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Computational Modeling of STHE

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Abstract*-Computational Fluid Dynamics (CFD) can very useful to gain visualize the flow and temperature fields on the shell side can simplify the assessment of the weakness. In this present study, attempts were made to investigate the impacts of various mass flow rates on fluid flow and the heat transfer characteristics of a shell-and-tube heat exchanger for 0°baffle inclination angle. The simulation results for different mass flow rates are compared for their performance. The study is concerned with a single shell and single side pass parallel flow heat exchanger. The flow and temperature fields inside the shell are studied using non-commercial computational fluid dynamics software tool ANSYS CFX 14.0.From the computational fluid dynamics simulation results, the shell side outlet temperature, pressure drop, recirculation near the baffles, optimal mass flow rate and heat transfer graph, mass and momentum graph, turbulence graph are determined for the given heat exchanger geometry.

Index Terms: **Computational fluid dynamics, computational modeling, momentum and mass graph, heat transfer graph, turbulence graph, pressure drop.**

I. INTRODUCTION

Shell and tube heat exchangers consist of a bundle of parallel tubes that provide the heat transfer surface separating the two fluid streams. The tube side fluid passes axially through the inside of the tubes. The shell side fluid passes over the outside of the tubes. The process fluid is usually placed inside the tubes for ease of cleaning or to take advantage of the higher pressure capability inside the tubes. The thermal performance of such an exchanger usually surpasses a coli type but is less than a plate type. Pressure capability of shell and tube heat exchanger is generally higher than a plate type. This heat exchanger shown in Fig.1 is generally built of a round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flows inside the tubes, the other flows across and along the tubes. The major components of this exchanger are tubes, shell, and front-end head, rear-end head, baffles and tube sheets.

A variety of different internal constructions are used in shell-and -tube exchangers, depending on the desired heat transfer and pressure drop performance and the methods employed to reduce thermal stresses, to prevent leakages.

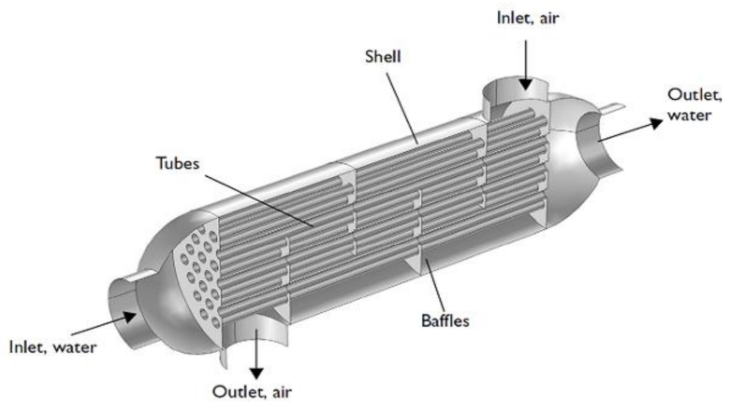

Fig.1: Shell and tube heat exchangers

Also used to provide for ease of cleaning, to contain operating pressures and temperatures, to control corrosion, to accommodate highly asymmetric flows and so on.

II. COMPUTATIONAL MODELING

A computational model is a mathematical model in computational science that requires extensive computational resources to study the behavior of a complex system by computer simulation. The system under study is often a complex nonlinear system for which simple, intuitive analytical solutions are not readily available.

Rather than deriving a mathematical analytical solution to the problem, experimentation with the model is done by adjusting the parameters of the system in the computer, and studying the differences in the outcome of the

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experiments. Operation theories of the model can be derived/deduced from these computational experiments. The computational modeling involves pre-processing, solving and post-processing.

A. Geometry modeling

The cavity model is designed according to TEMA (Tubular Exchanger Manufacturers Association) Standards Gaddis (2007), using Pro-E Wildfire-4 software. Design parameters and fixed geometric parameters have been taken similar to Ender Ozden et al.[9],as indicated in table.1.

