Final Project

Report on

Thermal and Hydraulic Design of Three Fluid Shell and Tube Heat Exchanger

Submitted in partial fulfillment of

the requirements for the award of the degree of

Masters of Technology

in

Thermal Engineering

Submitted by

Vishwam Shah

(16MMET25)

Under the guidance of

Mr M. Venkatesh

Prof. Niraj K. Shah

Department of Mechanical Engineering

Institute of Technology

NIRMA UNIVERSITY

Ahmedabad - 382 481

July 2017 to May 2018

Completion Certificate

This is to certify that **Vishwam Shah** Roll no. **16MMET25** has successfully completed the project as a part of Technical Internship Programme in our company as mentioned below.

Project Title: **Thermal and Hydraulic Design of Three Fluid Shell and Tube Heat Exchanger**

Date of Joining: 1st July 2017

Date of Completion of Project: 10th May 2018

Date of Submission of Report: 14th May 2018

In partial fulfilment for the award of the degree of **'Master of Technology'** in Thermal Engineering in the Department of Mechanical Engineering of Institute of Technology, NIRMA UNIVERSITY, Ahmedabad

Date:
Place:

Mr. M. Venkatesh
(Guide)
Head,
Product Design & Process
Technology (PDPT),
Heavy Engg. Division,
Larsen & Toubro Limited.
Mumbai – 400072

Certificate

This is to certify that the Major Project entitled **"Thermal and Hydraulic Design of Shell and Tube Heat Exchanger"** submitted by **Vishwam Shah (16MMET25)** , towards the partial fulfillment of the requirements for the award of Degree of **Master of Technology in Mechanical Engineering (Thermal Engineering)** of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination.The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institution for award of any degree.

Mr. M. Venkatesh
(Guide)
Head,
Product Design & Process Technology (PDPT),
Heavy Engg. Division,
Larsen & Toubro Limited.
Mumbai – 400072

Prof. Niraj K. Shah
(Guide)
Associate Professor
Department of Mechanical Engineering
Institute of Technology,
Nirma University
Ahmedabad, 382481

Dr. Vikas J. Lakhera
Professor & Head,
Department of Mechanical Engineering
Institute of Technology,
Nirma University
Ahmedabad, 382481

Dr. Alka Mahajan
(Director)
Institute of Technology,
Nirma University
Ahmedabad, 382481

ACKNOWLEDGEMENT

I would like to express my sincere gratitude to **Mr. M. Venkatesh**, Head, Product Design and Process Technology, L & T-Heavy Engineering Division for his valuable guidance and help extended at every stages of this M Tech project. I am deeply indebted to them for giving me a definite direction and moral support to complete the project successfully, and to **Prof. Niraj K. Shah**, Guide and Associate Professor, Department of Mechanical Engineering, Institute of Technology, NIRMA University, Ahmedabad for his valuable guidance

I also extend my sincere thanks to other members of the department for their valuable advice in every stage of successful completion of this Project.

Vishwam Shah
16MMET25

Abstract

Three Fluid Shell and Tube Heat Exchanger represents a new type of heat exchangers in the Shell and Tube Heat Exchanger (STHE) category. It consists of an outer shell and plurality of double pipe tubes incorporated inside the heat exchanger with three fluids exchanging heats compared to two fluids in a conventional STHE. In the Three Fluid STHE, the fluid flowing in the annulus between the inner and outer tubes exchanges heat simultaneously with the fluid on the shell side and the fluid flowing through the inner tubes. There are two heat conducting walls through which heat transfer takes place instead of a single heat conduction wall in STHE.

Three Fluid STHE can be used for many specific applications such as heat recovery system, power production, chemical processing, air conditioning, cryogenics, food processing, distillation, transport, pharmaceutical industry, etc. It has various advantages compared to conventional STHE in terms of compactness and overall performance.

There is no methodology available to carry out the design / check the performance of Three Fluid STHE. Hence, a methodology has been evolved to carry out the design of Three Fluid STHE starting with the fundamental concepts of heat transfer. The work includes understanding the design procedure of STHE with a parametric case study, then extending the work to design methodology for Bayonet Tube Heat Exchanger and finally to design of Three Fluid STHE. The calculations and performance characteristics are further elaborated during the analysis. Also the parameters during the calculations were validated using HTRI and HTFS software.

