as outlet. The temperature stream lines are used to represent the temperature variation as shown in Fig 22. The velocity variation of fluids in oval tube heat exchanger in Fig 23 shows that water which has 0.1 m/s initial velocity, is raised to 0.6 m/s due baffle and hot oil also increased from 0.02 m/s to 0.04m/s.

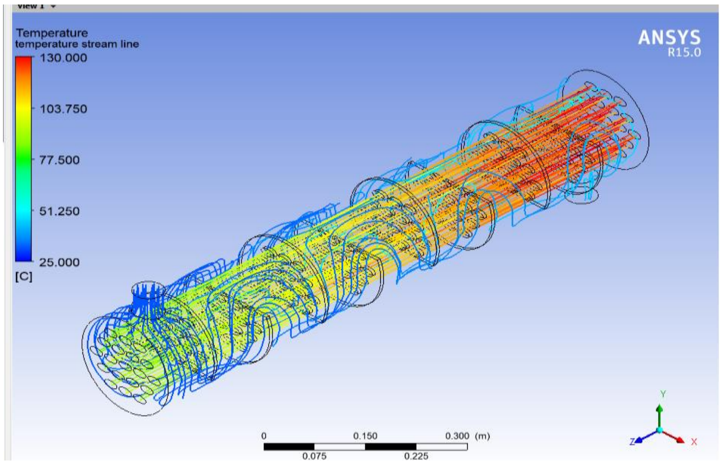


Fig. 22 temperature stream lines at 0^0

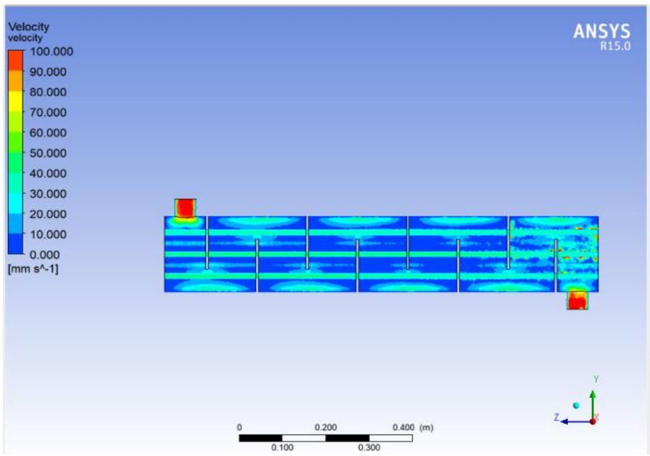


Fig. 23 velocity

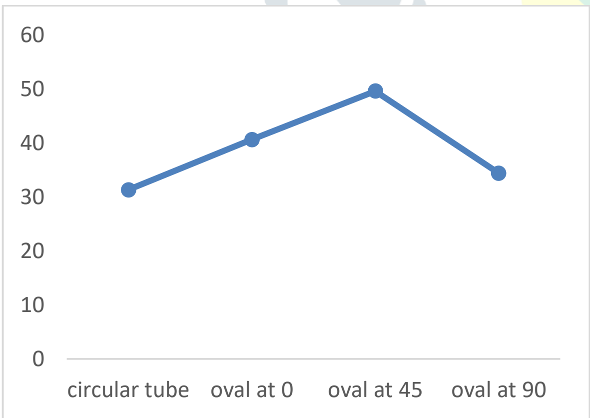
Fluid	Temperature [°C]		velocity $[m/s]$		Pressure $[Pa]$	
	Inlet	Outlet	inlet	Outlet	Inlet	Outlet
Water	30	50.46	0.1	0.67	134.76	0
Hot oil	120	79.4	0.02	0.0496	6005.8	351.30

Table 5 Oval tube heat exchanger at 0^0 orientation

By observing the tabular forms of results, temperature difference of hot oil and water, pressure difference of hot oil, velocity difference of hot oil and water are plotted in graphs.

