DESIGN AND DEVELOPMENTOF SHELL AND TUBE TYPE HEAT EXCHANGER BY USING CFD WITH FINAL RESULT

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ABSTRACT

The design and Analysis of Heat Exchanger is carried out with a major consideration in terms of Mass flow of the fluid to be distributed uniformly at the inlet of the exchanger on each side that is, fluid side and throughout the core. But, in actual practice, the flow Non-Uniformity is more common and significant which reduces the desired heat exchanger performance. Non- Uniformity is the non-uniform distribution of mass flow rate on one or both fluid sides in the heat exchanger core. The major feature of gross flow Non-Uniformity is that non uniform flow occurs at the macroscopic level due to poor header design or blockage of some flow passages during manufacturing, including brazing or operation. Non-Uniformity causes a significant increase in heat exchanger pressure drop and some reduction in heat transfer rate as well.

For Analysis and study purpose, the commercial Computational Fluid Dynamic (CFD) package, Fluent was utilized for modeling the tubular single pass heat exchanger with different tube arrangements namely four tube in-line arrangement, two tube in-line arrangement and square pitch tube arrangement. In each arrangement both flow Non-Uniformity and uniformmass flow distribution are considered. In heat transfer terminology there appear two mean temperatures namely cross-sectional mean temperature and adiabatic mean temperature. The cross-sectional temperature is the arithmetic mean of all tube side temperatures and the adiabatic mean temperature is weighted mean of tube side temperatures.

Investigation is carried out to study the cross sectional mean temperature and adiabatic mean temperature profiles in the computational domain, tubular single pass heat exchanger for flow Non-Uniformity or uniform mass flow distribution on tube side and ideal plug flow on shell side.It is investigated that for uniform mass flow distribution on tube side and ideal plug flow on shell side, there is no difference the cross-sectional mean temperature and adiabatic mean temperature. But for Non-Uniformity without back flow on tube side and ideal plug flow on shell side, the two mean temperatures have same value at the cross-section $\xi=0$. For Non-Uniformity with back flow on tube side and ideal plug flow on shell side, the temperature jump occurs at the beginning of the calculation domain.

The available computational fluid dynamics software package FLUENT is applied to determine the related problems. A large computational effort is involved for the memory access of the computers and computing time for the simulation of the complex geometries associated with the dense grids. Standard k - ε turbulence model is allowed to predict the three-dimensional flow and the conjugate heat transfer characteristics.

Keyword

1. *Validation of the fluent results with the experimental results.*

2. *Validation of obtained temperature profiles with the numerically solved temperature profiles for different flow regimes of Pressure difference or back Flow in the Exchanger tubes.*

2. INTRODUCTION

Heat exchange is One of the matter of interest and important processes in engineering between flowing fluids, and in various types of installations many types of heat exchangers are employed, as petro-chemical plants, process industries, pressurized water reactor power plants, nuclear power stations, building heating, ventilating, and air-conditioning and refrigeration systems. As far as construction design is concerned, the tubular or shell and tube type heat exchangers are widely in use, because of their efficiency, size and effectiveness along with ease of use.

The shell-and-tube heat exchangers are still the most common type in use. They have larger heat transfer surface area-to-volume ratios than the most of common types of heat exchangers, and they are manufactured easily for a large variety of sizes and flow configurations.They can operate at high pressures, and their construction facilitates disassembly for periodic maintenance and cleaning. The shell-and-tube heat exchangers consist of a bundle of tubes enclosed within a cylindrical shell. One fluid flows through the tubes and a second fluid flows within the space between the tubes and the shell. Typical Shell-and-Tube heat exchanger is shown in Figure 1.1.

Heat exchangers in general and tubular heat exchangers in particular undergodeterioration in performance due to flow Non-Uniformity. The common idealization in the basic tubular heat exchanger design theory is that the fluid is distributed uniformly at the inlet of the exchanger on each fluid side throughout the core. However, in actual practice, flow Non-Uniformity is more common and significantly reduces the idealized heat exchanger performance. Flow Non- Uniformity can be induced by the heat exchanger geometry, operating conditions (such as viscosity or density-induced Non-Uniformity), multiphase flow, fouling phenomena, etc. Geometry-induced flow Non-Uniformity can be classified into gross flow Non-Uniformity, passage-to-passage flow Non-Uniformity and manifold-induced flow Non-Uniformity.

The flows in shell-and-tube heat exchangers have only been investigated analytically [1,2, and 3] due to their complexity. Ranjit Kumar Sahoo , Wilfried Roetzel [1] and Chakkrit Na Ranong[2] carried out an analysis of the effect of Non-Uniformity on the thermal performance and the temperature distribution in shell and tube heat exchanger using a finite differencemethod.

A. Roy and S. K. Das [7] used the computer code STAR-CD to study the gross flow Non- Uniformity in an electrical heater. They found that reverse flows would occur for the poor header design and the perforated grid can improve the fluid flow distribution. Karno and Ajib S. [13] summarized various types of flow Non-Uniformity in heat exchangers and discussed the reason leading to flow Non-Uniformity. . Haren and R. Reddy [17] carried out an analysis of the effects of inlet fluid flow non-uniformity on the thermal properties and heat exchange variables. Performance and pressure drop in cross flow plate-fin heat exchangers by using a finite element method. However, few authors studied the fluid flow Non-Uniformity using the computational fluid dynamics (CFD) simulation technique, especially the effects of the configuration of header and distributor on the flow distribution in plate-fin heat exchangers. CFD simulation technique can provide the flexibility to construct computational models that are easily adapted to a wide variety of physical conditions without constructing a large-scale prototype or expensive test rigs. Therefore, CFD can provide an effective platform where various design options can be tested and an optimal design can be determined at a relatively low cost.

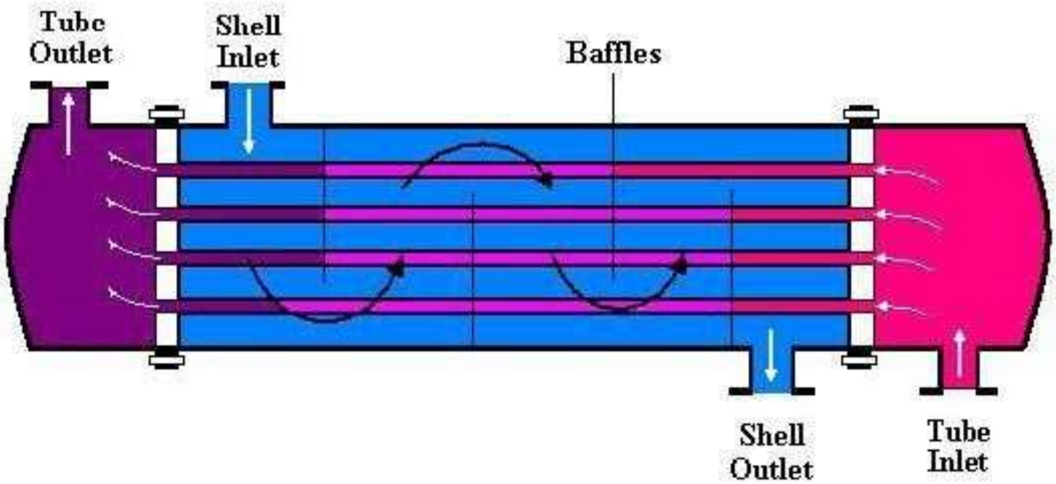


Fig 1.1 Shell-and-tube heat exchanger

Heat exchangers are widely used in petroleum refineries, chemical plants, petrochemical plants, natural gas processing, refrigeration, power plants, air conditioning and space heating. A heat exchanger is a device built for efficient heat transfer from fluid one to another, whether the fluids are separated by a solid wall so that they never mix, or the fluids are directly contacted. One common example of a heat exchanger is the radiator in a car, in which a hot engine-cooling fluid, like antifreeze, transfers heat to air flowing through the radiator.

3. LITERATURE REVIEW

The Flow Non-uniformity with or without back flow is major concerned for literature survey in this chapter which can be broadly classified under three categories.

The first part of the survey deals with the analytical solution for Non-Uniformity in shell and tube heat exchanger. Second part of the survey deals with the experimental and CFD analysis of Non-Uniformity in heat exchanger and third part of the survey deals with the analysis of Non- Uniformity in plate heat exchanger.

