Jimma Institute of Technology

Faculty of Mechanical Engineering

Masters of Science Program in Thermal System Engineering

Modeling the effect of helical baffles on the performance of shell and tube heat exchanger using CFD: (A case for Dashen Brewery Industry)

By: Workneh Asmare

A thesis Submitted to the School of Post graduate Studies in Partial Fulfillment of the Requirements for the Degree of Master of Science in Thermal Systems Engineering at Jimma Institute of Technology, Jimma University.

July, 2023

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Main Advisor: Balewgize A. Zeru (Asst. Professor)

Co- Advisor: Amanuel Tesfaye (MSc)

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Declaration

Workneh Asmare declare that this research entitled by *Modeling the effect of helical baffles on the performance of shell and tube heat exchanger using CFD in case of Dashen Brewery Industry* is my own original work and it is not been presented for a degree in any other University.

Workneh Asmare ______________ ____________

Done by Signature Date

Approved by

1. Balewgize A. Zeru (Asst. Professor) ______________ ____________

   Main Advisor Signature Date

2. Amanuel Tesfaye (MSc) ______________ ____________

   Co – Advisor Signature Date

3. Dr. Solomon T/MariamTeferi(PhD) ______________ ____________

   External examiner Signature Date

4. Nebiyu Bogale (Asst. Professor) ______________ ____________

   Internal examiner Signature Date

5. Ali Seid (MSc) ______________ ____________

   Thermal System Engineering Chair Signature Date

ABSTRACT

One of the many kinds of heat exchangers that are used in industry is the Shell and Tube Heat Exchangers (STHE). According to estimates, more than 35 to 40 percent from heat exchangers are shell and tube systems and mostly constructed with segmental baffle and without baffle. Design of STHEs basically determines the heat transfer rate, and check for pressure drops of both shell and tube side fluid stream. For efficient heat transfer rate the pressure drop must be below allowable pressure drop which guarantee turbulence. One of the factors which causes pressure drop is baffle arrangement. Baffle is the component of STHE that used to support and prevent vibration of the tubes. However, it causes a turbulence in shell side fluid flow and increases the heat transfer coefficient. The main purpose of this study was to analysis the effect of helical baffle on the performance of STHE using CFD for Dashen Brewery Industry. In Modeling and simulation analysis of STHE constructed with segmental baffle, helical baffle and without baffle was developed using ANSYS 19.2 and comparative and validation analysis have been carried out. The result of simulation revels the heat transfer rate and pressure drop in both segmental, helical and without baffle STHE. In the comparative analysis, the performance of STHE constructed with helical baffle was 12.43 % higher than segmental baffle and 20.08 % without baffle. To increase the performance helical baffle STHE optimization analysis also investigated by orientation of baffle with helix angle of 8°, 10°and 12° and 8° have 1.13 % effective than 10°. Also, from the validation analysis the current study analysis have been valid and the previous study have supports to the current study analysis. As a final conclusion, the capacity of STHE developed with helical baffles have 90.5 % higher than the existing STHE in Dashen Brewery Industry. And the current study of STHE constructed with continuous helical baffle have better performance than the previous study of dis continuous helical baffle STHE by Asghar Ali Ghoto et.al. Furthermore, the industry have gotten much profitable with a simultaneous decreasing of heat exchanger area, inlet flow rates and pressure drop.

Key words*: shell and tube heat exchanger, baffles, rate of heat transfer and pressure drop.*

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TABLE OF CONTENT

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

List of Acronyms

CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
CSDB	Combined Segmental-Disk Baffle
DB	Disk Baffle
DB-TR	Disk Baffle
FEM	Finite Element Method
SB	Segmental Baffle
STHE	Shell and Tube Heat Exchangers
PB	Plane Baffle
FB	Fold Baffle
CR	Circular Ribbed
TR	Triangular Ribbed
AB	Align Baffle
DCH	Dis - Continuous Helical baffle
LMTD	Log Mean Temperature Difference
E – NTU	Effective Number of Transverse Unit
EES	Engineering Equation Solver
HB	Helical Baffle
DBI	Dashen Brewery Industry
TEMA	Tabular engineering Manufacturing Association

List of Symbols

Q	Heat load (KW)
$\dot{m}$	Mass flow rate (kg/s)
A	Heat transfer surface area (m^2)
L	Length of tube (m)
N_t	Number of tube
P_t	Tube pitch (m)
d_o	Outside diameter of tube (m)
d_i	Inside diameter of tube (m)
D_b	Bundle diameter (m)
D_s	Shell diameter (m)
Gs	Shell side mass velocity ($kg/s\ m^2$)
u_s	Shell side velocity (m/s)
h_o	Shell side heat transfer coefficient ($W/m^2\ ℃$)
u_t	Tube side velocity (m/s)
N_{tpp}	Number of tube per pass
G_m	Tube side mass velocity (kg/sm^2)
h_i	Heat transfer coefficient for tube side fluid ($W/m^2\ ℃$)
U_i	Tube side overall heat transfer coefficient ($W/m^2\ ℃$)
U_o	Shell side overall heat transfer coefficient ($W/m^2\ ℃$)
B_s	Baffle spacing (m)
B_t	Baffle thickness (m)
N_b	Number of baffles

ΔP	Pressure drop (Pa)
j_f	Friction factor
$\Delta \mathrm{T}$	Temperature difference (℃)
$\mathrm{F_t}$	Temperature correction factor
$\mathrm{p_r}$	Prandtl number
N_u	Nusselt number
$\mathrm{R_e}$	Reynolds number
K	Thermal conductivity ($\mathrm{w/m\ k}$)
$\mathrm{c_p}$	Specific heat capacity ($\mathrm{J/kg.\ k}$)

Subscripts

h	hot side fluid
c	cold side fluid
hi	inlet temperature of hot fluid
ho	outlet temperature of hot fluid
ci	inlet temperature of cold fluid
co	outlet temperature of cold fluid
LMTD	log mean temperature difference
m	mean temperature difference
w	wall
s	shell side
t	tube side

Greek letters

ρ	Density (kg/m^3)
μ	Dynamic viscosity (Kg/m s)
α	Helix angle
ε	Heat transfer effectiveness

CHAPTER ONE

INTRODUCTION

1.1 Background of the Study

Heat is the energy transfer from a medium with a higher temperature to one with a lower temperature. The two mediums must establish thermal equilibrium for this energy flow to occur at a different temperature. There are three main ways in which heat can move from one place to another. Conduction, radiation, and convection are the three methods. Conduction is the direct transfer of heat from a hot medium to a cool medium. Convection is the movement of fluids to transfer heat from one place to another. Radiation is the transmission of heat by electromagnetic waves without the movement or interaction of physical objects.

Now a day, it is crucial to increase energy efficiency and practice conservation. Certain industries release a significant amount of heat energy into the atmosphere and may need a heat source that can be recovered. Engineers can find a heat exchanger in these situations [1].

A heat exchanger is an equipment in which heat is transferred from a hot fluid to a colder fluid without mixing of two fluids. Most frequently, one medium is cooled while the other is heated. They are mostly applicable in refrigeration, air conditioning, and space heating, and chemical processing. Moreover, it may on occasion be applied to marine, aircraft, or missile applications. The various heat exchanger types are depicted in the figure below from different perspectives [2].

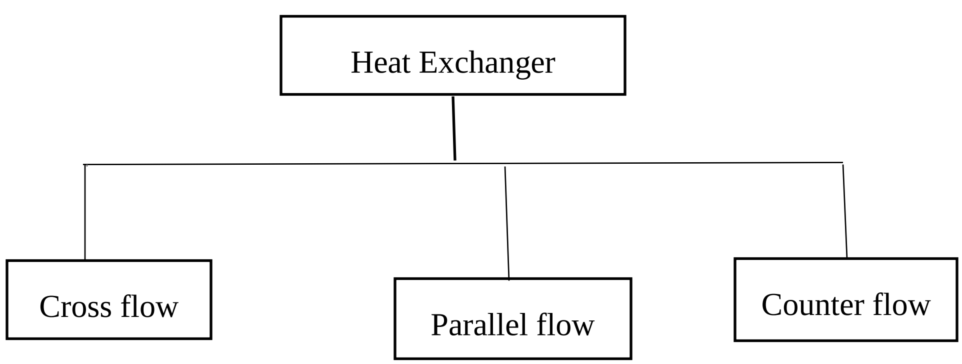


Figure 1.1 A): Classifications of heat exchanger based on fluid flow [2]

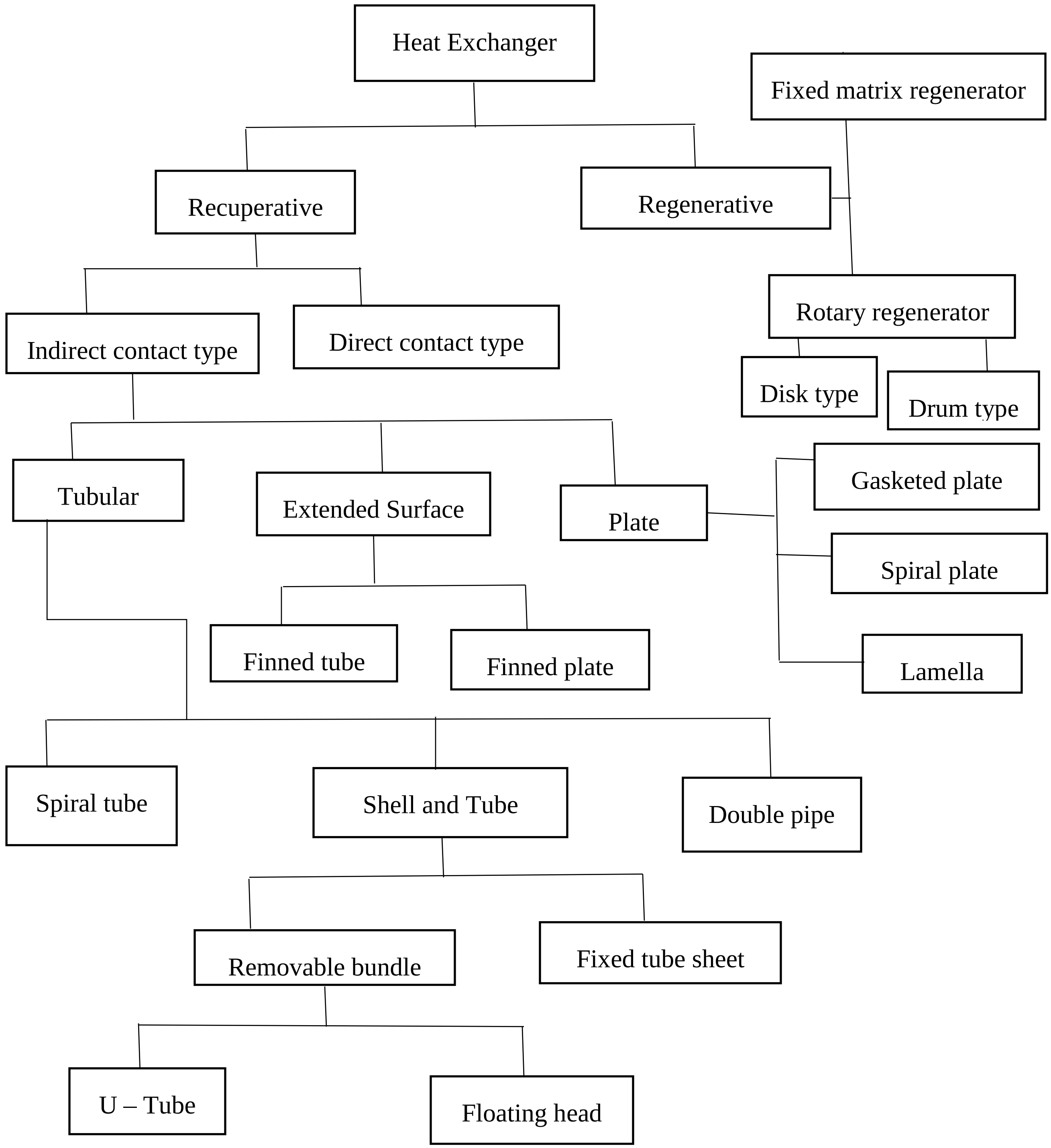


Figure 1.1 B): Classifications of heat exchanger based on applications and constructions [2]

One of the many kinds of heat exchangers that are used in industry is the shell-and-tube heat exchanger (STHE). According to estimates, more than 35 to 40 percent from heat exchangers are shell and tube systems. It is also the most popular kind of heat exchanger used for applications requiring higher pressures up to 552 bar. Because of its simple construction, little need for maintenance, and high rates of heat transmission to volume [3]. By taking into account their operability, maintainability, adaptability, and safety, those might be specially created. As a result, it is exceedingly durable and frequently utilized in industries [4]. However, DBI additionally utilized a shell and tube heat exchanger type to pre-heat the fuel oil and facilitate boiler combustion. This heavy fuel, although having a very high viscosity, is used to power the boiler's combustion process. Thus, a pre heater was utilized to lower the viscosity of heavy fuel oil until the temperature reached the flash point of 70 degrees.

As we can see from the graph above, shell and tube kind of heat exchanger (STHE) are a direct type under heat exchanger, meaning that the two media are kept apart by a wall through which heat is transferred so that they never mix. It had a variety of integrated parts, primarily shell, tube, baffles, tube sheet, etc. Engineers learned to enhance a STHEs performance in response to the need for these heat exchangers.

The STHE design basically determines the heat transfer coefficient and checks the liquid flow pressure drop on the shell as well as tube side. For efficient heat transfer, the pressure drop should be less than the allowable pressure drops to ensure turbulence. One factor that causes pressure drop is the location of the baffles [5].

Baffles are primarily used to protect the tubes from sagging and vibration and to direct the shell-side fluid flow into the tube bundle to create turbulence [6]. There are various types of baffles such as disc, segment and helical baffles. However, it is often used as a segment baffle.

Most shell-and-tube heat exchangers are designed with a segmented baffle configuration, and many researchers have investigated various factors to improve the performance of shell-and-tube heat exchangers.

This work focuses on modeling the effect of helical baffles on the performance of shell-and-tube heat exchangers using CFD. And the CFD analysis result of arranging baffles in a helical is said to optimize the performance of STHEs.

1.2 Statement of the Problem

Among the different types of heat exchangers, STHE are the standard and are used 35 - 40% in the industry[3]. Most tube and shell heat exchangers are designed with a segmented baffle arrangement.

Figure 1.2 Existing shell and tube heat exchanger in Dashen Brewery Industry

But in Dashen Brewery Industry, STHE are designed without baffles and a large amount of steam was needed to preheat the heavy oil to reduce the viscosity with a temperature of 70 degrees Celsius. First steam at 180 degrees Celsius, which is generated by an electric heater. Therefore, it requires a high amount of power and pump power, making its effectiveness low. In addition, the pipe length was 1.83 m and caused the sag effect. Therefore, the performance of STHEs can be improved by a different arrangement of the baffle, since the effectiveness of STHEs essentially depends on the baffle.

Ram kunwer[7] and many researchers are involved in designing new types of STHEs that use different baffles to increase performance. For example, the effect of baffle spacing on shell and tube heat exchanger performance has been studied experimentally and numerically. However, the current study was focused on by arranging baffle in a helical pattern, the effects of baffles on shell and tube heat are analyzed using CFD simulation to minimize the above-mentioned problems.

1.3 Objectives of the Study

1.3.1 General Objective

The objective of the research is modeling the effect of helical baffles on the performance of shell and tube heat exchanger using CFD in the Dashen Brewery Industry

1.3.2 Specific Objectives

- To collect data to set the capacity of tube and shell type heat exchanger.
- To model helical and segmental baffle shell and tube heat exchangers using CFD.
- To compare the effect of helical baffle with segmental type in shell and tube heat exchanger.
- To validate the CFD result from the available literatures data.

1.4 Research Questions

- What is the capacity of shell and tube heat exchanger?
- How much percent was increase the performance of STHE by the factor of helical baffle?
- What is the optimum helix angle to ensure maximum performance of STHE?
- What are the desired specifications (temperature and flow rates) to achieve the required capacity of STHE for Dashen Brewery Industry after optimization?

1.5 Significance of the Study

To perform company's operations, Dashen Beer Factory needs a sufficient amount of steam, which is produced by a boiler. Boilers are used to produce the necessary amount of steam energy through the combustion of fuel. High viscosity furnace oil, also known as heavy fuel, is the fuel utilized in this boiler, however combustion requires a lot of time. Pre heater is therefore linked with the boiler to boost efficiency. As a result, the importance of this study was to improve the pre heater heat exchanger's efficiency as a result of boiler efficiency.

1.6 Scope of the study

This research is focus on modeling STHE with helical and conventional segmental baffle arrangement with CFD and analysis the comparative study of them. Then after by optimizing the helical baffle STHE determine the design specification (Input parameters) for Dashen Brewery Industry of heat exchanger. Finally, investigates the validation analysis of the current work with the previous available literature work.

CHAPTER TWO
LITERATURE REVIEW

2.1 Introduction

Around the world, energy has been the backbone of technological development. So, many resear chers study the performance of heat exchangers to effectively utilize thermal energy.

Coulson et al., The most common kind of heat exchanger in industrial applications is the tube and shell type heat exchanger. It has surface densities between 50 and 500 m^2/m^3 and are easy to clean. Design codes and standards are available from TEMA (1999) - Tubular Exchanger Manufacturers Association [8].

They are commonly used in the chemical process industry, especially in petroleum refineries where large volumes of liquids need to be processed. Tube and shell type heat exchanger come in many variations to meet the process needs of nearly any industry or application. They provide reliable heat transfer performance by exploiting high turbulence and countercurrent flow and creating one or more passes [9].

Shell-and-tube heat exchangers can be designed to have a very large heat transfer area in a relatively small volume, are made of corrosion-resistant alloy steel, and can be used to heat, cool, and condense a wide variety of liquids.

2.2 Design components of shell and tube heat exchanger

Generally, components of tube and shell type heat exchanger are tubes, baffles, shells, front and rear headers, tube sheets and nozzles [8,9]. The selection criteria for these heat exchangers are operating pressure, temperature, thermal stress, liquid corrosiveness, fouling, clean ability and co st [8].

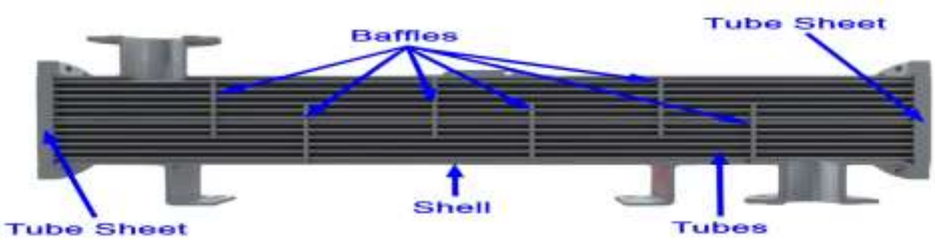


Figure 2. 1: Components of shell and tube heat exchanger [8]

2.2.1 Tube

Tubes are the basic component of a shell-and-tube heat exchanger, forming a heat transfer surface between one fluid flowing inside the tubes and another fluid flowing outside the tubes. Tubing can be seamless and is mostly made of steel alloys or copper. Depending on the application, other alloys such as nickel, titanium and aluminum will be used [10].

2.2.2 Shell

The shell is the outer cylinder that encloses the inner component, contains the outer tube flow, and forms the outer shell of the tube bundle. The main shell should protect against heat loss to the environment by the heat exchangers, resulting in efficient heat transfer rates through the insulation. The jacket diameter should be chosen so that the tube bundle is tightly secured. Different types of heat exchangers have different clearances between the tube bundle and the inner shell [9].

2.2.3 Tube sheet

A tube sheet is a plate or forging with holes for inserting heat exchanger tubes. Its main function is to keep mixing of tube-side and shell-side fluids. As a result, the fluids on the shell side and in the tube are effectively separated by tube sheet. In the case of a straight tube bundle heat exchanger requires a tube sheet heat exchanger, while a single tube sheet is sufficient for a U tube bundle. The tubes are embedded into gaps within the tube sheet and held in put by welding or hydraulic or mechanical stretching. The tube sheet thickness should always be greater than the tube outer diameter to achieve a perfect seal [9].

2.2.4 Tube bundle

A tube bundle is the basic component of a STHE used to join or confine tubes. There are two types of tube bundles, designed for your application. Hard Tube Plate Hard tube heat exchangers consist of straight tubes secured at both ends by tube plates welded to the shell. The tubes in U-tube shell and tube heat exchangers are U-bent and U-tube heat exchangers have only one tube sheet [12].

2.2.5 Heads or channel covers

A channel cover may be a circular plate that jolts to a conduit spine and can be expelled for pipe review without exasperating the pipe side run. For little warm exchangers, connections with flanged attachments or strung connections for pipe-side channeling are regularly utilized rather than channels or channel covers.

2.2.6 Baffle

Fundamentally, the performance of STHEs is affected by the properties of the baffles and liquid. Baffles are a component of shell-and-tube heat exchangers and are used to support tubes and keep tube misalignment. It is also important for disturbing and swirling the flow of liquid in the shell side of STHEs. Baffles are mainly used to induce cross-flow on the tubes, thereby improving heat transfer performance. Increased liquid flow turbulence raises heat transfer coefficient [1].

In Chapter 1, researchers described several ways to achieve efficiency in tube and shell type of heat exchangers. So, to investigate the study we should determine the design factors.

2.3 Design consideration of shell and tube heat exchangers

Initial Conditions are the basic design consideration which is specified by the researcher in order to design the required heat exchanger. Those conditions are: fluid allocation, tube arrangement type, fluid flow type, number of tube pass, and heat exchanger type [13].

2.3.1. Fluid allocation

This shows which liquid vapor flows through the tube and which in the shell side.
There are several factors to consider when considering tube and shell side fluid distribution. These factors are: Viscosity, corrosively, fouling tendency, pressure, flow rate, temperature range [14].

A. **Viscosity of fluids;** a major problem in heat exchangers is turbulence. Turbulence is characterized by Reynolds number and mass velocity, viscosity, and flow pattern. These are important parameters of turbulence. For a given velocity, the shell-side configuration results in much higher turbulence than the tube-side configuration. If the Reynolds number is high enough, tube-side mapping provides a sufficient heat transfer coefficient. Therefore, viscous liquids are better handled in the shell side.
B. **Corrosiveness,** classifying a fluid as corrosive requires knowing the corrosion resistance of the tube and shell side material components. Tube side components (tubes, ducts and duct covers, floating head covers, and tube side tube sheet surfaces) must be of good metallurgy. However, shell-side components (tubes, shells, shell covers, floating head covers, tube seat shell sides and

baffles) must have good metallurgical properties. Designs with corrosive jets in the shell side are more expensive if tube numbers, tube diameter, tube thickness, tube length and shell diameter are the same and the geometry is the same in both cases will be This is because cases, case covers and baffles cost more than ducts and duct covers. Therefore, it is preferable to place the highly corrosive liquid on tube side for this reason alone.

