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DESIGN OPTIMAIZATION OF A SHELL AND TUBE HEAT EXCHANGER BY CHANGING BAFFEL DESIGN

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ABSTRACT

Heat exchangers being one of the most important heat & mass transfer apparatus in industries like oil refining, power plants, chemical engineering etc. are designed with preciseness of optimum performance and long service life.

The Project deals with the design and analysis of the Shell and tube heat exchanger in which we derive the optimistic and reliable conditions for a heat exchanger. We know that segmental baffle is well known to many people and engineers. By replacing the segmental baffle we are introducing helical baffle to improve the heat transfer rate and flow rate inside the shell and tube heat exchanger. Here we are considering the helix angles of 30^o and 45^o .

The shell side zig-zag flow induced by the segmental baffle arrangement is completely eliminated in a helical heat exchanger. The flow pattern in the shell side of the continuous helical baffle heat

exchanger is rotational and helical due to the geometry of continuous helical baffles.

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In this project we design the 3d model of the shell and tube heat exchanger by solid works and analyze using ANSYS by using computational facilities.

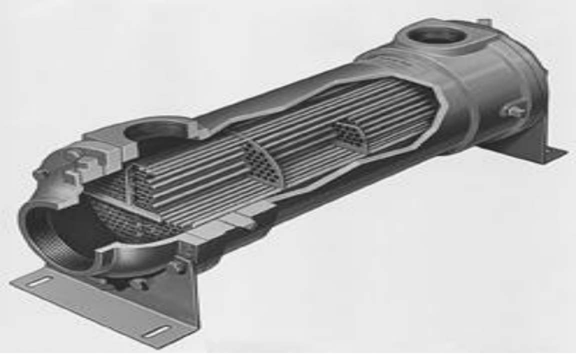
The results obtained in this project shows us that the desired properties from a heat exchanger i.e., high heat transfer coefficient and low pressure drop are more effectively obtained in a helix heat exchanger

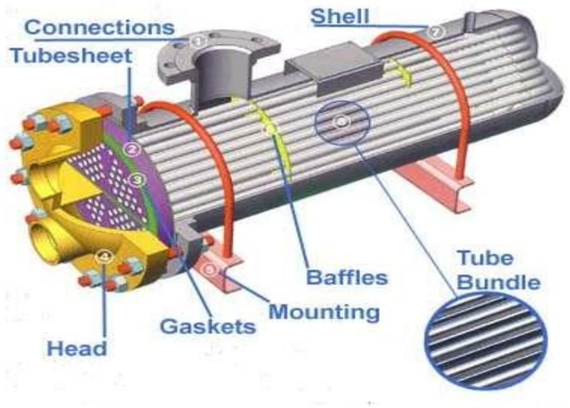
Introduction:

A heat exchanger is a device for transferring heat from one fluid to another, where a solid wall separates the fluids so that they never mix. They are widely used in refrigeration, air conditioning, space heating, power production, and chemical processing. One common example of a heat exchanger is the radiator in a car, in which the hot radiator

fluid is cooled by the flow of air over the radiator surface.

A **shell and tube heat exchanger** is a class of heat exchanger designs. It is the most common type of heat exchanger in oil refineries and other large chemical processes, and is suited for higher-pressure and higher-temperature applications. As its name implies, this type of heat exchanger consists of a shell (a large pressure vessel) with a bundle of tubes inside it. One fluid runs through the tubes, and another fluid flows over the tubes (through the shell) to transfer heat between the two fluids. The set of tubes is called a tube bundle, and may be composed by several types of tubes: plain, longitudinally finned, etc.





T Theory and Application:-

Two fluids, of different starting temperatures, flow through the heat exchanger. One flows through the tubes (the tube side) and the other flows outside the tubes but inside the shell (the shell side). Heat is transferred from one fluid to the other through the tube walls, either from tube side to shell side or vice versa. The fluids can be either liquids or gases on either the shell or the tube side. In order to transfer heat efficiently, a large heat transfer area should be used, leading to the use of many tubes. In this way, waste heat can be put to use. This is an efficient way to conserve energy.