Sahoo, R.K., and Wilfried Roetzel., [1] calculated the axial temperature profiles in a shell and tube heat exchanger by numerically for given Non-Uniformity on the tube side. For comparison the same Non-Uniformity are handled with the parabolic and hyperbolic dispersion model with fitted values for the axial dispersion coefficient and third sound wave velocity. The fundamental equations of hyperbolic model is derived 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 Non-Uniformity (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.

Wilfried Roetzel, and Chakkrit Na Ranong.,[2] tested and compared the newer hyperbolic dispersion model and *parabolic model considering the processes with pure Non-Uniformity (without back mixing) on the tube side of a shell and tube heat exchanger and plug flow on shell side . The boundary conditions of the model equations are discussed in detail for the steady state and equations of the axial temperature profiles are provided in the programmable form. For the hyperbolic model simple relationships between the model parameters are derived. Considering the transient adiabatic processes in the tube bundle a concept for the experimental determination of the model parameter M, the third sound Mach number, is developed. Authors concluded that for an overall consideration of a heat exchanger with Non-Uniformity the parabolic model is satisfactory. The parameter Pepar depend on both NTUs of the heat exchanger which makes the model difficult to handle. The advantage of the parabolic model is that only the only one parameter is needed. The hyperbolic model is superior to the parabolic model because it predicts the axial temperature profiles correctly, especially temperature jumps and (positive) slopes.*

Yimin xuan, wilfried roetzel, [3] Applied the dispersion model is to the description of the effects of shell and tube side flow Non-Uniformity. The method allows for effect of Non-Uniformity on transient process, influence of heat capacities of fluids and solid components, arbitrary inlet temperature variations and step disturbances of flow rates. General forms of initial conditions and two different flow arrangements are considered. By means of this model, an efficient and versatile method of predicting transient response of multi pass shell and tube heat exchangers is developed. A general form of the solution for steady-state and dynamic simulation is derived. The Peclet number has been used to quantitatively describe this kind of effect. The calculation has shown that the dispersion model should be applied instead of the plug-flow model ifPe<55.Temperature profiles are determined with numerical inversion of the Laplace transform. Some examples are calculated and the effect of Non-Uniformity is discussed. Flow Non-Uniformity hinders transient responses to any inlet changes and decreases thermal effectiveness of heat exchangers. Its effect becomes more remarkable with increasing NTC'.

Danckwerts, p. v., [4], analyzed when a fluid flows through a vessel at a constant rate, either "piston-flow" or perfect mixing is usually assumed. In practice many systems do not conform to either of these assumptions, so that calculations based on them may be in accurate. It is explained how distribution functions for residence-times can be defined and measured for actual systems.
Open and packed tubes are discussed as systems about which predictions can be made. . The use of the distribution functions is illustrated by showing how they can be used to calculate the efficiencies of reactors and blenders. It is shown how models may be used to predict the distribution of residence-times in large systems.

Žarko Stevanović., Gradimir Ilić., Nenad Radojković., Mića Vukić2, Velimir Stefanović2, Goran Vučković2,.[5] described , a numerical study of three-dimensional fluid flow and heat transfer in a shell and tube model heat exchanger. An iterative procedure for sizing shell- and-tube heat exchangers according to prescribed pressure drop is shown, then the thermo- hydraulic calculation and the geometric optimization for shell and tube heat exchangers on the basis of CFD technique have been carried out. Modeling of shell and tube heat exchangers for design and performance evaluation is now an established technique used in industry. . The baffle and tube bundle was modeled by the 'porous media' concept. Three turbulent models were used for the flow processes. The velocity and temperature distributions as well as the total heat transfer rate were calculated. The calculations were carried out using PHOENICS Version 3.3 code. The effect of different turbulence models on both flow and heat transfer is significant. Thisis due to the introduction effects of eddy-viscosity. It was concluded that Chen-Kim modification of the standard k-ε turbulence model give the best agreement to the experimental data of velocityfield.

Wilfried Roetzel and S. K. DAS., [6] Introduced a new concept of hyperbolic axial dispersion in fluid. This is an extension of the already established method of considering axial dispersion which takes the flow Non-Uniformity into account in the analysis of heat exchangers. The concept is introduced by analogical treatment of the axial dispersion with the fluid conduction. Hyperbolic conduction, which considers finite conduction wave propagation

Velocity is important only in special cases such as cryogenic temperatures or sudden incidence of high heat flux. On the other hand the similar propagation velocity of the dispersion wave appears to be a general phenomenon which affects the thermal performance of heat exchangers even for common applications. Based on the proposed theoretical foundation, the dynamic analysis of a U-type plate heat exchanger is presented for step and sinusoidal change in one of the inlet temperatures. For this purpose the traditional inlet boundary condition for the dispersion model has been extended to incorporate the effect of the finite propagation velocity of the dispersion wave. The method of Laplace transforms has been applied for the analysis, and the Laplace inversion is carried out numerically using fast Fourier transforms. The results indicate that the proposed concept of 'hyperbolic dispersion' can be developed as a powerful tool for the analysis of heat exchangers particularly in the transient regime of operation.

Anindya Roy, and Sarit K. Das., [7] utilized a modern technique based on the "axial dispersion model" has been to simulate the regenerative heat exchanger both in the warm-up and pseudo-steady state operation. The advantage of this model is that it takes all the flow Non- Uniformity and backmixing effects into consideration instead of idealizing the flow to be so called "plug flow". In contrast to previous studies with dispersion, in the present study the dispersion is considered to propagate with a finite propagation velocity following a hyperbolic law which is physically more consistent. The effect of different parameters on the cyclic response has been brought out and the results have been verified by comparing results of a rotaryregenerator. The technique utilized in the present study can act as a tool for modeling regenerators where non- uniformity in flow distribution is significant.

Roetzel, W., Spang, B., Luo, X., and Dash, S.K., [8] Proposed the emerging concept of dispersion of heat along the axial direction as a fluid flows through a passage bounded by solid wall has been presented with its most recent and remarkable advancement. This new proposition takes axial dispersion as a disturbance which propagates as a wave with a finite velocity. It has been proposed that this sound like propagation be named as the "third sound wave in flowing fluid". The fundamental analysis of this theory has been presented with particular emphasis on the boundary condition which plays a key role in the propagation of the wave. A general flux formulation has been used for this purpose. Analysis has also been presented for a two fluid situation. It has been found that the 'subsonic' and 'supersonic' flow with respect to third sound wave behave differently particularly at entry and exit. The theoretical background developed has been substantiated by three examples-one purely theoretical condition, one comparison with numerical analysis and finally application to a complete apparatus.

Zhe Zhang and YanZhong Li., [9] Used a computational fluid dynamics (CFD) program FLUENT has been used to predict the fluid flow distribution in plate-fin heat exchangers. It is found that the flow Non-Uniformity is very serious in the y direction of header for the conventional header used in industry. The results of flow Non-Uniformity are presented for a plate-fin heat exchanger, which is simulated according to the configuration of the plate-fin heat exchanger currently used in industry. The numerical prediction shows a good agreement with experimental measurement. By the investigation, two modified headers with a two-stage- distributing structure are proposed and simulated in this paper. The numerical investigation of the effects of the inlet equivalent diameters for the two-stage structures has been conducted and also compared with experimental measurement. It is verified that the fluid flow distribution in plate-fin heat exchangers is more uniform if the ratios of outlet and inlet equivalent diameters forboth headers are equal.

Xuan, Y., and Roetzel, W., [10] Developed a versatile and efficient method is developed for predicting dynamic performances of parallel and counterflow heat exchangers subject to arbitrary temperature variations and step flow disturbances, including the effect of flow Non- Uniformity and the influence of heat capacities of both fluids, shell wall and tube bank as well asnonzero initial temperatures. Two algorithms of numerical inversion of the Laplace transform areintroduced to determine the final temperature profiles in the realtime domain and some examplesare calculated with nonuniform initial conditions. The accuracy of the proposed method is demonstrated with the calculated results at new steady states. Experiments are carried out on a labor-sized heat exchanger to further examine the feasibility of this method and the comparison between calculated and measured temperature profiles is illustrated and discussed.

