CFD ANALYSIS OF SHELL AND TUBE HEAT EXCHANGER WITH SEGMENTAL BAFFLES

Aman Mishra

CFD ANALYSIS OF SHELL AND TUBE HEAT EXCHANGER
WITH SEGMENTAL BAFFLES

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by

Aman Mishra
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Under the guidance of

Dr. A. K. Satapathy

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NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA

CERTIFICATE

This is to certify that the thesis entitled **CFD analysis of shell and tube heat exchanger with segmental baffles,** submitted by **Aman Mishra** to National Institute of Technology, Rourkela, is an authentic record of bona fide research work carried under my supervision and I consider it worthy of consideration for the award of the degree of Master's of Technology of the Institute.

Date :

Dr. A. K. Satapathy
Associate Professor
Department of Mechanical Engineering
National Institute of Technology
Rourkela, 769008

DECLARATION

I certify that

1. The work contained in the thesis is original and has been done by myself under the general supervision of my supervisor.

2. The work has not been submitted to any other Institute for any degree or diploma.

3. I have followed the guidelines provided by the Institute in writing the thesis.

4. Whenever I have used materials (data, theoretical analysis, and text) from other sources, I have given due credit to them by citing them in the text of the thesis and giving their details in the references.

Aman Mishra

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I express my gratitude towards my supervisor Dr. A K Satapathy. He always listened to every little problem and suggested the remedy for it. His technical knowledge also the editorial advice helped me a lot in completing this thesis. I offer my sincere appreciation for all the support, patience and learning opportunities provided by him.

This project would not have been completed without all the help from my lab-mates. Any small technical issues discussed together were surely helpful for improving this thesis.

Date :

Place :

Aman Mishra

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LIST OF SYMBOLS AND ABBREVIATIONS

$C_{1\varepsilon}, C_{2\varepsilon}, C_{\mu}$	Turbulence model constants
L	Length of shell
D	Diameter of shell
d_s	Inlet and outlet diameter of shell
d_i	Internal diameter of tubes
t_s	Thickness of baffles
t	Thickness of tube
E	Total energy
k	turbulent kinetic energy
Re	Reynolds number,
p	pressure
T	temperature
u_i, u_j, u_k	dimensional velocities in x,y and z-directions respectively
Nu	Surface Nusselt number

Greek symbols

ρ	Density
ε	rate of dissipation of turbulent kinetic energy
μ	Dynamic viscosity

Abstract

Numerical investigation has been done for single pass shell and tube heat exchanger with segmental baffles. Heat transfer and flow pattern are numerically studied by varying number of baffles. Standard k-ε model is used for solving the above problem. Numerical simulation has been done for three different cases of baffle spacing with square bundle of tubes. Numerical solutions are obtained by solving 3D continuity, momentum, energy and turbulence (k-ε) equations using commercial solver CFD. To analyze the phenomenon, number of baffles is taken as varying parameter; other parameters like velocity, temperature, pressure and baffle cut are kept constant. Results obtained from numerical solution are analyzed extensively to get the effect of number of baffles on heat transfer rate and pressure drop on shell side.

Index Terms— shell and tube heat exchanger, segmental baffle, k-ε model.

CHAPTER 1

INTRODUCTION

1.1 Heat Exchanger

Heat exchangers are the apparatus which is widely used in various industries. Heat exchangers are used to transmit heat among two or more fluid streams. One can comprehend their application that any course of action which involve cooling, heating, condensation, boiling or evaporation will need a heat exchanger for this purpose. Fluids, usually are heated or cooled before the procedure or undergo a phase change. Various heat exchanger have taken the name according to the work they performed i.e. HE used to boil the fluid is called as boiler, HE used to evaporate the fluid is known as evaporator, HE which is used for condensing any fluid is known as condenser. Heat transfer and pressure drop are the measuring parameter for performance and efficiency calculation. It is better to calculate overall heat transfer coefficient, pressure drop and heat transfer area to represent the efficiency. It helps in calculation of size of HE required and its running cost etc. generally, there is plenty of literature, theories and papers to design a heat exchanger according to the necessities.

1.2 Types of Heat Exchanger

On the basis of nature of heat exchange procedure:

1. Direct contact heat exchanger: In this HE, fluids are mixed with each other and heat transfer will take place.

2. Indirect contact heat exchanger: In this HE, fluids are separated by a thin wall and heat transfer takes place by convection and conduction.

Indirect contact HE has two types:

(a) Regenerator type: In this HE, hot and cold fluid pass alternately through a space which contains solid matrix. When hot fluid pass through matrix then it heats the matrix and after that cold fluid pass through it then hot matrix transfer heat to the cold fluid.

(b) Recuperator type: It is most useful HE in which two or more fluid pass simultaneously but they are not allowed to be mixed. in this type of HE, heat transfer takes place in both side of dividing wall(pipes or tubes).

On the basis of relative direction of fluid motion:

i) Parallel flow heat exchanger: In this HE, both fluid flowing in the same direction.

ii) Counter flow heat exchanger: In this HE, both fluid flowing the opposite direction.

iii) Cross flow heat exchanger: In this HE, one fluid flows at some angle to the other fluid.

1.3 Applications of Heat Exchanger

1. IC engines and Automobile radiators.
2. Open hearth and glass melting furnaces.
3. Air heater and blast furnaces.
4. Oil coolers, Intercoolers, Air preheaters, Economizers, Super heaters, Condensers, and surface feed heaters of steam power plants.
5. Milk chiller and Pasteurizing Plant.
6. Evaporator of Ice plant etc.

1.4 Shell and Tube Heat Exchanger

For high pressure application, there is a heat exchanger called shell and tube heat exchanger is used. It is an indirect contact type heat exchanger under the subdivision of recuperator. It is a most versatile heat exchanger. By some modification, it can be act as a parallel flow, counter flow and cross flow as well. Shell and tube heat exchanger consists of a shell, tubes and headers. Tubes are generally made of cylindrical shape with circular cross section whereas shell has a wide variety according to the requirement. One fluid flows through the tubes and the other fluid pass through the shell. Heat exchanges between these two fluids are by convection-conduction- convection mode. Shell and tube heat exchanger has high value of log mean temperature difference correction factor.

A simple shell and tube heat exchanger can work as a parallel flow and counter flow heat exchanger but to make it a cross flow heat exchanger, some modification is to be done. Introducing baffles in the shell side is one of the modification.

1.5 Need of Baffles

Baffles are provided in the shell side.

1. To give structural rigidity.
2. To prevent vibration.
3. To prevent the tubes from sagging.
4. To divert the flow of shell side fluid.
5. Baffles are used to obtain higher coefficient of heat transfer.

1.6 Types of Baffles

There are various types of Baffles used in the industries

1.Plate Baffle

2. Helical Baffle
3. Segmental Baffle
4. Ladder type fold Baffle etc.

Spacing between the baffles affecting the heat transfer coefficient and heat transfer rate also. Baffle spacing the distance between two adjacent baffles. Segmental and plate Baffles are provided with cut called baffle cut to passing the fluid. During the fluid flow in shell side, recirculation and eddies are formed near the baffles. Other Baffles are expensive and difficult in manufacturing. Now a day researchers are trying to optimize the segmental baffles for better heat transfer and effectiveness.

1.7 Aims and Objective

The main objectives of this project are: Design and simulation of shell and tube heat exchanger with segmental baffle. Study the effect of baffle spacing on heat transfer and pressure drop.

1.8 Importance of the present work

Shell and tube heat exchanger with segmental baffles are easy to manufacture and less expensive as compare to other baffles but it has less effectiveness among its category. After knowing the effect of baffle spacing on heat transfer and pressure drop, it is helpful to optimize the heat exchanger and get higher effectiveness and heat transfer at low cost.

1.9 Outline of present work

Chapter 1 postulates the brief introduction of Heat exchanger and its types, need of baffles and objective of the present work.

Chapter 2 describes the previous work done in the field of development of shell and tube heat exchanger.