- **Temperature difference of hot oil and water:**

In this graph, oval at 45 degrees has more temperature difference which results to more heat transfer between the fluids when compared to other models. Circular tube has more temperature in water. The graph is as shown below



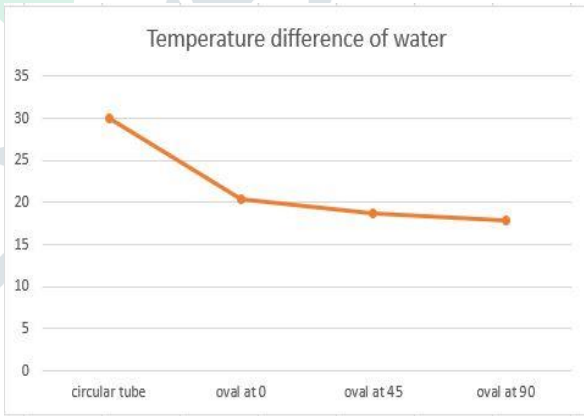


Fig 24. Temperature difference of oil Fig 25. Temperature difference of water

- **Velocity difference of oil and water:**

In this aspect, the initial velocity of oil & water is 0.02 m/s & 0.1 m/s respectively is given and final velocity gradually increases due to baffles but also leads to decreasing of velocity. Final velocity of oil of circular tube is less than the initial velocity of oil. So it declines to negative values. Higher velocity difference is observed in oval tube at 0 degrees. The low velocity difference of water is observed in oval tube at 45 degrees.

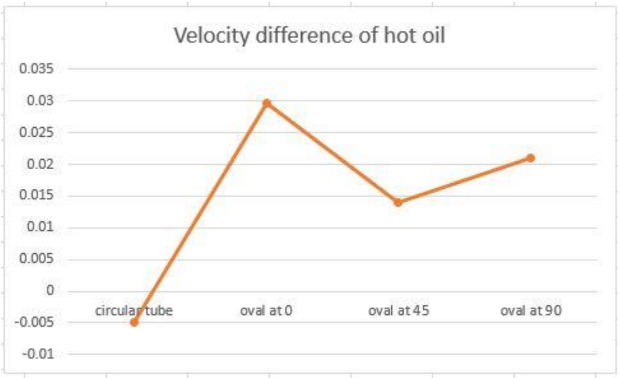


Fig. 26 Velocity difference of oil

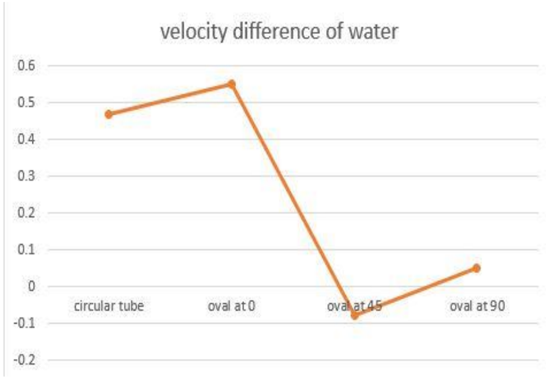


Fig. 27 Velocity difference of water

- **Pressure difference of hot oil:**

In this graph, more pressure difference is observed in oval tube heat exchanger at 0 and 45 degrees. The values are in Pascals (gauge pressure).