Wilfried Roetzel and Frank Balzereit., [11] 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 Peclet numbersis described and used to determine Peclet numbers for different shell-to-baffle clearances, numbers of baffles, and axial plug flow Reynolds numbers. Measurements with water reveal that Peclet numbers from 15 to 160 can occur and axial dispersion cannot be neglected in manycases.

Anil Kumar Dwivedi, and Sarit Kumar Das., [12] presented a predictive model suggest the transient response of plate heat exchangers, subjected to a step flow variation. The work also brings out the effect of the port to channel Non-Uniformity on the performance of plate heat exchangers under the condition of flow variation. The results indicate that flow Non-Uniformity affects the performance of the plate heat exchangers in the transient regime. A wide range of the parametric study has been presented which brings out the effects of NTU and heat capacity rate ratio on the response of the plate heat exchanger, subjected flow perturbation. To verify the presented theoretical model, appropriate experiments have been carried out. Experiments include the responses of the outlet temperatures subjected to inlet temperature transient in the circuit followed by a sudden change in flow rate in one of the fluids. Simulated performance has been compared to the performance measured in the experiments. Comparisons indicate that theoretical model developed for flow transient is capable of predicting the transient performance of the plate heat exchangers satisfactorily, under the given conditions of changed flow rates.

Karno, A., and Ajib, S., [13] prepared a new program for simulation and optimization of the shell-and-tube heat exchangers is to obtain useful results by employment of the computing technology fast and accurately. As an application of this program, the effects of transverse and longitudinal tube pitch in the in-line and staggered tube arrangements on the Nusselt numbers, heat transfer coefficients and thermal performance of the heat exchangers were investigated. The obtained values of the tube pitch were compared with literature values.

3 RESULT

Introduction

This chapter mostly deals with the results of the investigations of the area weighted and mass weighted temperature profiles for without back flow and with back flow in tubular heat exchangers.

Evaluation of mass weighted and area weighted average temperatures

FLUENT evaluates the mass weighted and area weighted temperatures as follows:

Mass weighted average temperature:

The mass-weighted average of a temperature is computed by dividing the summation of the product of the selected temperature and the absolute value of the dot product of the facet area and momentum vectors by the summation of the absolute value of the dot product of the facet area and momentum vectors (surface mass flux):

The standard k–ε model is adopted because it can provide improved predictions of near-wall flows and flows with high streamline curvature. The governing equations of the flow for mass, momentum, energy conservation and for k and ε as discussed earlier.

The fluid mass weighted temperature was calculated at any position of the calculation domain.

$$T_{ad} = \frac{\int T\rho\vec{u}\cdot d\vec{A}}{\int \rho\vec{u}\cdot d\vec{A}}$$

$$T_{ad} = \frac{\sum_{i=1}^{n} T_i \rho_i \vec{u}_i \cdot \vec{A}_i}{\sum_{i=1}^{n} \rho_i \vec{u}_i \cdot \vec{A}_i}$$

Area weighted average temperature:

The area-weighted average of a temperature is computed by dividing the summation of the product of the selected temperature and facet area by the total area of the surface:

$$T_m = \frac{1}{A}\int T\,dA$$

$$T_m = \frac{1}{A}\sum_{i=1}^{n} T_i |A_i|$$

single-pass shell-and-tube exchanger with four channels

Case1: with pure axial plug flow on the shell side and Non-Uniformity without Back flow on the tube side:

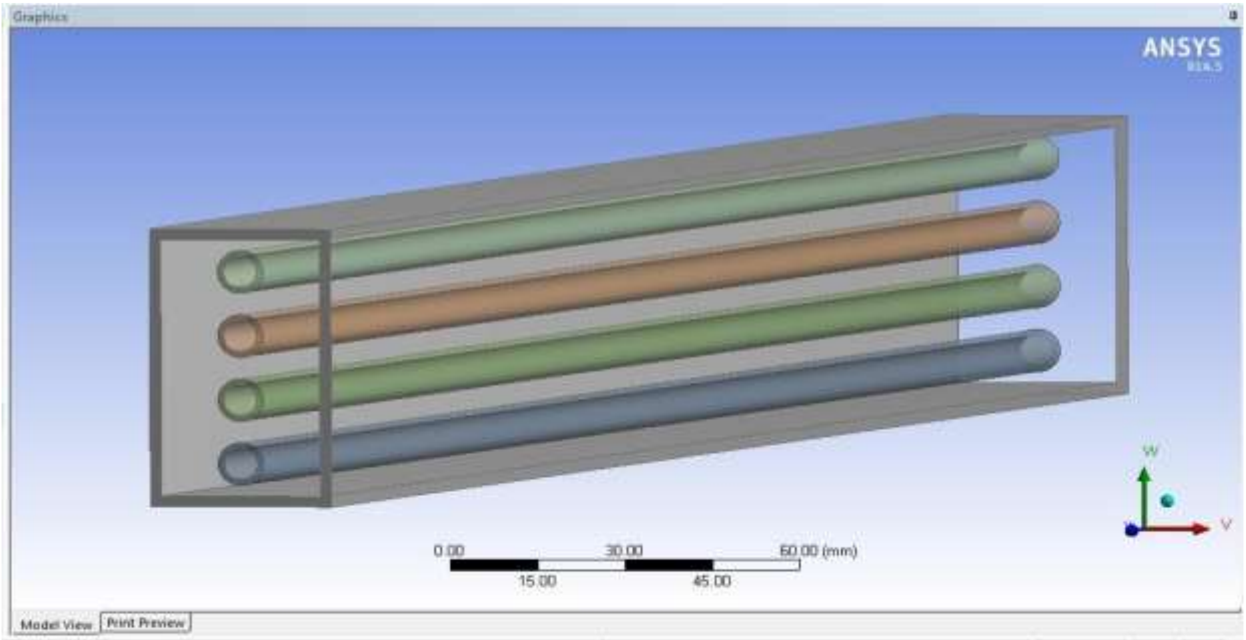


Fig: 6.1: Computational Geometry for four tube in-line arrangement

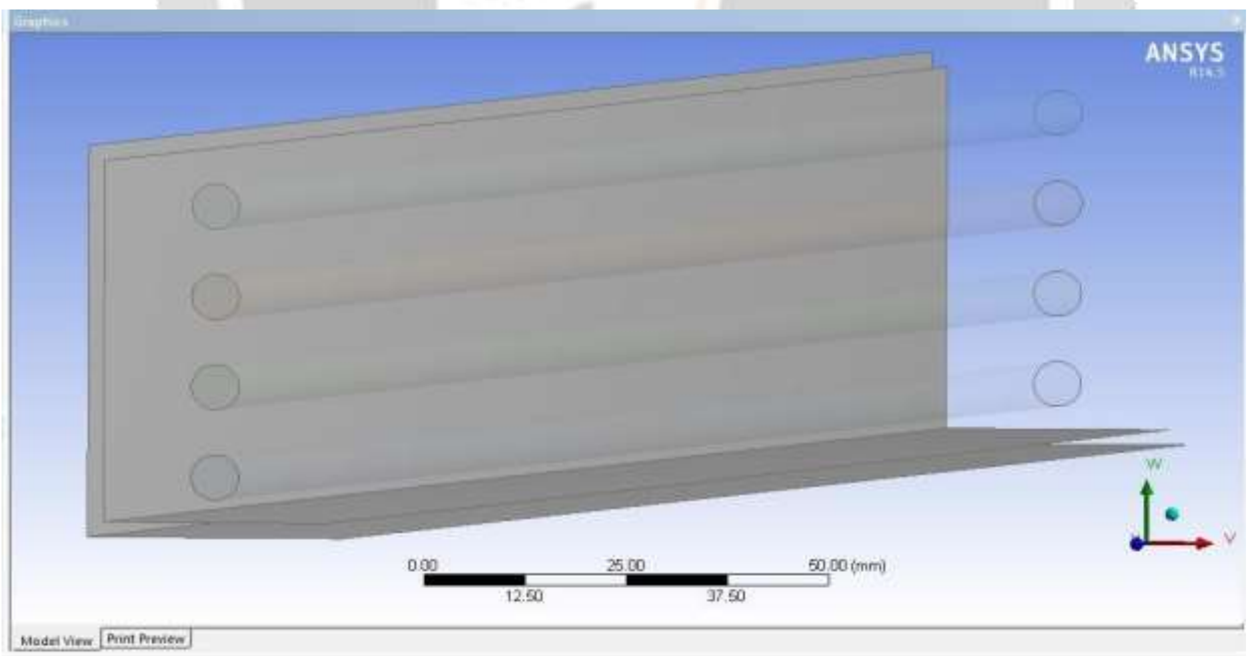
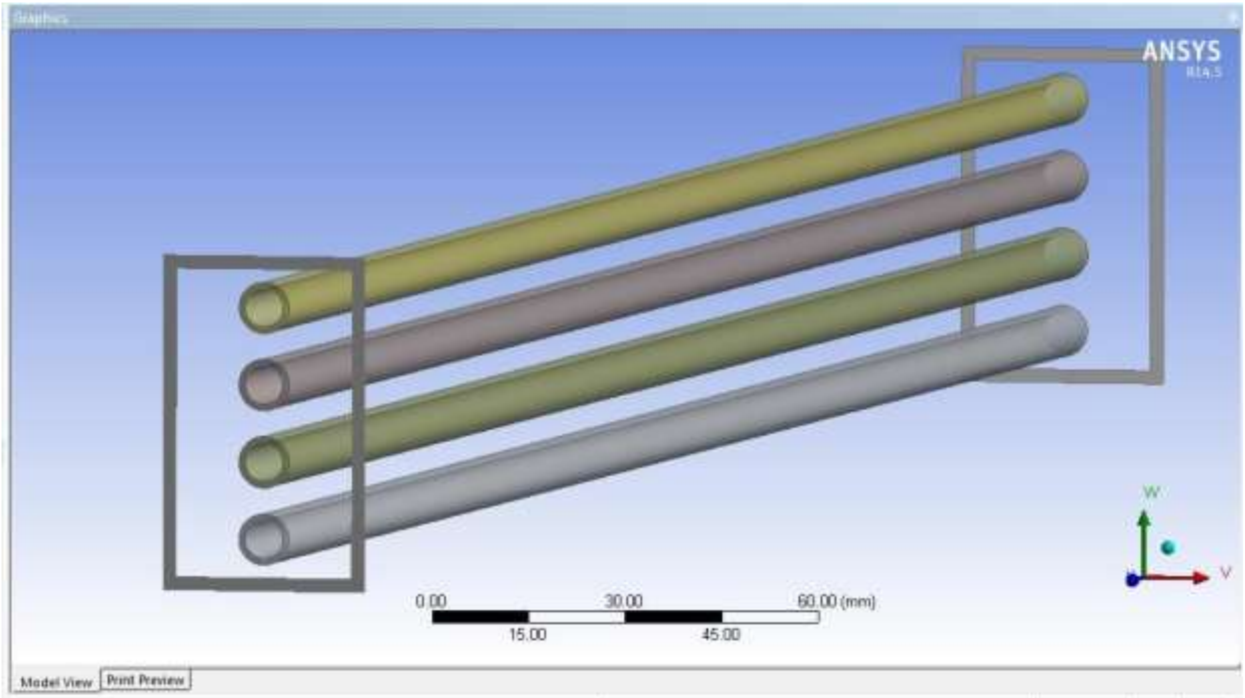


Fig: 6.2: Computational Geometry of four tube in-line arrangement



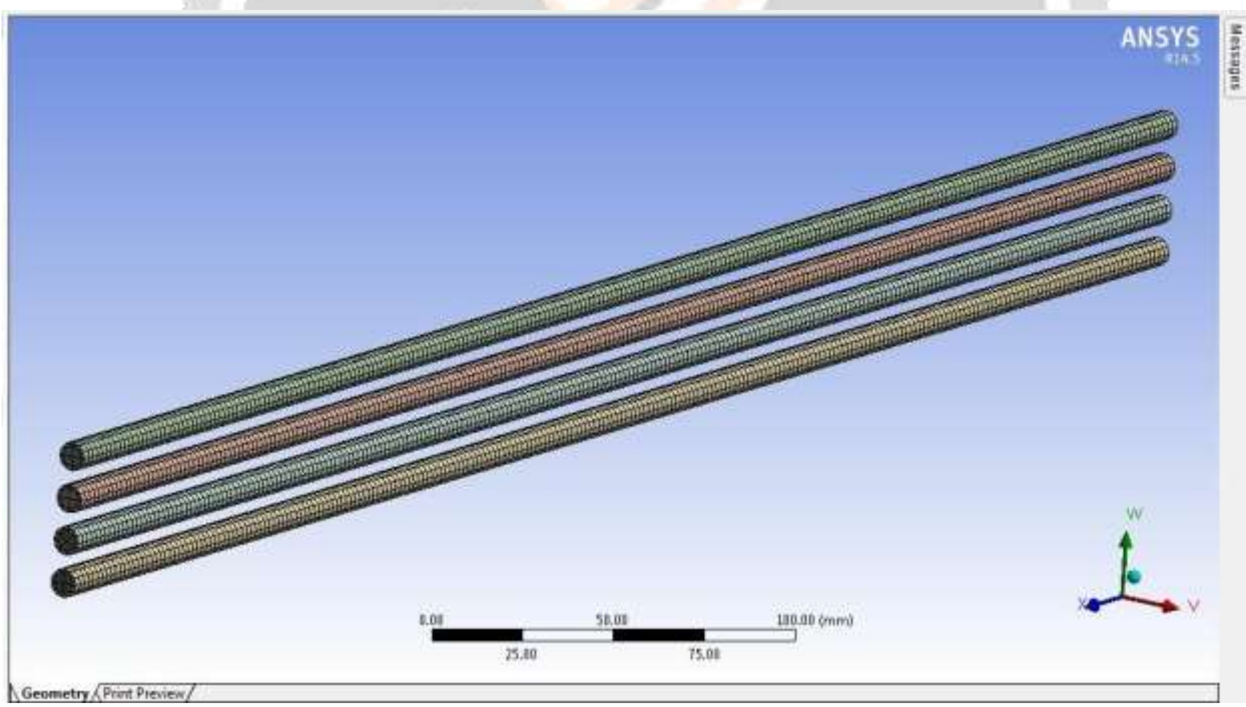


Fig: 6.3: Computational Geometry of four tube in-line arrangement

Fig:6.4 : Meshing of four tube in-line arrangement

Dimensions of the exchanger:

Length of the exchanger (L)	*560 mm*
Width of the exchanger (b)	*32 mm*
Height of the exchanger (h)	*52 mm*
Thickness of Shell wall	*02 mm*
Tube Outer diameter (D)	*08 mm*
Tube Inner diameter (d)	*06 mm*

Boundary conditions:

Channel Number	*Channel 1*	*Channel 2*	*Channel 3*	*Channel 4*
Mass Flow Rate in (Kg/sec)	*3.78*	*2.52*	*1.26*	*0*