Keywords:

Three Fluid Shell and Tube Heat Exchanger, STHE , Three Fluids, heat conducting walls, shell, double pipe, Bayonet Tube Heat Exchanger, HTRI, HTFS

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Chapter 1

INTRODUCTION

Heat Exchangers are basically heat transfer equipment that exchanges heat with two or more number of fluids either through direct contact or indirectly through a solid wall to avoid mixing of fluids.

Heat Exchangers are critical equipment that are extensively used in the process, power, transportation, air-conditioning and refrigeration units, cryogenic, heat recovery; alternate fuels, and manufacturing industries, as well as being key components of many industrial products available in the marketplace. Also it has wide application for energy conservation, conversion, recovery, and successful implementation of new energy sources.

There are numerous types of Heat Exchangers used depending on its area of application. Also Shell and Tube Heat Exchangers are the workhorse of the industry. Till date large number of researchers has tried to overcome drawbacks of the Shell and Tube Heat Exchanger. There are various improvements and modifications done by employing rod baffles, helical baffles, flower baffles, helical tubes, plate baffles, and special inserts inside the tubes.

So an effort is being carried out for optimization of the Shell and Tube Heat Exchanger by incorporation of three fluids instead of two fluids used in a conventional Shell and Tube Heat Exchanger.

1.1 Shell and Tube Heat Exchangers (STHEs)

Shell-and-tube heat exchangers (STHEs) [7] are the most common and widely used unfired heat transfer equipment in the chemical processing industries. They are also used extensively in , nuclear, coal and gas-based, geothermal, and ocean power generation facilities. STHE consists of a shell and encompassing a multitude of tubes . One fluid flows through the tubes (tubeside) and the other fluid flows over the tube bundles on the shellside.

Though there are many other types of Heat exchangers design available that are more efficient and better than STHE, still STHEs holds the large percentage of world market of heat exchangers. The following are the reasons why they are most commonly used.

1. It has high versatility. It is very flexible in size and can vary from less than one square meter to a thousand square meters and even more.

2. They are mechanically robust to withstand normal shop fabrication stresses, stresses due to transportation and erection, as well as the stresses of normal and abnormal operating conditions.

3. Can be designed for almost any duty with a very wide range of temperatures and pressures.

4. Range of pressure drops can be accommodated.

5. Tube leaks can be easily located and plugged since pressure test is comparatively easy.

6. Can be built in many materials and maintenance can be done by non-skilled workers as well.

7. Design methods and mechanical codes have been established from many years of experience.

Besides this, by the use of special inserts, helical baffles and other special designs, the efficiency of the STHE can be more optimised.