C. Fouling tendency

Fouling means that a dirty stream or liquid accumulates to some extent and reduces the heat transfer coefficient. Once dirt builds up to a certain level, it is imperative to clean the heat transfer surfaces. Mechanical cleaning inside of the tube (by water jet, steam, or stab) is much easier than cleaning the outside of tube. Unfortunately, as explained above, dirty streams are always viscous, and viscous streams are better handled by the shell side. Therefore, in this situation, fouling tendency and viscosity impose conflicting demands on the side allocation. Dirty viscous flow to the shell side results in higher heat transfer coefficients, lowering the initial cost of heat exchanger, but requiring more frequent (and more difficult) cleaning, reducing operational cost will be higher. However, if the dirty and viscous flow is passed through the pipe side, the initial cost will be higher, but the running cost will be lower. Therefore, a route with a lower overall cost (purchasing cost plus operating cost) is a more economical alternative. Note here that the initial or fixed cost depends not only on the viscosity of the stream, but also on other parameters currently being discussed. According to this reasoning, the guide fluid preferably passes through the tube side.

D. Pressure: Compared to the diameter and thickness of the shell and tube, the sides of the tube are smaller than the shell, so the area is also smaller and more pressure is required. Higher pressure flowing through the tube side minimizes fouling factors inside the tube. Steam and heavy oil are the working fluids in this design. Heavy oil has a higher viscosity than steam, which causes fouling and takes longer to consider cleaning. However, steam is less viscous than heavy oil, making it a less corrosive factor and less cleaning mechanism. Therefore, it is better to assign it to the tube side.

2.3.2 Tube arrangement layout

There are many different tube assembly patterns. These are:

Triangle (30°), rotated triangle (60°), square (90°), rotated square (45°). A triangle or rotated triangle pattern accommodates more tubes than a square or rotated square pattern. Additionally, the triangular pattern creates high turbulence resulting in a high heat transfer coefficient. However, at a typical pipe slope was 1.25 times the pipe outer diameter, mechanical pipe cleaning is not possible because there is no access road. The triangle layout is therefore limited to clean shell-side services. Services that require mechanical cleaning through the case side should use the square pattern. Since dry cleaning does not require access lanes, the Triangle Array can be used for dirty hull service where dry cleaning is appropriate and effective. As previously mentioned, the design of fixed tube sheet is typically used for service on the clean case side. Therefore, a triangular tube pattern preferably used for fixed tube sheet heat exchangers and a square (or rotated square) pattern also be used for floating head heat exchangers. For U-tube heat exchangers, a triangular pattern can be used if the shell-side flow is clean, and a square (or rotated square) pattern can be used if it is dirty. In STHEs, the tubes are typically arranged in an equilateral, square, or rotated square pattern. Basically, the placement of the tube pattern has a unique effect on the heat transfer coefficient and pressure drop in the heat exchange process. Therefore, the pattern arrangement of triangles and rotated squares provides high heat transfer coefficients. Of these patterns, the equilateral triangular configuration has a higher pressure drop than the square pattern. A square or twisted square arrangement is used for heavily contaminated liquids to mechanically clean the outside of the tube. There is also straggled tube pattern, but in this arrangement tubes are positioned at irregular manner. In triangular tube arrangement follows a specific pattern with equal spacing between tubes. Therefore, in this design we choose a triangular tube arrangement.

2.3.3 Number of tube pass

Moreover, shell-and-tube heat exchangers are categorized based on their functions or uses. In general, a service could be single-phase and focus on cooling or heating streams (a liquid or gas). In a single phase, neither the state nor the phase change. Yet, in a two-phase system, a phase transition occurs (such as condensing or vaporizing). There are several service combinations possible because a STHE has two sides. Since the preheater's primary function is stream heating, it depends on this phase identification. We have to choose between one shell and two tube pass heat exchangers based on this straightforward complexity.

2.3.4 Shell and tube heat exchanger type

According to tube sheet or tube bundle, the heat exchanger can be categorized.

A. Sheets of fixed-tube heat exchanger

Straight tubes fastened to tube sheets joined to the shell at both ends define as heat exchanger with a fixed-tube sheet. The structure may have integral tube sheets, bonnet-style channel coverings, or removable channel covers.

The main benefit of a fixed-tube sheet construction is that it is inexpensive and has the most straightforward design. In fact, the fixed tube sheet was the slightest costly kind of development, given an extension joint is not necessary. Extra benefits incorporate the capacity to mechanically clean the tubes contribute since they are open after evacuating channel cover and overhauling.

Offers most extreme security against spillage of the shell side liquid as there are no flanged joints, an advantage for deadly or inflammable administrations.

The drawbacks of a fixed-tube sheet development are: As the bundle is "fixed" to the shell and cannot be expelled, the exterior tubes are not be cleaned mechanically. Since mechanical cleaning (by rodding or jetting) is typically used, this poses a limitation in that fluid in the shell side must be clean in order to perform the necessary cleaning of the tubes' exterior. It should be noted, though, that chemical cleaning is a possibility. Hence, if a palatable chemical cleaning program was determined and utilized, fixed-tube sheet development may be chosen for fouling administrations in the shell side. Chemical cleaning is not very common because it is labor-intensive and challenging to use.

Within the occasion of a huge differential temperature between the tubes and the shell, the tube sheets will be incapable to assimilate the differential push, in this manner making it vital to consolidate a development joint on the shell. This takes absent the advantage of low taken a toll to a critical degree.

B. U-tube heat exchanger

As the name implies, the tubes of a U-tube heat exchanger are bent in the shape of a U. Evidently, there is only one tube sheet in a U-tube heat exchanger. However, the twisting of tubes speaks to an extra cost. Further, the minimum U-bend diameter is usually three times exterior diameter of tube. So that the central pass-partition lane is considerably larger in a U-tube heat exchanger than in one having straight tubes. Hence, for a given tube numbers it will have a greater shell diameter. Evidently, this difference in shell diameter will be larger for small shell diameters where the central U-bend lane will represent a larger fraction of the shell cross- sectional area and thereby result in a larger reduction in the tube numbers that can be accommodated.

The additional cost of bending the U-shaped tubes and the larger diameter of the sleeve will more or less offset the cost savings resulting from the removal of the tube sheet. Therefore, the cost of a U-tube heat exchanger is comparable to that of a fixed tube plate heat exchanger.

The advantage of a U-tube heat exchanger is that, being free at one end, it allows the bundle to expand or contract depending on how differential stresses develop. Therefore, no detailed calculations for design conditions and many other conditions (start-up, disturbance, shutdown, etc.) are needed to determine the requirements for expansion joints in the hull. In addition, it allows cleaning of the outside of the tube because the tube bundle can be disassembled.

The disadvantage of the U-tube construction is that the inside of the tube cannot be cleaned effectively, as the U-bending would require a drill shaft with a flexible tip for cleaning. Therefore, the U-tube heat exchanger should not be used for services with dirty liquid inside the tube. This places a very serious limitation on U-tube heat exchangers for oil refining services, which often have dirty streams on both the tube side and the shell side. This is mainly why fixed-tube or U-tube plate heat exchangers are rare in refineries.

C. Floating-head heat exchanger

The floating head is the most versatile and expensive type of STHE. It is so named because while one tube plate is fixed relative to the hull, the other is free to "float" inside the hull. Thus, it allows free expansion of the tube bundle and also allows cleaning both inside and outside of the tube. It can therefore be used for services where the shell side and tube side fluids are dirty and, therefore, it is the standard construction type used in dirty services such as refineries.

The higher cost of a floating-head heat exchanger is due to the fact that this type of construction has more components than fixed plate or U-tube types. In addition, the sleeve diameter in the floating head structure is larger because it must be larger than the floating tube plate to be able to disassemble the tube bundle. R. Mukherjee was selected from these types of fixed tube plate heat exchangers.

2.3.5 Fluid flow type

The operating modes and efficiency of the heat exchanger are largely determined by the direction of the fluid flow. The most common arrangements for flow paths in heat exchangers are reverse flow and parallel flow. The counter flow heat exchanger is the flow direction of one of the working fluids opposite to the flow direction of the other fluid. However, in a parallel flow exchanger, the two exchanger fluids circulate in the same direction. The figure below shows the flow direction of liquid in parallel and counter current flow exchangers. Under comparable conditions, more heat is transferred in the upstream arrangement than in the parallel flow heat exchanger.

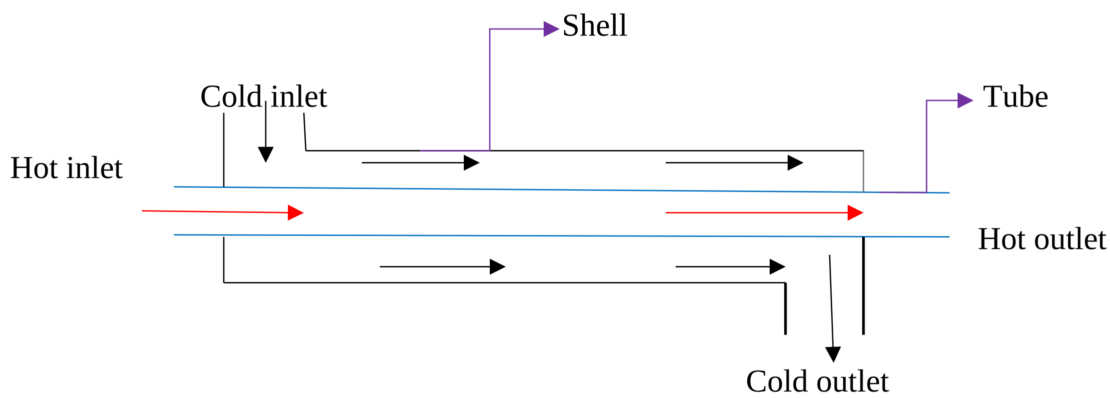


Figure 2. 2: Parallel flow heat exchanger [15]

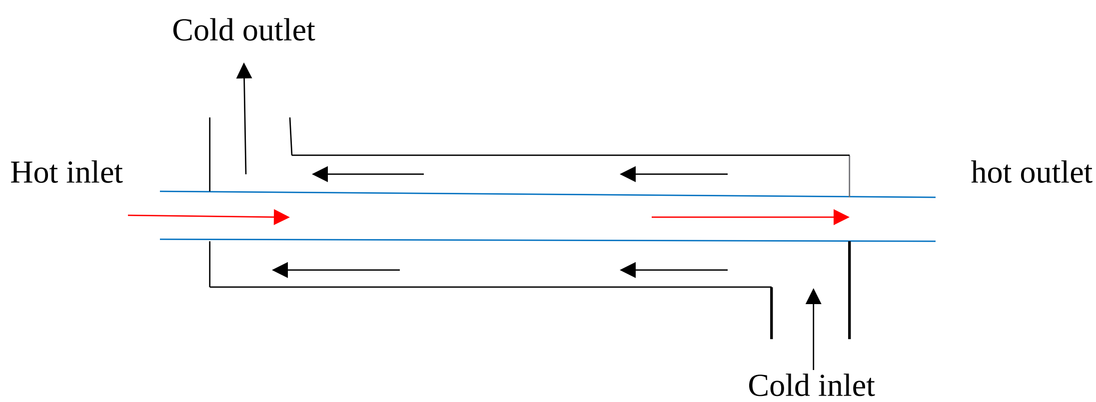


Figure 2. 3: Counter flow heat exchanger [15]

The temperature profile of the two heat exchangers has two major drawbacks in the parallel flow plane. For starters, the widening temperature contrast at closed doors causes enormous thermal stress. The growth and compression of the growing materials is limited because different fluid temperatures can lead to tissue failure. Currently, the temperature of the cold liquid leaving the heat exchanger never exceeds the lowest temperature of the hot liquid. This relationship can be a problem especially in cases where the reason for the plan is to increase the temperature of the cold liquid.

A parallel flow heat exchanger design is advantageous when two fluids need to be at approximately the same temperature. The Counter-flow heat exchanger has three main advantages over parallel-flow designs. First, the thermal stress across the heat exchanger is minimized due to the more uniform temperature difference between the two fluids. Second, the exit temperature of the cold liquid can approach the maximum temperature (inlet temperature) of the hot liquid. Third, a uniform temperature difference results in a uniform heat transfer coefficient across the heat exchanger.

To achieve higher efficiency and higher heat transfer, it is recommended to design the heat exchanger to counter-flow [15].

2.4 Related works

Bassel A. Abdelkader et al. [16] Studied the effects of tube layout, fluid characteristics, mass flow rate, baffle cut, and baffle number. To anticipate the heat-transfer coefficient and pressure drop on the shell side of a STHX, a model that solved the governing equations using the Kern, Bell-Delaware, and flow-stream analysis (Wills Johnston) programs was created using the Engineering Equations Solver application. According to the experiment, adding more baffles increases the pressure drop and heat-transfer coefficient on the shell-side. While the heat transfer coefficient rises along with the mass flow rate, the pressure drop does so more quickly. As the number of baffles grows, the pressure drop reduces while the baffle cut remains constant.

Dr Jian Wen et al. [17] To enhance the refrigeration performance in the chillers, this study numerically examined the heat transfer characteristics, resistance characteristics, and stress characteristics of shell and tube heat exchangers (STHXs) with various baffles (segmental baffles (SB), plane helical baffles (PB), and fold helical baffles (FB)). The findings indicate that STHXsSB has the highest shell-side flow velocity and temperature differential, whereas

STHXsPB has the lowest. With an increase in the shell-side volume flow, STHXs' heat transfer coefficient and pressure drop steadily rise, whereas h/P gradually falls. STHXsHB h/P grows by 62.2% to 65.7% compared to STHXsSB, and STHXsFB increases by 73.9% to 76.1%. The tensions around the baffles' perforations are greater than those in other areas. Additionally, temperature has a significantly bigger impact on the baffle stress distribution than pressure does. Despite the fact that STHXsFB has the highest maximum stress, it falls within the permitted stress range for the material. Thus, due to STHXsFB great all-around performance, it is ideal for water chillers.

Asghar Ali Ghoto et al. [5] Examined the influence of baffle design in shell and tube heat exchangers. As we all know, the efficiency of the shell and tube heat exchanger is the output of pressure drop and heat transfer. In addition, the pressure drop and heat transfer depend on the design of the baffles. In this study, different baffles were simulated to evaluate performance and determine which baffles are suitable for high-volume flow in shell and tube heat exchangers. Basically, from this work single segment, double segment, spiral, and single segment plus spiral baffle are observed modeled, and simulated using Solid Works CAD software. As a result, the spiral single-segment baffle has less pressure drop and a higher heat transfer rate.

M. A. Zebua et al. [1] The performance of shell-and-tube heat exchangers was evaluated experimentally and numerically in this work to determine the impact of baffle distance. It is expected that the parameters for the heat exchanger shell length and diameter are 700 mm and 156 mm, respectively. There are eight tubes, each measuring 11.6 and 13 millimeters in diameter, respectively. The baffle distance in this investigation ranged between 40 mm, 50 mm, and 60 mm. Using numerical and experimental data, we then looked at the thermal efficiency and heat transfer coefficient. The findings indicate that when baffle spacing rises, heat transfer efficiency declines. As a result, the baffle distance is reduced while the heat transmission coefficient increases.

Swanand Gaikwad et al. [3] A segmental baffled shell and tube heat exchanger (STHX) has undergone a three-dimensional simulation. The investigation makes use of the ANSYS CFX commercial code, which uses modeling, meshing, and the Finite Element Method to produce numerical findings. There is a lot of research in the literature on the effects of baffle cut and baffle spacing when considered separately, but there is significant uncertainty when these two factors are combined. The purpose of this study was to determine the best combination of baffle cut and baffle spacing for a shell and tube heat exchanger's efficient operation. The heat exchanger heat exchanger spacing to accommodate 6, 8, or 10 baffles and the baffle cuts at 30, 35, and 40% of the shell inner diameter are tested. In the shell and tube heat exchanger, the pressure drop and shell side heat transfer coefficient were numerically calculated results that showed the impact of the researched factors. According to the study, switching from 6 to 8 baffle designs results in an increase in the shell side heat transfer coefficient of 13.13 percent, while switching from 6 to 10 baffle configurations results in an increase of 23.10 percent for a particular baffle cut. Additionally, evidence suggests that pressure losses occur between 44 and 46.79 percent of baffle spacing, which can be handled by regulating the baffle cut.

Badhan Saha et.al. [18] A numerical investigation was performed on a single-pass shell and single-pass tube heat exchanger with a variable number of segmental baffles and baffle spacing using the commercial program ANSYS FLUENT. The amount of heat that can be transmitted while utilizing a minimum amount of electricity determines how well-optimized a shell and tube heat exchanger is. The rate of heat transfer increases when baffle space and baffle number are surpassed. However, the pressure loss also increases as a result of the faster heat transfer rate. This investigation's main objective is to ascertain how pressure drop impacts power usage and heat transfer rate in order to lower both.

Asif Ahmed et al. [19] Two tube bundle heat exchangers with standard segmented baffles and continuous helical baffles were compared using the finite element technique (FEM). 37 tubes are housed in a 200 mm in diameter by 500 mm long shell with a fluid mass flow range of 0.5 kg/s to 2 kg/s on the shell side. First, mathematical and physical models are developed and statistically simulated using the finite element method (FEM). The computational model is validated using the shell-side average Nusselt number (Nus), which is calculated from the simulation results and compared with the available test results. Comparative studies have shown that continuous helical baffles have 72-27% better heat transfer coefficients per unit pressure drop than traditionally segmented baffles for the same shell side mass flow rate. Additionally, compared to conventional segmented baffles, continuous spiral baffles have a shell-side pressure drop that is 59–63% lower. As a result, it has been found that helical baffle shell and tube heat exchangers work better at attaining high-efficiency heat transfer and minimal pressure loss. However, we should explain our decision to use a traditionally segmented baffle to confirm our findings. Researchers are also looking at how often this spiral baffle needs to be maintained and cleaned.

Ram Kunwer et al. [7] Using numerical and experimental techniques, the authors examined segmented and aligned baffles of shell and tube heat exchangers with 30° and 50° shear profiles. Also included was a test comparison comparing continuous spiral baffles with segmented baffles that had a 25o torsion angle. The number of shell and tube heat exchangers with various configurations was studied, and the results revealed that the AB-30 structure is more effective than others. The numerical and experimental results of AB-30 differed by 10.1%, according to Kunwer et al.'s 2020 study.

P.S.P.Amirtharaj et al. [20] It has been done to improve heat transfer performance using mathematical and CFD analysis. The Kern technique is used to calculate the math. Two models of shell and tube heat exchangers, one with segmental baffles and the other with 20° inclined baffles, were subjected to a CFD analysis. By substituting inclined baffles for segmental baffles shell and tube heat exchanger performance in terms of heat transfer is improved and pressure drop is decreased.

Bin Gao et al. [21] At the five helix angles of 8°, 12°, 20°, 30°, and 40°, experimental research was done to examine the flow resistance and heat transfer of different shell-and-tube heat exchangers with discontinuous helical baffles. The effects of the baffle helix angle on the irreversible loss of heat exchangers are examined using thermodynamic analysis based on the second law. The findings show that for a given shell-side volume flow rate, heat transfer coefficient, and shell-side pressure drop are higher for heat exchangers with smaller helix angles than for those with larger helix angles. However, the flow resistance with a greater helix angle is lower and the heat transfer performance is better under the same shell-side Reynolds number conditions. Among the five heat exchangers under examination, the one with helical baffles at a 40° helix angle exhibits the best overall performance. Additionally, in some instances of low shell-side Reynolds numbers, heat exchangers with helical baffles perform better.

G.D. Chen, et al. [22] In this work, the helical baffled mixed multiple (CMH) shell pass, continuous or combined helical baffles, and discontinuous helical baffles (DCH) are all explored.

Numerous experimental and numerical data show that the helical baffled STHXs perform better in terms of flow and heat transfer than the traditional segmental baffled (SB) STHXs.

Usman Salahuddin et al. [23] This report reviews the major helical baffle research that has been conducted to improve the effectiveness of shell and tube heat exchangers. Some of the important factors that affect how shell and tube heat exchangers work are discussed. A comparison of the two is also given to show that helical baffles are preferable to segmental baffles. The best results are often achieved with a mix of discontinuous, folded, sextant helical baffles, 40° baffle inclination angle, and short baffle spacing. Also, continuous helical baffles eliminate dead zones.

Dipankar De et al. [24] Using ANSYS FLUENT software tools, a helical baffle was designed for a shell and tube-type heat exchanger and was compared to a straight baffle. This research examines how, when the flow velocity stays constant, different helix angles affect the pressure drop and overall heat transfer coefficient. Due to the shape of the continuous helical baffles, the flow pattern on the shell side of the heat exchanger with continuous helical baffles is forced to be rotating and helical, increasing the heat transfer coefficient per unit pressure drop in the heat exchanger.

I M Arsana et al. [25] this study conducts experimental study to determine the effect of baffle spacing to the effectiveness of the STHE with helical baffle. The parameters that baffle spacing are 4, 8 and 12 cm. As a result, the largest effectiveness which is 39.86% and heat transfer rate of 4867.4 Watts with 4 cm baffle spaces and the effectiveness and heat transfer rate of baffle spacing at 8 cm are 35.23% and 3,709.12 Watts respectively. Finally, the baffle spacing of 12 cm leads to the lowest effectiveness, 29.59% with a heat transfer rate only 3301.21 Watts.

Vivek Singh Parihar et al. [26] The aim of this study was to simulate the performance of helical shell and tube heat exchanger trough several optimization techniques using CFD. The first optimization technique is replacing the straight tube by helical tube in the heat exchanger and we used 10, 12, and 14 number helical baffles with 50% baffle cut. It also found that the high heat transfer is obtained with increased number of baffles.

Mohammad Ramin Daneshparvar et al. [27] investigates the thermal-hydraulic performance in the shell and tube heat exchanger with helical baffles using CFD method. Basically this study focused on the effects of baffle pitch and baffle angle on heat transfer coefficient and pressure drop. Finally, the optimum geometries of the investigated helical baffles were suggested using multi-objective genetic algorithm (MOGA) optimization.