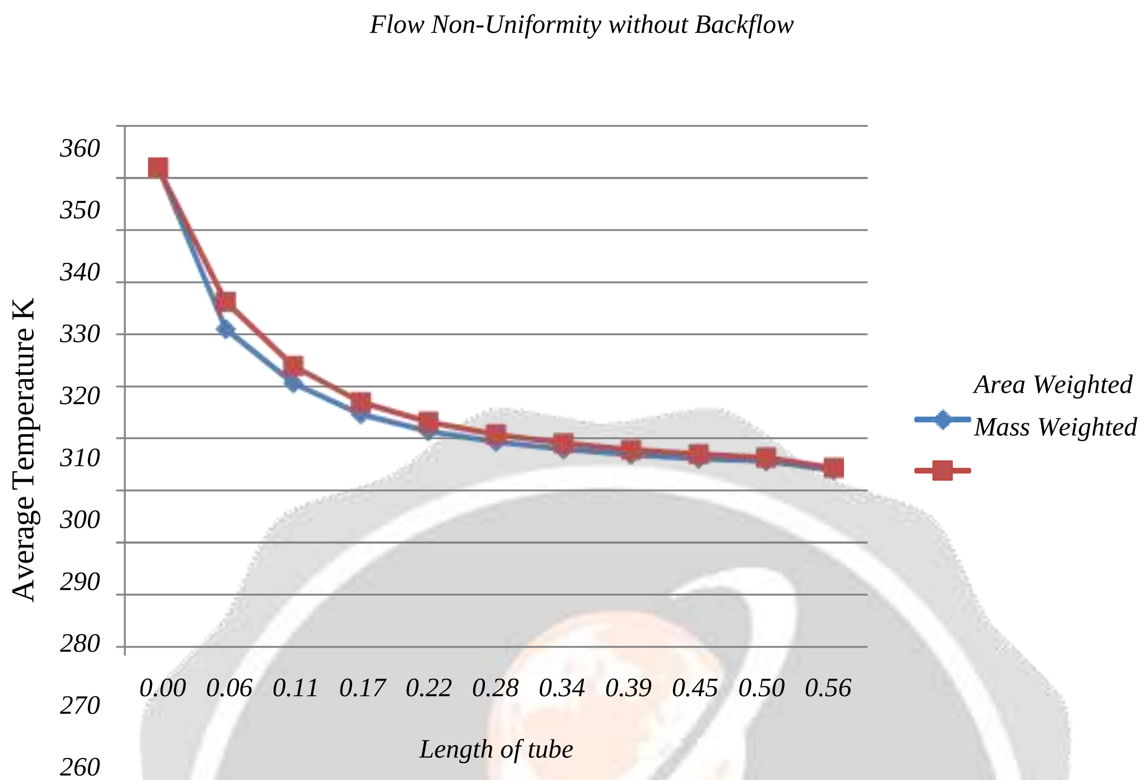


Fig. 6.6: Average temperature profiles for Non-Uniformity without back flow for Four tube in-line arrangement

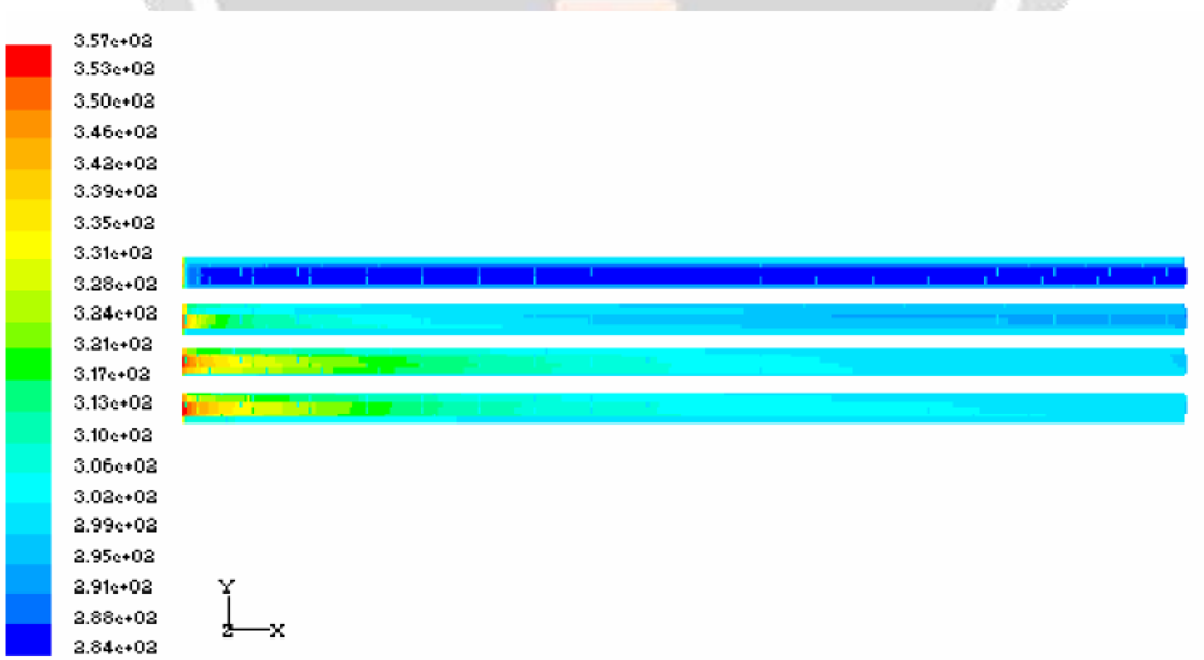


Fig. 6.7: Temperature contours for Non-Uniformity without back flow for four tube Inline arrangement

6.2.2 Case2: The flow velocity is the same for all the channels, i.e. tube side plug flow. The constant mass flow rate, 2.52 Kg/sec is given in each of the four channels. In this case both area weighted average and mass weighted average temperatures are identical which are listed in the table (6.6).

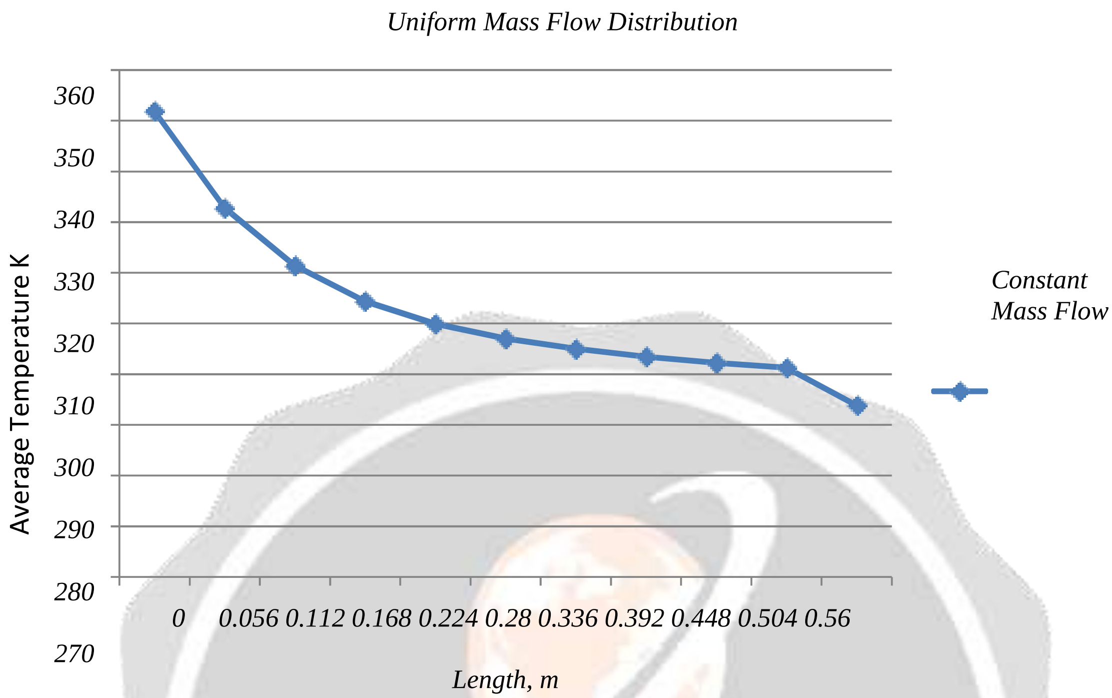
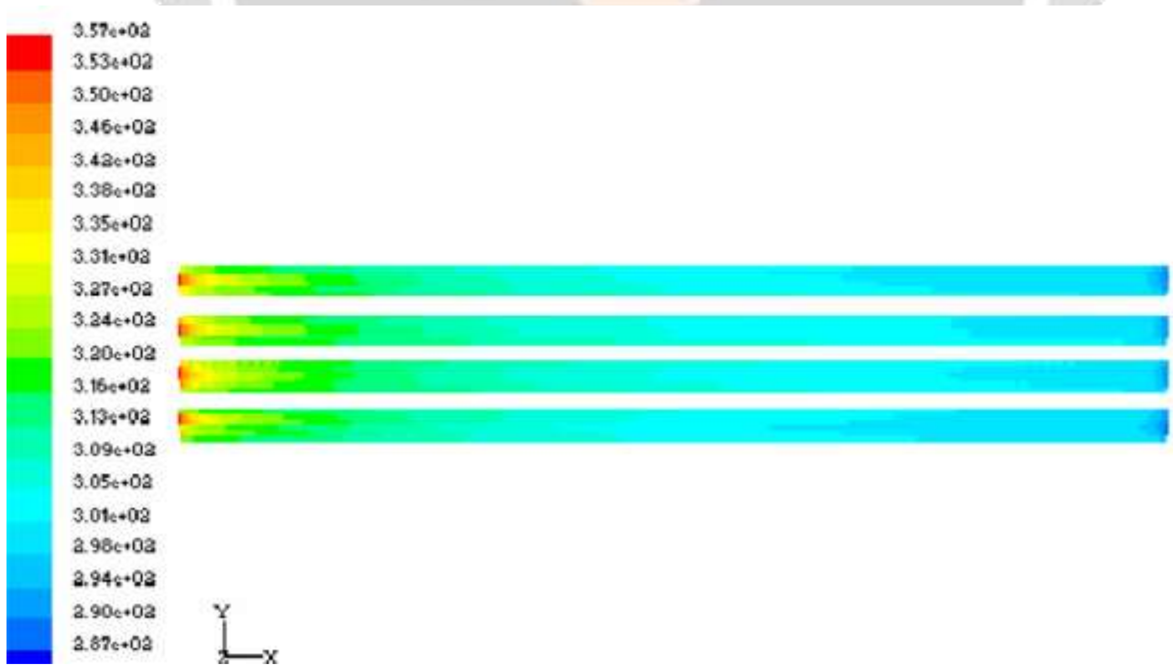