1.2 Shell and Tube Heat Exchanger Construction

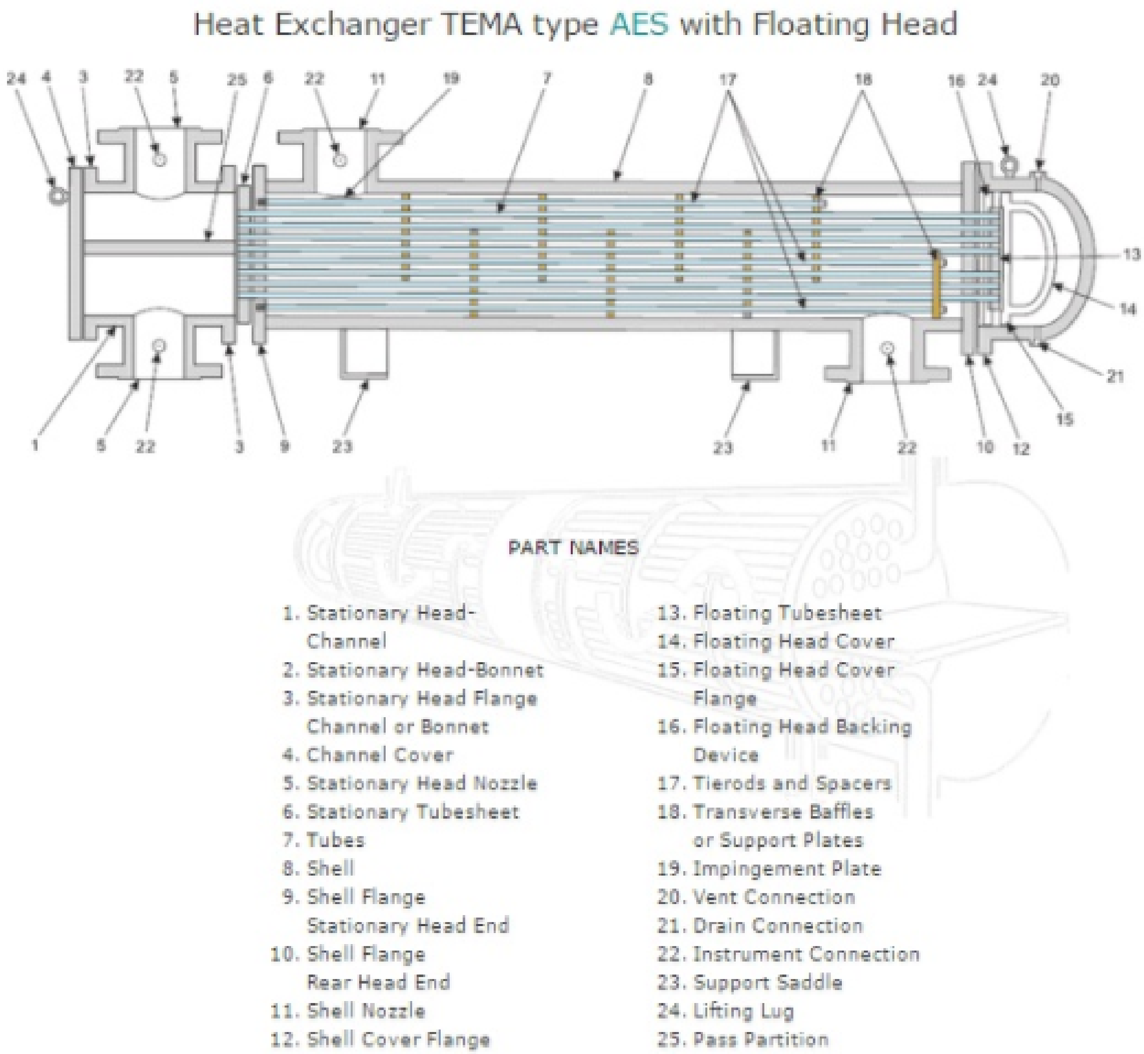

Figure 1.1: Shell and Tube Construction [2]

The Standards of the "Tubular Exchanger Manufacturers Association" (TEMA) [2] classifies the STHEs based on the following:

1. Front End
2. Shell
3. Rear End

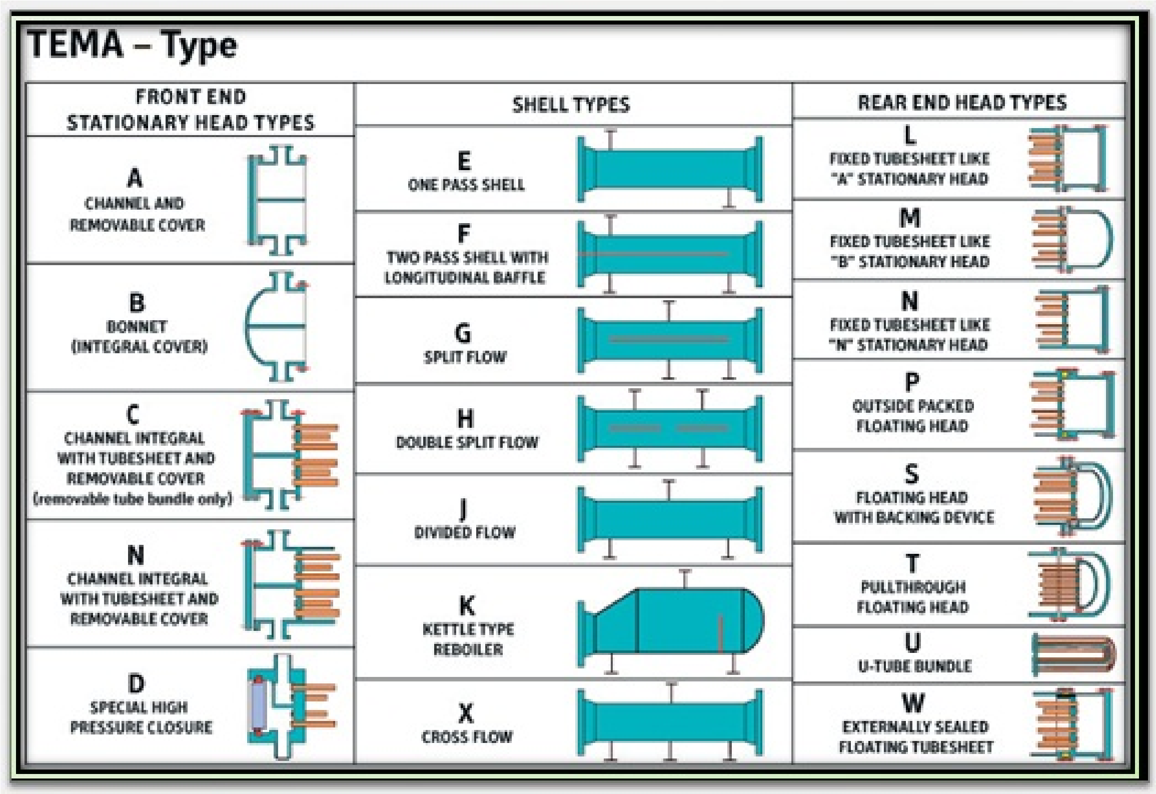

Figure 1.2: Shell and Tube Construction [2]

1.2.1 Front End Types

1. A – Type Head

In this type , the channel barrel is flanged at both ends. The tube sheet is bolted to one flange and a flat channel cover to the other. Thus only the channel cover has to be removed for cleaning of the tubes by rodding or hydro-blasting; the channel and connecting piping are not disturbed.

2. B – Type Head

In this construction, the channel barrel is flanged at one end only, the other end being welded to a semi-elliptical bonnet or dished end. This type is lighter and cheaper than the A type, especially at high pressures, as the thickness of the bonnet is considerably less than that of a flat cover plate. Here the entire bonnet has to be removed for cleaning even the inside of tubes, which means that the channel piping connections have to be dismantled.

3. C – Type Head

This is similar to the A type in that there is a flat channel cover. However, the tubesheet end of the channel is not flanged to the tubesheet but welded to it. The tubesheet itself is extended as a flange and bolted to the shell. Evidently, this type of construction is intended only for removable bundles.

4. N – Type Head

As in C-type, flat channel cover is provided. However, the tubesheet is integral to both the channel and the shell and can evidently be used only for fixed-tubesheet heat exchangers.

5. D – Type Head

For channels for high-pressure services on the tubeside (design pressure ¿ 2133 psig or 150 kg/m^2 (g)), specially designed and non-bolted closures are employed. Many of these are patented. These special high-pressure channels are generally referred to as D-type closures.

6. Conical head

Although not formally designated in the TEMA nomenclature, these heads are used quite often in single tube pass heat exchangers, especially in vertical thermo syphon reboilers. These are simply conical sections, flanged at both ends with the flange at the larger diameter bolted to the tubesheet and the flange at the smaller diameter bolted axially to the piping. In services handling low pressure gases or vapours on the tubeside, or simply very high flow rates, where the nozzle size is greater than 50 percent of the channel diameter, conical nozzles offer a convenient solution.

1.2.2 Shell Types

Shell Types TEMA Standards defines various shell types [8](E, J, F, G, H, K, and X) depending upon the fluid flow through the shell .

1. E – Type Shell

A TEMA E shell is a single-pass shell, where the fluid on the shellside enters at one end and leaves from the other end. This is the most commonly used type of shell in the chemical process industries.

2. F – Type Shell

A TEMA F shell is a 2-pass shell. In this shell type, a longitudinal baffle is provided, which divides shellside flow into two passes viz. an upper pass and a lower pass. The shellside fluid enters the shell at one end in either the upper half or the lower half and flows along the length of the the first shell pass and turns around at the end of the longitudinal baffle. The fluid then flows through thesecond shell pass, and exits through the other half of a shell.