2.5 Research gap

Many researchers recently conducted studies to improve the performance of shell and tube heat exchangers by investigating different factors that affects the heat exchangers. From those study the comparison analysis of parallel and counter flow of fluid, material type of tube, triangular and square tube arrangement and baffles. Specially they analysis of the effect of different baffles. Most of the study was conducted the baffle type effect especially on segmental, disk and also few as helical baffles. Also in helical baffle the pattern or structure of baffles helically was discontinuous as well as joining of segmental baffle and provides an optimized output. But there was a limitation to study the helix angle effect on the continuous helical baffle shell and tube heat exchanger. So, researcher will propose to analysis the effect of continuous helical baffle in shell and tube heat exchanger with optimization of angle orientation in CFD simulation in order to perform the effectiveness of shell and tube heat exchanger.

CHAPTER THREE
RESEARCH METHOD AND MATERIALS

3.1 Research design and approach

In this section focused on the design methodology and the techniques of the data collection mechanisms. The general fame work of this study is designed based on the objectives or goals of this research to address the research questions. In order to simple and clarify the work, the activities can be classified in to three phases. The first phase concerns on the literature review, and the second is the main or the detail works on research (data collection, design methods, comparison as well as validation analysis and result and conclusions). In the final phase conducts the outcome of the research (conclusions) are carried out.

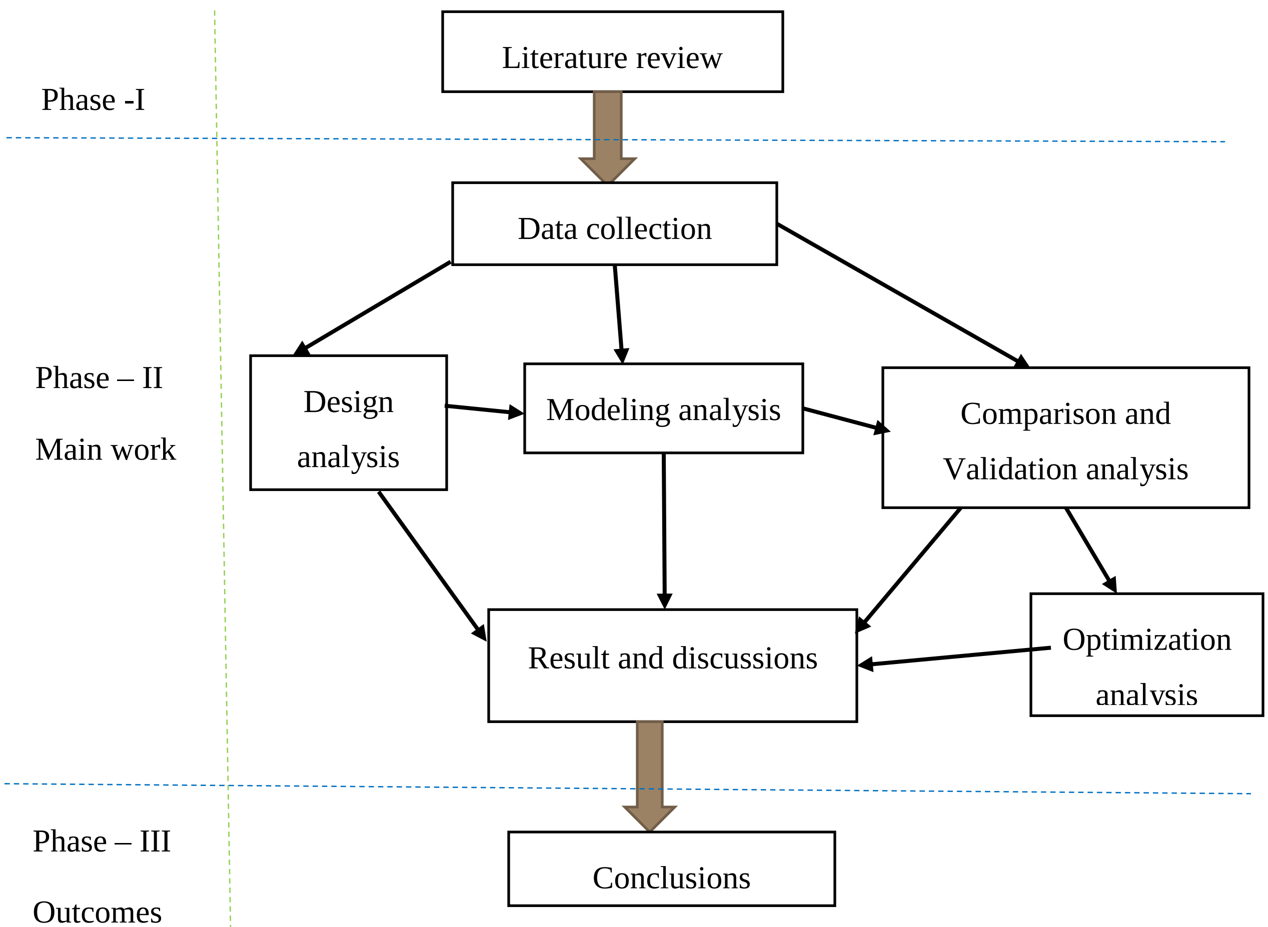


Figure 3.1 General work flow of methodology

To address the research questions, we need to collect the appropriate data using quantitative approach. Because this approach focused on collects the numerical data through structured surveys.

3.2 Research method and analysis

Research method is the systematic approach used to investigate and solve problems related to the objectives of the research. Mostly, in this discipline experimental and analytical methods are adopted [6,24,28]. So this study involves analytical method is conducted, because both mathematical and simulations are carried out to analyze the study.

3.3 Material selections

In this analysis of materials for shell and tube heat exchanger design, there are liquid and solid materials. These fluid substances are heavy oil, to which high-viscosity fluid and saturated steam are assign on the shell or tube side. The solid materials are aluminum for the tube and baffle while carbon steel is in the shell. These materials are the existing STHE for the Dashen brewery industry.

3.4 Mathematical analysis

Shell and tube heat exchangers are designed by using either Kern's in other words logarithmic mean temperature difference (LMTD), or Bell-Delaware which is the effective number of transverse units (E-NTU) analysis method. Kern's method is mostly used for the preliminary design and provides conservative results whereas; the Bell- Delaware method is more accurate method and can provide detailed results. It can predict and estimate pressure drop and heat transfer coefficient with better accuracy [28].

The LMTD method is convenient for sizing problems where at least one of the fluid outlet tempe ratures is a given parameter. It is less well suited for rating problems, where the task is to find the outlet temperatures on the basis of known heat exchanger type and fluids entering the exchanger. However, the inlet or outlet temperatures of the fluid streams are not known, a trial-and-error procedure could be applied for using the LMTD method in the thermal analysis of heat exchangers. The converged value of the LMTD will satisfy the requirement that the heat transferred in the heat exchanger must be equal to the heat convected to the fluid. In these cases, to avoid a trial-and-error procedure, the method of the number of transfer units (NTU) based on the concept of heat exchanger effectiveness may be used. This method is based on the fact that the inlet or exit temperature differences of a heat exchanger are a function of *UA/Cc* and *C c/Ch.*

In this design, logarithmic mean temperature difference (LMTD) design method is used to order thermal analysis depending on the input parameters information. to do this analysis we should follow the design procedures or steps[13,29].

To perform the mathematical analysis of this study, input parameters and thermophysical properties of the fluid are specified.

A. Input parameters

Input parameters are the initial data or specification that used to emphasis the design analysis. In this study the design input parameters are taken from Dashen Brewery Industry listed in table 3.1.

Table 3. 1 Design input parameters (Dashen Brewery Industry)

Parameters	Unit	Hot side/ Tube side Steam	Cold side/ Shell side heavy fuel oil
Inlet temperature	℃	180	25
Outlet temperature	℃	90	70
Max. allowable pressure	bar	20	
Operating (allowable)pressure	bar	10	
Mass flow rate	kg/s	0.02586	

B. Thermophysical properties of fluid materials

In this study the thermophysical properties of fluid materials are known by using Engineering Equation Solver (EES) software. Because the thermophysical properties of heavy fuel oil can vary depending on the specific composition or formulation and grade of the oil being used. Those properties have determined for both tube and shell side fluids at mean temperature and operating pressure of both fluids, tabulated in bellow the table 3.2.

Table 3. 2: Thermophysical properties fluids (steam and heavy fuel oil)

Physical properties	Unit	Steam @T_{mean} = 135 ℃	Furnace Oil (heavy fuel oil) @ T_{mean} = 47.5 ℃
Density, (ρ)	kg/m^3	931	680.4
Thermal conductivity, (K)	w/mk	0.6833	0.122
Specific heat, (c_p)	$J/kg.k$	4269	2309
Dynamic viscosity, ($\mu * 10^3$)	Kg/m s	0.2048	0.4547
Prandtl number, p_r		1.28	8.61

3.4.3 Determining shell and tube heat exchanger capacity without baffle configuration

The design analysis of shell and tube heat exchangers is essentially focused on improving the efficiency of the heat exchanger. To improve the efficiency or capacity of the heat exchanger, we determine the overall heat transfer coefficient U, the overall heat transfer area A, and the average temperature difference ΔT. Because these terms are directly proportional to the efficiency of the heat exchanger. Shell and tube heat exchanger capacities are expressed by heat load Q.

Note: It is used for preheating purposes in the Dashen brewing industry and built without partitions. The heat load was depending on the area and temperature difference of the two fluid streams. So, according to the following mathematical analysis steps, we can determine the area A, and heat transfer, Q of the heat exchanger[31].

Step 1: Determine the rate of heat transfer, Q

The amount or quantity of heat required to raise the temperature of cold fluid in the same manner it is the amount of heat loss from hot fluids.

To determine the rate of heat transfer, we consider the flow condition systems which is steady flow system. In steady flow system the mass flow rate in one inlet and one exit valve must be equal. So, by considering the assumptions we have to apply the energy balance equation.

$$\text{Heat load} = \mathrm{Q} = \dot{m}_h c_{ph}(T_{hi} - T_{ho}) = \dot{m}_c c_{pc}(T_{co} - T_{ci}) \qquad (3.1)$$

Where, Q = heat load or heat transfer rate

$\dot{m}_h$ = Mass flow rate of Steam

$\dot{m}_c$ = Mass flow rate of fuel oil

c_{ph} = Specific Heat for Steam

c_{pc} = Specific Heat for fuel oil

T_{hi} = Inlet Temperature of Steam,

T_{ci} = Inlet Temperature of fuel oil

T_{ho} = Outlet Temperature of Steam

T_{co} = Outlet Temperature of fuel oil

From the saturated steam at mean temperature and operating pressure we can read the thermophysical properties. Furthermore, we can calculate values using Engineering Equation Solver (EES), energy balance equation is stated that in steady state condition energy loss and energy gain are equal:

$$Q_{loss} = Q_{gain} = Q$$

$$\dot{m}_h \, c_{ph}(T_{hi} - T_{ho}) = \dot{m}_c c_{pc}(T_{co} - T_{ci})$$

$$Q = \dot{m}_h c_{ph} \, (T_{hi} - T_{ho})$$

$$Q = 0.02\dot{5}86 \times 4269 \, (180 - 90)$$

$$Q = 9{,}935.671 \text{ W} = 9.935671 \text{ KW}$$

Step 2: Mass flow rate of heavy fuel oil (cold fluid)

From the energy balance equation (3.1) heat loss and heat gain are equal. So the equation becomes:

$$Q_{gain} = \dot{m}_c \, c_{pc}(T_{co} - T_{ci})$$

$$\dot{m}_c = \frac{Q}{c_{pc}(T_{co} - T_{ci})} \qquad (3.2)$$

$$\dot{m}_c = \frac{9.935671}{2309 \, (70 - 25)} = 0.096 \;\; \text{Kg/s}$$

Therefore, the flow rate of shell side fluid is 0.096 Kg/s.

Step 3: Logarithmic mean temperature difference (LMTD),

The logarithmic mean temperature difference is obtained by tracing the actual temperature profile of the fluids along the heat exchanger and is an exact representation of the average temperature difference between the hot and cold fluids.

$$\Delta T_{LMTD} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}} \quad (3.3)$$

$$= \frac{(180 - 70) - (90 - 25)}{\ln \frac{180-70}{90-25}} = 85.54\ ℃$$

The appropriate average temperature difference between the hot and cold fluids is equal to 85.54℃.

Step 4: Determine correction factor, F_t

The correction factor is depending on the geometry of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. In the design of shell and tube heat exchanger, true temperature difference is estimated from the logarithmic mean temperature by applying a correction factor.

The correction factor (F_t) is a function of the shell and tube side fluid temperature and the number of tube and shell passes. It is normally corrected as a function of two dimensionless temperature ratios and also common shell-and-tube heat exchanger configurations.

For one shell pass heat exchanger, the temperature correction factor is plotted in R versus S as in appendix 1, figure A.

$$R = \frac{(T_{hi} - T_{ho})}{(T_{co} - T_{ci})} = \frac{(180 - 90)}{(70 - 25)} = 2.0 \quad (3.4)$$

$$S = \frac{(T_{co} - T_{ci})}{(T_{hi} - T_{ci})} = \frac{(70 - 25)}{(180 - 25)} = 0.29 \quad (3.5)$$

From appendix C, temperature correction factor (F_t) corresponding to R and S is 0.86.

Step 5: Mean Temperature Difference, ΔT_m

$$\Delta T_m = F_t \Delta T_{LMTD} \tag{3.6}$$

Were, ΔT_m = true temperature difference

F_t = temperature correction factor

$$\Delta T_m = F_t \Delta T_{LMTD} = 0.86 \times 85.54 = 73.56℃$$

Step 6: Estimate the overall heat transfer coefficient U.

To estimate the overall heat transfer coefficient U, for heat exchanger design, we will consider the fluid type in both shell and tube side, and identify the purpose. So, the fluids are steam and fuel oil in the hot and cold side respectively used for heating purpose.

From appendix D, the estimated overall heat transfer coefficient (U - values) in steam (hot fluid) to heavy oil (cold fluid) is 60- 450 $W/m^2℃$.

So, in this analysis we can estimate the value of overall heat transfer coefficient U which is 71 $W/m^2℃$ by result of some iteration. After estimating this value, we have to determine the overall surface area of heat exchanger.

Step 7: Determine Provisional heat transfer surface Area

Provisional heat transfer surface area is area covering surface by shell side fluid which is total outside surface area of tubes. It can be determined using the calculated value of rate of heat transfer, Q mean temperature difference, and the estimated value of heat transfer coefficient, U_a

$$A = \frac{Q}{U_a \Delta T_m} \tag{3.7}$$

$$= 9935.671/(71 \times 73.56) = 1.902 \text{ m}^2$$

So, the total outside surface area of tubes is 1.902 m^2.

To calculate the surface area of a single tube, we need to know the outer diameter of the tube. The heat transfer increases with a more compact tube arrangement, which requires a smaller outer diameter to achieve a higher rate of heat transfer. However, the cleaning mechanism for this arrangement is slightly difficult and involves the use of chemical cleaning. Based on this, we have selected an outer diameter (d_o) of 20 mm and an inner diameter (d_i) of 16 mm for the tube. Correspondingly, the length of the heat exchanger is 1.83 m, as provided by Dashen Brewery Industry.

Step 8: The surface area of one tube, $\mathrm{A_0}$

This is the fluid flow area that covers in a single tube surface through the total length of the tube, which is the product of tube diameter with its length.

$$\mathrm{A_0} = \pi d_o \mathrm{L} \tag{3.8}$$

$$= 3.14 \times 20 \times 10^{-3} \times 1.83 = 0.115\ \mathrm{m^2}.$$

Step 9: determine number of tubes($\mathrm{N_t}$): it refers to the number of tubes in the heat exchanger which defined as the ratio of the total outside surface area of tubes (provisional area) to outside surface area of one tube.

$$\mathrm{N_t} = \frac{\mathrm{A}}{\mathrm{A_0}} \tag{3.9}$$

Were, $\mathrm{N_t} =$ number of tubes,

$\mathrm{A} =$ Total outside surface area of tubes, and

$\mathrm{A_0} =$ Surface area of one tube.

$$\mathrm{N_t} = \frac{1.902}{0.115} = 16.5 = 16$$

Step 10: Tube pitch, $\mathrm{P_t}$

Tube pitch is the center-to-center distance between two adjacent tubes and more explain in appendix H from the side view. Tube pitch for triangular tube arrangement can be defined as:

$$\mathrm{P_t} = 1.25 \times \mathrm{d_o} \tag{3.10}$$

$$= 1.25 \times 20 = 25\ \mathrm{mm} = 0.025\mathrm{m}$$

Step 11: Determine bundle diameter, D_b

Then we can obtain the estimate value of bundle diameter D_b from the equation bellow which is an empirical equation based on standard tube layouts. This equation is applicable for triangular and square patterns, but in this paper, we have to select the triangular pitch in order to use triangular tube layout patterns. Because in triangular tube layout patterns the tubes are more compact than other arrangements as a result, it increases heat transfer rate.

$$D_b = d_o \left(\frac{N_t}{K_1}\right)^{1/n_1} \tag{3.11}$$

Where, D_b = bundle diameter in mm, d_o = outside diameter of tubes in mm

K_1 = Constant of specific triangular tube arrangements N_t = number of tubes.

n_1 = Exponent to specific triangular tube arrangements

Both K_1 and n_1 =the constant of convective heat transfer coefficient between the fluid inside the tubes and outside of the tubes in the table 3.3.

In triangular pitch, as the shell side fluid is relatively clean, we use 1.25 triangular pitch [13].

Table 3. 3: constant values of K_1 and n_1 [13]

Triangular pitch, $P_t = 1.25 \times d_o$					
Number of passes	1	2	4	6	8
K_1	0.319	0.249	0.175	0.0743	0.0365
n_1	2.142	2.207	2.285	2.499	2.675
Square pitch, $P_t = 1.25 \times d_o$					
Number of passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

By inserting the value of constants, $K_1 = 0.319$ and $n_1 = 2.142$ from the above table to the equation (3.11) the value of bundle diameter becomes,

$$D_b = 20(\frac{16}{0.319})^{1/2.142}$$

$$= 124.4 \text{ mm} = 0.1244 \text{ m}$$

Step 12: shell diameterD_s: it is the basic parameter to analysis and design of shell and tube heat exchanger which is directly proportional to the area of heat exchanger.

$$D_s = \text{bundle diameter} + \text{shell bundle clearance} \tag{3.12}$$

Where, $D_s =$ the inner diameter of shell

From the appendix B, we can take the value of bundle diametrical clearance, using the bundle diameter 0.1244 m m and outside packed head heat exchanger which is approximately 38 mm.

So,$D_s = \text{bundle diameter} + \text{shell bundle clearance} = 0.1244 \text{ m} + 0.038 = 0.163 \text{ mm} = 0.163\text{m}$.

To set the outer diameter of shell we consider the thickness of the shell. From TEMA standard the thickness of the shell is 9.5 mm for over the 0.33m outer diameter and 11.1mm for over 0.9m outer dimeter. And also less than the 0.33 m outer diameter, indicates that the operating pressure is high [8]. In case of this paper the operating pressure or allowable pressure is 10 bar which is high operating pressure and taken as a minimum thickness of 9.5mm, so the outer diameter of shell is 0.172 m.

Step 14: Area of cross flow

It is the shell side fluid covering area inside heat exchanger, which defined as:

$$A_s = \frac{(P_t - d_o) \times D_s}{P_t} \tag{3.13}$$

$$= \frac{(0.025 - 0.02) \times 0.1624}{0.025} = 0.03248\mathrm{m}^2$$

Step 15: Shell side mass velocity, Gs

So, we can determine the fuel oil mass velocity:

$$\text{Fuel oil mass velocity(Gs)} = \frac{\text{Fuel oil flow}}{\text{cross flow area}} \tag{3.14}$$

$$= \frac{0.096}{0.03248} = 2.96\ \mathrm{Kg/sm^2}$$

Step 16: Shell side velocity($\mathrm{u_s}$): it is the maximum fluid flow velocity in the shell side, which determined as:

$$\text{Fuel oil velocity}(\mathrm{u_s}) = \frac{\text{Fuel oil mass velocity}}{\text{Fuel oil density}} = \frac{2.96}{680.4} = 0.0044\ \ \mathrm{m/s} \tag{3.15}$$

Step 17: Shell equivalent diameter for a triangular pitch arrangement

$$\mathrm{d_e} = \frac{1.1}{\mathrm{d_o}}\left(\mathrm{P_t}^2 - 0.907\mathrm{d_o}^2\right) \tag{3.16}$$

$$= \frac{1.1}{0.02}\left(0.025^2 - 0.907 \times 0.02^2\right) = 0.0144\mathrm{m}.$$

Step 18: Shell – side Reynolds number,

$$\mathrm{R_e} = \frac{\mathrm{Gs}\ \times\ \mathrm{d_e}}{\mu} \tag{3.17}$$

$$= \frac{\mathrm{Gs}\ \times\ \mathrm{d_e}}{\mu} = \frac{2.96\ \ \times\ 0.0144}{4.547\ \times 10^{-4}} = 93.74$$

$R_e < 2300$, Therefore, flow inside shell is laminar flow.

Step 19: Shell side Prandtl number,

$$P_r = \frac{\mu \times C_p}{k_f} \tag{3.18}$$

$$= \frac{0.0004547 \times 2309}{0.122} = 8.61$$

Step 20: Shell side Nusselt number,

$$N_u = \frac{h_o \times d_e}{k_f} = 0.36 R_e^{0.55} P_r^{0.4} (\frac{\mu}{\mu_w})^{0.14} \tag{3.19}$$

$$N_u = 0.36 \times 93.74^{0.55} \times 8.61^{0.4} (\frac{0.0004547}{0.0002048})^{0.14} = 11.57$$

Step 21: Shell side Heat transfer coefficient,

$$h_o = \frac{N_u \times k_f}{d_e} \tag{3.20}$$

$$= \frac{11.57 \times 0.122}{0.0144} = 98.03 \ \mathrm{W/m^2\,°C}$$

Step 23: Number of tubes per pass

It is the number of tubes in each pass,

$$N_{tpp} = \frac{N_t}{N_p} \tag{3.21}$$

Were, N_{tpp} = Number of tubes per pass

N_t = Number of tube

N_p = Number of passes,

Since the flow direction of fluids inside the tubes are passes only in one direction which means there is no back or return flow, the number of passes called single passé ($N_p = 1$).

$$N_{tpp} = \frac{16}{1} = 16$$

Step 24: Tube side mass velocity

$$G_m = \frac{\dot{m}_t}{N_{pp} \times \pi d_i^2/4} \tag{3.22}$$

Were, G_m = Tube side mass velocity

$\dot{m}_t$ = Tube side flow rate

d_i = Inner diameter of tube

$$G_m = \frac{0.02586}{16 \times 3.14 \times 0.016^2/4} = 8.04\ Kg/sm^2$$

Step 25: The average tube side velocity; it is the fluid flow velocity inside the tube and can be determined as:

$$u_t = \frac{G_m}{\rho} \tag{3.23}$$

Were, u_t = Tube side velocity

ρ = Tube side fluid density

$$u_t = \frac{8.04}{931} = 0.0086\ m/s$$

Step 26: Tube side Prandtl number and Reynolds number

$$P_r = \frac{\mu \times C_p}{k_f} \tag{3.24}$$

Were, P_r = tube side prandtl number

C_p = Specific heat capacity of tube side fluid

k_f = thermal conductivity of tube side fluid

μ = tube side fluid viscosity

$$P_r = \frac{2.048 \times 10^{-4} \times 4269}{0.6833} = 1.28$$

Tube side Reynolds number can be determined as

$$R_e = \frac{G_m \times d_i}{\mu} \tag{3.25}$$

$$R_e = \frac{8.04 \times 0.016}{2.048 \times 10^{-4}} = 628.33$$

Since, $R_e < 2100$ the flow becomes laminar.