Heat exchangers with only one phase (liquid or gas) on each side can be called one-phase or single-phase heat exchangers. Two-phase heat exchangers can be used to heat a liquid to boil it into a gas (vapor), sometimes called boilers, or cool a vapor to condense it into a liquid (called condensers), with the phase change usually occurring on the shell side. Boilers in steam engine locomotives are typically large, usually cylindrically-shaped shell-and-tube heat exchangers. In large power plants with steam-driven turbines, shell-and-tube surface condensers are used to condense the exhaust steam exiting the turbine into condensate water which is recycled back to be turned into steam in the steam generator.

LITERATURE REVIEW

The literature reviewed in this chapter can be broadly classified under three categories. The first part of the survey deals with the analytical solution for maldistribution in shell and tube heat exchanger. Second part of the survey deals with the experimental and CFD analysis of maldistribution in heat exchanger, and third part of the survey deals with the analysis of maldistribution in plate heat exchanger.

WilfriedRoetzel, Chakkrit Na Ranong., [1] calculated the axial temperature profiles in a shell and tube heat exchanger by numerically for given maldistributions on the tube side. For comparison the same maldistributions are handled with the parabolic and hyperbolic dispersion model with fitted values for the axial dispersion coefficient of third sound wave velocity.

The analytical results clearly demonstrate that the hyperbolic model is better suited to describe the steady state axial temperature profiles. For a global consideration of a heat exchanger with maldistribution the parabolic model is satisfactory.

The parameter Pepar depends on the NTU of the maldisributed flow stream and on the NTU of the transversely mixed flow stream which makes the model

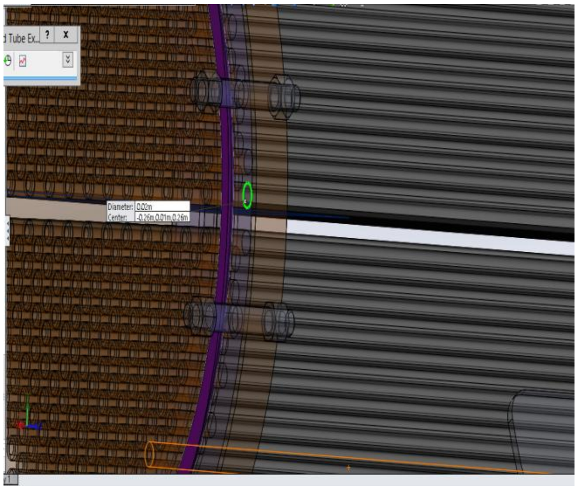
difficult to handle. The hyperbolic model predicts the axial temperature profiles correctly, especially temperature jumps and positive slopes. The third sound Mach number M characterizes the type of Temperature maldistribution and is independent of both NTUs. For a given type of relative Temperature maldistributionPehyp is proportional to the NTU of the maldistributed flow stream but does not depend on the other NTU: Sahoo, R.K., and WilfriedRoetzel., [2] derived The fundamental equations of hyperbolic model and its boundary conditions in terms of cross-sectional mean temperature from the basic equations of heat exchanger The traditional parabolic model and the proposed hyperbolic model which includes the parabolic model as a special case can be used for dispersive flux formulation. Instead of using the heuristic approach of parabolic or hyperbolic formulation, these models can be quantitatively derived from the axial temperature profiles of heat exchangers. In this paper both the models are derived for a shell-and-tube heat exchanger with pure maldistribution (without back mixing) in tube side flow and the plug flow on the shell side. The Mach number and the boundary condition which plays a key role in the hyperbolic dispersion have been derived and compared with previous investigation. It is observed that the hyperbolic model is the best suited one as it compares well with the actual calculations.

This establishes the hyperbolic model and its boundary conditions.