3. G – Type Shell

A TEMA G shell is a split flow shell configuration. This construction is usually used in horizontal thermosyphon reboilers. There is only one central support plate which splits the shellside fluid and there are no baffles present.

4. H – Type Shell

A TEMA H shell basically comprises of two G type shells placed in parallel adjacent to each other. This type is called as double split flow type. This shell type consists of two support plates. . This type is most commonly used for horizontal thermo syphon reboilers. The advantage with TEMA G and H shells is that the pressure drop is reduced drastically. Since there are no pumps in thermo syphon reboiler circuits, the pressure drop has to be restricted to a bare minimum; hence, these configurations are employed. Besides, and no less importantly, the natural tendency of the two phases to separate is minimized.

5. J – Type Shell

A TEMA J shell is a divided flow type shell where the fluid on the shellside enters at the centre of the tube bundle (along the length) and divides into two streams. Half of the shellside fluid flows to the left and the other half to the right and leaves separately.

6. K – Type Shell

A TEMA K shell is a special shell type employed for kettle reboilers having an integral vapour disengagement space in the shape of an enlarged shell.

7. X – Type Shell

A TEMA X shell is a pure cross flow shell where the fluid enters from the top (or bottom) and flows across the tubes and exits from the opposite end of the shell. Pressure drop for this shell type is extremely low. Also, majority of the shellside pressure drop occurs in the shellside inlet and outlet nozzles .The pressure drop in the tube bundle is negligible.

1.2.3 Rear End Types

There are eight rear head types: L, M, N, P, S, T, U, and W, which correspond in practice to only three general construction types, namely fixed-tubesheet, U-tube, and floating-head.

1. L/M/N – Type

Rear head L type is identical to a front head A type, and rear head M type is identical to a front head B type, while rear head N type is identical to front head N type.. These three rear head types are used with fixed tubesheet heat exchangers.

2. P – Type Head

It is the outside packed floating head. In this type the tubesheet is sealed to the shell by compressing the packing material contained between rear head and the extended shell flange by means of a ring.

3. S – Type Head

It is a type of floating head with a backing device. It is usually referred as the split ring floating head . Also it has a backing ring to which the floating head is bolted. The backing ring is made up in two halves so that the floating tubesheet can be pulled through the shell.

4. T – Type Head

It is a pull through floating head type of rear end. It is usually preferred when there is a large pressure difference between the shellside and the tubeside.

5. U – Type Head

This type is normally used for high pressure applications on tubeside and with clean fluids on the tubeside. U-tube construction eliminates the rear tubesheet and the rear head channel and results in a very cost-effective design. U-tube bundles are removable and facilitate mechanical cleaning of the shellside of the tube bundle if square tube layout pattern is provided.

6. W – Type Head

It is an externally sealed floating head. It has O ring that seals floating tubesheet, shell and channel together respectively. It is suitable for low pressure, non-hazardous fluids.

1.2.4 Tube Layout Patterns

There are four tube layout patterns:

- Triangular
- Rotated triangular
- Square
- Rotated Square

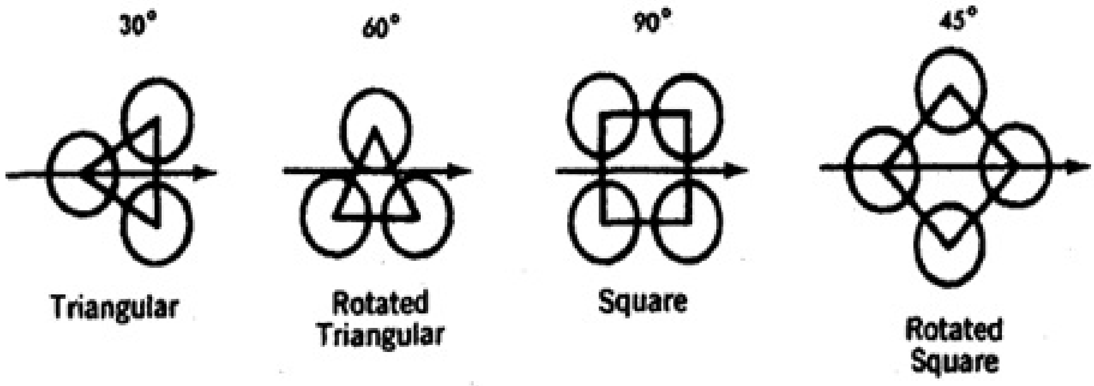

Figure 1.3: Tube Layout [2]