Fig: 6.8: Average temperature profile for uniform mass flow distribution for four tube in-line



arrangement

Fig.6.9: Temperature contours for uniform mass flow distribution for four tube in- line arrangement

Table 6.1: Average temperatures for Non-Uniformity without back flow and Uniform mass flow distribution

Length	Area weighted average temperature for without back flow (K)	Mass weighted average temperature for without back flow(K)	temperature for Uniform form mass flow distribution (K)
0	351.84171	351.84171	351.84171
0.056	321.01426	325.87114	332.92410
0.112	309.93627	313.72236	321.42662
0.168	304.25413	306.96709	314.21540
0.224	300.80452	303.20198	309.92462
0.28	300.63951	300.81297	307.02865
0.336	297.82236	299.15826	305.21971
0.392	296.98425	297.95812	303.62893
0.448	296.13891	297.21227	302.62719
0.504	295.71336	296.41318	301.82112
0.560	293.96291	294.74133	293.91846

6.3. single-pass shell-and-tube heat exchanger with square in-line tube Arrangement

Case1: with pure axial plug flow on the shell side and Non-Uniformity without back flow on the tube side:

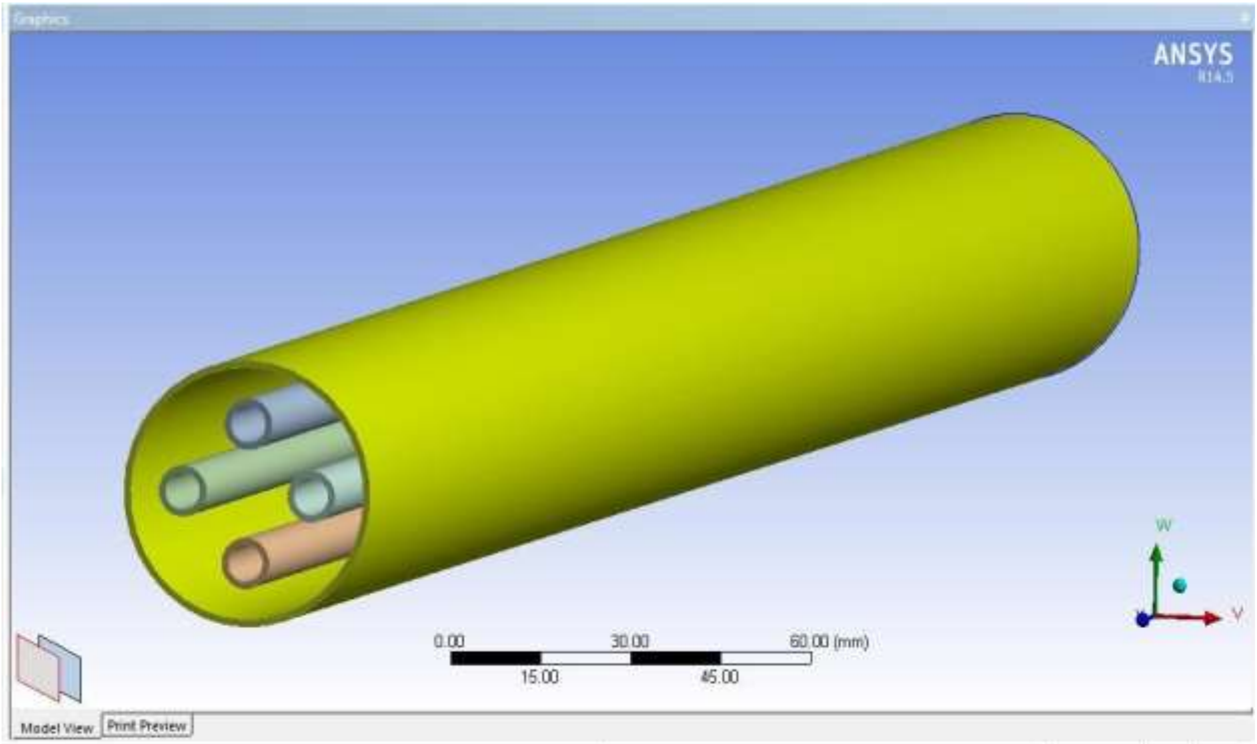


Fig: 6.11. Computational domain for square inline-tube arrangement

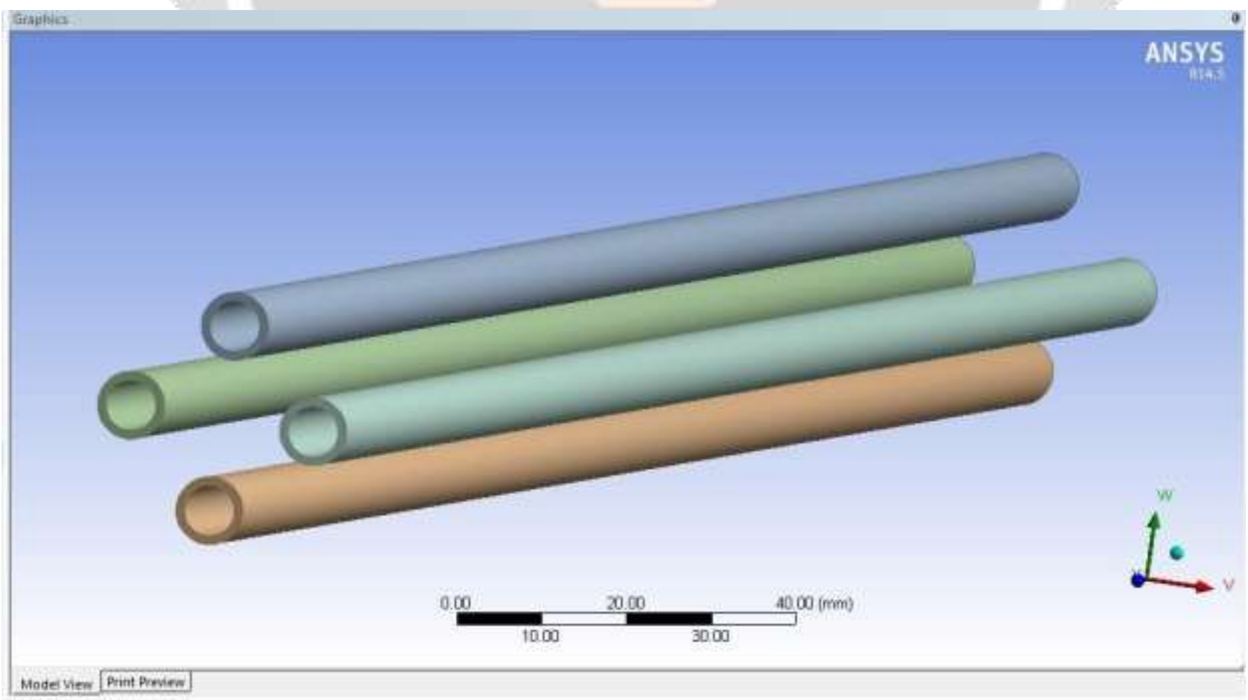


Fig: 6.12. Computational domain for square inline-tube arrangement

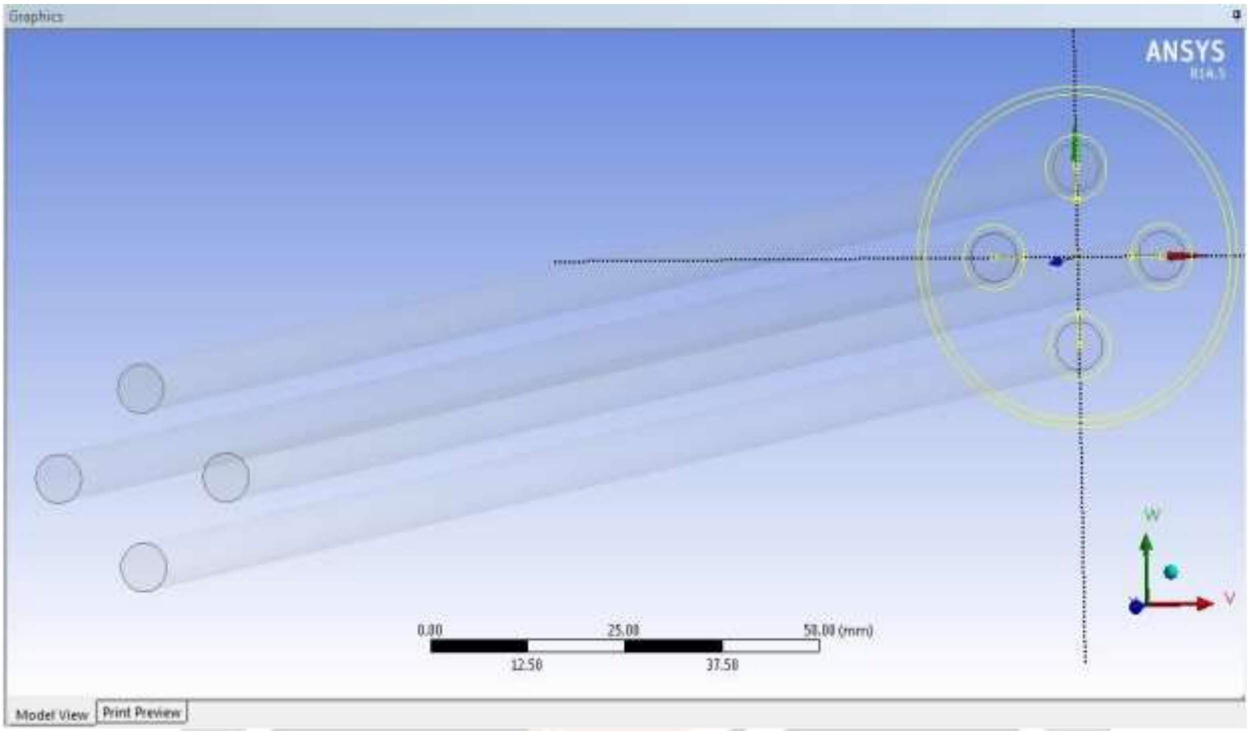


Fig:6.13 . Computational domain for square inline-tube arrangement

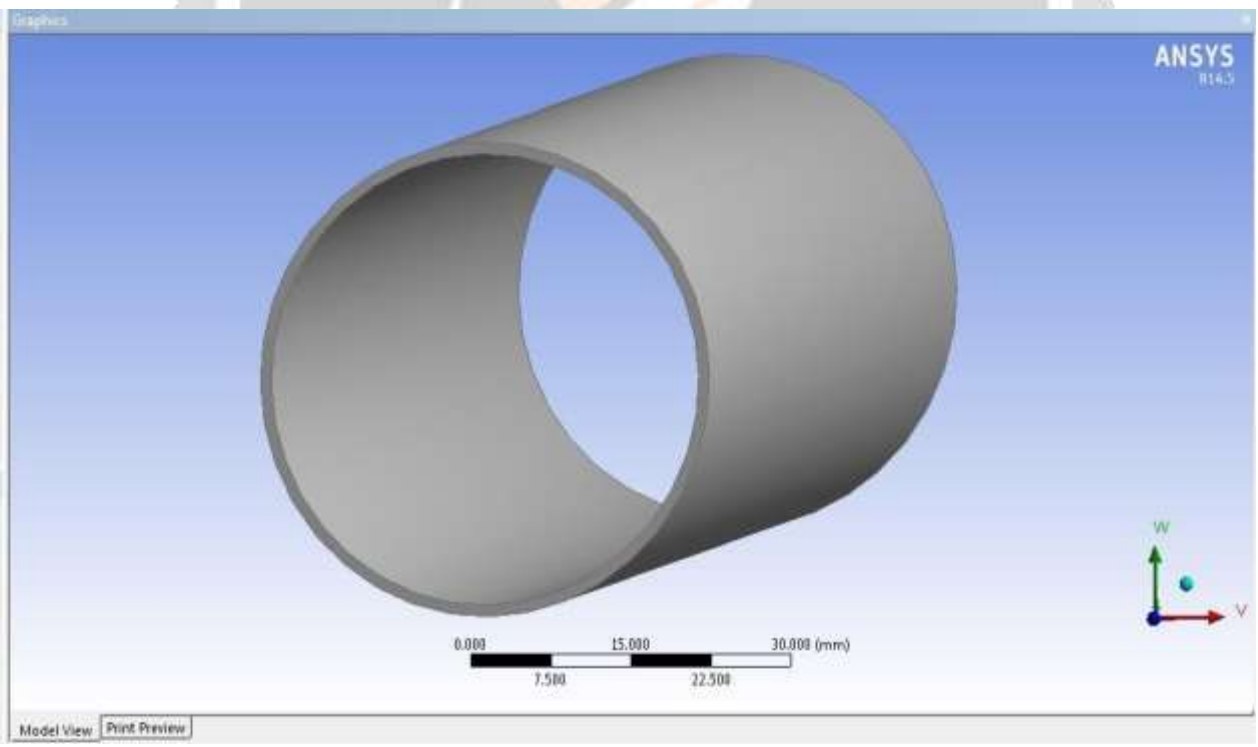


Fig:6.14 . Computational domain of Shell for square inline-tube arrangement

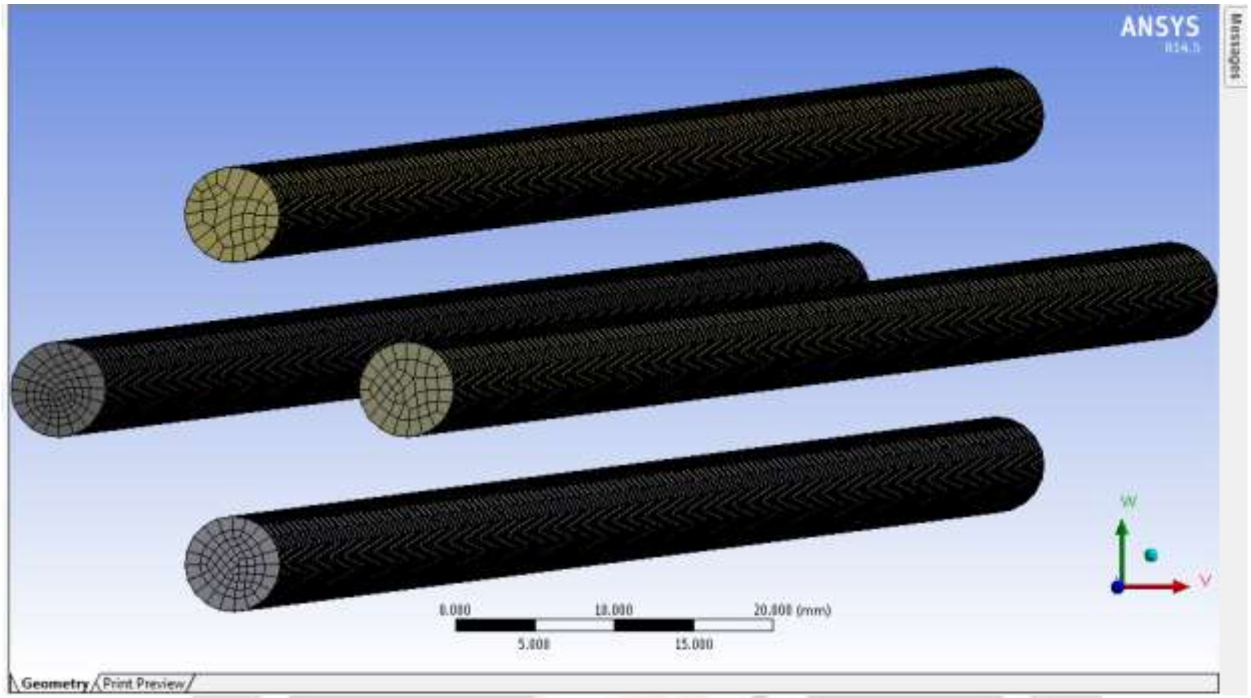


Fig6.15 Meshing of Four Square Inline Tube Arrangement

Dimensions of the computation domain:

Length of the exchanger (L)	*560 mm*
Shell Outer diameter (Ds)	*42 mm*
Shell Inner Diameter (ds)	*38 mm*
Tube Outer diameter (D)	*8 mm*
Tube Inner Diameter (d)	*6 mm*

Boundary conditions:

Channel Number	*Channel 1*	*Channel 2*	*Channel 3*	*Channel 4*
Mass Flow Rate in (Kg/sec)	*3.78*	*2.52*	*1.26*	*0*

Table 6.2: Average temperatures for Non-Uniformity without back flow in Square inline arrangement

Length (m)	Area weighted average temperature for without back flow (K)	Mass weighted average temperature for without back flow(K)	Temperature for uniform form mass flow distribution (K)
0	353.42662	353.42662	353.42662
0.056	317.96126	321.92419	328.72231
0.112	309.41316	313.15286	320.21989
0.168	303.87122	305.19846	313.21318
0.224	301.74318	302.91846	308.26289
0.28	300.21541	301.92416	305.72189
0.336	299.95814	300.41318	304.82112
0.392	299.98436	299.96291	302.80452
0.448	298.13891	298.98891	301.96236
0.504	298.84123	298.25362	300.81452
0.56	295.98429	295.58126	294.98149

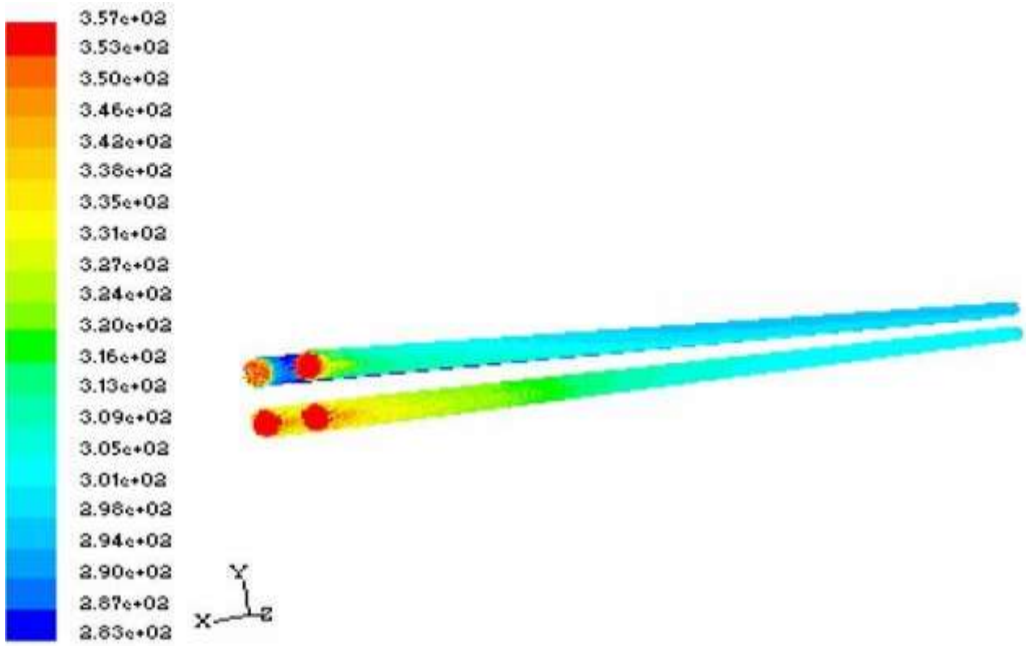


Fig: 6.16. Temperature contours for Non-Uniformity without back flow for Square in-line tube arrangement

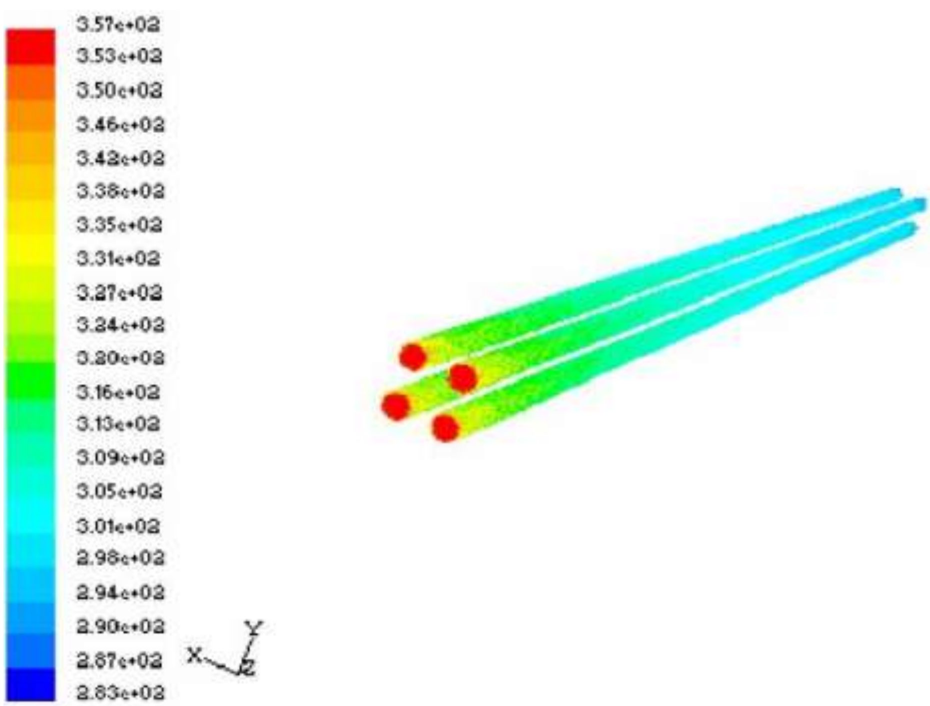


Fig: 6.18. Temperature contours for uniform mass flow distribution for Square in-line tube arrangement

Single-pass shell-and-tube heat exchanger with four tube inline arrangement and square inline tube arrangement:

Tube side Non-Uniformity without back flow and shell side ideal plug flow in four tube inline Arrangement:

In this arrangement mass weighted average temperature and area weighted temperature have same value at the position L=0, Fig.6.6. The area weighted temperature profile falls below the mass weighted temperature profile along the length of the exchanger. But in the case of uniform mass flow distribution both average temperature profiles are identical along the exchanger. In the comparison between the linear velocities distribution of mass flow and Uniform mass flow rate the slope of the area weighted temperature and Mass weighted temperature maintains a constant difference throughout the length of the exchanger.

Tube side Non-Uniformity without back flow and shell side ideal plug flow in four tube Square inline Arrangement:

In this arrangement mass weighted average temperature and area weighted temperature have same value at the position L=0, Fig.6.17. The area weighted temperature profile falls below nearly equal to the mass weighted temperature profile along the length of the exchanger. But in the case of uniform mass flow distribution both average temperature profiles are identical along the exchanger. In the comparison between the linear velocities distribution of mass flow and Uniform mass flow rate the slope of the area weighted temperature and Mass weighted temperature maintains a constant difference for half the length of exchanger and converges to common values at the end of length of the exchanger.

5. CONCLUSIONS

1. *Analysis and Validation done numerical using Computational Fluid Dynamics and it is observed that due Non-Uniformity without back flow on tube side and ideal plug flow on shell side, the area weighted temperature profile falls below the mass weighted temperature profile. When we compared the Non-Uniformity without back flow with an ideal uniform mass flow distribution, the temperature profiles of Non-Uniformity withoutback flow falls below the temperature profiles of uniform mass flow distribution.*

2. *The effect of Flow Non-Uniformity is similar in four tube in-line arrangement and square inline tube arrangement for same mass flow rate and equal dimensions of the exchanger.*

Stress Distribution in the Gasket for Three Cases:

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