Step 27: Tube side heat transfer coefficient [28]

The flow of tube side fluid is turbulent then the heat transfer coefficient is calculated as:

$$h_i = 1.86 \frac{k_f}{d_i} (R_e P_r)^{0.33} \left(\frac{d_i}{L}\right)^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{3.26}$$

Were, h_i = heat transfer coefficient for tube side fluid

L = Length of tube

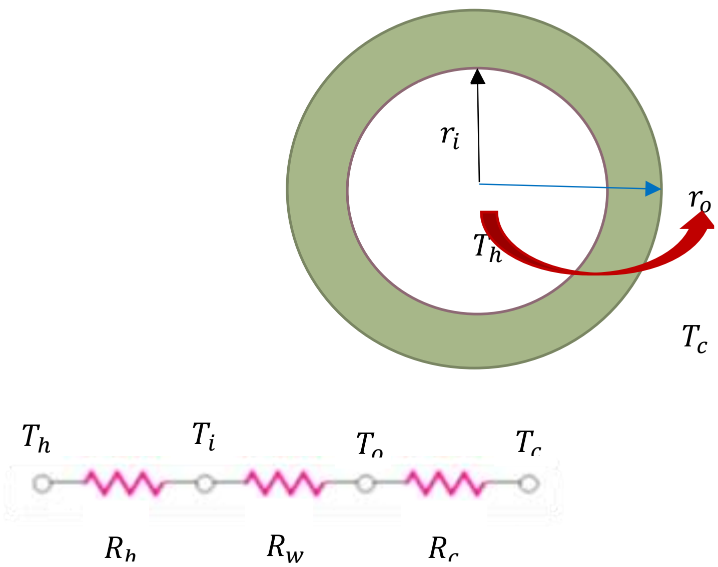
$$h_i = 1.86 \times \frac{0.6833}{0.016} (628.33 \times 1.28)^{0.33} \left(\frac{0.016}{1.83}\right)^{0.33} \left(\frac{0.0004547}{0.0002048}\right)^{0.14}$$

$$= 168.81 \ \mathrm{W/m^2\ ^\circ C}$$

Step 28: Overall heat transfer coefficient

The heat is transfer in radial direction which is flows from hot temperature (tube side) to the cold temperature (shell side). In this process conduction and convection heat transfer mechanism is takes place.

First heat is transferred by hot fluid (steam) to inner wall of tube by convection, then through the wall of the tube by conduction and finally from the outer wall of tube to cold fluid (heavy fuel oil) by convection.



Where, T_h = the tube (hot) side temperature

T_c = the shell (cold) side temperature

r_i, r_o = inner and outer radius of tube respectively

Q = heat transfer rate

Figure 3.2 Thermal resistance of shell and tube heat exchanger

Where, R_h = Thermal resistance through hot fluid, $R_h = {}^{1}/_{h_i A_i}$

R_w = Thermal resistance through tube thickness, $R_w = \frac{\ln({}^{r_0}/_{r_i})}{2\pi kL}$

R_c = Thermal resistance through hot fluid, $R_c = {}^{1}/_{h_o A_o}$

So, the overall heat transfer coefficient is determined as convection – conduction – convection as follows.

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{d_i \ln(d_o/d_i)}{2k_w} + \frac{d_i}{d_o h_o}} \tag{3.27}$$

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{d_o \ln(d_o/d_i)}{2k_w} + \frac{d_o}{d_i\, h_i}} \tag{3.28}$$

Were, $U_i =$ tube side overall heat transfer coefficient

$U_o =$ Shell side overall heat transfer coefficient

$d_i =$ Inner diameter of the tube

$d_o =$ Outer diameter of the tube

$k_w =$ Thermal conductivity of the tube material

$h_i =$ Heat transfer coefficient on the tube side flow

$h_o =$ Heat transfer coefficient on the shell side flow

Substituting those values in equation 3.27 and 3.28 we can determine the overall heat transfer coefficient of heat exchanger.

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{d_i \ln(d_o/d_i)}{2k_w} + \frac{d_i}{d_o\, h_o}}$$

$$= \frac{1}{\frac{1}{168.81} + \frac{0.016 \ln(0.02/0.016)}{2\times 202.4} + \frac{0.016}{0.02\times 98.03}} = 71.5 \quad \mathrm{W/m^2\,°C}$$

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{d_o \ln(d_o/d_i)}{2k_w} + \frac{d_o}{d_i\, h_i}}$$

$$= \frac{1}{\frac{1}{98.03} + \frac{0.02 \ln(0.02/0.016)}{2\times 202.4} + \frac{0.02}{0.016\times 168.81}} = 58.83 \ \mathrm{W/m^2\,°C}$$

Therefore, the tube and shell side overall heat transfer coefficient are $71.5\ \mathrm{W/m^2}$ °Cand $58.83\ \mathrm{W/m^2}$ °Crespectively.

The calculated value of Overall heat transfer coefficient has approximately the same and greater value than the estimated value, so the sizing of heat exchanger is valid.

Step 29: Tube side pressure drop [8]

In a shell and tube heat exchanger, the pressure drop in the tube side determines the loss due to friction and reversal fluid in the side flow. From the above Reynolds number analysis, flow inside the tube is laminar. So, pressure loss due to reversal flow is negligible and the pressure drop in the tube side can determine as follows.

$$\Delta P_t = 4j_f(\frac{L}{d_i})(\rho\, u^2)(\frac{\mu}{\mu_w})^m \quad 3.29$$

Where, $\Delta P_t =$ Tube side pressure drop — $u =$ the average tube side velocity

$j_f =$ Dimensionless friction factor — L = Tube length

$\rho =$ Tube side fluid density — $d_i =$ Inner diameter of tubes

$$m = \begin{cases} -0.25\,, for\ laminnar\ flow \\ -0.14, for\ turbulent\ flow \end{cases}$$

So, $\Delta P_t = 4j_f(\frac{L}{d_i})(\rho\, u^2)(\frac{\mu}{\mu_w})^m$

$$= 4 \times 0.014(\frac{1.83}{0.016})(931 \times 0.0086^2)(\frac{0.0002048}{0.0001024})^{-0.25}$$

Since, $j_f = 0.014$ from the chart in appendix E.

$\Delta P_t = 0.38Pa$, Smaller than operating pressure.

Step 30: Shell side pressure drop of without baffle shell and tube heat exchanger [13,14]

The total shell side pressure drops are the pressure loss due to frictional factor, change in flow direction (return loss) and nozzles. In case of single shell pass flow the return loss becomes zero and the pressure loss due nozzles are negligible. So, the total shell side pressure drops are only the frictional loss and determine as follows.

$$\Delta P_s = f\left(\frac{L}{d_h}\right)\frac{\rho\, u^2}{2} \quad 3.30$$

Where, $\Delta P =$ shell side pressure drop

$f =$ Friction factor, $f = \exp(0.576 - 0.19\, ln\, R_e)$ for laminar flow

So, $f = 1.332$

$d_h =$ Hydraulic diameter, the same as tube inner diameter in circular flow.

$u =$ Shell side average velocity

$$\Delta P_s = 1.332 \times \left(\frac{1.83}{0.016}\right) \times \frac{680.4 \times 0.0044^2}{2} = 1.003\ Pa.$$

It is much smaller value than the allowable operating pressure.

3.4.4 Thermal analysis of heat exchanger with segmental baffle configuration

Those equations are adopted for shell and tube heat exchanger analysis [24].

Step 1: Baffle spacing; it is the distance between each baffle

$$B_s = 0.6 \times \mathrm{D_s} \tag{3.31}$$

$$= 0.6 \times 0.163 = 0.0978\mathrm{m}$$

Step 2: Bundle cross flow area

$$A_s = \frac{(\mathrm{D_s} \times C \times B_s)}{P_t} \tag{3.32}$$

Where, C = tube clearance, $C = P_t - \mathrm{d_o} = 0.025 - 0.02 = 0.005m$

$$A_s = \frac{(0.163 \times 0.005 \times 0.0978)}{0.025} = 3.19 \times 10^{-3}\mathrm{m}^2$$

Step 3: Mass velocity of shell side fluid (G_s)

$$G_s = \frac{\dot{\mathrm{m}}_\mathrm{c}}{A_s} = \frac{0.096}{3.19 \times 10^{-3}} = 30.1\ \mathrm{kg/sm^2}$$

Step 4: the maximum Velocity of shell side fluid

$$u_s = \frac{G_s}{\rho} = \frac{30.1}{680.4} = 0.044 \text{ m/s}$$

Step 5: Reynolds number (R_e)

$$R_e = \frac{\rho_s \times u_s \times d_h}{\mu} \qquad 3.33$$

Where,

$\rho_s =$ Density of shell side fluid $\qquad \mu_s =$ Dynamic viscosity of shell side fluid stream

$u_s =$ Shell side velocity

$d_h =$ Hydraulic diameter of the shell can determine,

$$d_h = \frac{4 \times (A_{shell} - A_{tubes})}{P}$$

Where, A is the cross- sectional area of flow inside the tube

P is the wetted perimeter of flow inside the shell

$$R_e = \frac{\rho_s \times u_s \times d_h}{\mu} = \frac{680.4 \times 0.044 \times 0.1614}{0.0004547} = 10{,}626.63$$

Since, $R_e > 2300$ the flow becomes turbulent flow.

Step 6: Heat transfer coefficient (h_o)

$$N_u = \frac{h_o \times d_e}{k_f} = 0.36 R_e^{0.55} P_r^{0.4} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$N_u = 0.36 \times 10{,}626.63^{0.55} \times 8.61^{0.4} \times \left(\frac{0.0004547}{0.0002048}\right)^{0.14} = 156.07$$

$$h_o = \frac{N_u \times k_f}{d_e} = \frac{156.07 \times 0.122}{0.0144} = 1{,}322.26 \text{ W/m}^2\,°\text{C}$$

Step 7: Number of baffles (N_b)

$$\mathrm{N_b} = \frac{L}{B_s + B_t} \tag{3.34}$$

Were, L =length of tube, B_s = baffle spacing and B_t = baffle thickness

$$\mathrm{N_b} = \frac{1.83}{0.0978+0.015} = 16.22 = 16$$

Step 8: Pressure drop, $\Delta \mathrm{P_s}$ [32]

Pressure drop is an important parameter in heat exchanger design. The heat exchanger should be design in such a way that unproductive pressure drop should be avoided to maximum extent in area like inlet and outlet bends, nozzles and manifolds by reducing the shell side pressure drop the pumping power can be saved, more turbulence can be maintain at shell side resulting in less fouling and higher heat transfer coefficient.

$$\Delta P_s = 8 j_f \left(\frac{\mathrm{D_s}}{\mathrm{d_e}}\right)\left(\frac{L}{B_s}\right)\left(\frac{\rho_s {u_s}^2}{2}\right)\left(\frac{\mu_s}{\mu_t}\right)^{-0.14} \tag{3.35}$$

Were, ΔP_s = pressure drop with segmental baffle configuration heat exchanger

j_f = Friction factor which is 0.05 from the Moody's chart in appendix F.

ρ_s = Density of shell side fluid

$\mathrm{d_e}$ = Equivalent diameter of shell

μ_s = Dynamic viscosity of shell side fluid stream

μ_t = Dynamic viscosity of tube side fluid stream

L = Length of tube or heat exchanger

B_s = Baffle spacing

D_s = Shell diameter

u_s = Shell side velocity

After substituting those value in the above equation 3.32 we can get the shell side pressure drop in segmental baffle heat exchanger.

$$\Delta P_s = 8 \times 0.05 \left(\frac{0.163}{0.0144}\right)\left(\frac{1.83}{0.0978}\right)\left(\frac{680.4 \times 0.044^2}{2}\right)\left(\frac{0.0004547}{0.0002048}\right)^{-0.14}$$

$$= 68.89\ Pa.$$

3.4.5 Thermal analysis of heat exchanger with helical baffle configuration

The following equations are determines the thermal analysis of helical baffles [24].

Step 1: Baffle spacing

$$B_s = \pi \times \mathrm{D_s} \times \tan(8°) \tag{3.36}$$

For optimization analysis the helix angle have specified from 8° to 12°. So we select the inclination angle analysis with initially from this angle value.

$$B_s = 3.14 \times 0.163 \times \tan(8°) = 0.072m$$

Step 2: Bundle cross flow area

$$A_s = \frac{(\mathrm{D_s} \times C \times B_s)}{P_t}$$

$$= \frac{(0.163 \times 0.005 \times 0.072)}{0.025} = 2.35 \times 10^{-3}\mathrm{m}^2$$

Step 3: Mass velocity of shell side fluid (G_s)

$$G_s = \frac{\dot{\mathrm{m}}_\mathrm{c}}{A_s} = \frac{0.096}{2.35 \times 10^{-3}} = 40.85$$

Step 4: Maximum Velocity of shell side fluid

$$u_s = \frac{G_s}{\rho} = \frac{40.85}{680.4} = 0.06\,\mathrm{m/s}$$

Step 5: Reynolds number ($\mathrm{R_e}$)

$$\mathrm{R_e} = \frac{\rho_s \times u_s \times \mathrm{d_h}}{\mu} = \frac{680.4 \times 0.06 \times 0.1614}{0.0004547} = 14{,}490.86$$

Step 6: Heat transfer coefficient ($\mathrm{h_o}$)

$$N_u = \frac{h_o \times \mathrm{d_e}}{k_f} = 0.36\mathrm{R_e}^{0.55}\mathrm{P_r}^{0.4}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$N_u = 0.36 \times (14{,}490.86)^{0.55} \times (8.61)^{0.4} \times \left(\frac{0.0004547}{0.0001846}\right)^{0.14} = 187.81$$

$$h_o = \frac{N_u \times k_f}{d_e} = \frac{187.81 \times 0.122}{0.0144} = 1{,}591.17\ \mathrm{W/m^2\ ^\circ C}.$$

Step 7: Number of baffles

$$N_b = \frac{L}{B_s + B_t} \tag{3.37}$$

Where, $L =$ length of tube, $B_s =$ baffle spacing and $B_t =$ baffle thickness

$$N_b = \frac{1.83}{0.072 + 0.015} = 21$$

Step 8: pressure drop

By applying the above pressure drop formula (equation 3.35) we can also determine the shell side pressure drop of shell and tube heat exchanger constructed with in helical baffle.

$$\Delta P_s = 8 j_f \left(\frac{D_s}{d_e}\right)\left(\frac{L}{B_s}\right)\left(\frac{\rho_s {u_s}^2}{2}\right)\left(\frac{\mu_s}{\mu_t}\right)^{-0.14}$$

$$j_f = 1.58\,[\ln(R_e - 3.28)]^{-2} = 1.58[\,\ln(1293.68 - 3.28)]^{-2} = 0.017$$

$$\Delta P_s = 8 \times 0.017\left(\frac{0.163}{0.0144}\right)\left(\frac{1.83}{0.072}\right)\left(\frac{680.4 \times 0.06^2}{2}\right)\left(\frac{0.0004547}{0.0002048}\right)^{-0.14}$$

$$= 59.16\ Pa$$

As result of mathematical analysis the pressure drop with helical baffle STHE is less than segmental baffle STHE.

NB: From the above shell side convection heat transfer coefficient analysis of STHE constructed with segmental and helical baffle have significantly increase from the without baffle STHE. It is a result of turbulence cases, baffles are significantly diverts the flow of fluids and makes turbulence then, the ratio of inertia force to viscous force (R_e) becomes high, crosspondly the nusselt number was increases as a result the heat transfer coefficient (h_o) is increase.

3.5 Modeling (simulation) Analysis

This section is focus on the modeling of the study regarding with the governing equation and the CFD analysis process flow. Governing equations are non-linear partial differential equation which governs the flow of the fluid inside the fluid domain. Generally the governing equation expresses the conservation of mass, momentum and energy in a fluid of heat exchanger analysis. Which states that the total (mass, momentum and energy) of the fluid entering in a control volume must equal to the (mass, momentum and energy) leaving the control volume. This equation is used to model the flow of fluid through the heat exchanger and to calculate the temperature distribution and heat transfer rate. Those are:

Continuity equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \tag{3.38}$$

X – Momentum equation

$$\nabla.(\rho uV) = -\frac{\partial P}{\partial x}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z} \tag{3.39}$$

Y - Momentum equation

$$\nabla.(\rho uV) = -\frac{\partial P}{\partial y}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z} \tag{3.40}$$

Z - Momentum equation

$$\nabla.(\rho uV) = -\frac{\partial P}{\partial z}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z} \tag{3.41}$$

Energy equation

$$\nabla.(\rho eV) = -P\nabla V+\nabla(K\nabla T)+\emptyset \tag{3.42}$$

Where, u, v, and w, are the velocity components of x, y, and z direction

V is the velocity vector, ∇ is the del operator and ∅ is dissipation rate.

P, T and K are pressure, temperature and thermal conductivity of the fluid of the fluid respectively.

Modeling analysis also a mandatory check of the system, beginning with the generation of the geometry and ending with the spatial results using a computer program [32, 33]. To model the heat exchanger using CFD properly, we follow the appropriate CFD flow chart shown below.

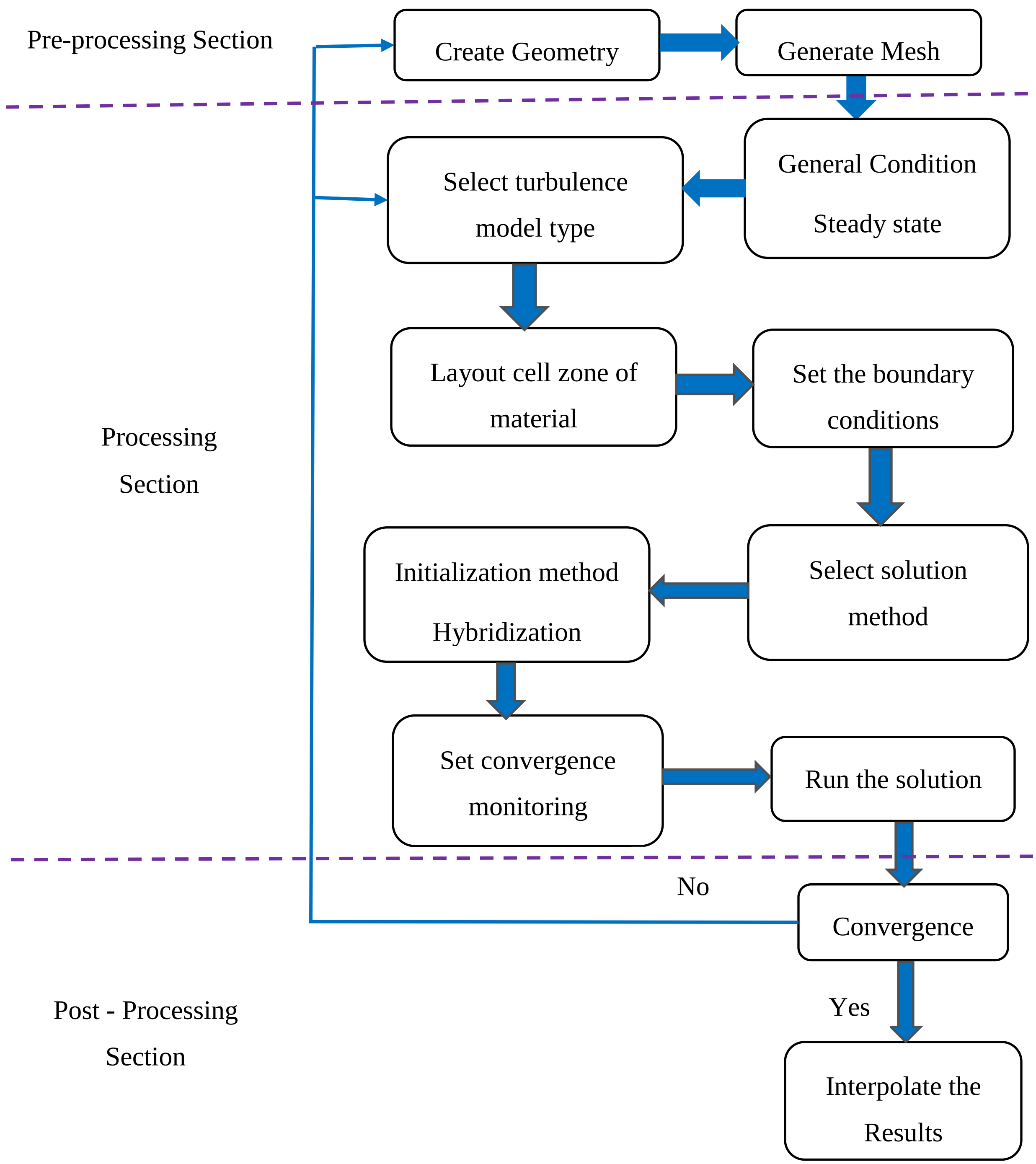


Figure 3.3 General procedures of CFD simulation work

This typically looks at fluid flow design work to delineate shell and tube heat exchangers from fractional baffles and create helical baffles. In the case of this fragment, it is also difficult to reconstruct this heat exchanger with a CFD part. This shows that it takes time to capture simulation results. So, in order to organize questioning it and cut the additional time short, we have to use small judgments. The geometric details shown in table 3.4 below can be clear, geometrically clear to achieve the research purpose.

3.5.1 Geometry Construction

Using the ANSYS Fluent Design Module of ANSYS 19.2, a Shell and Tube Heat Exchanger of 0.6 m length, 0.09 m shell diameter, seven tubes, and six baffles is built in this study. Additionally, the tube's inner and outer diameters are, respectively, 0.017 and 0.02 meters. In comparison to square or other types of tube arrangements, a shell and tube heat exchanger's tube layout with a triangle (45°) tube arrangement pattern is preferable [35]. This configuration uses an E type heat exchanger with a single shell and tube pass. The detailed modeling requirements are shown below table 3.4 [36].