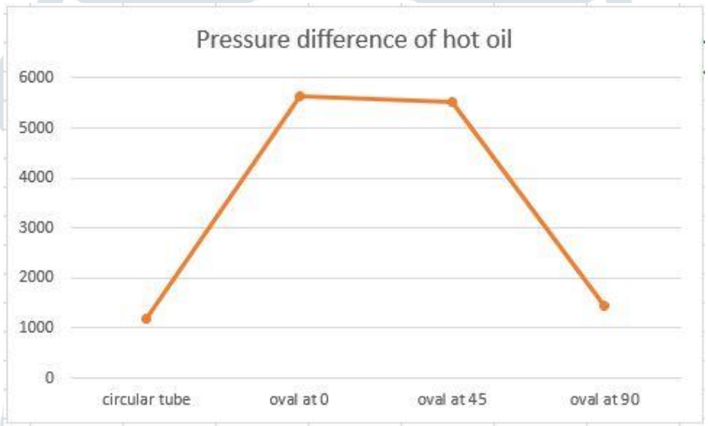


Fig. 28 Pressure difference of oil

- **Temperature variation along length:**

Temperature variation of fluids in heat exchangers along its length as shown in Graph 6 & 7. In this graph, oval at 45^0 has more temperature drop than the others regarding to oil. Whereas circular tube heat exchanger has greater increase in temperature in regrading with water. Overall performance in temperature variation, the oval at 45^0 and oval at 0^0 are better than the other heat exchangers.

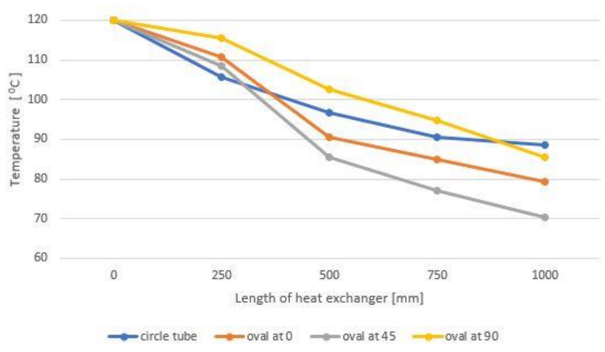


Fig 29. Temperature variation of hot oil

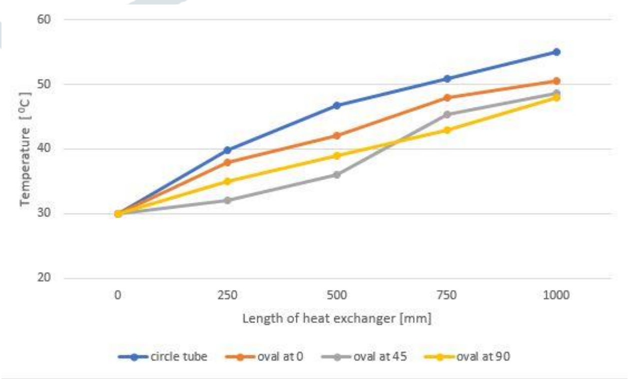


Fig 30. Temperature variation of water

- **Velocity variation along length:**

Velocity variation of fluids in heat exchanger along its length shown in Graph 8 & 9. Circular tube heat exchanger velocity drop in hot oil and velocity raise in water have observed. The oval at 0^0 heat exchanger high raise in velocity of oil and also in water have observed in the results. All the fluids in heat exchanger increased the velocities according to the input conditions.