Chapter 3 shows the identification of problem and its formulation with boundary condition.

Chapter 4 deals with introduction of CFD and its elements. it gives over view of fluent package and knowledge of CFD procedures and solve the problem numerically.

Chapter 5 deals with the results obtained from the CFD solver along with some discussion about it.

Chapter 6 gives the conclusion of the entire project and future scope for further work.

Chapter 7 gives the details of references used for this project work.

CHAPTER 2

LITERATURE SURVEY

A considerable amount of work has been done by G.V. Srinivasa Rao et al [2]in the field of analysis of shell and tube heat exchanger. They evaluated that a shell and tube heat exchanger with square pitch bundle arrangement. They have taken the temperature as input parameter and observed the behavior of heat transfer coefficient, coefficient of friction and pressure drop for different combination of fluids by the use of C coding. They got to know that Nu is increasing with increase in Re in tube side. For SO2 -Steam combination, curve is steeper as compared to CO2 - Steam combination. They also found that friction factor decreases with increase in Re in tube side. The decrement in friction factor for both fluid combination i.e. (CO2 - steam and SO2 - steam) are almost same.

A research has been done by Yonghua You et al [3] in the field of performance improvement of shell and tube heat exchanger. They used trefoil hole baffles to make the heat exchanger cross flow type. By introducing baffles, heat transfer area increased so heat transfer also increases. The effect of baffle distance is also investigated by them. They validated their numerical solution with experimental data. They used 12.8 mm internal diameter stainless steel tubes, low carbon steel baffles aligned in staggered manner. They found that pressure loss in shell side increases with increase in shell side Reynolds number. They also found that heat transfer per unit area initially decreases with increase in shell side Reynolds number but after some value it starts increasing with increase in Reynolds number. Convective heat transfer coefficient increases with increase in shell side Reynolds number. With decrease in number of baffles, heat transfer coefficient and heat flux decreases. They got to know that when number of baffles is decreased from 6 to 3 then pressure loss is decreased by 35%. They found that overall thermo hydraulic performance is decreasing with increase in number of baffles.

A research has been -Yan Zhou et al [4] to study the flow characteristics in shell and tube heat exchanger with trefoil hole baffle. A numerical investigation on shell side flow and heat transfer based on RNG k-ε model has been done. They used 12.2 mm internal diameter tubes which are arranged in rotational regular triangle pattern. They validated their results with experimental data. They found that fluid flow on shell side of shell and tube heat exchanger is periodic due to the structural characteristics. The fluid gets accelerated and jet and swirl flow generated near the baffles because of decrement in area of the flow. Gradual decrement of velocity in radial direction takes place which leads to decrement in average heat transfer coefficient. They found that pressure drop and coefficient of heat transfer vary periodically along the longitudinal direction. Secondary flow is developed both sides of the baffles which decreases the thickness of boundary layer and increase the heat transfer.

A research has been done by Jie Yang et al [5] to investigate the novel shell and tube heat exchanger with plate baffle. They found that Nusselt number increases with increase in Reynolds number for both shell and tube heat exchanger i.e. with rod baffle and with plate baffle but Nu of shell and tube heat exchanger with plate baffle is more than that of shell and tube heat exchanger with rod baffle. They also found that the pressure drop in shell side increases with increase in Re but pressure drop is more for shell and tube heat exchanger with plate baffle when comparing to shell and tube heat exchanger with rod baffle. They also got to know that outlet temperature and pressure gradient of shell side cold fluid is more in shell and tube heat exchanger with plate baffle as compare to shell and tube heat exchanger with rod baffle.

A research has been done by Jian Wen et al [6] to compare the shell and tube heat exchanger with two different baffles i.e. plain helical baffle and ladder type fold baffle. They studied and analyzed the performance of the shell and tube heat exchanger with these two baffles. The operating conditions were same. They found that in plain helical baffle heat exchanger, there are triangular leakage zones which caused loss of heat and reduction in effectiveness of the heat exchanger. This leakage can be eliminated by using ladder type fold baffles. They got to know that shell side heat transfer coefficient and the overall heat transfer coefficient both are increasing with increase in volume flow rate but these two coefficients are more in case of ladder type fold baffle as compare to plain helical baffle. They also found that pressure drop is greater in case of shell and tube heat exchanger with ladder type fold baffle.

Ender Ozden et al [7] investigated the dependencies of coefficient of heat transfer, pressure drop on baffle spacing, baffle cut and shell diameter. They performed the simulation for single shell single pass shell and tube heat exchanger. They found that with decrease in baffle spacing, more cross flow and heat transfer area achieved. When more spacing is provided then after striking to baffle, flow direction changed and some recirculation zone is developed so area behind the baffles are not so effective in heat transfer process but if baffle spacing is less then after changing the direction of flow fluid again strikes the back face of previous baffles so effective heat transfer area increased. So they observed that with increase in number of baffles more heat transfer occurs.

Avinash D Jadhav et al [8] numerically investigated the dependency of heat transfer coefficient and pressure drop on baffle spacing and baffle cut. They compared their results with the Bell-Delaware method results. They performed simulation for two values of baffle cut and different varying flow rate.

Chetan Namdeo Patil et al [9] numerically investigated the effect of baffle cut on heat transfer coefficient and pressure drop with constant baffle spacing. They found that for 30% baffle cut pressure drop is less and heat transfer coefficient is almost same for 30% and 25% baffle cut.

Su Pon Chit et al[10] investigated the effect of baffle spacing on heat transfer and pressure drop. They postulates that pressure drop increases at a faster rate than heat transfer coefficient when baffle spacing decreases. Optimum baffle spacing is 0.4 to 0.6 time of shell diameter.

B. Peng et al [11] investigated the effect of different baffles on heat transfer coefficient keeping the pressure drop constant. They have found that for same pressure drop, shell and tube heat exchanger with segmental baffle has lower heat transfer coefficient as compared to shell and tube heat exchanger with helical baffles. They have also found the correlation between Nusselt number and Reynolds number as well as friction factor and Reynolds number.

Edward S. Gaddis et al [12]proposed a procedure to find the pressure drop. Their equation consists of correlation factor which is influenced by leakage and bypass stream. They have validated their equation by comparing their results with experimental data.

H.D. Li et al [13] obtained the local heat transfer coefficient by the use of mass transfer measurement. Mass transfer coefficient has been transformed by the use of an analogy between heat and mass transfer. Abdur Rahim et al investigated the effect of baffle inclination on various parameters of shell side fluid flow and heat transfer keeping the baffle cut constant. They found that 4% decrease in pressure drop in case of 10^{o} baffle inclination and 26% decrease in pressure drop in case of 20^{o} baffle inclination.

Kiran K et al [14]simulated the shell and tube heat exchanger with baffles and investigated the effect of baffle spacing on heat transfer, pressure drop and outlet temperature of shell side. They postulated that outlet temperature does not affect significantly with baffle spacing whereas it has considerably changed with mass flow rate. They also found that the pressure drop changes appreciately with mass flow rate and baffle spacing.

Prasanna J et al [15] analyzed the hydrodynamic and heat transfer effect on shell and tube heat exchanger with different baffle cut and spacing. They found that for 25% baffle cut, results are slightly better. As baffle spacing decreases, the heat transfer is improved.

CHAPTER 3

PROBLEM IDENTIFICATION

3.1 Introduction

A shell of diameter (D) equals to 80 mm is taken as model which has inlet and outlet of circular cross section of diameter (ds) equals to 20 mm. The length of the shell is taken as 400 mm. There are 4 tubes having square bundle with internal diameter (di) 16mm and thickness 2mm. Shell consists of 8 segmental baffles which are equispaced. But, in the analysis, 3 different numbers of baffles are used keeping other parameters and boundary conditions constant, to observe the effect of number of baffles on heat transfer and pressure drop of shell. Constant fluid properties are assumed. After the creation of three geometric models, each model is analyzed with constant velocity and temperature inlet condition. The flow condition is taken as turbulent. Table 3.1 shows the boundary conditions of the problem.