Table 3. 4 Modeling specification of segmental and helical baffle shell and tube heat exchanger [5]

Specification variables	Value
Shell and tube heat exchanger length, L	0.60 m
Shell diameter, D_s	0.09 m
Inner tube diameter, d_i	0.017 m
Outer tube diameter, d_o	0.02 m
Number of tube, N_t	7
Tube pitch, P_t	0.025 m
Tube pattern angle	Triangular, 45°

Number of baffles, N_b	6
Baffle thickness, B_t	0.015 m
Baffle spacing, B_s	0.085 m
Baffle cut for segmental baffle	25%
Inlet and outlet header length	0.05
Inlet and outlet nozzle diameter of shell, d_s	0.04
Inlet and outlet nozzle diameter of tube,d_t	0.03
Helix angle, α	8°, 10°, 12°,

To study the simulation characteristics, like mesh independent test, mesh quality, comparison analysis of three STHE types (without baffle, segmental baffle and helical), the geometry shall be simple and save computational time in CFD analysis to get a better results. And after identifying the comparison result, in order to optimize the helical baffle we select the helix angle variation since the helix angle variation, the number of helical baffles are also changed.

The three dimensional feature of shell and tube heat exchanger in the above specifications are shown in the figure below.

The size or dimensions of three STHEs are similar except the baffles orientations and types.

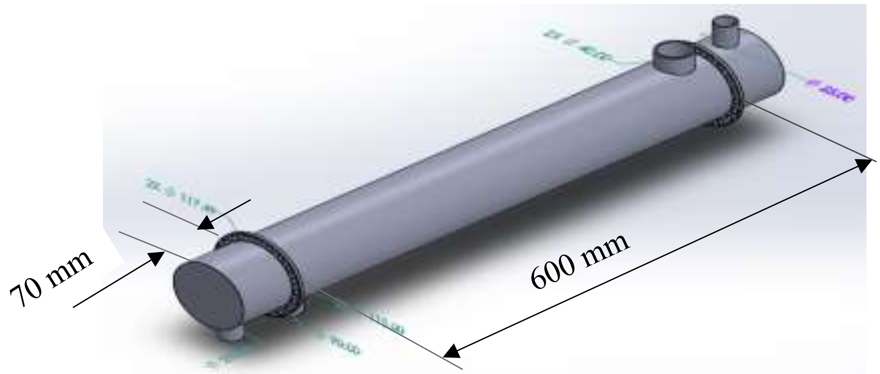


Figure 3.4 3-D Model of helical baffle shell and tube heat exchanger

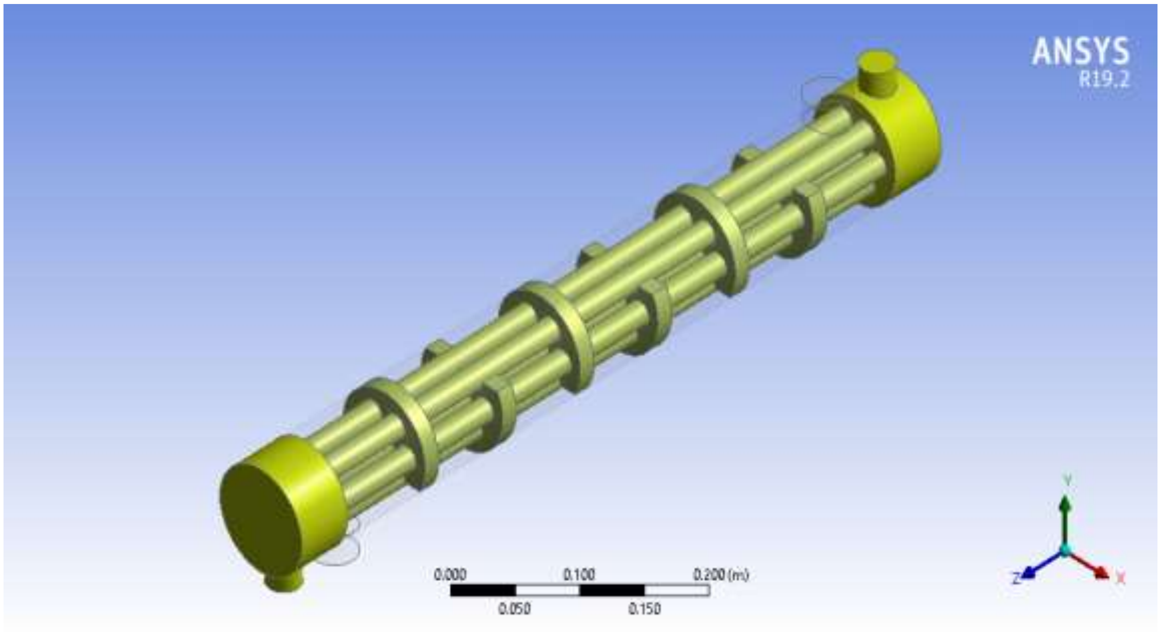


Figure 3.4 A) Simulation model of segmental baffle shell and tube heat exchanger

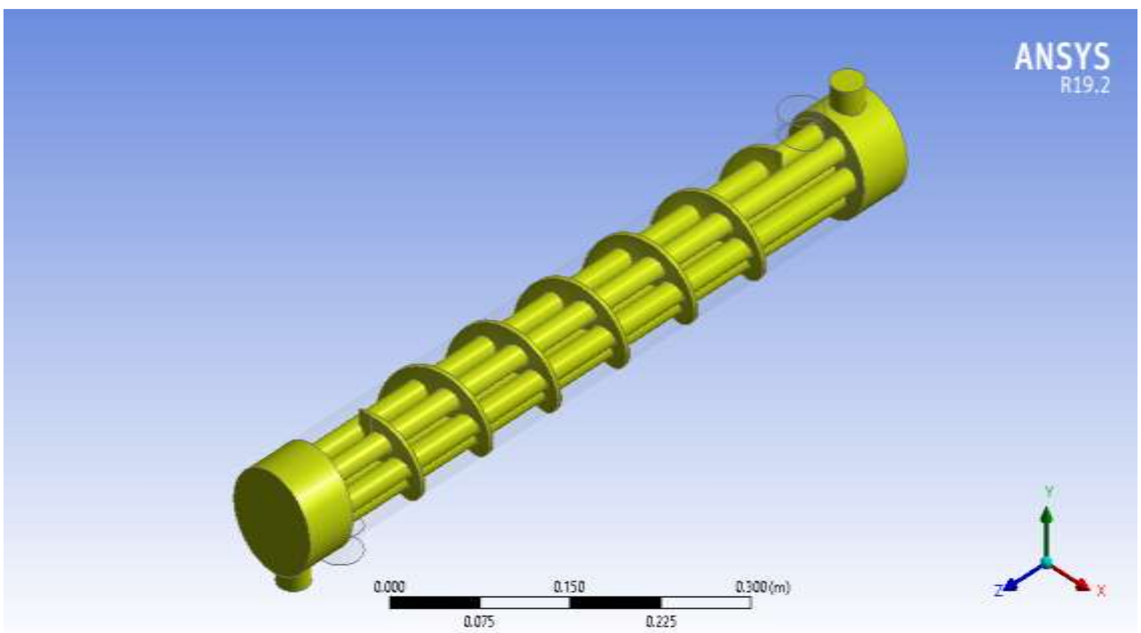


Figure 3.4 B) Simulation model of helical baffle shell and tube heat exchanger

3.5.2 Mesh Criteria

Meshing is the process of discretization a model or geometry in to elements to perform an accurate simulation result.

3.5.2.1 Mesh type

Tetrahedral element meshing is used in this model's meshing in order to have a complex geometry, particularly in the helical baffle arrangement. Tetrahedral and hexahedral mesh element types are employed in shell and tube heat exchanger design study, but tetrahedral element meshing type is the most common [35].

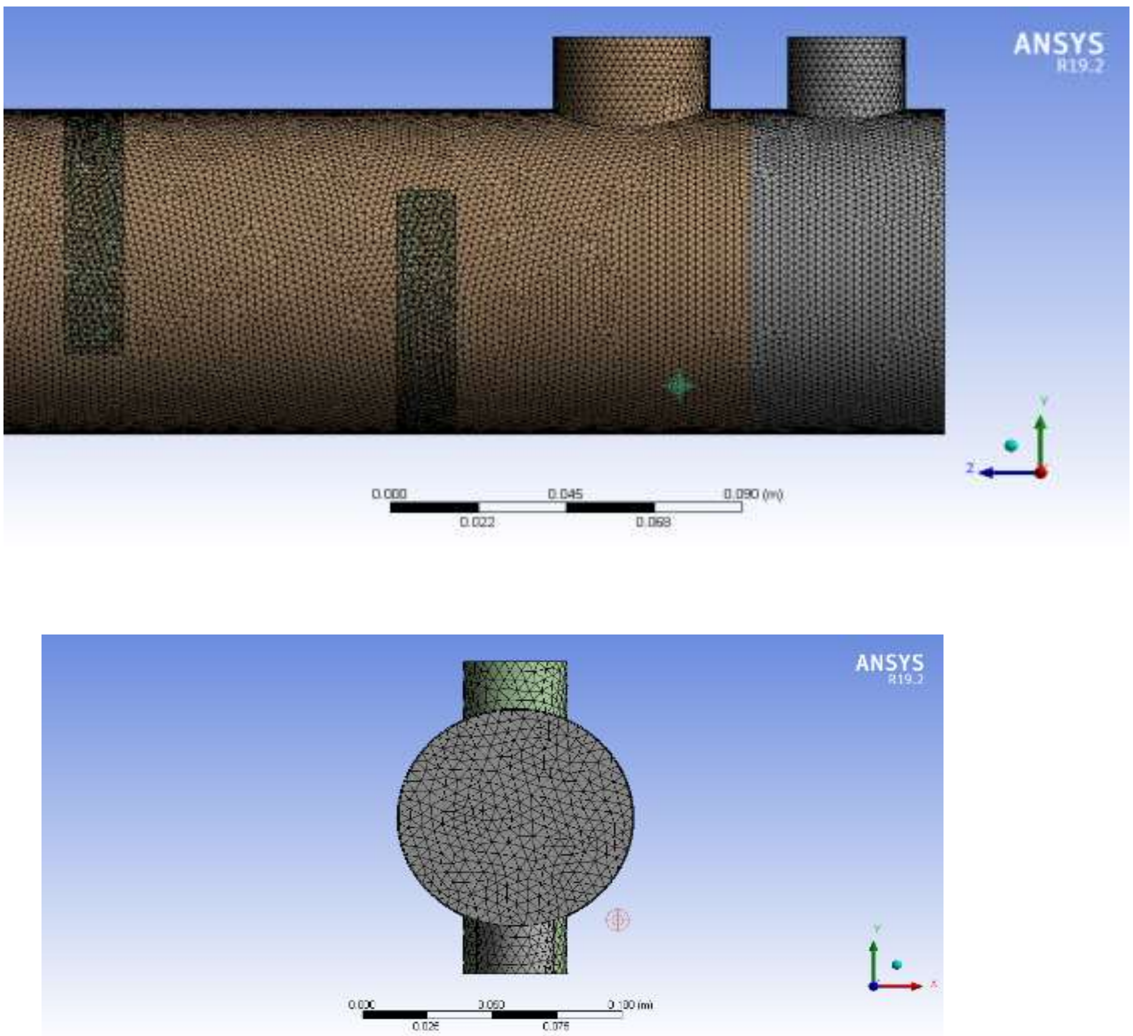


Figure 3.5: A) Fine mesh for segmental baffle shell and tube heat exchanger

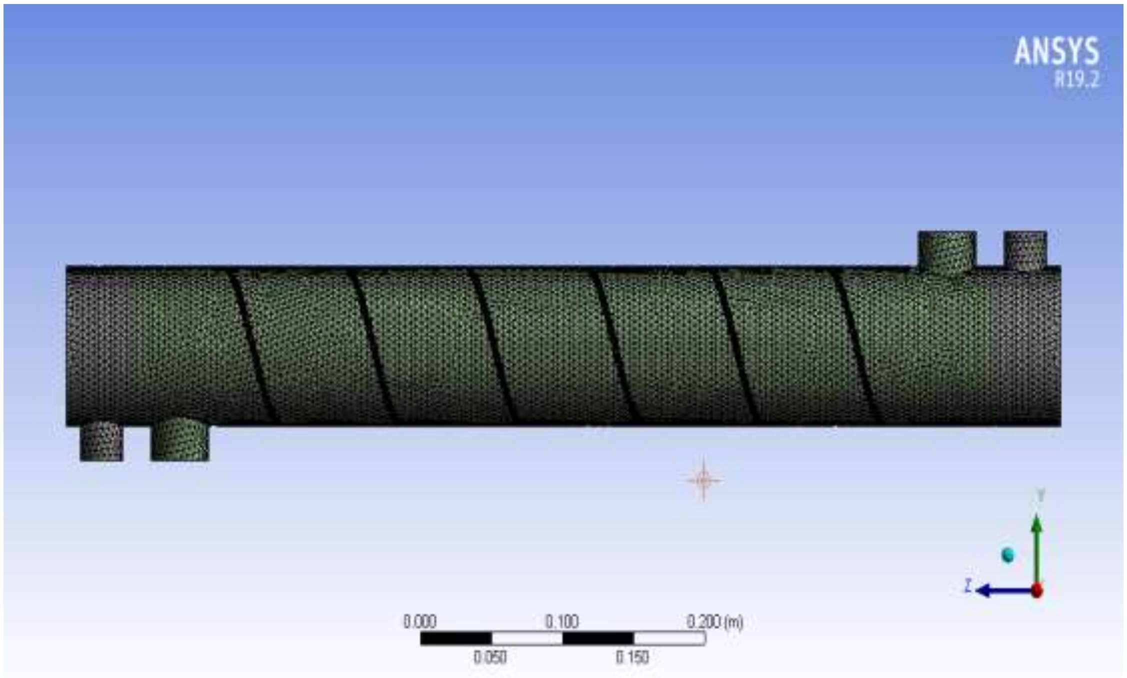


Figure 3.5: B) Fine mesh for helical baffle shell and tube heat exchanger

3.5.2.2 Mesh independence test

It is the process of determining the characteristics of a result on the dependence of the mesh density. Mesh density means the mesh quality of an object with different element sizes. The quality of the mesh increases with decreasing the element size .so the mesh independence test identifies the element size, and the result becomes constant or independent of the element size. For this study to check the grid independence we have to select the element size from the default (coarse) to $1/8^\wedge th$ of the default size (very fine). As shown in the figure, approximately we have similar results in different element sizes. But for a better result achievement and minimizing computational time, we select the finer mesh size.

The mesh independent have also consider the number of nodes and elements in each element size. So in this analysis the number of nodes and elements are recorded in the table 3.5 bellow.

Table 3. 5 Number of nodes and elements

Element size	Number of nodes	Number of elements	Mesh quality
0.6089725	289 966	1 214 937	Coarse
0.017945	289 998	1 215 456	Medium
0.0044625	300 644	1 265 872	Finer
0.00223125	636 767	2 881 873	Finest

Table 3. 6 Result on mesh independent test

Flow rate (kg/s)	Shell side outlet temperature (K) @ a given inlet hot temperature and different flow rate			
	Coarse (Default size)	Medium (1/2 Default size)	Finer (1/4 Default size)	Finest (1/8 Default size)
0.01293	356.07	357.32	358.40	358.94
0.02586	389.04	390.26	391.84	392.06
0.05172	458.23	459.31	460.00	460.62

As we observe from the table above 3.6 and figure 3.5 C the mesh quality of coarse medium finer and also finest have an approach value. If the mesh quality increases the solution becomes convergence. In this case the solution was significantly does not dependent on the mesh density. But to get more accurate result we select finer mesh quality and it doesn't also need to mesh with finest.

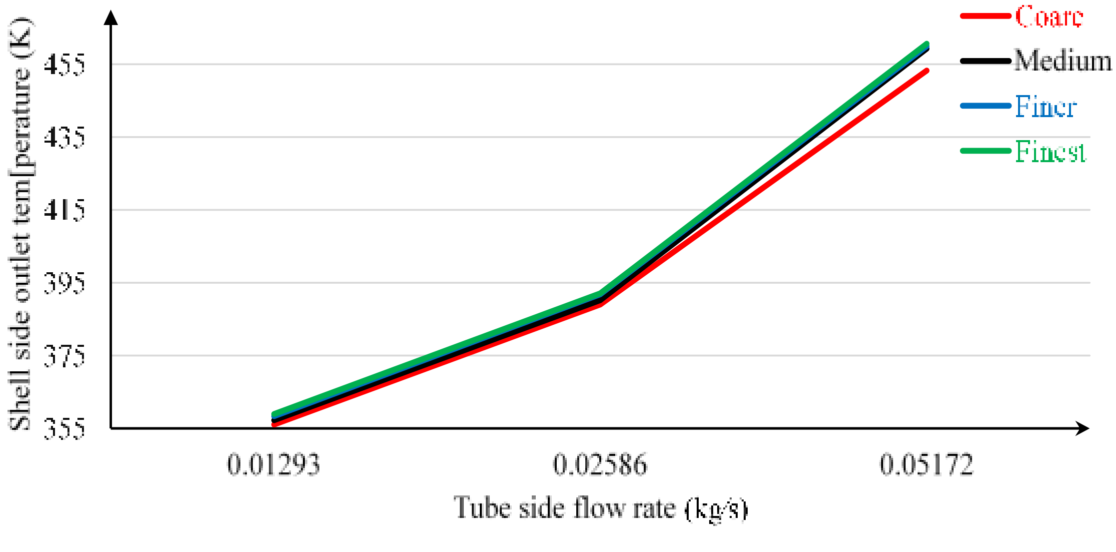


Figure 3.5 C) Mesh independence test

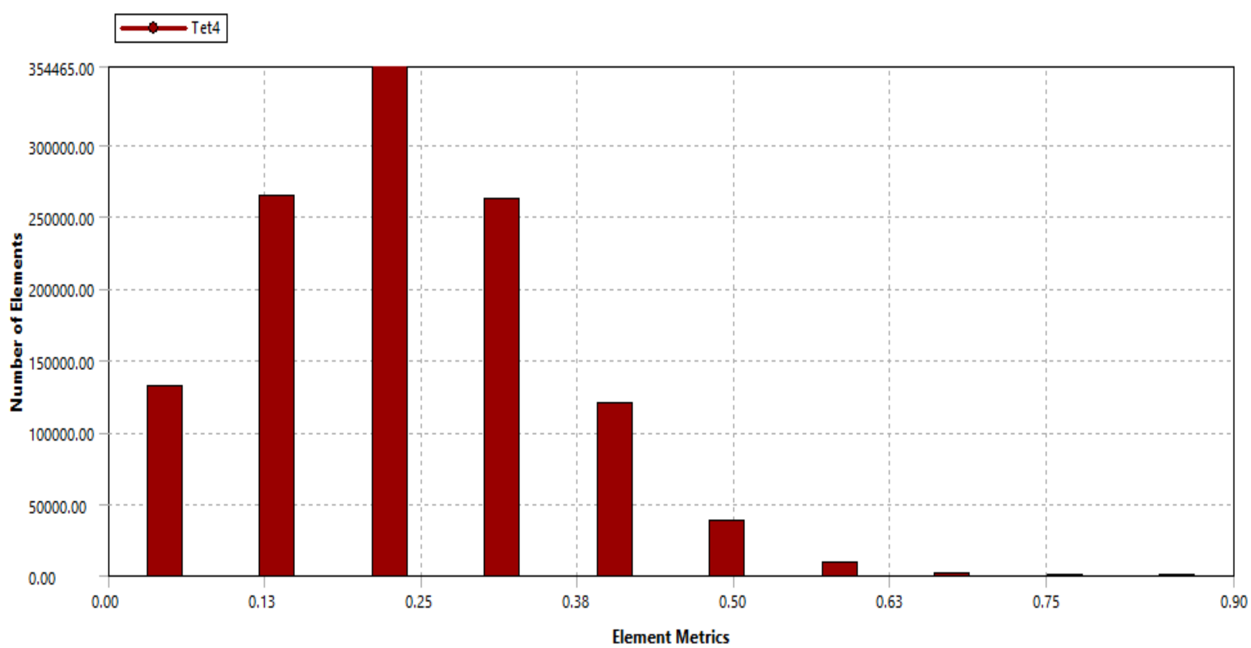


Figure 3.5: D) Mesh metrics quality of shell and tube heat exchanger

3.5.3 Setup Condition

For CFD analysis to generate the desired or predicted result, this section is a necessary. It looks into various setup circumstances.

3.5.3.1 General Condition

It is evident that the design analysis is based on pressure, density, and whether or not time is a factor. The design analysis in this work is based on pressure and steady state conditions. Heat exchanger analysis typically includes a pressure factor when it is pressure-based. Steady state circumstances reveal that it is not impacted by the passage of time. Because it's time independent.

3.5.3.2 Turbulence model type

This section introduces the model study of the fluid flow in the heat exchanger that is turbulent. In CFD, there are various turbulence models, including SpalartAllmaras, k-εstandard, and k-εrealisable. It is advised for turbulent viscosity since the SpalartAllmaras model is used or specified on flow models that only solve one transport equation. It has been demonstrated to produce good results for boundary layers subjected to adverse pressure change since wall-bounded flows were specifically intended for aeronautical applications.

When the flow occurs under convection and diffusion conditions that affect turbulence, the k-ε model is provided. It focuses on the system affecting the kinetic energy of turbulent flow. In general, it only applies to continuous flows with single-phase conditions and turbulent flow conditions. Thus, it appears that the standard and the realizable k-ε model are the two most relevant for the study of shell and tube heat exchangers. Therefore, in order to frame as a general purpose model, we must use the standard k-ε model [3, 6, 28, 37].

3.5.3.3 Materials of geometry domain

We configured the working fluid and solid domains in this part to carry out the heat exchanger study. Heavy fuel oil, saturated steam, and aluminum are the working fluids in this study's fluid and solid domains, respectively. These materials were chosen from the Dashen brewing industry.

3.5.3.4 Cell zone conditions

It supports the fluid stream allocation in the heat exchanger study. Therefore, in this research, saturated steam is in the tube side stream and heavy fuel fluid is assigned to the shell side stream. Also labeled in the solid domain is aluminum.

3.5.3.5 Boundary conditions

Boundary conditions are an essential component of simulation setup. This is due to the fact that boundary conditions reflect the influence of additional components or structures that were not directly modeled in our investigation. It also incorporate the design assumptions. The basic boundary conditions for a shell and tube heat exchanger analysis are fluid flow direction, flow conditions, and inlet and output parameters (flow rate or velocity, temperature, and pressure of the two fluid streams). Therefore, in this study, we conduct the counter flow direction under steady state conditions while assuming that the shell is made of an insulated material. Additional details about boundary conditions are included in the table below.