WilfriedRoetzel and Frank Balzereit., The effect of the deviation of the actual three-dimensional flow field on the shell-side from the frequently assumed one-dimensional uniform axial plug flow can be taken into account by superimposed axial dispersion in the fluid. The measure for axial dispersion is the Peclet number which can vary from infinite (no dispersion) to zero (complete axial mixing). For the fast and more reliable calculation of transient processes with the axial dispersion model,

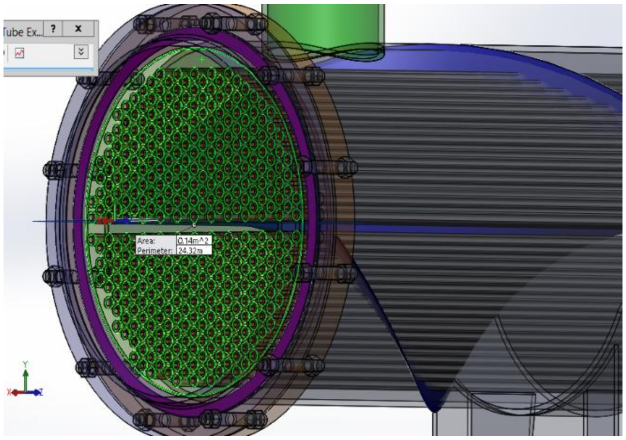
the Péclet number has to be known. A residence time distribution measurement technique for the determination of shell-side dispersive of baffles, and axial plug flow Reynolds numbers.

Drafting details of heat exchanger:



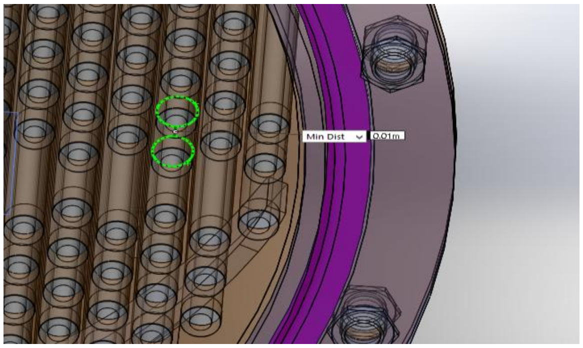
Diameter: 0.02m
Center: -0.26m,0.01m,0.26m