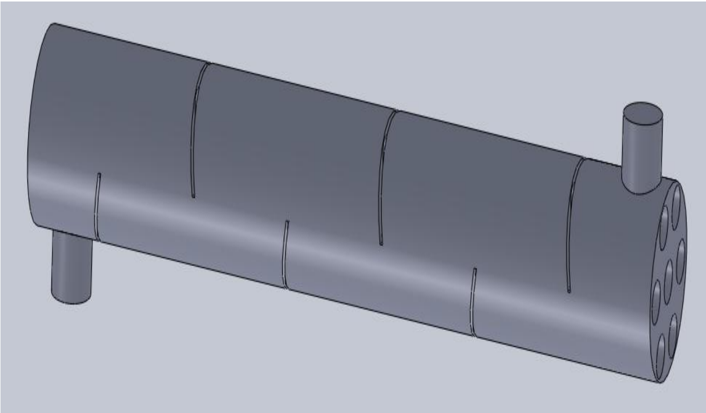

Fig-2: Cavity Model of Shell and Tube Type Heat Exchanger at 0° baffle inclination

Table-1: Design parameters of shell and tube heat exchanger

Heat exchanger length, L	600 mm
Shell inner diameter, D_i	90 mm
Tube outer diameter, d_o	20 mm
Tube bundle geometry and pitch triangular	30 mm
Number of tubes, N_t	7
Number of baffles, N_b	6
Central baffle spacing, B	86 mm
Baffle inclination angle,ϴ	0°,10° and 20°

B. Mesh Generation

The partial differential equation that governs fluid flow and heat transfer are not usually amenable to analytical solutions, except for very simple cases. Therefore, in order to analyze fluid flows, flow domains are split into smaller sub domains (made up of geometric primitives like hexahedra and tetrahedral in 3D, and quadrilaterals and triangles in 2D) and discredited governing equations are solved inside each of these portions of the domain. Mesh generation is the practice of generating a polygonal or polyhedral mesh that approximates a geometric domain.

The entire geometry is divided into three fluid domains Fluid Inlet, Fluid Shell and Fluid Outlet.

Details of Meshing

Type of Analysis: - 3D

Type of Element: - Tetrahedral (10 Node)

Physical preference-CFD

Solver preference-CFX

Use advance size of function- On Curvature

Relevance centre- Coarse

Smoothing- Medium

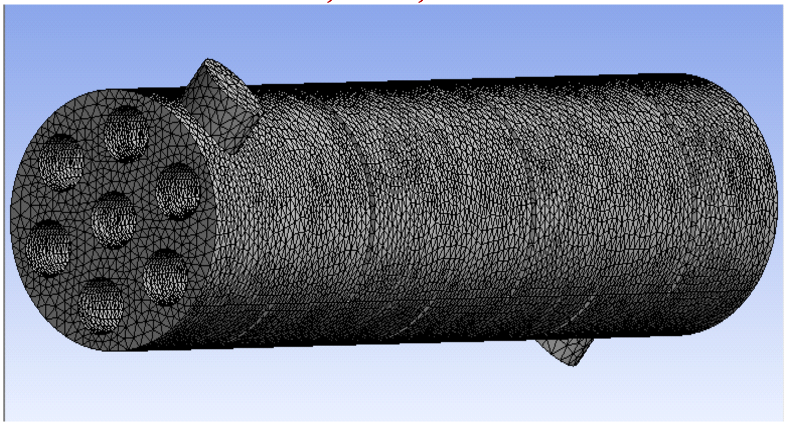

Fig-3: Meshing of Heat Exchanger

Governing equations

The governing equations of the flow are modified according to the conditions of the simulated case. Since the problem is assumed to be steady, time dependent parameters are dropped from the equations are:

Conservation of mass: $\nabla(\rho V_r) = 0$

X-momentum :

$$\nabla.(\rho u V_r) = -\frac{\partial p}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z}$$

y-momentum:

$$\nabla.(\rho v V_r) = -\frac{\partial p}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + \rho g$$

z-momentum:

$$\nabla.(\rho w V_r) = -\frac{\partial p}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + \rho g$$