Table 3. 7 Boundary condition for simulation

	B.C Type	Tube side	Shell side
Inlet	Mass – flow -inlet	0.02586 kg/s	0.096 kg/s
Outlet	Pressure - outlet	0	0
Temperature	Inlet Temperature	453 K	298 K

3.5.4 Solution setup

According to the turbulence models, discretization plays an important role. Here the SIMPLE algorithm for calculating the pressure and the velocities is selected. The momentum equations must be calculated sequentially. In the SIMPLE algorithm, an initial guess is made for the pressure and velocity components to discretize the momentum equation. In the second step, it solves the pressure correction equation. After correcting for the pressures and velocities, the algorithm jumps to solve all other discretized transport equations. Otherwise, when the simulation reaches convergence, it stops and another iteration follows. For the heat exchanger analysis, a first as well as a second order discretization scheme is alternatively implemented. Both first and second order discretization are carefully chosen for kinetic energy, momentum and dissipation rate. Choosing first order discretization offers the advantage of better convergence compared to second order, while second order discretization reduces the discretization error. Depending on this, we chose SIMPLE and first-order discretization in this model analysis.

The convergence condition and initialization procedure must be expressed in this section.

3.5.4.1 Initialization method

Initialization is the process of giving the flow variables their initial values and starting the solution with those values. In ANSYS Fluent, there are two different types of initialization methods: standard initialization methods and hybrid initialization method. Their primary differences are Standard initialization just assigns constant values to the field characteristics, but hybrid initialization solves a number of iterations of a reduced equation system and, as a result, typically yields a more accurate prediction of the flow variables, particularly the pressure field.

3.5.4.2 Convergence criteria

The convergence requirement called for residuals of 0.001 in all continuity, velocities, turbulent viscosity, and disputation rate, but 0.00001 for energy in order to converge quickly. After several iterations, the solution converged, thus we must now interpolate the results, which is conducted in the chapter 4 [1].

CHAPTER FOUR

RESULT AND DISCUSSION

Different results from CFD simulation have to be configured. The study looked at the impact of baffles, particularly helical baffles. We also need to simulate the typical segmental shell and tube heat exchanger as we investigate the comparative study. The heat exchanger model's temperature and pressure differences are shown in the figure and table below.

4.1 Comparison of Segmental and Helical baffle shell and tube heat exchanger

As we investigate the comparative analysis, we incorporate the result from the figure and table. Those figures shown in terms of temperature and pressure contour as well as streamline.

4.1.1 Shell and tube heat exchanger without baffle

As we observed in the temperature contour figure the outlet temperature of shell side fluid was heated from 298 K to 371 K. Similarly, the tube side temperature condenses from 453 K to 314.26K. As a result of this the effectiveness and heat transfer rate becomes 89.51 % and 15.763 KW respectively.

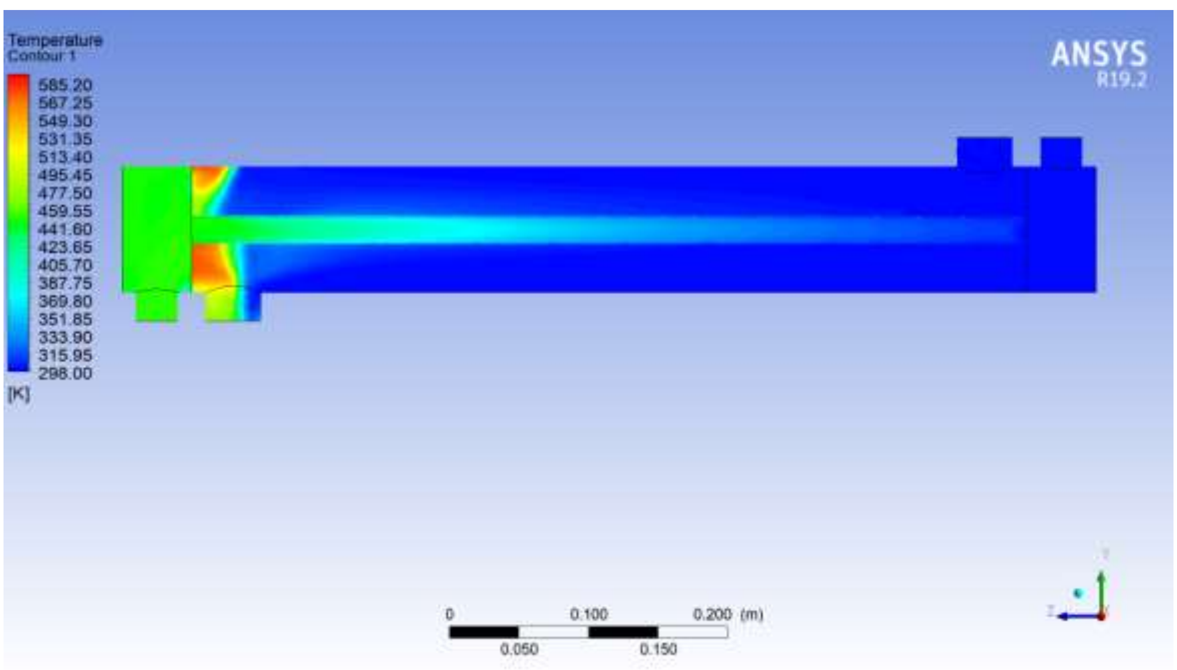


Figure 4.1 A) temperature contour of without baffle shell and tube heat exchanger

From the pressure contour figure, there was high amount of pressure in the inlet and similar value in out of inlet and outlet valve and the pressure drop in this heat exchanger was 7.02 Pa.

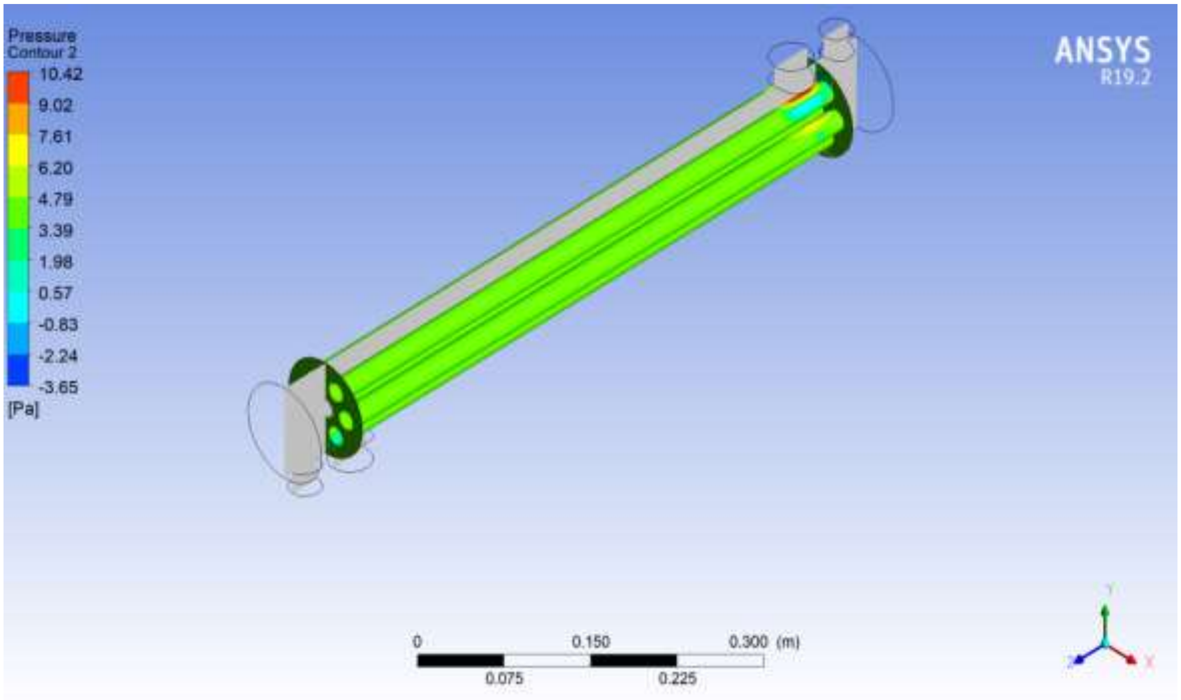


Figure 4.1 B) Pressure contour of without baffle shell and tube heat exchanger

As we observed from the velocity stream line in the figure 4.1B, there was a high velocity at the nozzle or inlet and outlet valves is both shell and tube sides.

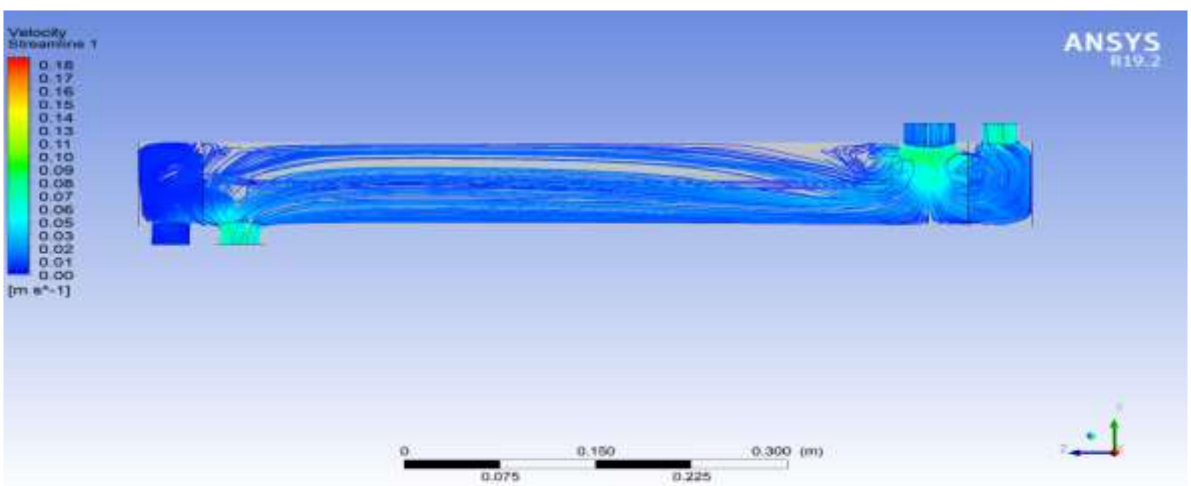


Figure 4.1 C) Velocity stream line of shell and tube heat exchanger without baffles

4.1.2 Shell and tube heat exchanger with segmental baffle

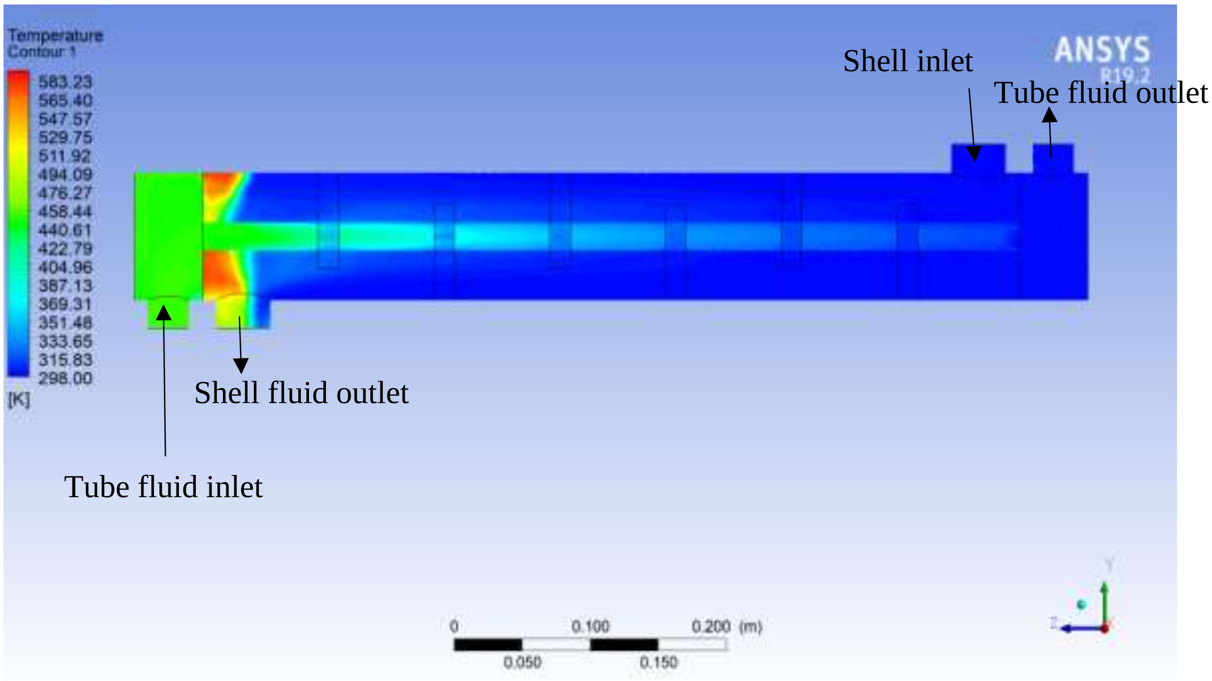


Figure 4.2 A) Temperature contour of segmental baffle shell and tube heat exchanger

According to the figure and table 4.1, the segmental baffle shell and tube heat exchanger's shell side outlet temperature is 375.37 k at a given inlet mass flow rate and temperature. As a result of this the capacity of heat exchanger and heat transfer effectiveness have 16.835 KW and 96.54 % respectively. In this heat exchanger the heat transfer rate was 6.8 % higher than that of the heat exchanger without a baffle and 7.85 % of heat transfer effectiveness.

And also, in the pressure drop analysis for segmental baffle STHE, shows in the figure 4.2 B gives the result of 7 pa. As we compared with STHE without baffle configuration they have almost similar value. Also the outlet pressure is less than the inlet nozzle in both of shell and tubes.

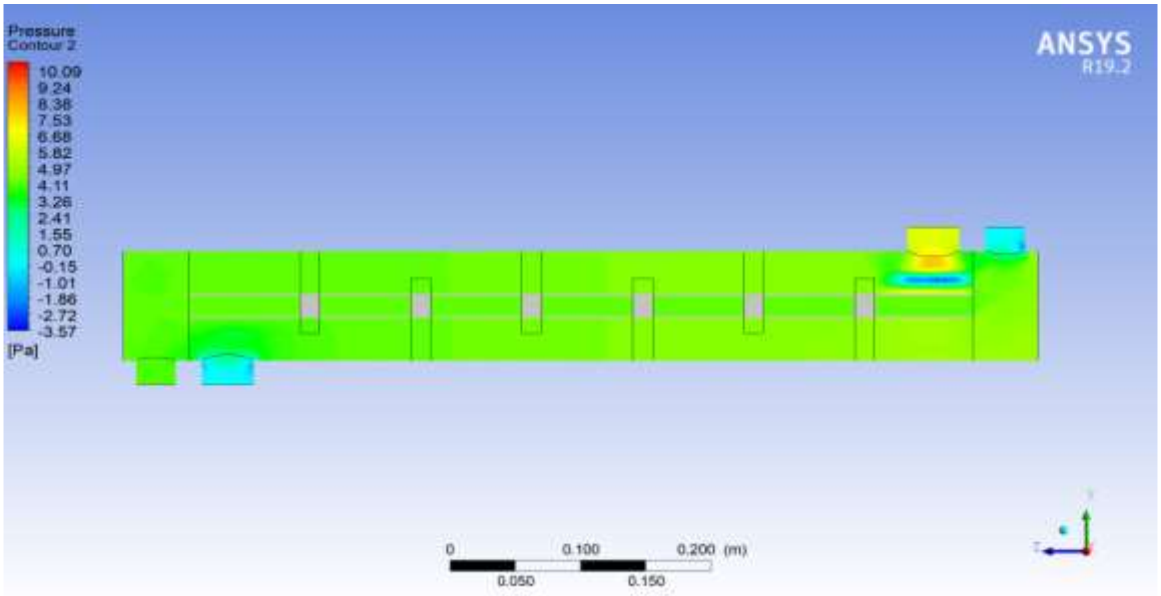


Figure 4.2 B) Pressure contour of segmental baffle shell and tube heat exchanger

The velocity stream line of segmental baffle STHE are creates up and down or zig zag movements as we observe from the figure 4.2 C bellow. The velocity distribution through surface of the heat exchanger are less than the inlet and the outlet of nozzles.

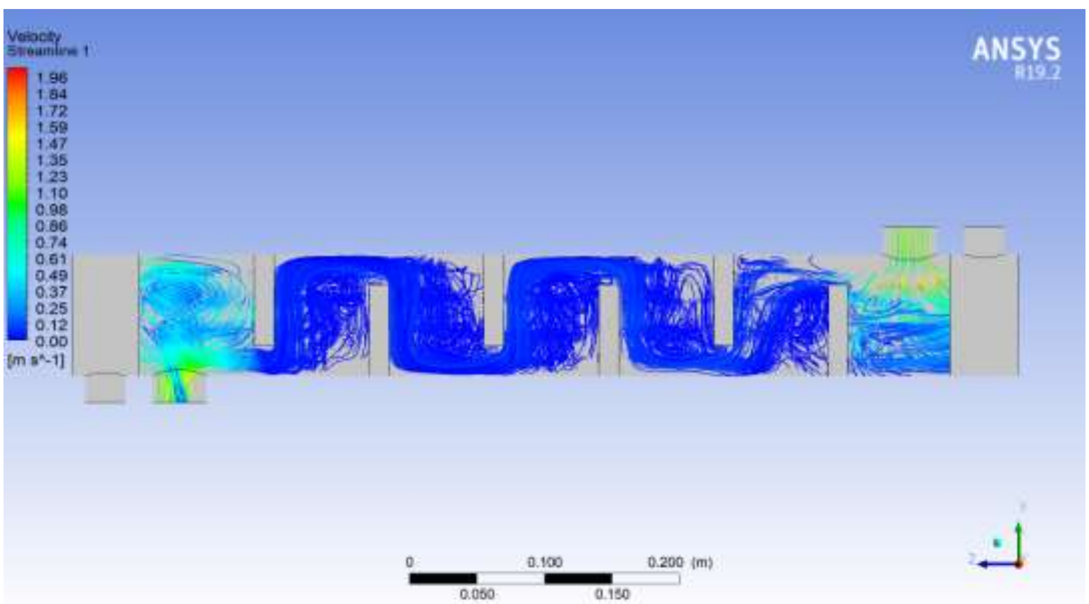


Figure 4.2 C) Velocity stream line of shell and tube heat exchanger with segmental baffles

4.1.3 Shell and tube heat exchanger with helical baffle

From the modelling analysis of helical baffle STHE regarding with the given input parameters (boundary conditions) we have recorded the output results.

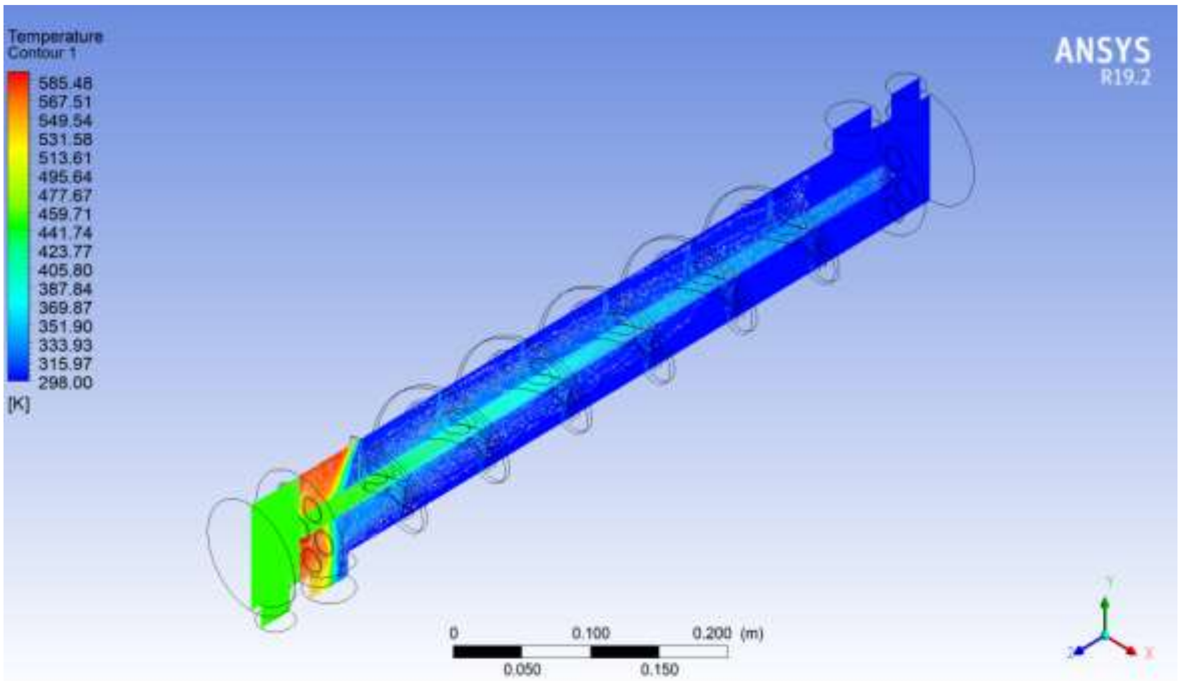


Figure 4.3 A) Temperature contour of helical baffle shell and tube heat exchanger

From the figure 4.3 A) shows the temperature distribution through the heat exchanger is different. The temperature in the shell side is rises to 392.06 k from 298. Whereas the tube side reduces from 453 k to 298.95 k. It shows that the heat transfer rates is different through the whole length and have greater value at the middle of heat exchanger due to counter flow heat exchanger. In this heat exchanger the rate of heat transfer is 18.928 KW and the heat transfer effectiveness becomes 99.28 %.

To conclude the temperature contour, as we observe in the figure 4.1 A, 4.2 A and 4.3 A at the left side or the tube fluid inlet and shell fluid outlet region have a high temperature due to the low flow rate of tube fluid relative to the volume of tube head cover. And also the tube sheet conducts heat from accumulated inlet header fluid to the shell fluid.