Diameter of tube present inside the baffle is 0.02m



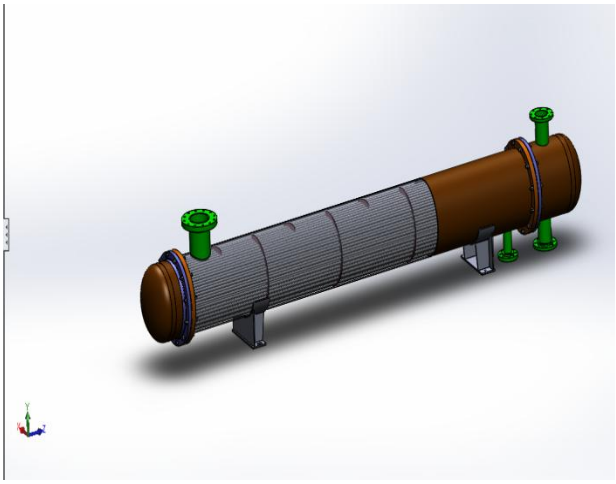
Area: 0.14m^2
Perimeter: 24.32m

The minimum distance between tubes is 0.01m

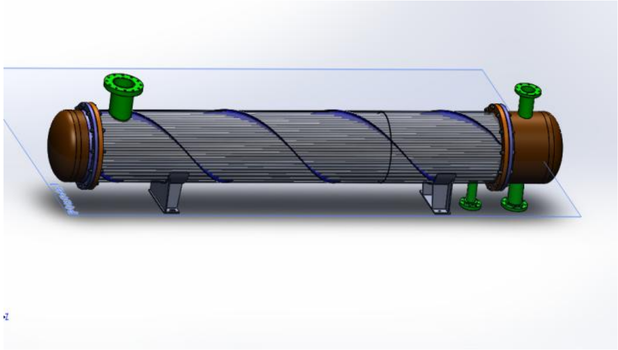


Perimeter of heat exchanger is 24.32m

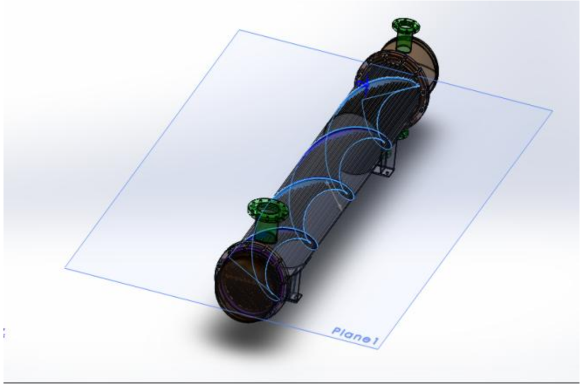
Design of a 3d model in solid works



3d designed solid works model with segmental baffles



Spiral baffle heat exchanger



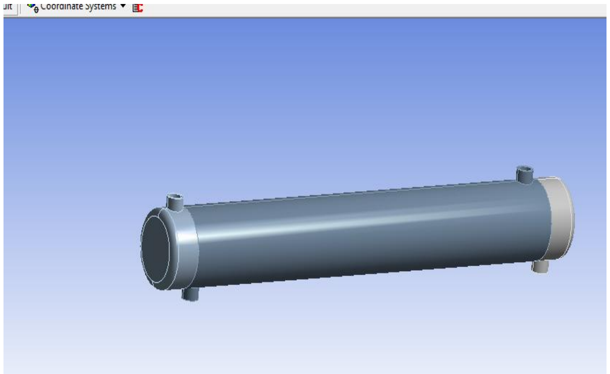
Spiral drawn with the twisting angle of 30 degrees

Analysis of the shell and tube heat exchanger:-

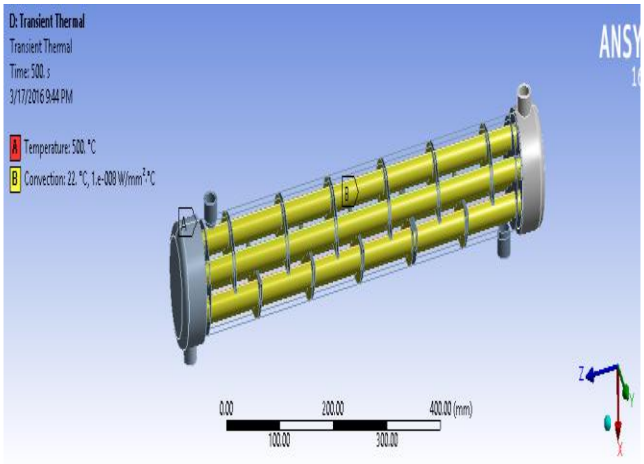
Structural and thermal analysis is where the industrial application of FEA methods began and is still the most common application today. Although many 3D CAD systems have basic capabilities embedded within their products, the key to successful analysis is understanding how to set up your models and interpret the results correctly – including the importance of quality meshing.

When implemented correctly, FEA can yield many benefits over traditional hand calculation methods or prototype testing in terms of productivity and innovation. However, inaccurate or wrong results due to poor guidance from the FEA software or your training provider can lead to uninformed decision making or much worse. Wilde has been involved in the assessment of stresses within structures since formation in 1980.