It is evident that the triangular or rotated triangular tube pattern permits more number of tubes to be accommodated for a given diameter of the shell compared to square or rotated square tube pattern arrangements. Also, the triangular arrangement have tubes more closely packed compared to square arrangement producing higher turbulence , therefore, a higher heat transfer coeffcient. However, a triangular (or rotated triangular) pattern does not allow mechanical cleaning on shellside for the normally used tube pitch as access lanes for cleaning are not available. For fouling services on the shellside, which require mechanical cleaning of the shellside of the tube bundle, square (or rotated square) pitch has to be used

A rotated triangular pattern does not offer any advantage over a triangular pattern in the conversion of pressure drop to heat transfer and, hence, its use is rare.

1.2.5 Types of Baffles

Baffles is an important component of the STHE construction. Baffles are used to:

a) Support tubes

b) Enable a desirable velocity to be produced for the shellside fluid.

c) Prevent failure of tubes due to flow-induced vibration.

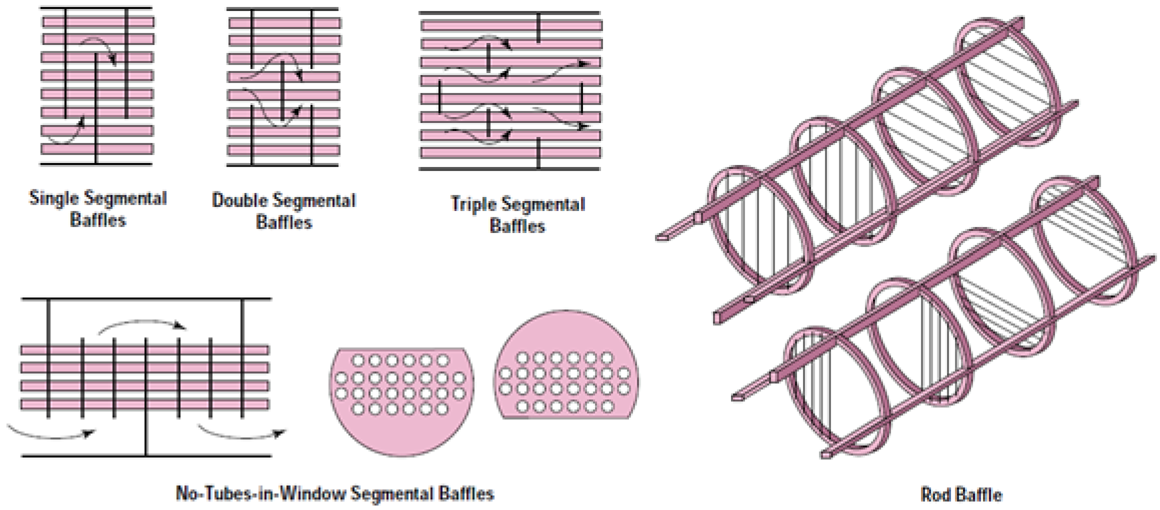

Figure 1.4: Baffles [2]

There are two principal types of baffles: plate type and rod type. Plate baffles may be of single segmental, double segmental, or triple segmental type. It is very important to select appropriate baffle type for the optimum performance of STHE.

Segmental baffles are metallic plates in the form of disc on the shellside that guide the shellside fluid flow from inlet to the outlet. These discs are provided with either horizontal or vertical cut that creates turbulence and enhances the shellside heat transfer coefficient.

Rod baffles are rod grids on the circular disc that extends along the length of the shell. These type of baffles produce less pressure drop compared to single segmental baffles and also, the dead spots behind the baffles are eliminated.