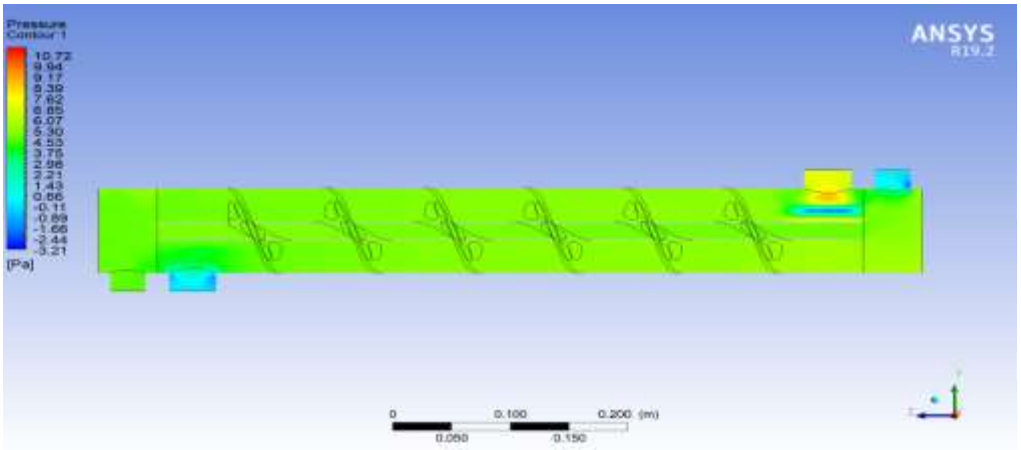


Figure 4.3 B) Pressure contour of helical baffle shell and tube heat exchanger

According to the contour diagram of pressure, in figure 4.3 B) we can see that the profile of pressure. In both streams which is shell and tube side the pressure is build up at the inlet and reduces trough the outlet fluid stream. In helical baffle STHE the pressure drop was reduces by 7% from the STHE without baffle configuration. In order to emphasis the comparison analysis of segmental and helical baffle STHE was illustrated in the figures bellow.

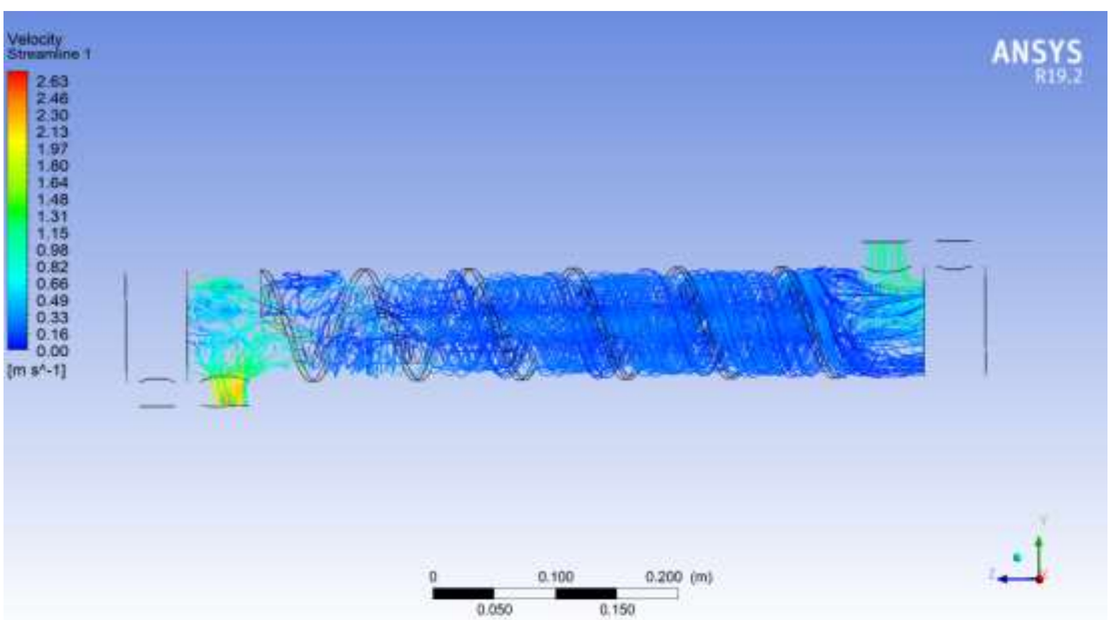


Figure 4.3 C) Velocity stream line of shell and tube heat exchanger with helical baffles

As we observe from the above figure 4.3 C the velocity distribution have makes a swirl motion through all the surfaces of the outer tube which reduces the dead zone as compared to the segmental baffle STHE.

Table 4. 1 CFD Result of segmental and helical baffle shell and tube heat exchanger with different mass flow rate

Output parameters	Segmental baffle			Helical baffle		
	0.01293	0.02586	0.05192	0.01293	0.02586	0.05192
Shell side outlet Temperature (T_{co}), K	358.94	375.37	455.42	360.46	392.06	460.62
Tube side outlet Temperature (T_{ho}), K	298.15	303.36	301.12	298.03	298.95	299.11
Heat transfer effectiveness (E), %	99.74	96.54	97.90	99.9	99.28	99.28
Shell side pressure drop , Pa	5.96	7.00	9.42	4.89	6.95	7.74

To analysis the comparison of the two heat exchanger type by the effectiveness versus flowrate, we demonstrate the effectiveness of heat exchanger. Effectiveness of heat exchanger introduces the efficiency of heat exchanger [38].

$$\varepsilon = \frac{Q}{Q_{max}} \tag{4.1}$$

Where, Q = the product of C_{min} and there cross ponding temperature difference. And C_{min} is the minimum value of product of flow rate and specific heat capacity of fluid stream. Also Q_{max} is the product of C_{min} and the maximum temperature difference in the heat exchanger. So,

$$C = \dot{m}_h c_{ph} \text{ , } or \text{ } \dot{m}_c c_{pc} \tag{4.2}$$

If $\dot{m}_h c_{ph} < \dot{m}_c c_{pc}$ then $C_{min} = \dot{m}_h c_{ph}$ and the reverse is true.

In this study C_{min} is equal to $\dot{m}_h c_{ph}$, so we can analysis the effectiveness.

$$\varepsilon = \frac{(T_{hi} - T_{ho})}{(T_{hi} - T_{ci})} \tag{4.3}$$

So, for segmental baffle shell and tube heat exchanger:

$$\varepsilon = \frac{(453 - 303.36)}{(453 - 298)} = 0.9654$$

For helical baffle shell and tube heat exchanger:

$$\varepsilon = \frac{(453 - 298.95)}{(453 - 298)} = 0.9939$$

Therefore, the efficiency of heat exchanger constructed with segmental baffle and helical baffle are 96.54 % and 99.39 % respectively.

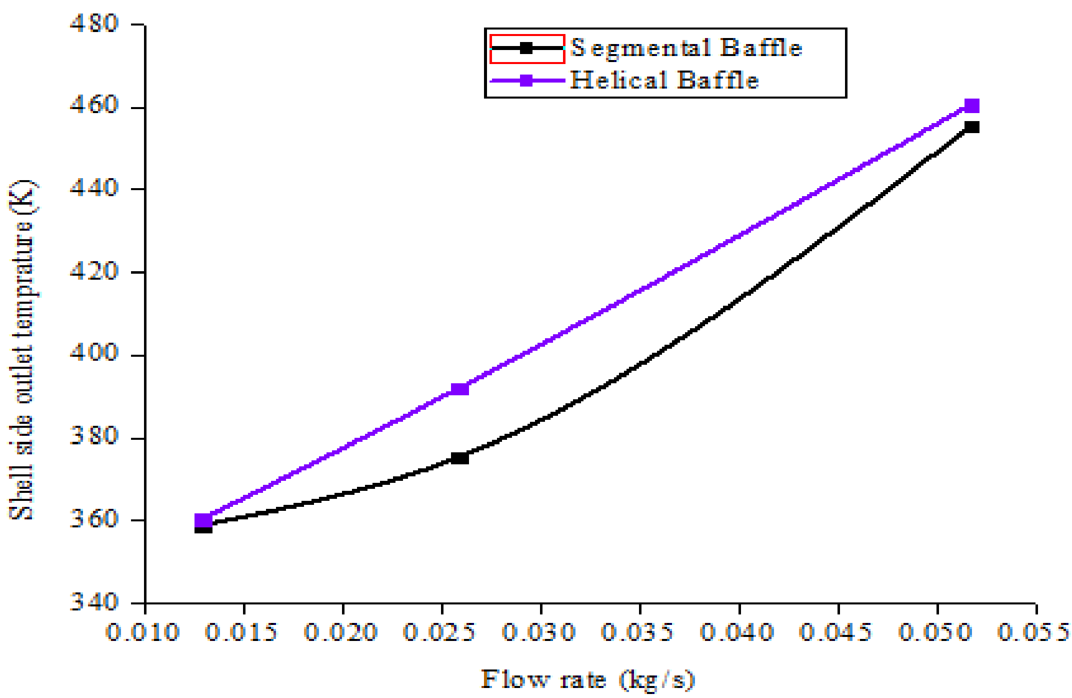


Figure 4.4 A) Flow rate versus shell outlet temperature

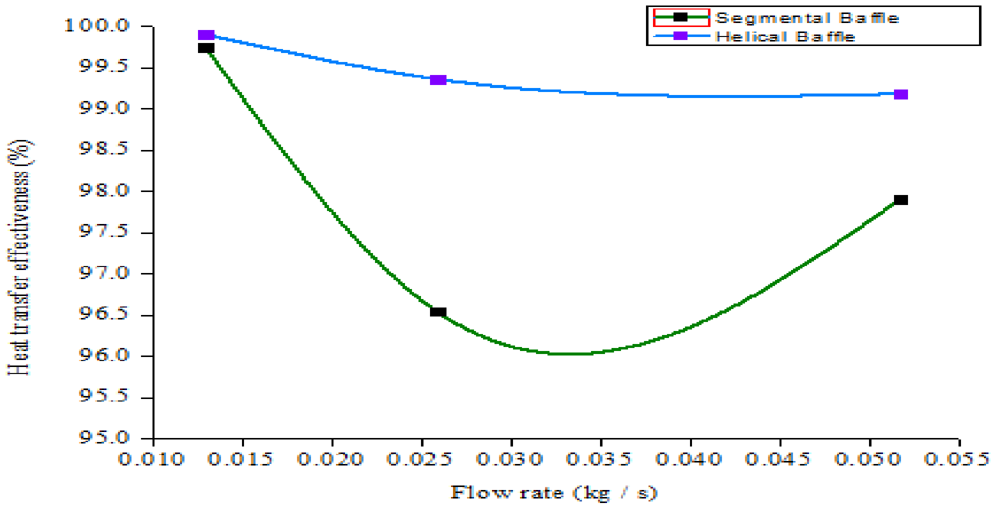


Figure 4.4 B) Flow rate versus Heat transfer Effectiveness

As we observe the above figure 4.4 B) when the flow rate increases the effectiveness of helical baffle STHE have decreases through gradually due to the flow distribution, velocity effect, residence time and fouling factors. Those factors are inverse relation to effectiveness of shell and tube heat exchangers. But in segmental baffle STHE the effectiveness makes up and down at a certain flow rate value and at that value there is maximum effectiveness value and after that the effectiveness becomes reduces significantly as compared to helical baffle.

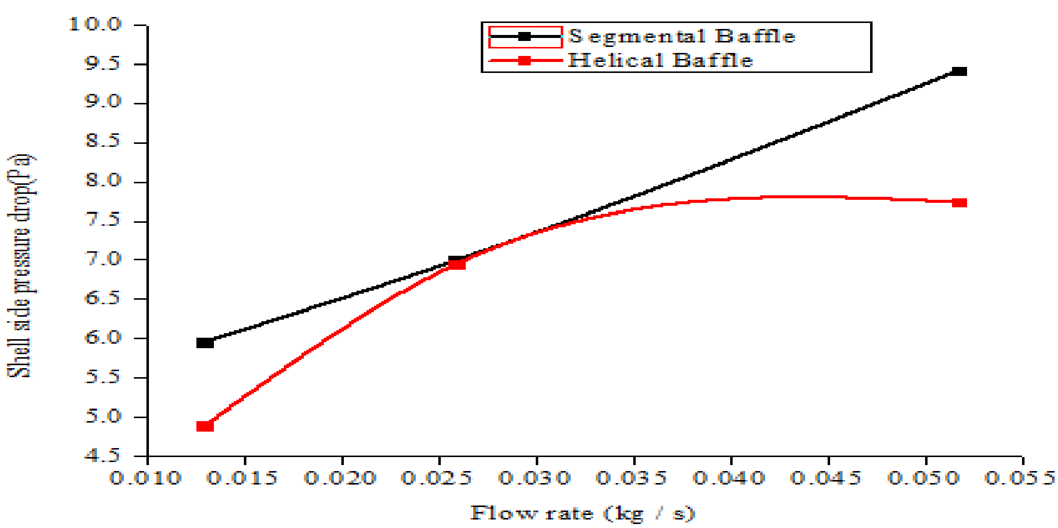


Figure 4.4 C) Flow rate versus Pressure drop

From the above figure 4.4 result shows, the SB and HB heat exchanger of different output parameters of shell side outlet temperature, heat transfer rate and effectiveness. In figure 4.4 A and B, the highest outlet temperature and heat transfer rate is recorded with HB shell and tube heat exchanger. It was efficient than segmental baffle STHE by 3 % and 11.04 % efficient than STHE without baffle.

Table 4. 2 CFD Result of helical, segmental and without baffle STHE at a given input parameters

Outlet parameters	Without baffle, WB	Segmental baffle, SB	Helical baffle, HB
Shell outlet temperature	371 K	375.37 K	392.06 K
Heat transfer effectiveness	89.51 %	96.54 %	99.28%
Heat transfer rate	15.763 KW	16.835 KW	18.928 KW
Shell side pressure drop	7.02 Pa	7 Pa	6.95 Pa

4.2 Optimization of Helical baffle shell and tube heat exchanger

Optimization is the increasing of the performance of heat exchanger by different parameter study analysis [39]. The aim of optimization in this study, to re design the shell and tube heat exchanger for Dashen Brewery Industry for developing the effectiveness of heat exchanger.

To optimize the shell and tube heat exchanger, some researchers are try to study the effect of helix angle. Helix angle is the pitch of baffle configuration of helical baffle in shell and tube heat exchanger. The previous work was study the helix angle effect by selecting the helix angle parameter of 0°, 10° and 20°. As a result of this study 20° helix angle have better rather than 10° [28]. Starting from this the current study have try to optimize the helix angle by selecting parameters of 8°, 10°and 12°degree. After getting the optimized angle we try to determine the design parameters for the DBI.

Table 4. 3 Result of optimized parameters

Helix Angle	Effectiveness (%)	Shell side outlet Temperature (K)
0°	89.51	371.0
8°	99.35	394.6
10°	98.24	393.34
12°	97.40	392.84

From the table 4.3 and figure 4.5 the shell side outlet temperature and effectiveness of heat exchanger have approximately the same value. But at the helix angle of 8° baffle configuration have better outlet temperature and effectiveness than 10°. So in order to optimize the shell and tube heat exchanger in case of Dashen Brewery Industry, we have to select the helix angle of 8° baffle configuration and determine the design parameters of this heat exchanger.

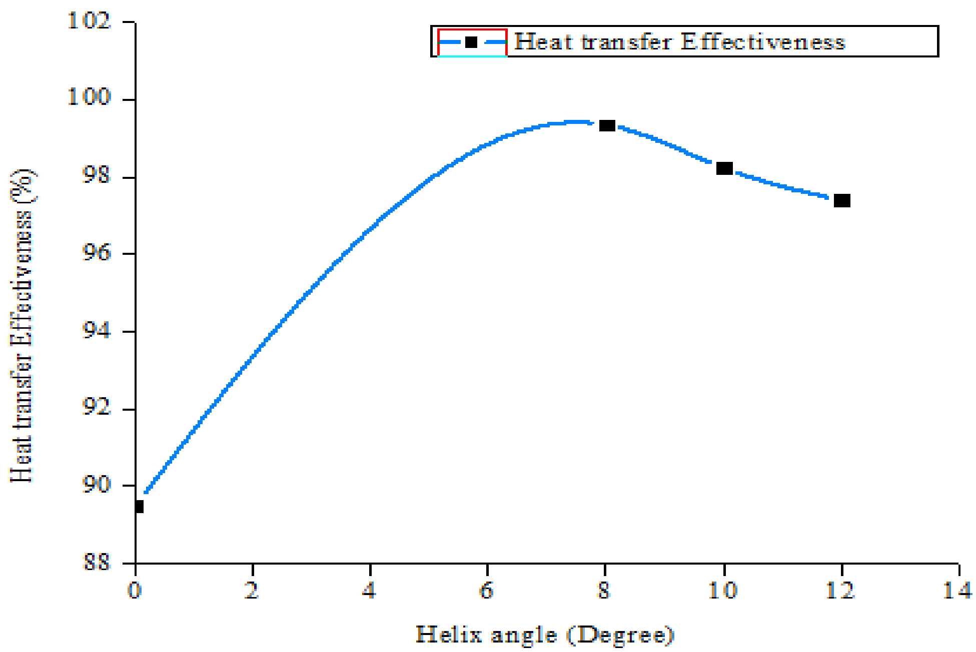


Figure 4.5 Helix angle versus heat transfer effectiveness

4.3 Sizing shell and tube heat exchanger for Dashen Brewery Industry

In this section we demonstrate the appropriate heat exchanger area (size) and input parameters. So, as we stated before in the heat exchanger capacity of DBI is 9.93571 KW heat is generated. Initially, to generate this het energy constructed without baffle. This amount of heat energy is generated by STHE without baffle, which is currently used in DBI. But STHE constructed with in HB, the capacity of heat exchanger becomes 18.928 KW that means increased by 90.5%. It is beyond the required heat energy. So, to save or utilize the energy we need to set the amount of flowrate and inlet temperature of steam at a given inlet temperature and flowrate respectively. In order to specify the required input temperature and flow rate, we should to simulate the optimized STHE with a constant flow rate by varying the value of steam temperature until to achieving the required outlet temperature. The design required input temperature of steam is approximately 374 K at a given flowrate of $0.02586 \mathrm{kg/s}$. But by reducing the given flow rate by half, the required input temperature was 407 K illustrated below in the table 4.4 and figure 4.6.

Table 4. 4 specification of hot inlet temperature at constant flow rate

Hot inlet temperature, (K) at $\dot{m}_h = 0.02586\ \mathrm{kg/s}$	Shell outlet Temperature, (K)
373	341.46
378	350.00
393	359.00
423	376.88

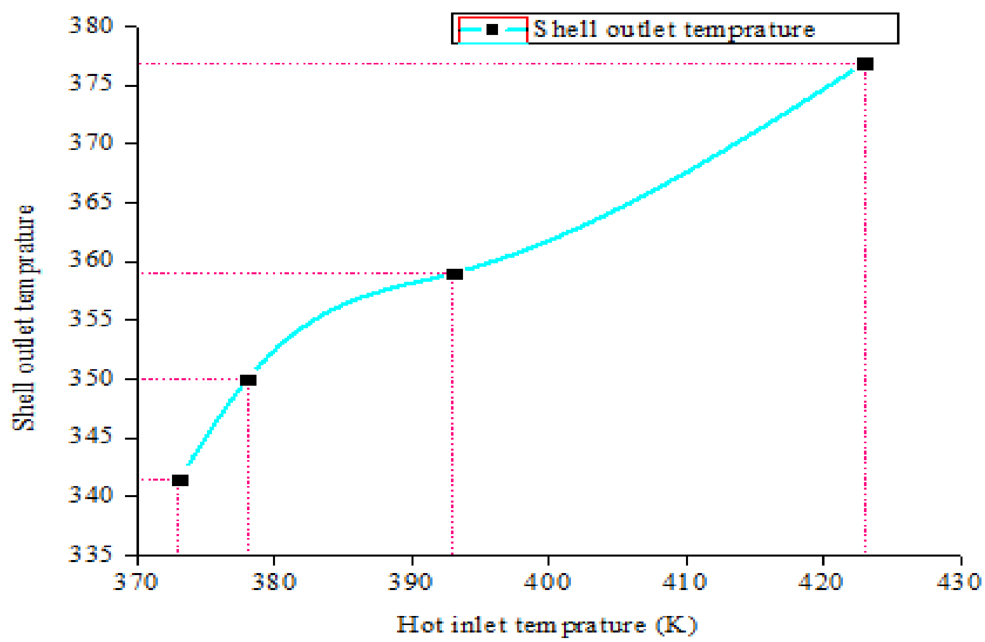


Figure 4.6 Hot inlet temperatures versus shell side outlet temperature

4.4 Validation Analysis

Validation analysis refers to a process of evaluating the accuracy and reliability of the research findings or results. Different types of validations were carried out for different study analyses. In this study, we investigate the CFD validation analysis and the cross-validation analysis.

4.4.1 CFD Validation analysis

CFD validation analysis is concerned with the identification of accuracy or the errors of the previous work result to the current CFD results which means measuring the percentage of the approach value. The error or the average deviation was less than 10% [6].

The study analysis of Asghar Ali Ghoto et.al [5] was similar to geometry specifications in the present work but with different boundary conditions. Asghar Ali Ghoto et.al [5] studied the factors of baffles and have gotten a result in terms of pressure drop. For this validation have taken the simulation result of helical baffle analysis with a flow rate of 0.0383 to 0.08) kg/s. The boundary conditions are listed in the table below.

Table 4. 5 Boundary conditions for CFD validation analysis

Parameters	Shell side	Tube side
Fluid	Cold water	Hot water
Inlet temperature, K	300	340
Flow rate, Kg / s	0.0386, 0.04, 0.05, 0.06, 0.07, 0.08	0.78441
Pressure outlet	Atmospheric pressure	Atmospheric pressure

As we observe from the figure 4.7 below, the simulation result of the Asghar Ali Ghoto et.al [5] work and the present work have recorded an approach value. As a result, it is valid in CFD validation analysis because the average deviation was less than 10 % which is 5.81 %. It indicates the present work simulation method have to be valid or acceptable, so we have to expect a true result.