Energy: $\nabla(\rho e V_r) = -p\nabla V_r + \nabla(k\nabla T) + q\varphi$ ----(1)

In Eq.(1), φ is the dissipation function that can be calculated from

$$\varphi = \mu[2[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial z}\right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)^2] + \lambda(\nabla.V_r)^2$$

C. Turbulence Model

The realizable k-ε model is a relatively recent development and differs from the standard k-ε model in two important ways:

- The realizable k-ε model contains a new formulation for the turbulent viscosity.
- A new transport equation for the dissipation rate, ε, has been derived from an exact equation for the transport of the mean-square vortices fluctuation.

Transport equations for the Realizable k-ε model

The modeled transport equations for k and ε in the realizable k-ε model are

$$\frac{\partial}{\partial t}\rho k + \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j}[\left(\mu + \frac{\mu t}{\sigma_k}\right)\frac{\partial k}{\partial x_j} + G_k + G_b - \rho\varepsilon - Y_M + S_k$$

and

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \frac{\partial}{\partial x_j}(\rho\varepsilon u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{v\varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b + S_\varepsilon$$

where $C_1 = \mathrm{Max}\left[0.43, \frac{n}{n+5}\right], n = \frac{Sk}{\varepsilon}$, $s = \sqrt{2S_{ij}S_{ij}}$

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In these equations, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients, G_b is the generation of turbulence kinetic energy due to buoyancy, Y_M represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate.
C_2 and $C_{1\varepsilon}$ are constants., σ_k and σ_ε are the turbulent Prandtl numbers for **k** and **ε**, respectively. S_k and S_ε are user-defined source terms.

Boundary conditions:-

1. The working fluid of the shell side is water,
2. The shell inlet temperature is set to 300 K,
3. The constant wall temperature of 450 K is assigned to the tube walls,
4. Zero gauge pressure is assigned to the outlet nozzle,
5. The inlet velocity profile is assumed to be uniform,
6. No slip condition is assigned to all surfaces,

III. RESULT AND DISCUSSION

A. Validation

Simulation results are obtained for different mass flow rates of shell side fluid ranging from 0.5 kg/s,1 kg/s and 2 kg/s. The simulated results for different mass flow rates for model with 0°baffle inclination angle are validated with the data available in [9].

It is seen that the temperature gradually increases from 300k at the inlet to 340 k at the outlet of the shell side. The average temperature at the outlet surface is nearly 326.22 k for this model. The pressure drop is less for 2kg/s mass flow rate compared to other two mass flow rates. The maximum velocity is 3.02841 m/s for 2 kg/s mass flow rate at the inlet and exit surface and velocity magnitude reduces to zero at the baffle surface. It can be compared that 2 kg/s mass flow gives more heat transfer than other two mass flow rates. From the stream line contour, it is found that recirculation near the baffles induces turbulence eddies which would result in more pressure drop for this model. From the result table it is found that the shell outlet temperature decreases with increasing mass flow rates. Increment in mass flow rate gives also increment in velocity. For given model, the mass flow rate must be below 2 kg/s, if it is increased beyond 2 kg/s the pressure drop increases rapidly.