3.2 Governing Differential Equation

Continuity Equation:

Table 3.1: Boundary conditions for the model

Fluid	Inlet Velocity	Inlet Temperature	Outlet
Shell Side	1.2 m/s	363 K	Pressure Outlet
Tube Side	0.4 m/s	300 K	Pressure Outlet

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0$$

Momentum equation:

$$\frac{\partial}{\partial t}(\rho u_i u_j) = -\frac{\partial \rho}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_i}$$

where

$$\tau_{ij} = \mu \left(\frac{\partial u_i}{\partial x_i} + \frac{\partial u_j}{\partial x_j} \right) - \frac{2}{3} \frac{\mu \partial u_k}{\partial x_k} \partial_{ij}$$

Energy equation:

$$\frac{\partial}{\partial t}(\rho E) + \frac{\partial}{\partial x_i}(u_i(\rho E + \rho)) = \frac{\partial}{\partial x_i} \left(k \frac{\partial T}{\partial x_i} \right)$$

where E is the total energy and k is the thermal conductivity

Turbulent kinetic energy equation:

$$\frac{\partial \rho k}{\partial t} + \frac{\partial \rho u_j k}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\mu \frac{\partial k}{\partial x_j} \right) - \frac{\partial}{\partial x_j} \left(\frac{\rho}{2} u_j u_i u_i + p u_j \right) - \rho u_i u_j \frac{\partial u_i}{\partial x_j} - \mu \frac{\partial u_i}{\partial x_k} \frac{\partial u_i}{\partial x_k}$$

Rate of dissipation equation is:

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho u_j \varepsilon)}{\partial x_j} = C_{\varepsilon 1} P_k \frac{\varepsilon}{k} - \rho C_{\varepsilon 2} \frac{\varepsilon^2}{k} + \frac{\partial}{\partial x_j} \left(\frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right)$$

CHAPTER 4

CFD MODELING

4.1 Introduction

The innovation of rapid computerized PCs, joined with the improvement of exact numerical strategies to tackle physical issues, has altered our way of study and practice liquid flow and heat exchange. That methodology is termed as CFD to put it plainly, and it has prepared it conceivable to break down multifaceted stream geometries no sweat as that confronted while tackling glorified issues utilizing ordinary strategies. CFD may in this way be viewed as a region of learn joining liquid flow and numerical examination. Verifiably, the prior improvement of cfd in the 1960s and 1970s was driven by the need of the aviation commercial enterprises. Current CFD, be that as it may, has applications over all controls – common, mechanical, electrical, gadgets, substance, aviation, sea, and biomedical building being a couple of them. CFD substitutes testing and experimentation, and lessens the aggregate time of testing and outlining.

4.2 CFD Programs

All established CFD software contain three elements (i) a pre-processor, (ii) the main solver, and (iii) a post-processor

4.2.1 The preprocessor

Pre-processing is the first step of CFD analysis in which the user

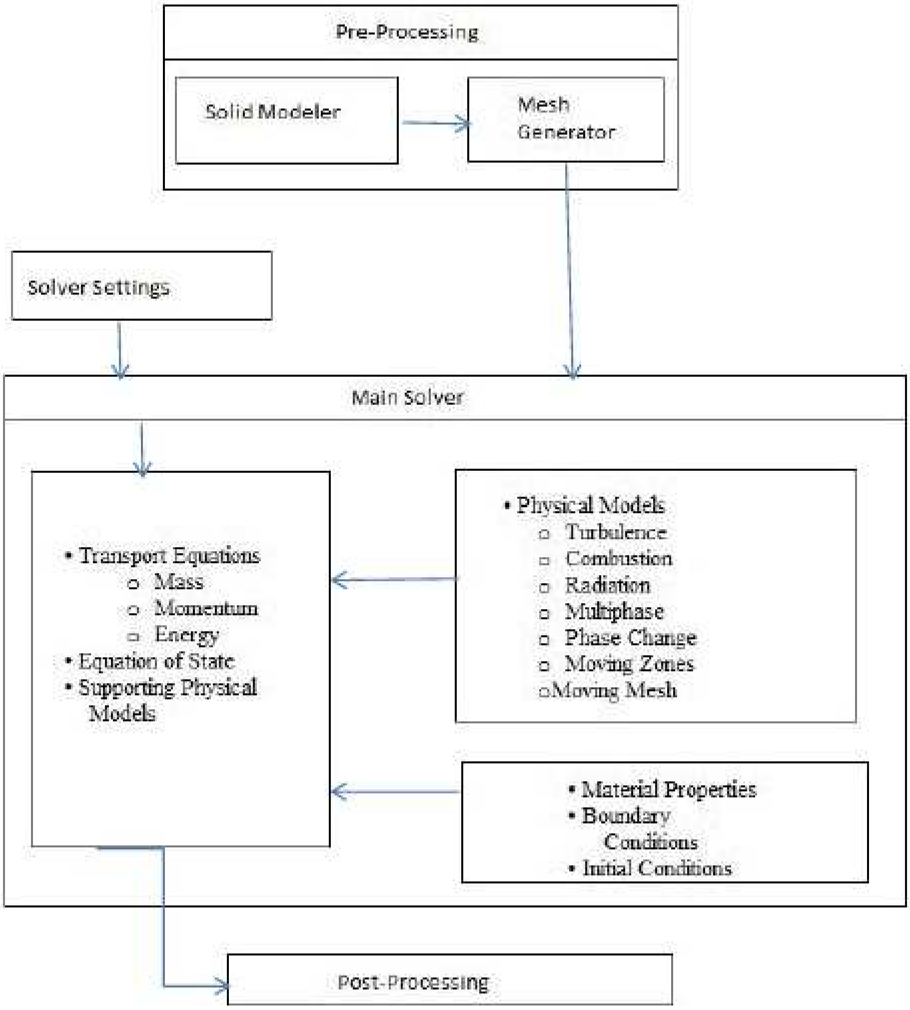


Figure 4.1: Overview of CFD[1]

(a) characterizes the displaying targets,

(b) Distinguishes the computational area, and

(c) Outlines and makes the framework

4.2.2 The main solver

The main solver does following functions: Selection of physical model.

(a) Definition of material properties

(b) Define boundary conditions

(c) Solution initialization

(d) Setting of relaxation factor

(e) Setting of convergence criteria

(f) Run calculation

(f) Saving results

4.2.3 The post processor

The post-processor is the last some portion of CFD programming. It helps the client to investigate the outcomes and get valuable information. The outcomes might be shown as vector plots of vector amounts like speed, form plots of scalar variables, for instance weight and temperature, streamlines and liveliness if there should arise an occurrence of insecure reproduction. Worldwide parameters like skin grinding coefficient, lift coefficient, Nusselt number and Colburn variable and so on might be registered through suitable recipes. This information from a CFD post-processor can likewise be sent out to perception programming for better show and to programming for better diagram plotting.

4.3 CFD Procedure

There are five steps are used to solve the problem in CFD.

They are:

(a) Geometry Development

(b) Mesh Generation

(c) Specification of flow condition

(d) Calculation and numerical solution

(e) Results

A 3-D analysis of Shell and tube heat exchanger has been done in ANSYS FLUENT 15.0 [16]. The above 5 steps have been done using CFD tools.

4.3.1 Geometry Development

Geometries are made on ANSYS WORKBENCH 15.0. There are three different geometries, having different number of baffles. They are as shown in figures 4.2b, 4.2c and 4.2d.

4.3.2 Mesh Generation

Meshing has been done using ANSYS meshing tool. Meshing of the geometry is as shown in figures 4.3a and 4.3b.