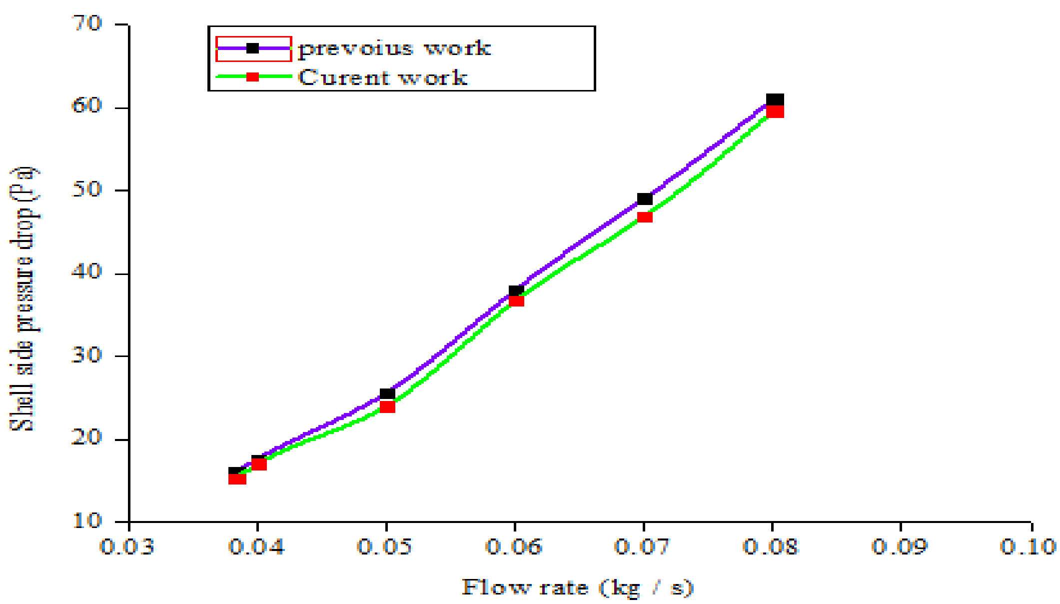


Figure 4.7 CFD validation analysis of present work with Asghar Ali Ghoto et.al [5]

4.4.2 Cross validation analysis

In this analysis focused on checking the result of the previous study was supports or contradict to the current work.

The previous study investigates the effect of baffles in STHE basically on the pressure drop with different shell mass flow rates. The modelling baffles are single segmental, double segmental helical and a combination of single segmental and helical baffles using CAD software. Finally the combination of single segmental and helical baffle have less pressure drop rather than others. But the baffle models in helical and combination of two are arranging the segmental baffle shape and size helically.

The current work focused on the effect of continuous sweep helical baffle of STHE on the pressure drop and heat transfer coefficient. To investigate this analysis the comparison of segmental and helical baffles have carried out by the modeling and simulating of CFD ANSYS 19.2 fluent. As a result helical baffle have less pressure drop than segmental by reducing dead zone, fouling effect and others. Those studies are used different modeling software, boundary conditions, and internal geometry construction but they have similar geometry specifications.

To investigate this analysis, we have to interpolate the data from the resulting graph of flow rate versus pressure drop of helical baffle [5] with the corresponding data of the present work. Those data are listed in Table 4.6 below.

Table 4. 6 Validation analysis of present work with Asghar Ali Ghoto et.al [5]

Mass flow rate (kg/s)	Shell side pressure drop	
	Asghar Ali Ghoto et.al [5] work	Present work
0.0386	16.24	6.98
0.04	18.65	7.35
0.05	24.31	11.60

0.06	33.14	22.15
0.07	48.05	29.13
0.08	60.72	37.40
0.096	71.20	41.00

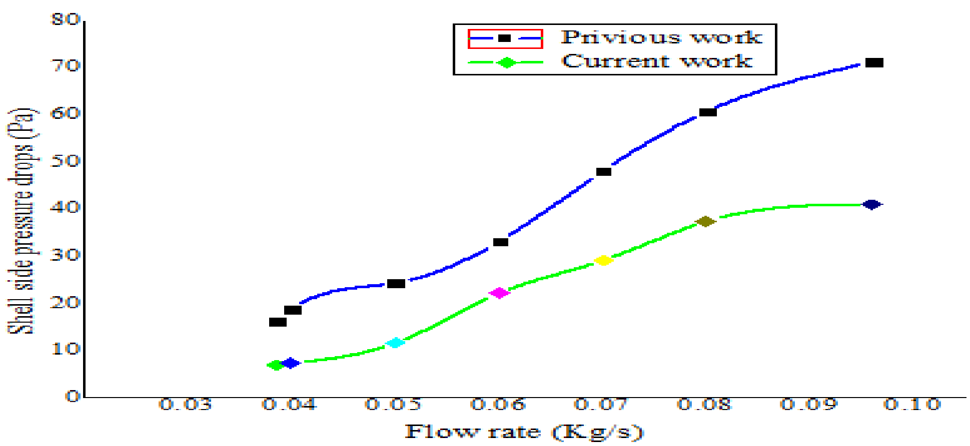


Figure 4.8 Cross validation analysis of present work with Asghar Ali Ghoto et.al [5]

As we observe from Table 4.6 and Graph 4.8 the result was recorded at the same flow rate with different input parameters and simulation methods. And the previous work study and the current study have raises the pressure drop with increasing the flow rate of shell side fluid. It implies that the previous work Asghar Ali Ghoto et.al [5] have supports the current study analysis. But there is some variation on the values of results according to the parameter and boundary condition changes.

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

In this study, segmental baffle and continuous helical baffle shell and tube heat exchanger was developed by using ANSYS 19.2 and illustrates the compression analysis in terms of heat transfer effectiveness (ε),heat transfer rate (Q),and pressure drop (ΔP_s). After compression analysis, we try to optimize the better helix angle to construct the helical baffle shell and tube heat exchanger. Using the optimized helix angle we determine the preferable design input parameters for the case of Dashen Brewery Industry. As a result of this investigation we have the following conclusions.

- For the comparison analysis, STHE constructed with helical baffle have better efficiency than segmental baffle as well as without baffle. Which indicates the capacity or heat load of this STHE was increased by 12.43 % and 20.08 % from segmental and without baffle STHE respectively.
- From the optimization of helix angle investigation, 8°, 10°and 12° with the same initial parameters apply 8° have better than 10° which is 1.13 % effective than 10°.
- From the design specification analysis for the case of Dashen brewery industry shell and tube heat exchanger, mass flow rate and tube side inlet temperature with 8° helix angle configuration of shell and tube heat exchanger to achieve the required shell side outlet temperature of 70 ℃ and the required heat transfer rate was 374 K with 0.02586 $\mathrm{kg/s}$ or 407 K with 0.01293 $\mathrm{kg/s}$.
- As validation analysis, the previous work [5] have supports the current study analysis.
- Generally in this study analysis, the performance of shell and tube heat exchanger for this industry increased by 90.5% from the previous existing heat exchanger. And the industry have gotten much profitable with simultaneous decreasing of heat exchanger area and pressure drop.

5.2 Recommendations

This study has been analysis the effect of helical baffle on the shell and tube heat exchanger performance using CFD simulation. And validate the current work result to the previous works done by using simulation analysis. Furthermore, the recommendation for the future study in the field of heat exchanger can be done with experimental investigation on helical baffle shell and tube heat exchanger.

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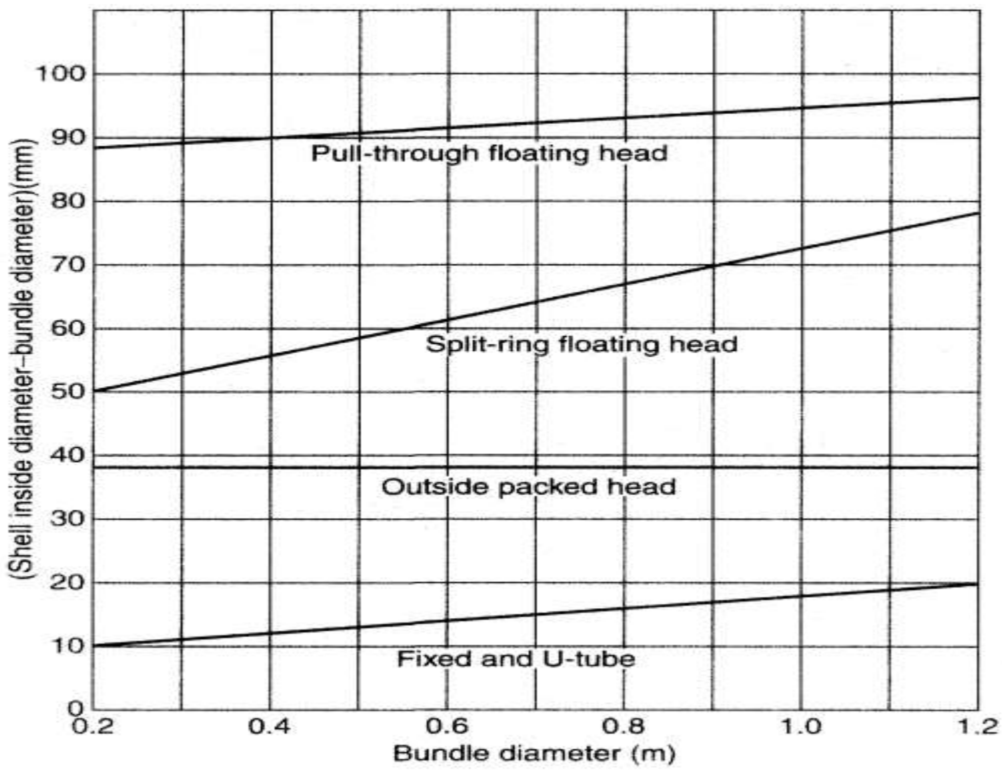
APPENDIX'S

Appendix A: Tabulated result for design analysis

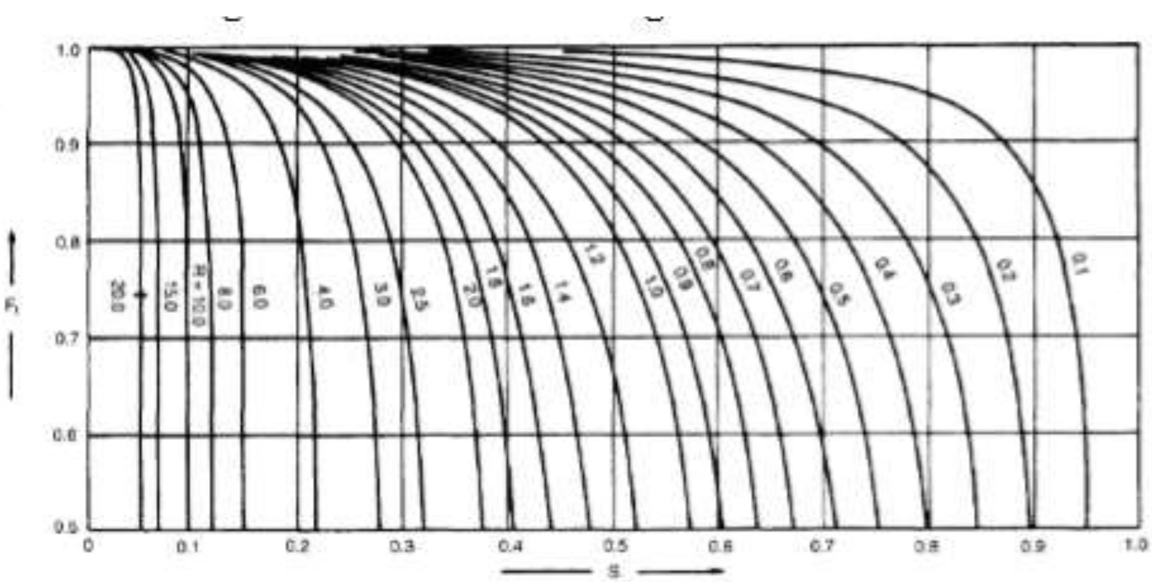
SI. No.	Design parameters	Symbol	Value	Unit
1	Heat load	Q	9.936	Kw
2	Mass flow rate of shell side fluid	$\dot{m}_c$	0.096	kg/s
3	Log mean temperature	ΔT_{LMTD}	85.54	℃
4	Mean temperature difference	ΔT_m	73.56	℃
5	Provisional heat transfer surface area	A	1.902	m^2
6.	Surface area of a single tube	A_0	0.115	m^2
7	Tube length	L	1.83	m
8	Number of tubes	N_t	16	
9	Tube pitch	P_t	0.025	m
10	Bundle diameter	D_b	0.125	m
11	Inner Shell diameter	D_s	0.163	m
12	Thickness of shell	t	0.0095	m
13	Area of cross flow	A_s	0.03248	m^2
14	Shell side velocity	u_s	0.0044	m/s
15	Shell equivalent diameter	D_E	0.0144	m

16	Shell side heat transfer coefficient	h_o	98.03	W/m^2 ℃
17	Tube side velocity	u_t	0.0086	m/s
18	Tube side heat transfer coefficient	h_i	168.81	W/m^2 ℃
19	Overall heat transfer coefficient	U	71.5	W/m^2 ℃
	For segmental baffle shell and tube heat exchanger			
20	Baffle spacing	B_s	0.0978	m
21	Cross flow area	A_s	0.00319	m^2
23	Shell side velocity	u_s	0.044	m/s
24	Heat transfer coefficient	h_o	1,322.26	W/m^2 ℃
25	Number of baffles	N_b	16	
26	Pressure drops	ΔP_s	68.89	Pa
	For helical baffle configuration of shell and tube heat exchanger			
27	Baffle spacing	B_s	0.072	m
28	Cross flow area	A_s	0.00235	m^2
29	Shell side velocity	u_s	0.06	m/s
30	Heat transfer coefficient	h_o	1,591.17	W/m^2 ℃
31	Number of baffles	N_b	21	
32	Pressure drops	ΔP_s	59.16	Pa

Appendix B: Shell-bundle clearance [13].



Appendix C: Temperature correction factor: one shell pass; two or more even tube 'passes [40]

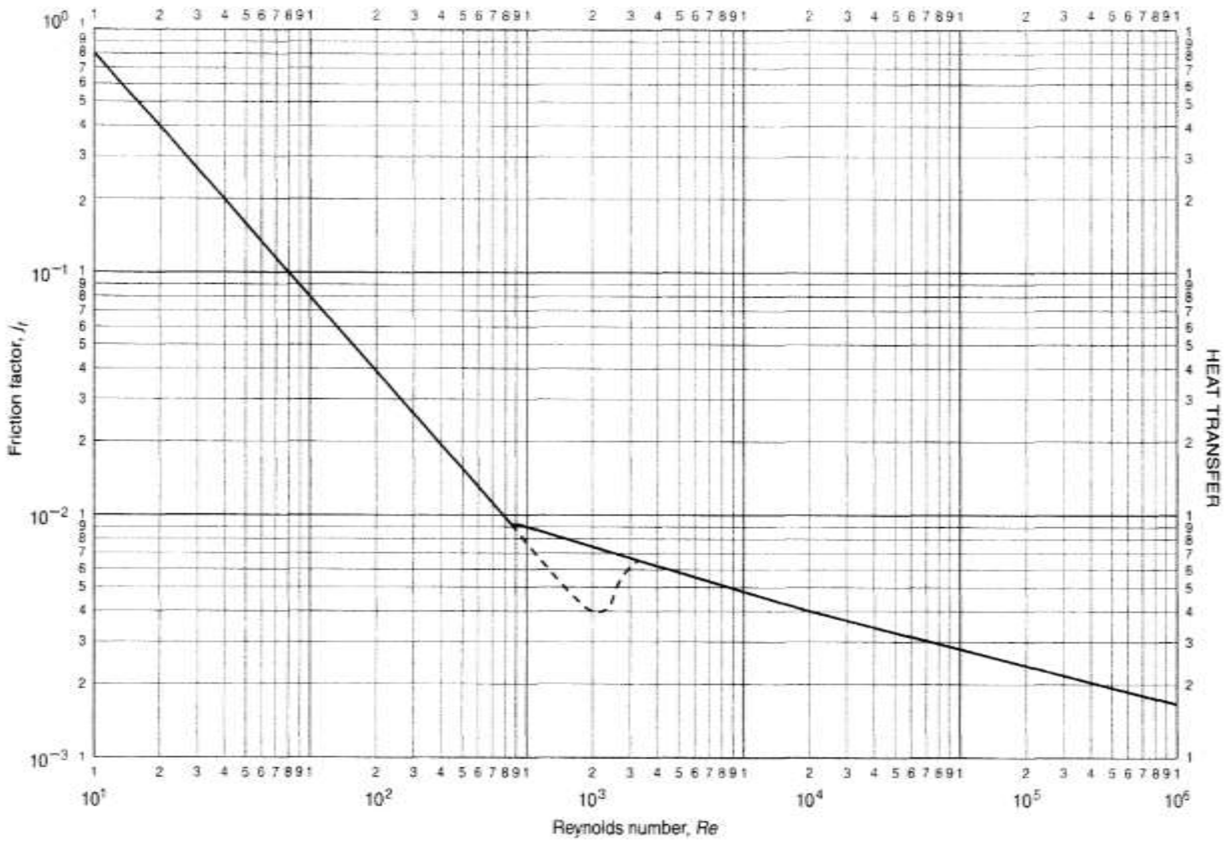


Appendix D: Overall heat transfer coefficient for different combination [8].

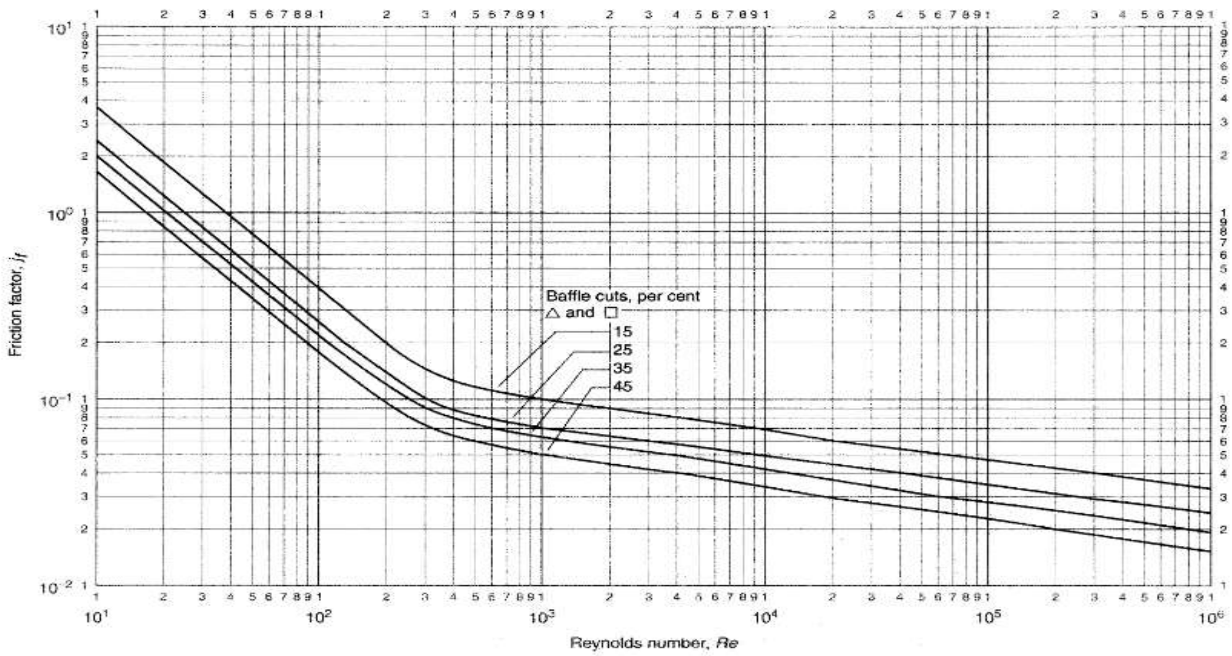
Shell and tube heat exchangers		
Hot fluid	Cold fluid	U ($w/m^2 °C$)
	Heat exchangers	
	heaters	
steam	Water	1500 -4000
Steam	Organic solvents	500-1000
Steam	Light oils	300-900
steam	Heavy oils	60- 450
steam	Gases	30- 300
Dowtherm	Heavy oils	50- 300
Dowtherm	Gases	20 – 200
Flue gases	steam	30 – 100
Flue	Hydrocarbon vapours	30 - 100
	Coolers	
Organic solvents	water	250 – 750
Light oils	Water	350 – 900
Heavy oils	Water	60 – 300
Gases	water	20 – 300

Organic solvent	Brine	150 -500
Water	Brine	600 – 1200
Gases	Brine	15 – 250

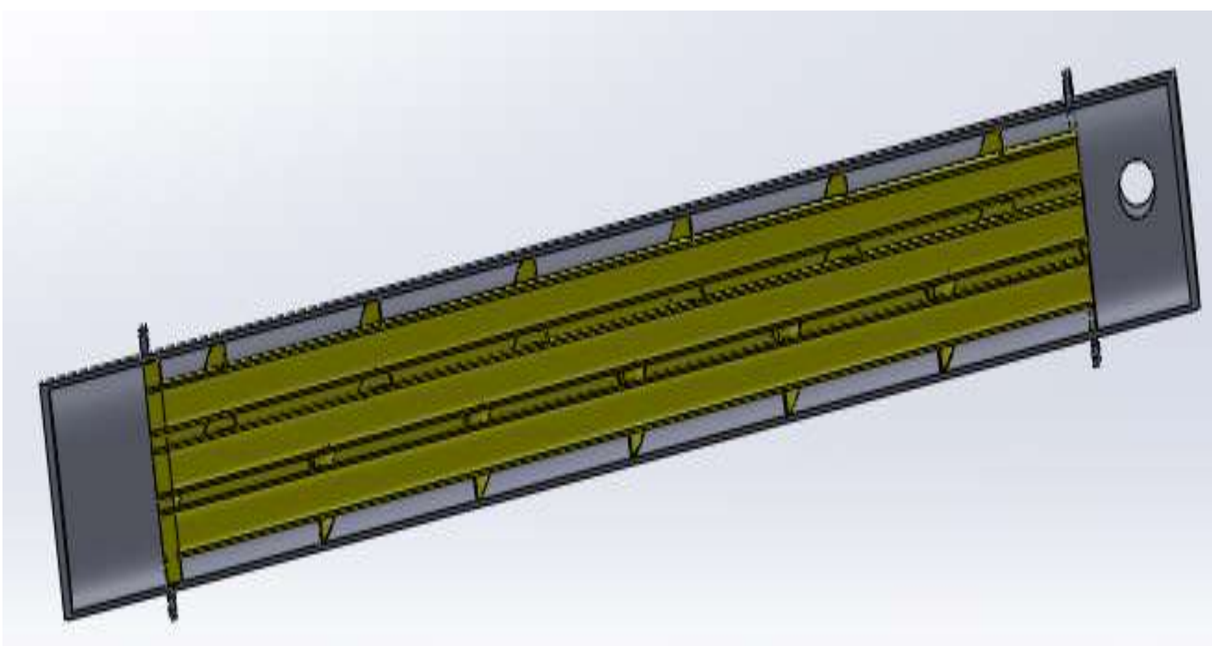
Appendix E: Tube side friction factors [8]



Appendix F: Shell side friction factors with segmental baffles[8]



Appendix G: Sectional views of helical shell and tube heat exchanger



Appendix H: Isometric views of helical tube bundle in STHE