For 0.5 kg/s mass flow rate

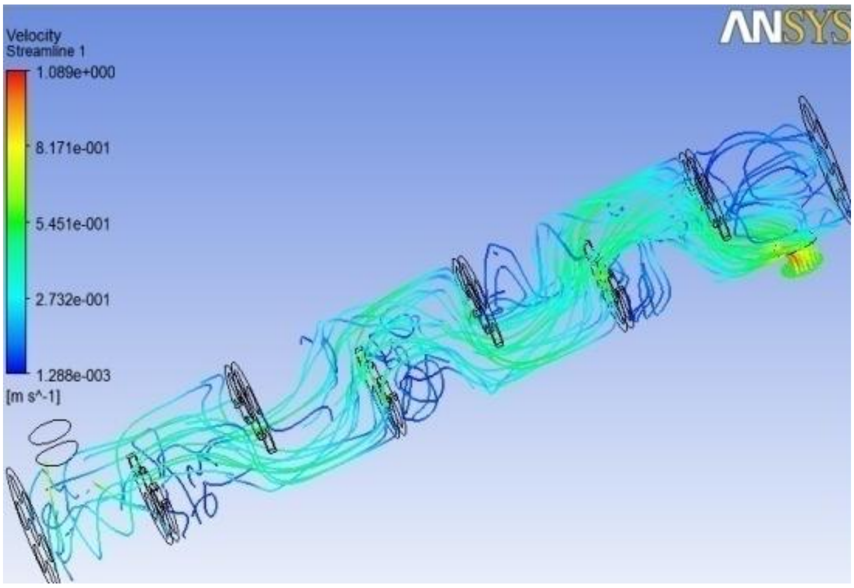

Fig-4: Velocity streamline contour

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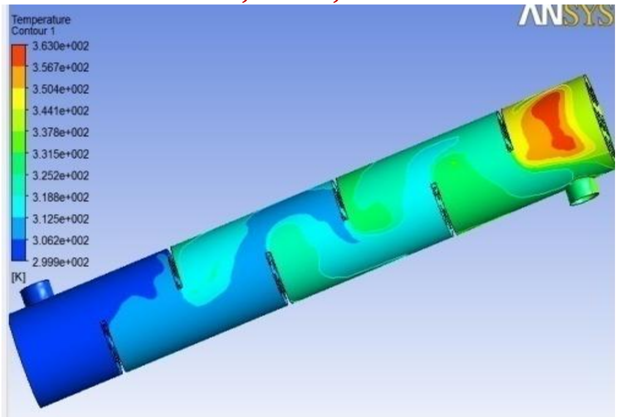