4.3.3 Specification of flow condition

Assumptions

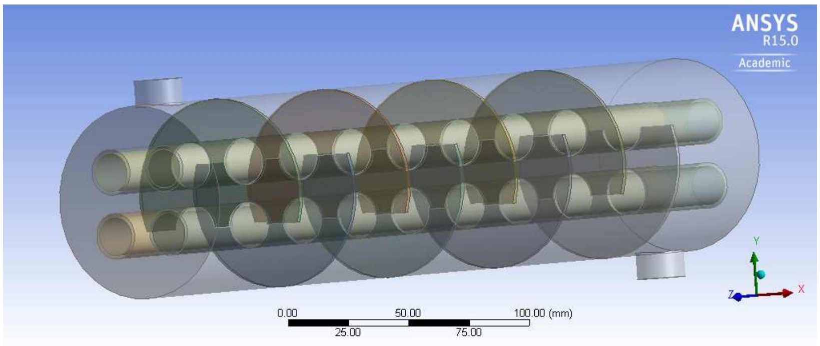
(a) Flow is incompressible i.e. density of the fluid does not change.

(b) Leakage from the gap between tube and baffle is neglected.

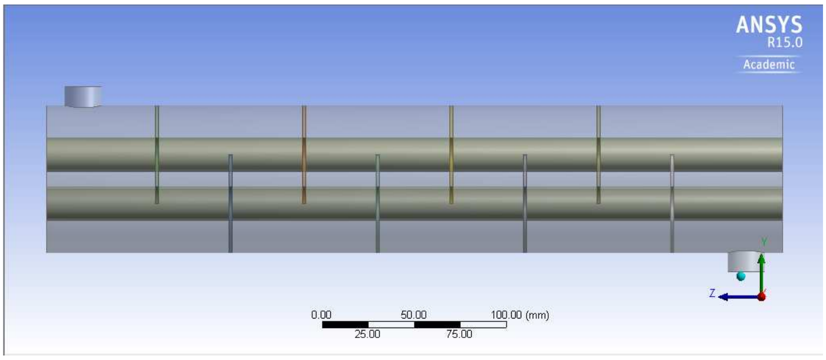
(c) Heat transfer to the baffles is neglected.

(d) Water is taken as working fluid and its properties are considered to be constant.

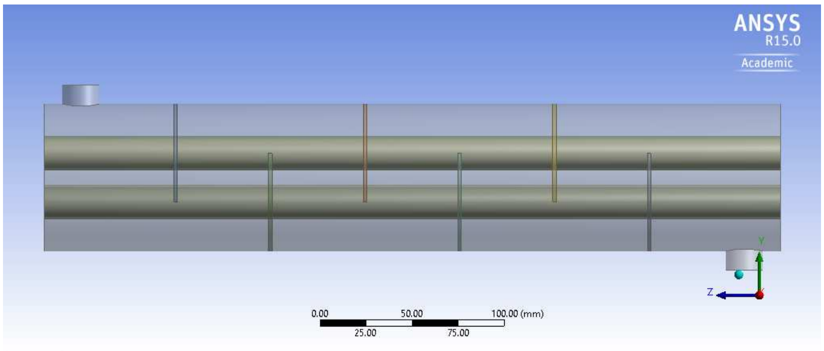
(e) Header effect is neglected



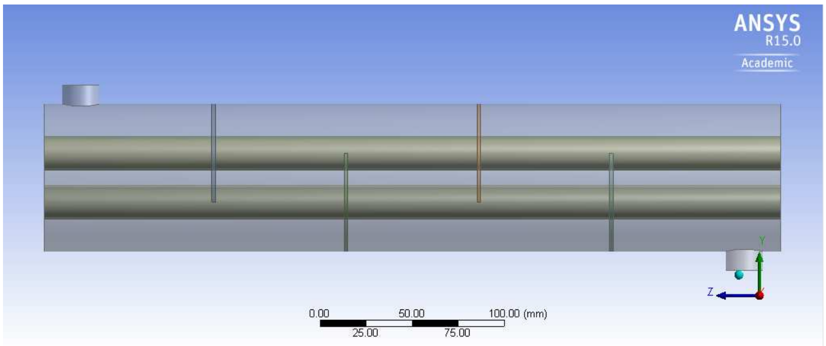
(a) Overall geometry of shell and tube heat exchanger with 8 baffles



(b) Front view of geometry with 8 baffles

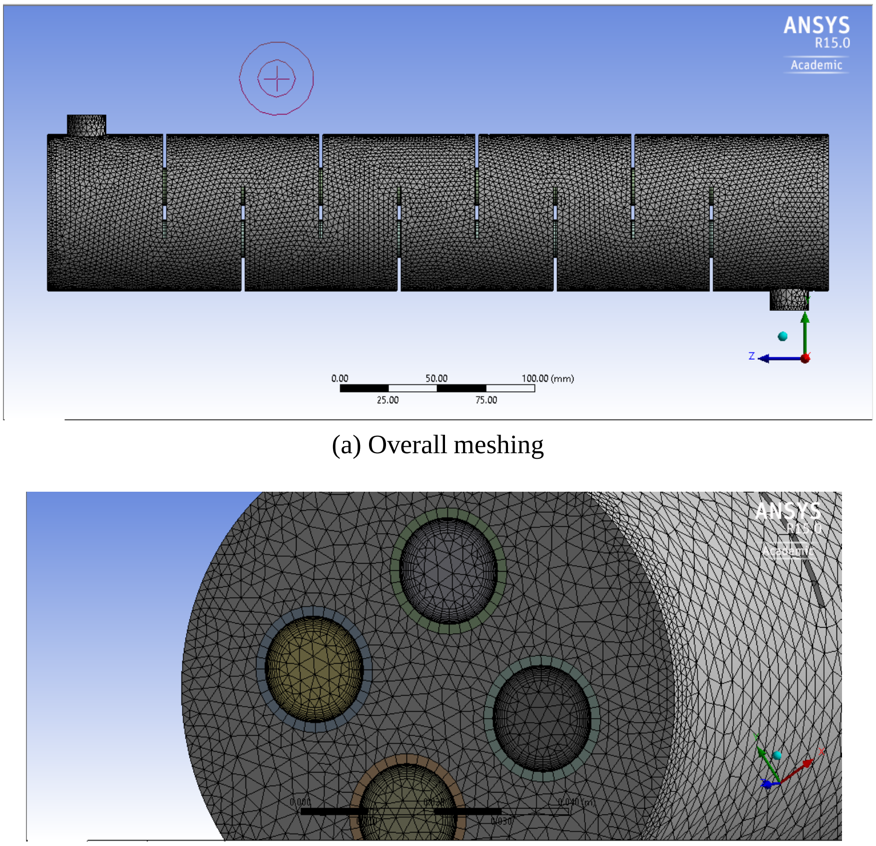


(c) Front view of geometry with 6 baffles



(d) Front view of geometry with 4 baffles

Figure 4.2: Geometry



(a) Overall meshing

(b) Magnified view of meshing

Figure 4.3: Meshing

Table 4.1: Details of Solver, Model and Materials

Solver	Model	Material
Pressure Based, steady state	Standard k-ε	Fluid-water, solid- copper

Table 4.2: Properties of water

Density	998.2 kg/m^3
Specific Heat	4182 J/kg-K
Thermal conductivity	0.6 W/m-K
Viscosity	0.001003 kg/m-s

Table 4.3: Properties of copper

Density	8978 kg/m^3
Specific Heat	381 J/kg-K
Thermal conductivity	387.6 W/m-K

Table 4.4: Solution methods

Pressure-velocity coupling	SIMPLE scheme
Gradient	Green-Gauss Node Based
Pressure	Second order
Momentum	Second order upwind
Turbulent kinetic energy	First order upwind
Turbulent dissipation rate	First order upwind
Energy	Second order upwind

CHAPTER 5

RESULT AND DISCUSSION

5.1 CASE 1

5.1.1 Velocity contour

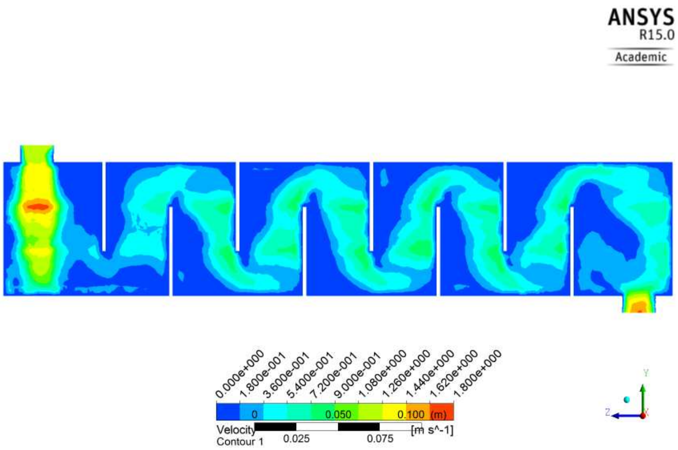
Velocity contour, streamline and velocity vector plots showing that when hot fluid enters the shell and passing through the baffles, there is a formation of recirculation zone in the back face of the baffles. In that region, flow is less as shown in figures 5.1a5.1b and 5.1c respectively.