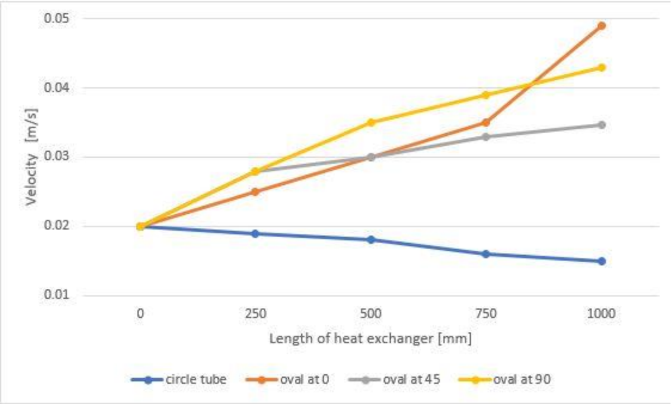


Fig 31. Velocity variation of oil

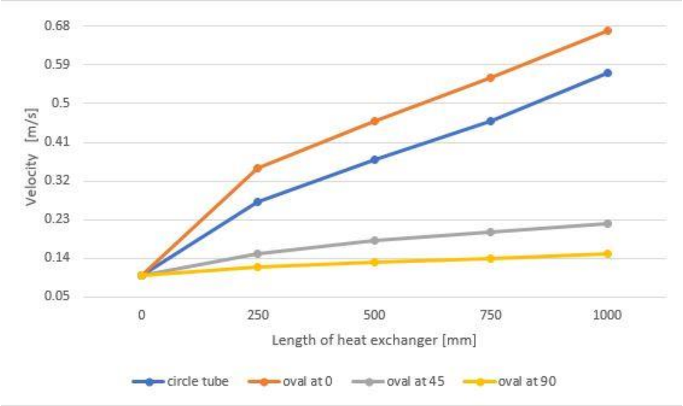


Fig 32. Velocity variation of water

- **Pressure variation in heat exchanger:**

The pressure variation of fluids in heat exchanger along its length as shown in Graph 10. Large pressure variation is observed in oval tube at 0^0 and oval tube at 45^0. Circular tube and oval tube at 90^0 has smaller pressure variation.

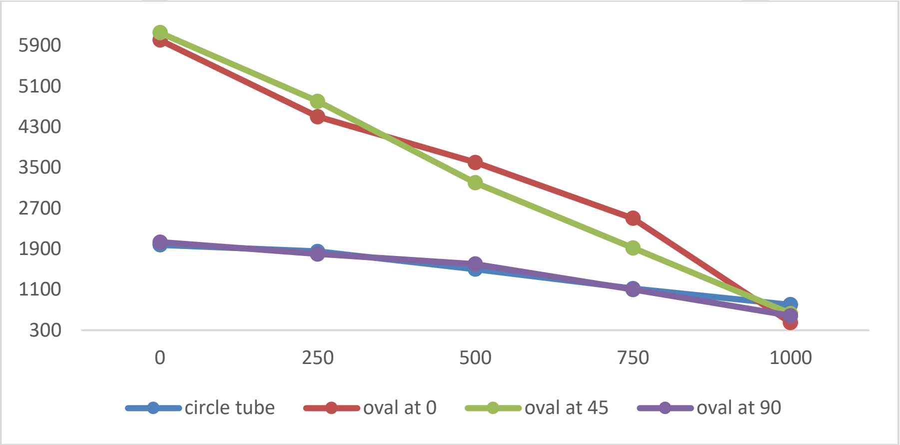


Fig 33. Pressure variation of oil

7. CONCLUSION:

The main object of the project is to design oval tube heat exchanger and compare results with the circular tube heat exchanger. Generally, shell and tube heat exchangers are mostly used in heat transfer purpose. Initially, the basic model of shell and circular tube is designed as per the TEMA and IS standards in order to accommodate the purpose of oil coolers in power plant industry. The thermal design is calculated and mechanical design is taken from the standards.

Based on the above results, circular has oil outlet temperature 88.6 °C and water outlet temperature 60°C. Oval at 0^0 has temperature oil outlet 79.4°C, oval at 45^0 has oil outlet temperature 71 °C and oval at 90^0 has oil outlet temperature of 75°C. When comparing these results, oval at 45^0 heat exchanger gives more temperature difference. It can be used in power plant industries as oil coolers and it can replace the circular tube heat exchanger. By enhancing the oval tube heat exchanger by using helical baffles, increasing the diameter and the length, it will increase the temperature difference and also efficiency increases. So that oval tube heat exchanger perform better than the circular tube heat exchanger regarding to this design. With this design, twisted tube heat exchanger can be developed which results increase in heat transfer and overall efficiency. There is wide scope in development of heat exchangers of oval tube and twisted tube for better results.

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