Fig-5: Temperature Contour

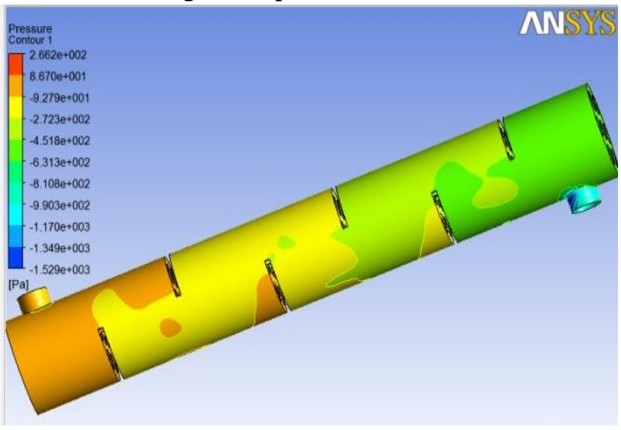

Fig-6: Pressure Contour

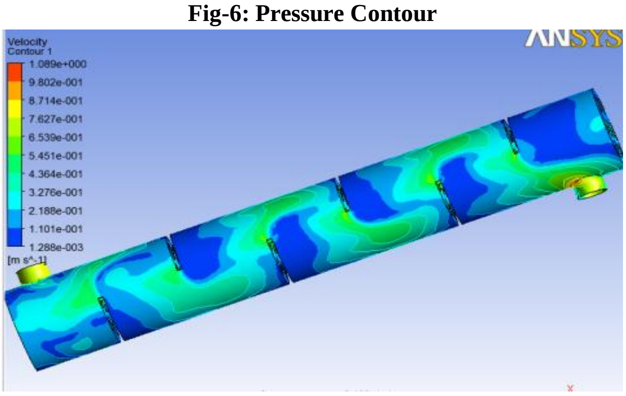

Fig-7: Velocity Contour

For 1 kg/s mass flow rate

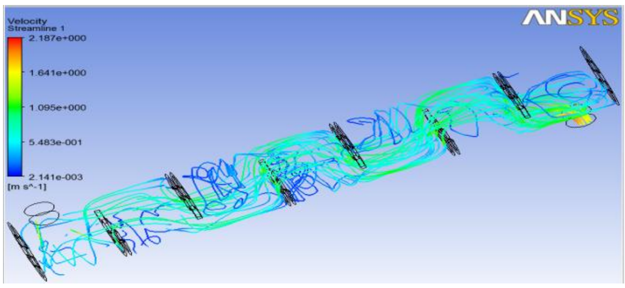

Fig-8: Velocity streamline contour

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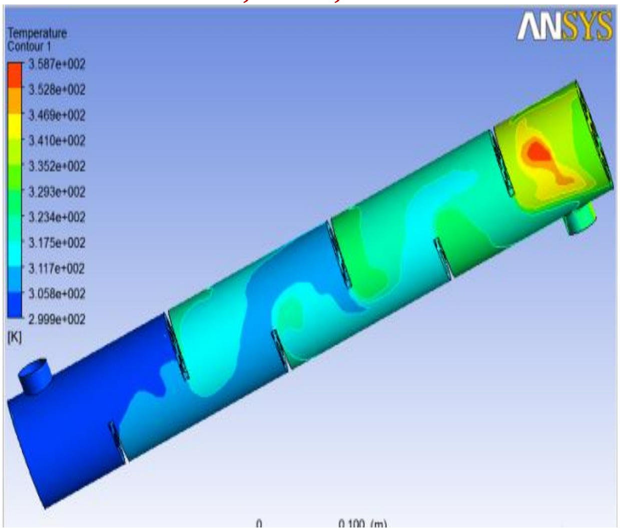