5.1.2 Temperature contour

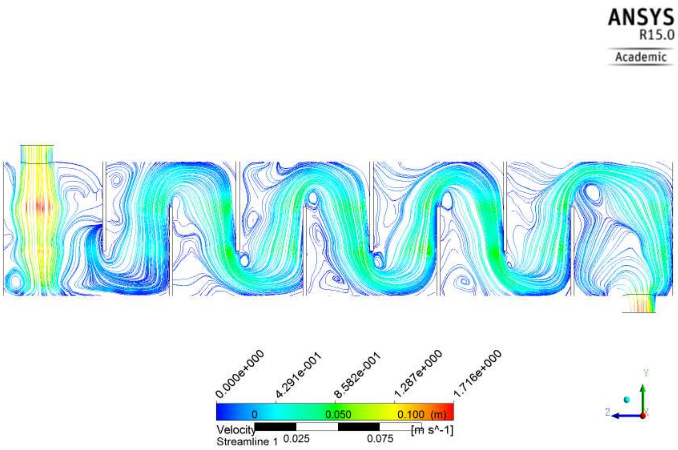
Temperature contour as shown in figure 5.2 shows how temperature varying in different zones. From velocity contour it is known that in back face of baffles flow is very less so the temperature in the back face of the baffles are also less as compare to their front face.

5.1.3 Pressure Contour

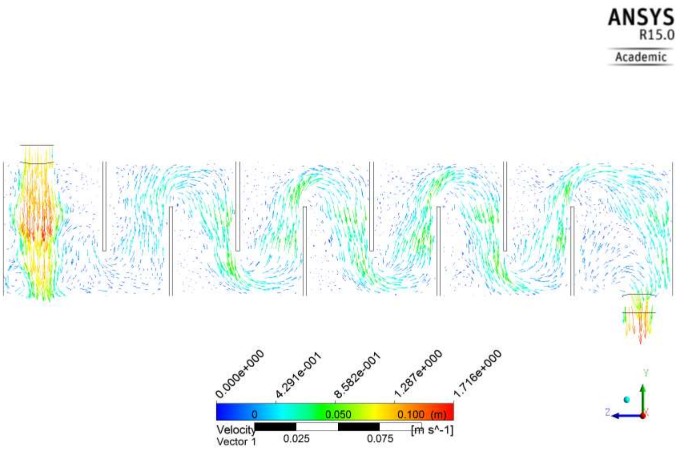
Figure 5.3 shows that pressure is gradually decreasing as fluid flows through the shell as well as pipe. This pressure drop is the deciding parameter for the pumping requirement. If pressure drop is more then more pumping power is required.



(a) velocity conour



(b) streamlines



(c) velocity vector

Figure 5.1: Velocity conour, streamline and velocity vector for 8 baffles

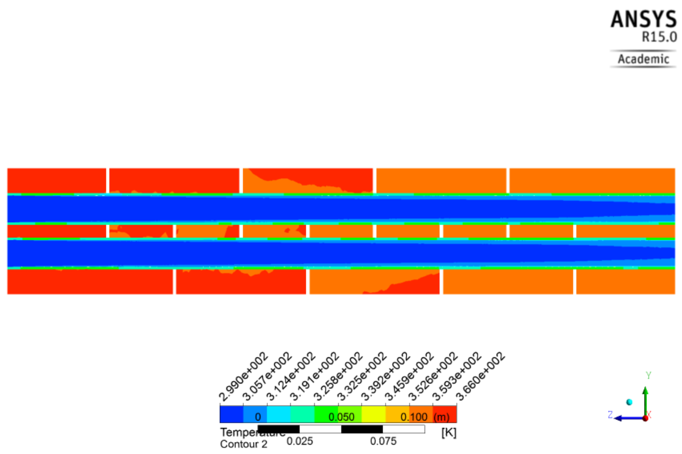


Figure 5.2: temperature contour

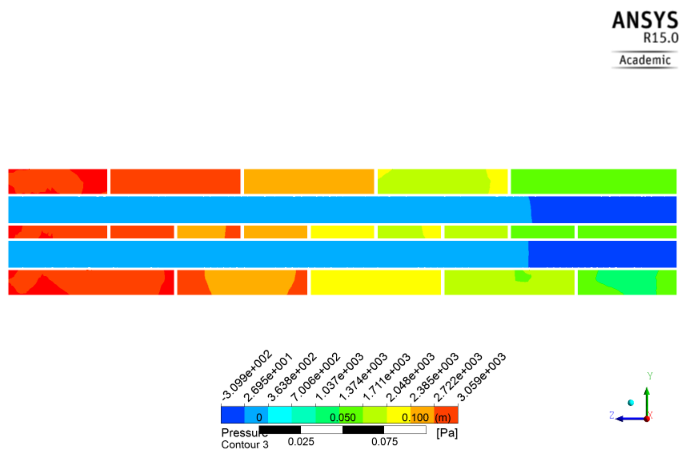


Figure 5.3: pressure contour

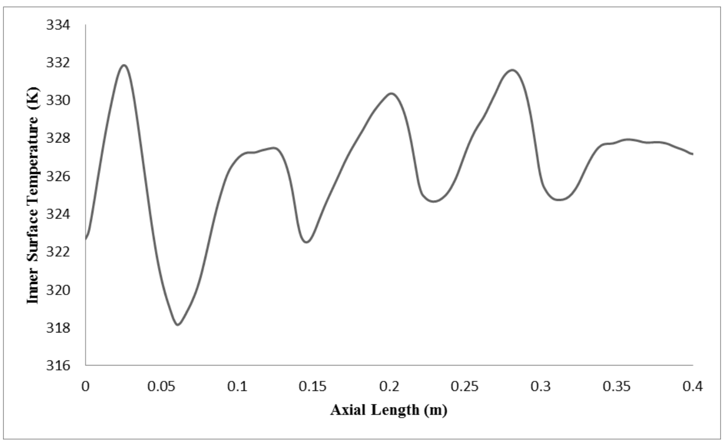


Figure 5.4: Inner surface temperature v length

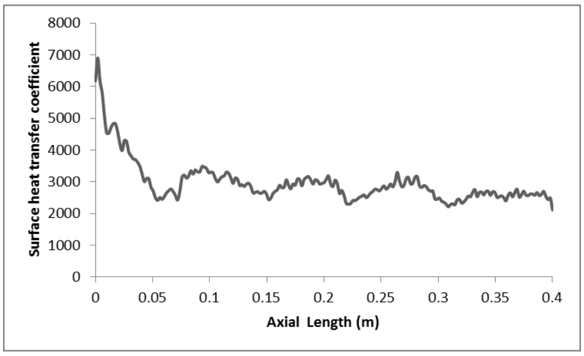


Figure 5.5: surface heat transfer coefficient vs length

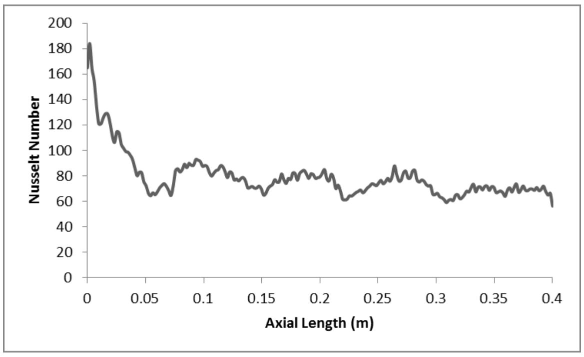


Figure 5.6: Nusselt number vs length

5.1.4 Variation of properties along length

Graph 5.4 demonstrate that the area of the pipe which is just backside of baffles have minimum temperature. In this case four minimum points are there because alternate baffles (not all baffles) are in contact with the selected pipe. Minimum temperature at each backside of baffle is more as compared to the previous one because of the heat transfer. At the end portion of the shell, temperature of the surface of pipe is less because hot fluid temperature is low as compared to other portion and heat is transferred to cold fluid from the pipe.