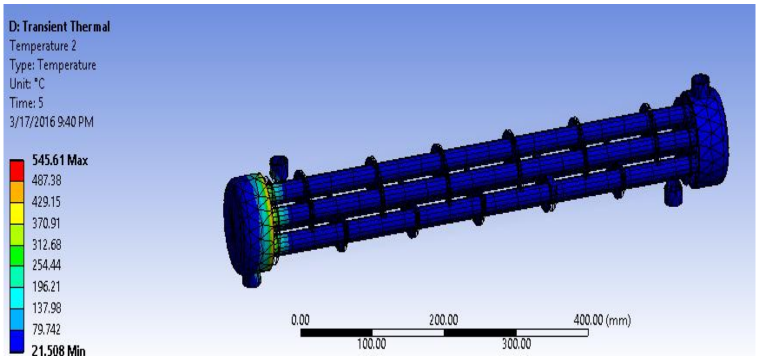
Today, we provide a powerful combination of leading software and a large, highly skilled and experienced technical services team offering support, consulting and training.



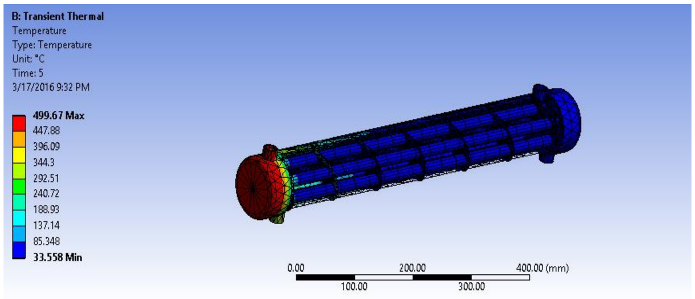
Optimized design for structural thermal analysis imported into Ansys design modeler



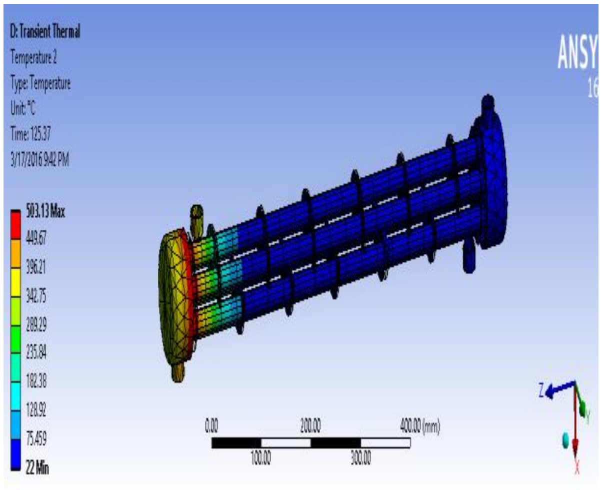
Boundary condition for segmental baffle



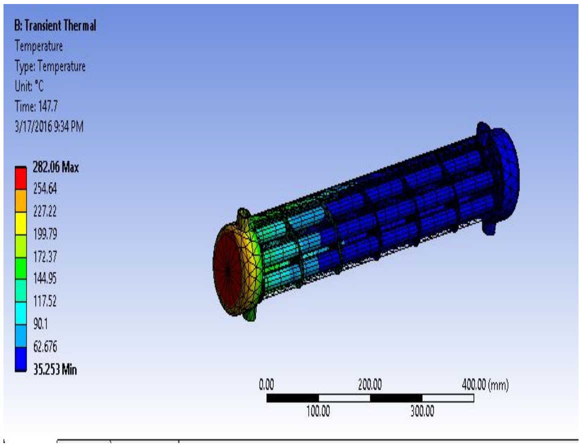
Temperature distribution segmental baffle at 5sec



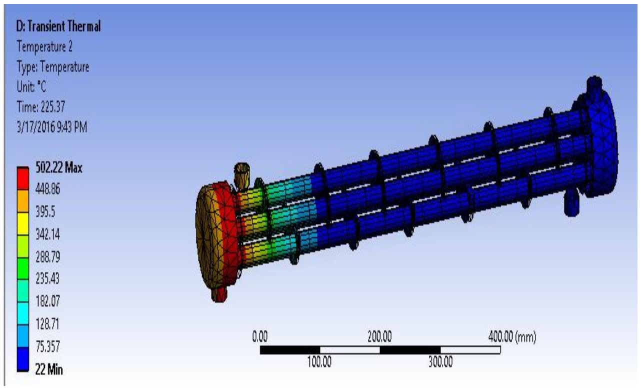
Temperature distribution in helix baffle at 5 sec