Fig-9: Temperature Contour

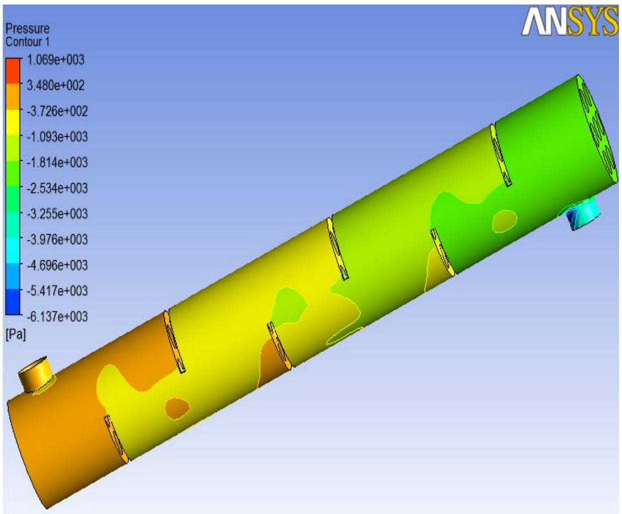

Fig-10: Pressure Contour

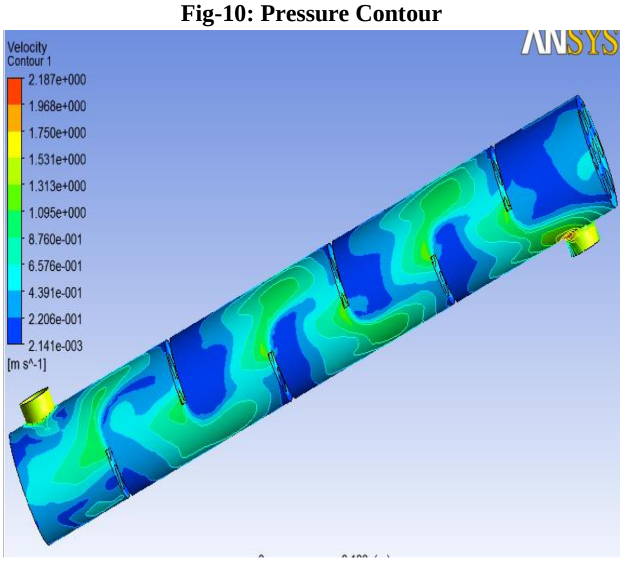

Fig-11: Velocity Contour

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For 2 kg/s mass flow rate

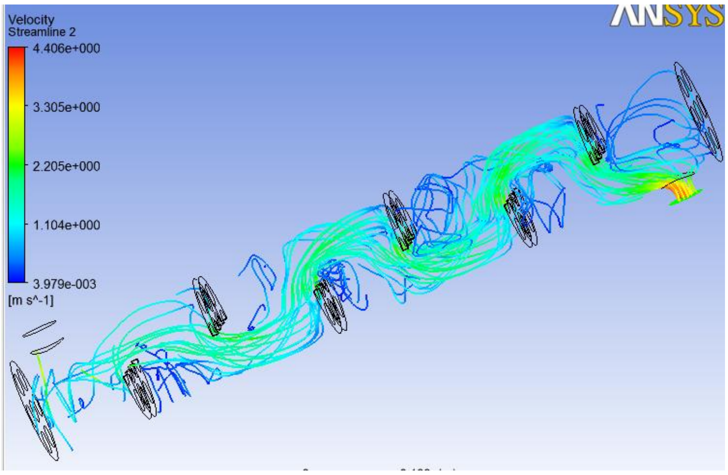

Fig-12: Velocity streamline contour

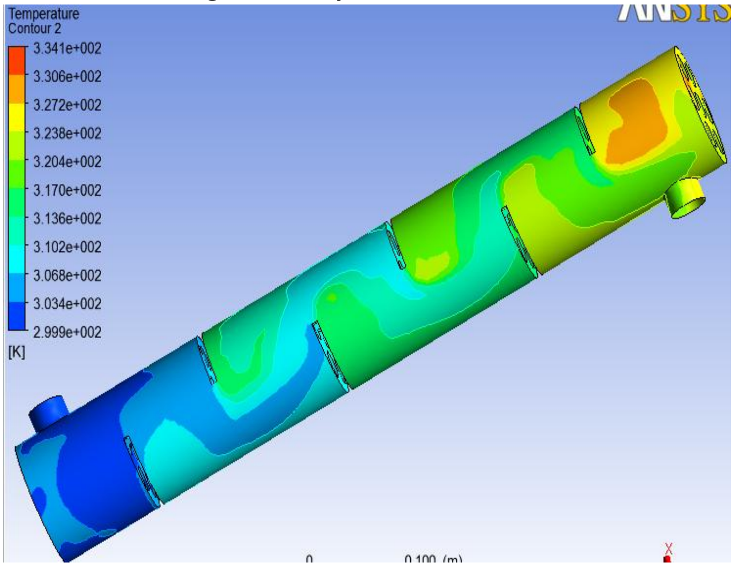

Fig-13: Temperature Contour

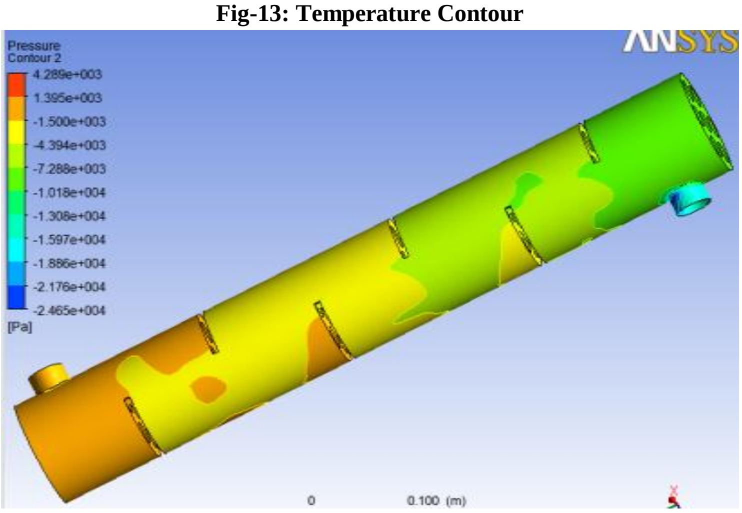

Fig-14: Pressure Contour

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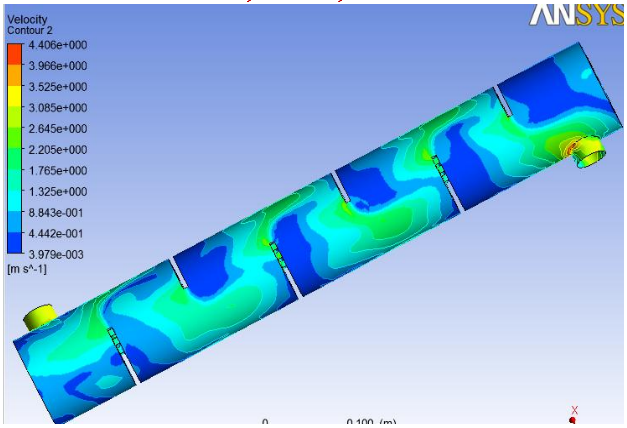