Graph 5.5 gives the variation of surface heat transfer coefficient in axial direction. We know that heat transfer coefficient is dependent on velocity of the fluid and geometry. The graph shows that heat transfer coefficient continue fluctuation. This is due to the high turbulence of fluid. At the back face of the baffle there is less fluid flow so there is less heat transfer also, so the heat transfer coefficient at that area is less as compared to the other areas. After that area, heat transfer coefficient starts increasing. In last portion of heat exchanger, there is less heat transfer so the heat transfer coefficient is also less.

The graph 5.6 demonstrates the variation of surface Nusselt number in axial direction in the inner surface of the pipe. This curve is same as heat transfer coefficient versus length because Nusselt number is directly proportional to heat transfer coefficient.

5.2 CASE 2

5.2.1 Velocity contour

Velocity contour, streamline and velocity vector plot postulates that as the spacing between baffles increasing or in other words, number of baffles are decreasing the recirculation zone became wider. Since recirculation zone is more, the contact between hot fluid and tubes are less as shown in figures 5.7a, 5.7b and 5.7c respectively.

5.2.2 Temperature contour

As previously discussed, in this case 5.8 recirculation zone is wider so the contact between hot fluid and the tube is less as compared to the previous case. Since contact is less, heat transfer is also less. Due to less heat transfer the temperature of the hot fluid decreasing at slower rate as compared to the previous case.

5.2.3 Pressure Contour

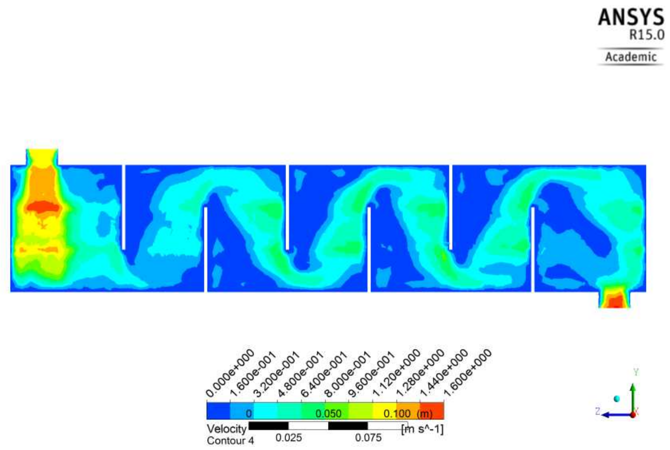
The contour 5.9 shows that as the both fluid flow forward, pressure is decreasing continuously. But the pressure drop in this case is less as compared to the previous case. Since the shell pressure drop is less so the pumping power required maintaining the flow is also less.

5.2.4 Variation of properties along length

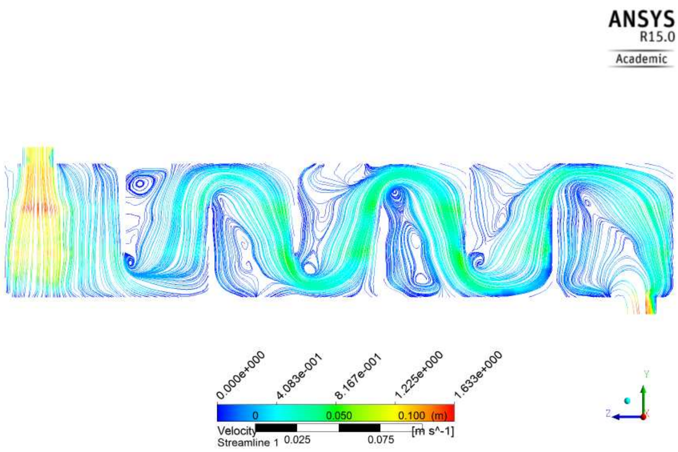
As discussed in case 1, temperature is less at the back face of baffle. In this case5.10 tube which is selected for observation has three contacts between tubes and baffles. As fluid flows in forward direction, heat is transferred to the cold fluid. Due to this heat transfer the temperature at each back face region is greater than the temperature of the previous baffle's back face region.

As previously discussed in case 1, heat transfer coefficient is dependent on velocity of fluid and nature of the surface. In the back face of the baffles as shown in figure 5.11, there is a recirculating region so the fluid is less in that region. Since the fluid flow is less in that region, heat transfer as well as heat transfer coefficient is also less in that region.

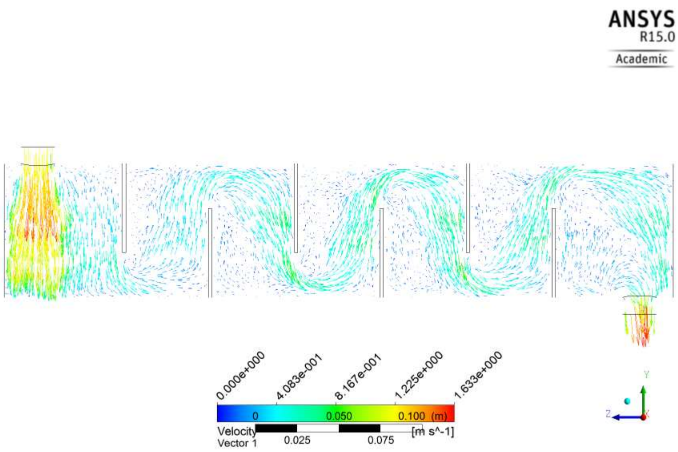
Nusselt number varies linearly with heat transfer coefficient (Nu= h*Dh/K) as shown in figure 5.12. So the pattern of the curve is same as heat transfer coefficient but the value has been changed by a factor Dh/K.