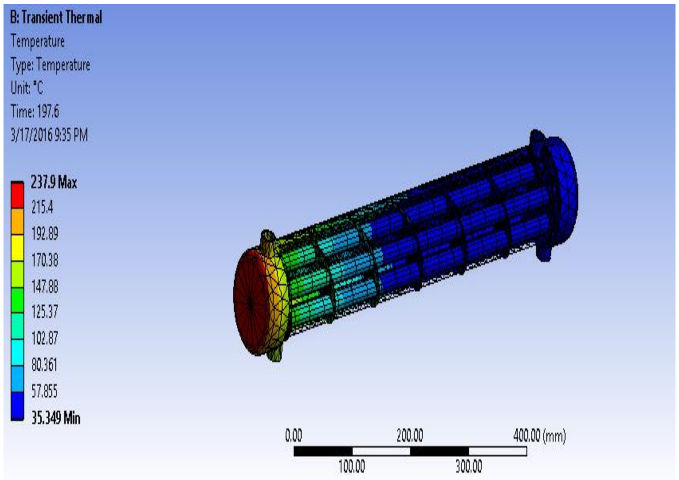
Temperature distribution in segmental baffle at 125sec



Temperature distribution in helix baffle at 125sec

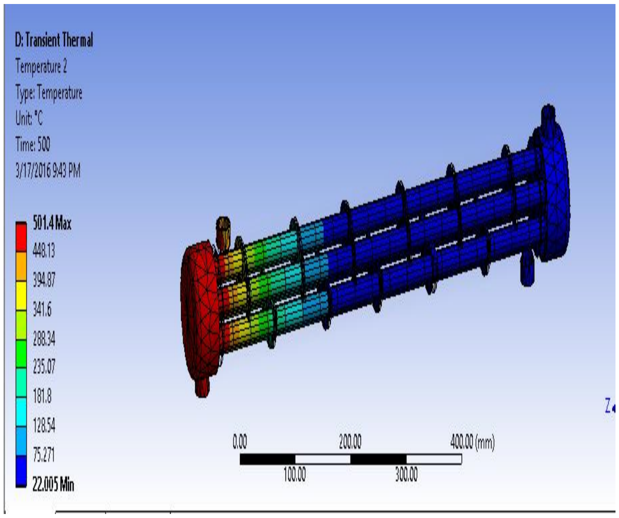


Temperature distribution in segmental baffle at 200sec



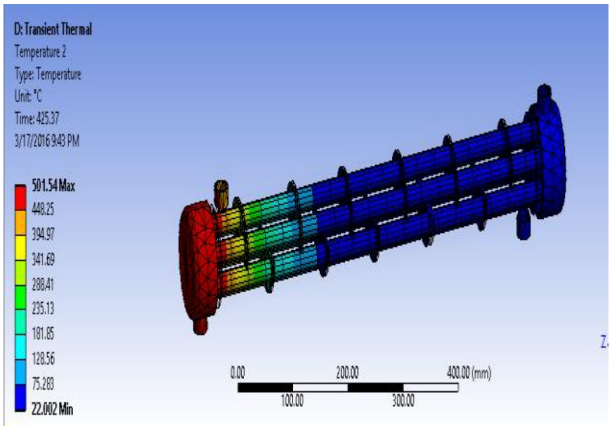
Temperaturedistribution in segmental baffle