Fig-15: Velocity Contour

Table-2: CFD Result

Mass flow rate[kg/s]	Outlet Temperature [K]	Pressure difference [Pa]	Velocity [m/s]
0.5	330.032	976.836	0.755108
1	326.627	3948.86	1.51358
2	322	15528.7	3.02841

For 0.5 kg/s mass flow rate

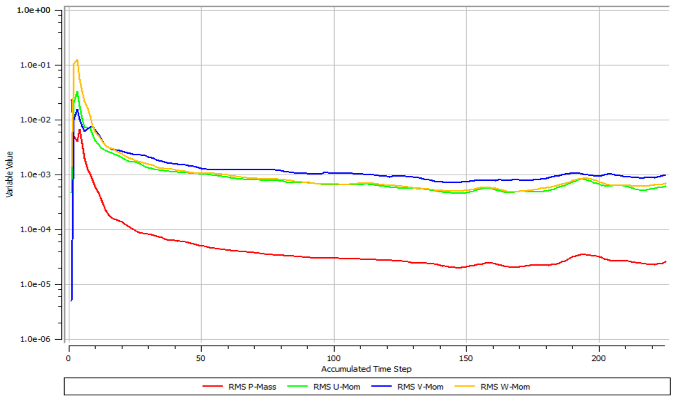

Fig-16: Momentum and mass graph

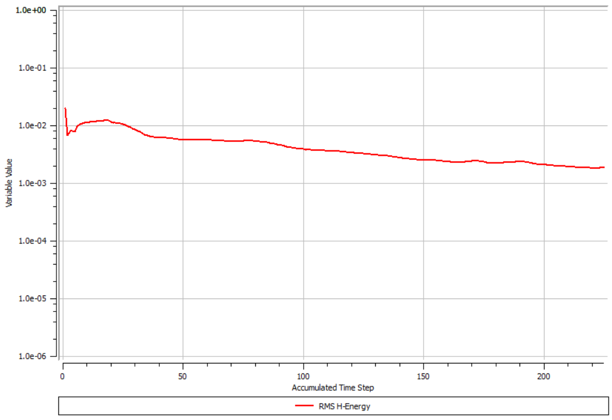

Fig-17: Heat transfer graph

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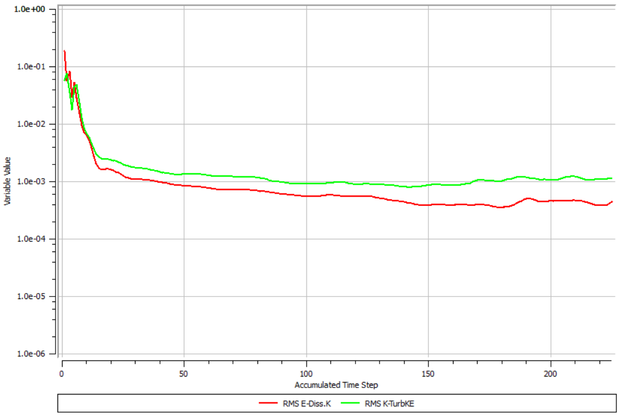