(a) velocity contour



(b) streamlines



(c) veclocity vector

Figure 5.7: Velocity contour, streamline and velocity vector for 6 baffles

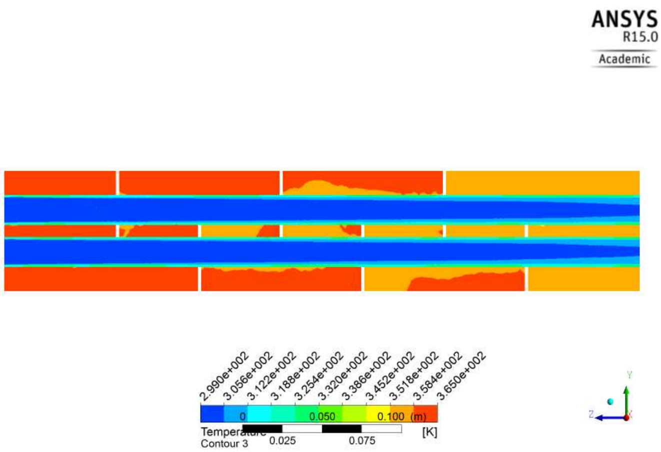


Figure 5.8: temperature contour

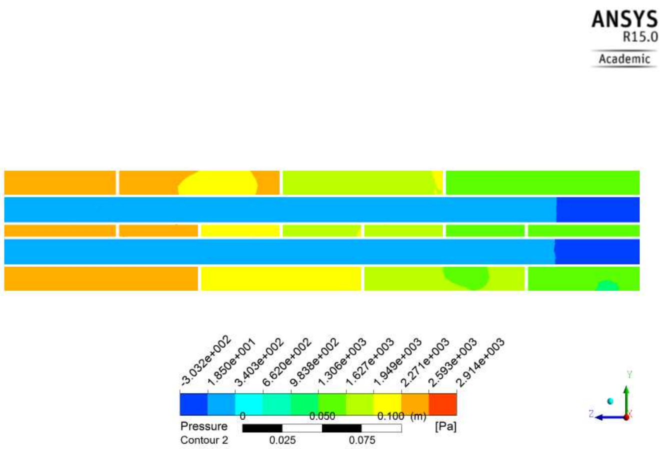


Figure 5.9: pressure contour

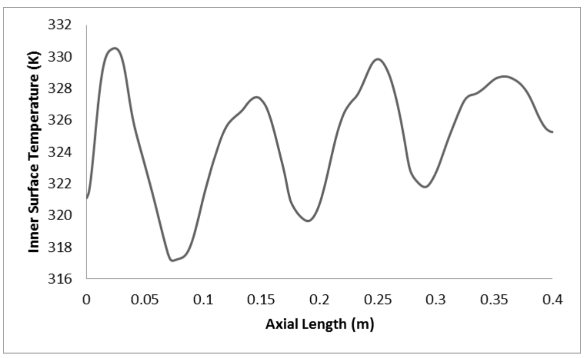


Figure 5.10: inner surface temperature vs length

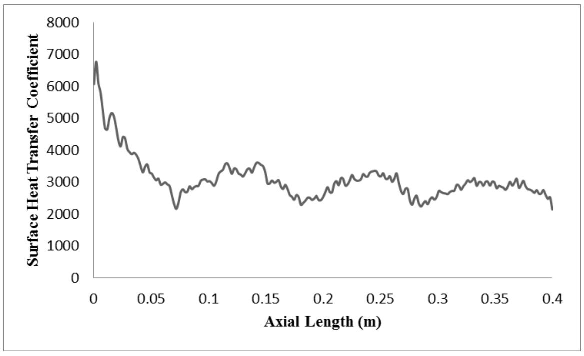


Figure 5.11: surface heat transfer coeffiecient vs length

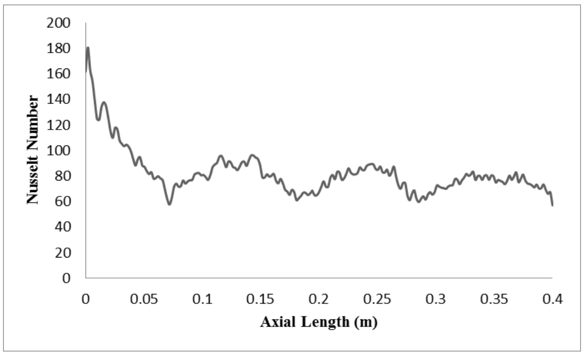


Figure 5.12: Nusselt number vs length

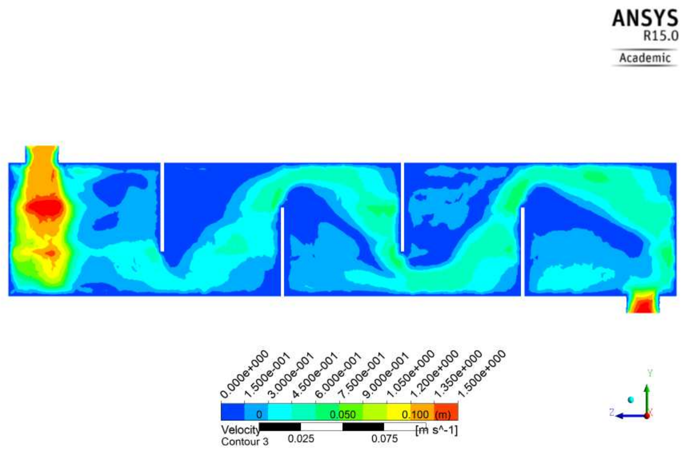
5.3 CASE 3

5.3.1 Velocity contour

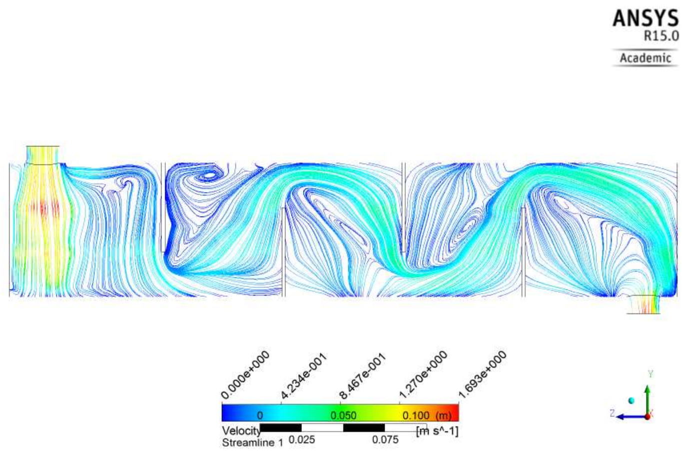
Velocity contour, streamline and vector plot shows that in this case recirculation zone is more as compared to previous cases. Since recirculation zone is more so the contact between hot fluid and tubes are lesser than previous two cases as shown in figures 5.13a, 5.13b and 5.13c respectively.

5.3.2 Temperature contour

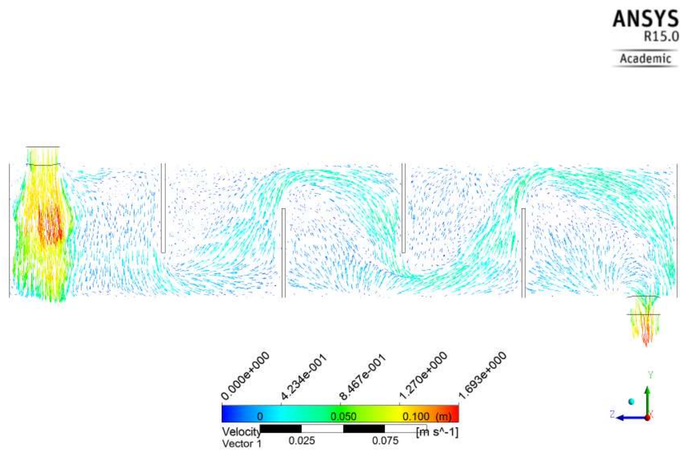
Temperature contour as shown in figure 5.14stipulates that most of the heat transfer take place in end portion. It is because of the less contact between hot fluid and tubes. As discussed above that recirculation zone is more in this case so the heat transfer is lesser in this case as compared to the previous two cases.