At 200 sec



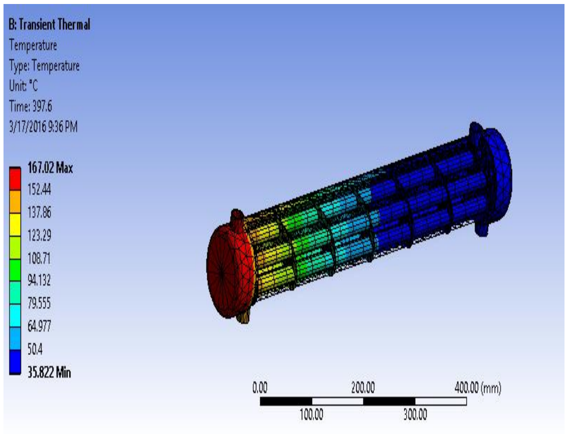
Temperature distribution in helix baffle

At 200 sec



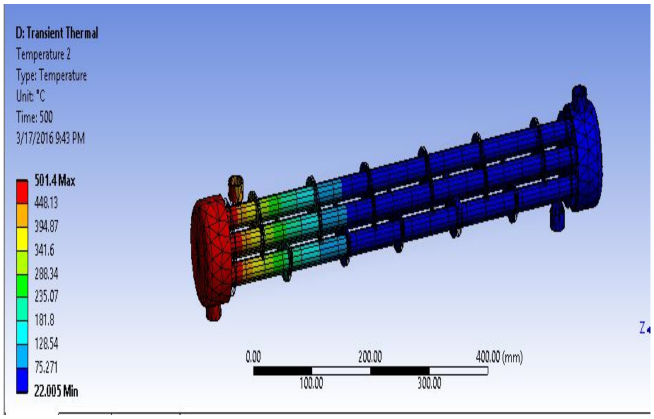
Temperature distribution in segmental baffle

At 400sec

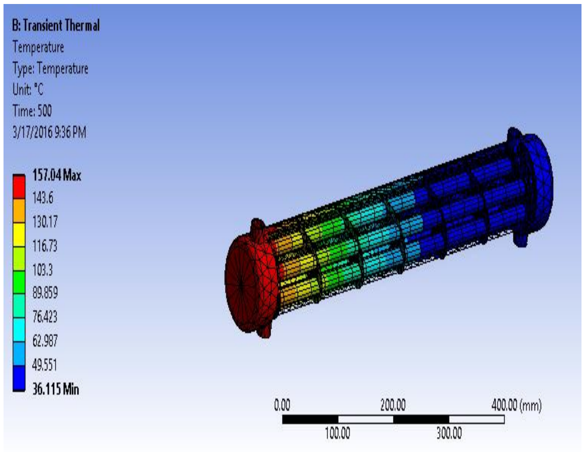


Temperature distribution in segmental baffle

at 400sec



Temperature distribution in segmental baffle at 500sec



Temperature distribution in helix baffle at 500sec

RESULT &DISUSSION :

Model (C4, D4) > Transient Thermal (D5) > Solution (D6) > Temperature

Time [s]	Minimum [°C]	Maximum [°C]
5	21.508	545.61
9.025	21.917	532.97
10.822	21.971	526.12
12.619	21.986	521.01
16.465	21.995	515.12
22.612	21.999	510.72
32.909	22.	507.6
50.889		505.46
81.336		504.04
125.37		503.13
175.37		502.57
225.37		502.22
275.37		501.97
325.37		501.79
375.37	22.001	501.65
425.37	22.002	501.54
475.37	22.004	501.44
500.	22.005	501.4

Model (A4, B4) > Transient Thermal (B5) > Solution (B6) > Temperature

Time [s]	Minimum [°C]	Maximum [°C]
5.	33.558	499.67
6.6667	34.064	499.51
7.7944	34.498	499.37
8.9222	34.999	499.18
11.844	35.	498.21
16.595	35.003	494.89
24.516	35.012	486.51
38.39	35.039	462.3
63.14	35.107	411.02
101.35	35.176	342.5
147.7	35.253	282.06
197.6	35.349	237.9
247.6	35.454	208.35
297.6	35.569	188.7
347.6	35.691	175.67
397.6	35.822	167.02
447.6	35.962	161.29
473.8	36.037	158.91
500.	36.115	157.04

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