Fig-18: Turbulence graph

1-250 iterations for 1 kg/s mass flow rate
250-450 iterations for 2 kg/s mass flow rate

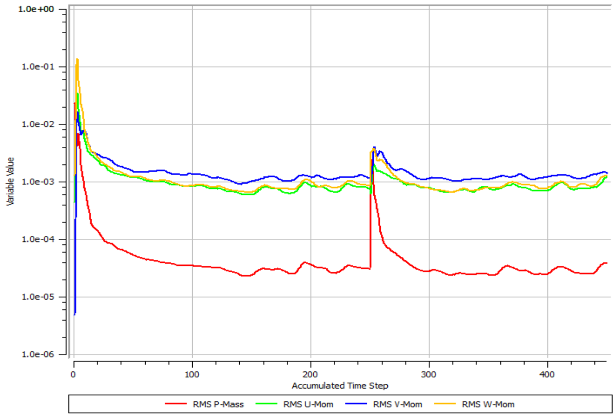

Fig-19: Momentum and mass graph

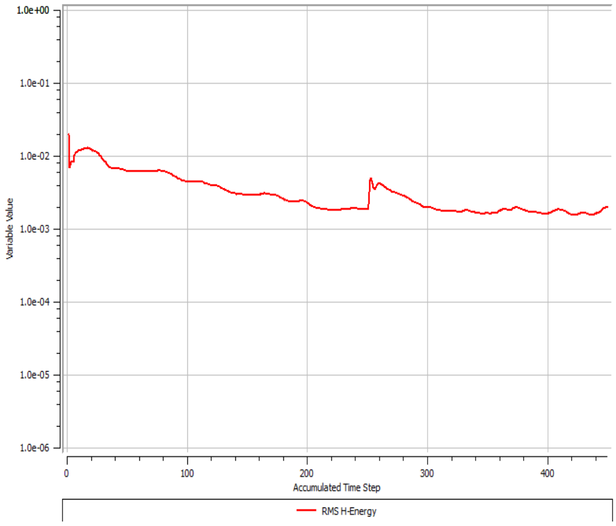

Fig-20: Heat transfer graph

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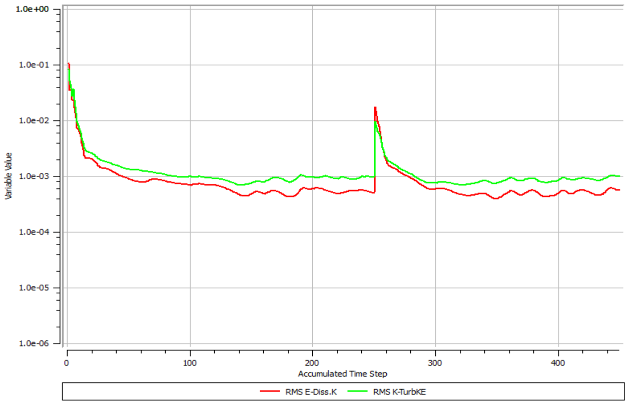

Fig-21: Turbulence graph

IV.CONCLUSION

Shell and tube heat exchanger for shell side study of the fluid flow with zero baffle inclination angle is modeled in the Pro-E Software. The initial CFD analysis has been performed in the ANSYS Software. The k-ε turbulence model is used for the simulation based on the literature survey. The initial simulation results agree with the fundamental physics of the heat transfer in the heat exchanger. The results show the following points.

- CFD has emerged as a cost effective alternative and it provides speedy solution to heat exchanger design and optimization.
- The k - ε turbulence model has been most widely employed for heat exchanger design optimization
- From the CFD analysis it is found that the shell outlet temperature decreases with increasing mass flow rates.
- Increment in mass flow rate gives also increment in velocity.
- For the given model, the mass flow rate must be below 2 kg/s, if it is increased beyond 2 kg/s the pressure drop increases rapidly.

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