(a) velocity contour



(b) streamlines



(c) velocity vector

Figure 5.13: Velocity contour, streamline and velocity vector for 4 baffles

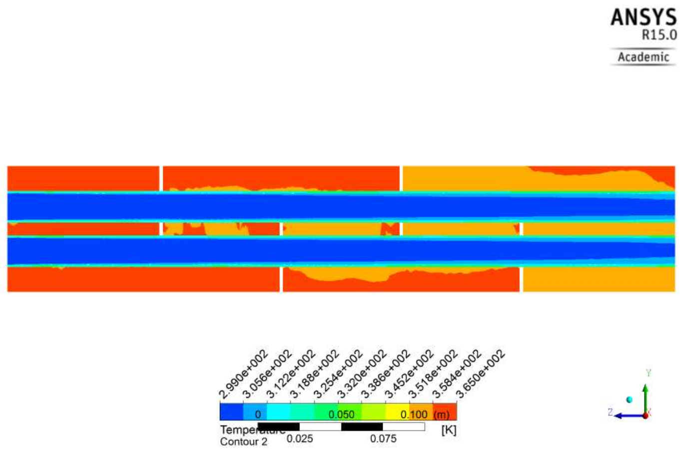


Figure 5.14: temperature contour

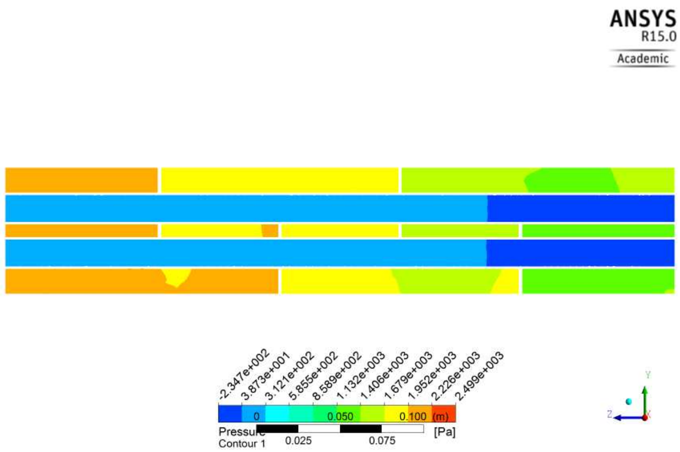


Figure 5.15: pressure contour

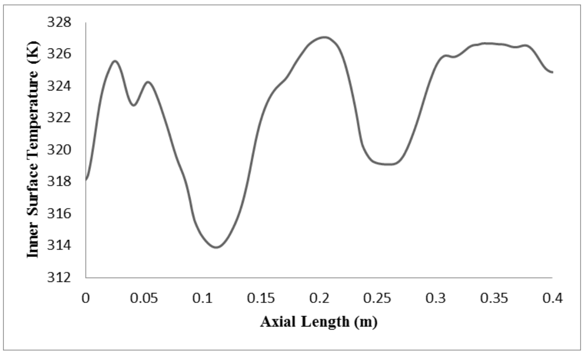


Figure 5.16: Inner surface temperature vs length

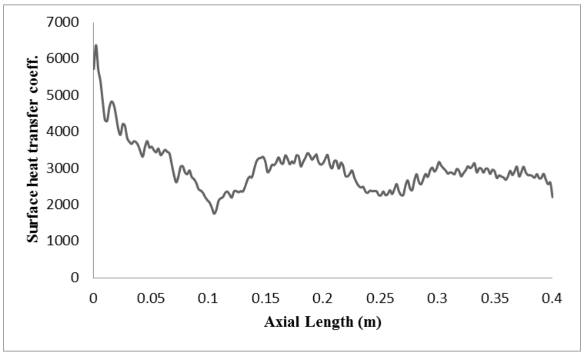


Figure 5.17: surface heat transfer vs length

5.3.3 Pressure Contour

From the contour 5.15it is clear that pressure in both shell side and tube side decreasing continuously as the fluid flows in forward direction. But the pressure drop on shell side is low as compared to the previous two cases. Since pressure drop is less so the pumping power required maintaining the flow is also less as compared to the previous cases.

5.3.4 Variation of properties along length

The graph 5.16 shows the variation of temperature in the inner surface of the tube. In this case four baffles are used. The surface which is selected for observation has two baffles connected with it so two back faces will appear in this case. Temperature of the two back face region is less as compared to other portion as shown in the graph. Temperature at the back face of second baffle is more as compared to the temperature of the back face of the first baffle due to the heat transfer.

The graph 5.17 demonstrates the variation of surface heat transfer coefficient with axial length. The heat transfer coefficient is continuously fluctuating because it depends on velocity of fluid. In the recirculating region fluid flow is less so the heat transfer as well

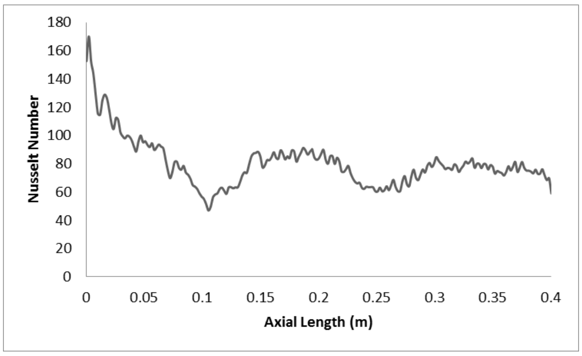


Figure 5.18: Nusselt number vs length

Table 5.1: Heat transfer rate in various cases

Number of baffles	Heat transfer rate (W)
4	8043.8305154292
6	9884.910562944
8	11053.37143

as heat transfer coefficient is also less. In this case only two back faces are there. So two craters formed.

Graph 5.18shows the variation of Nusselt number with length. The curve is same as heat transfer coefficient but values has been changed because Nu is dependent on heat transfer coefficient.

5.4 Heat transfer rate in various cases

The graph 5.19postulates that heat transfer increases with increase in number of baffles. The heat transfer increases with increase in number of baffles because the effective area of heat transfer increases as the number of baffle increases.

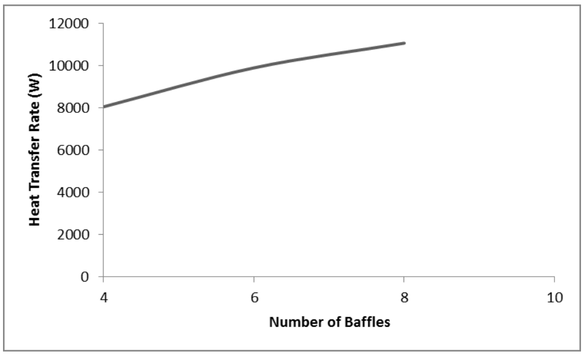


Figure 5.19: Heat transfer rate vs number of baffles

Table 5.2: Shell side pressure drop in various cases

Number of baffles	Shell side pressure drop (Pa)
4	2116.44502
6	2695.90933
8	3160.72314

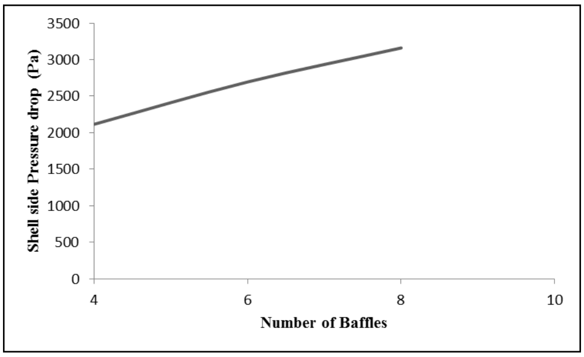


Figure 5.20: Shell side pressure drop vs number of baffles

5.5 Shell side pressure drop in various cases

The graph 5.20 demonstrates the variation of pressure drop in shell side with the number of baffles. As the number of baffles increases, shell side pressure drop increases. If pressure drop increases then required pumping power is also increase.

CHAPTER 6

CONCLUSION AND FUTURE SCOPE

6.1 CONCLUSION

Effect of number of baffles on heat transfer

As the number of baffles increasing for same length of shell, then spacing between the baffles are decreasing. Since the spacing between the baffle is less so the area of recirculation is less and high turbulence is developed. In high turbulence region, heat transfer rate is also high. Another point of interest is that due to increase in number of baffles, fluid has to cover more distance in the shell so the effective heat transfer area increases which is results in high heat transfer rate. Heat transfer is 37.414 % more in case of 8 baffles as compared to 4 baffles.

Effect of number of baffles on shell side pressure drop

As the number of baffles increasing, pressure drop in shell side increases. Since increasing in number of baffles means decreasing the space between baffles so the path for fluid flow becomes narrow. When fluid passes through narrow path, its pressure decrease and kinetic energy increase. So pressure drop increases with increase in number of baffles. As the pressure drop increase pumping power required to maintain the flow is also increases. Pressure drop is an adverse phenomenon which should be taken into consideration while designing the shell and tube heat exchanger. Pressure drop in shell side is 49.34% in case of 8 baffles as compared to 4 baffles.

6.2 FUTURE SCOPE

(a) Optimization of baffle spacing, to reduce the pressure drop in shell side and increase the heat transfer rate.

(b) Optimization for baffle cut.

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