Chapter 2

Literature Review

With an increase in the use of Shell and Tube heat Exchangers in the process plant industries, it becomes important to get the best possible performance of the STHE with the minimum optimal cost, space. There are many methods already being used to increase the performance and capacity of the STHE.

For this purpose, the thermal design of three fluid Shell and Tube Heat exchanger has been developed. As the name suggests, it consists of three fluids exchanging heat simultaneously in a Shell and tube heat exchanger as compared to two fluids that are conventionally used in a Heat Exchanger. Also, heat transfer occurs across two heat transfer walls compared to a single thermal wall in Shell and Tube Heat Exchanger.

2.1 Objective of Three Fluid Shell and Tube Heat Exchanger

With a Three Fluid STHE, various performance parameters of the conventional shell and Tube Heat Exchanger can be increased:

1. Increase in heat duty and the capacity can be achieved within a limited space.

2. Higher heat transfer coeffcient and lower pressure drop compared to conventional Heat Exchanger.

3. Reduction in number of connecting piping with corresponding reduction in the capital and operational cost.

4. Reduction in the overall size of the heat exchanger compared to the conventional STHE for a certain heat duty.

5. The foot print area for series of heat exchangers can be reduced.

The Thermal Design of Three Fluid Shell and Tube Heat Exchanger, being complex, the design methodology has to be derived starting from the basic fundamentals.

The methodology for carrying out the thermal design is already well established in detail for the STHE. Hence, carrying out the thermal design for a case study on the STHE would be the first step in developing the design methodology. Subsequently, the work will be further extended to thermal design of a Bayonet Tube Heat Exchanger wherein there will be two heat transfer walls as compared to only one heat transfer wall in the STHE for the two fluids used.

For the Bayonet Tube Heat Exchanger, the results obtained will be compared with the results from the literature. The next step will be to derive methodology for three fluids operating in single phase for a shell and tube heat exchanger. And finally, the main objective of the project is to evolve thermal design methodology for the Three Fluid Shell and Tube Heat Exchanger, wherein the hot fluid on the in the annulus will be condensing. The heat released will be used to vaporize the fluid on the shellside and heat the single-phase fluid on the tubeside.

2.2 STHE Selection Considerations

According the design reports [9] , guidelines have been set for the starting the design procedure. Before carrying out the actual thermal design of the STHE, correct selection of the shell type and allocation of the fluids is very important for the best and optimum design and working of the heat exchanger. An incorrect selection can lead to huge losses and also can be hazardous. So the procedure to be followed while design of STHE is:

1. Allocation of fluids to the shellside and the tubeside

2. Selection of the Shell type
3. Selection of the tube bundle type
4. Selection of the head types
5. Selection of the Exchanger geometry

2.2.1 Allocation of fluids to the shellside and the tubeside

Certain guidelines are to be followed for the allocation of the fluid on the shellside and tubeside as this has a considerable impacts on the heat exchanger design.

The primary considerations that must be taken care of are:

1. Safety
2. Reliability
3. Ease of Maintenance
4. Cost

The other important factors that also need to be considered are:

1. Nature of the fluids :

The fluids can be hazardous, corrosive, fouling in nature

2. State of the Fluids :

The fluids can be in single or multi-phase, at different temperatures and pressures

Table 2.1: Allocation of Fluids

Tubeside fluid	**Shellside fluid**
Corrosive fluid	Condensing vapour (unless corrosive)
Cooling water	More Viscous Fluid
Fouling fluid	Fluid with large temperature difference (>40°C)
Less viscous fluid	
High-pressure fluids	

2.2.2 Selection of the Shell type

Based on the application of the heat exchangers, there are various types of Shell geometry available in the TEMA Standards. So the selection of the shell type is based on this only. Various factors that must be considered while selection is to be done such as :

1. Smallest numbers of shells in series that are required to handle the temperature cross.

2. Smallest numbers of shells in parallel that are required for the allowable pressure drop.

2.2.3 Selection of the Head type

The front end and the rear end types also need to be selected carefully according to the applications and the purposes for which heat exchanger is defined. Certain factors that affect the choice of the types of head based on TEMA Standards are :

1. Need for mechanical cleaning of the shellside : If the shellside fluid is prone to fouling, then it becomes necessary to mechanically clean the shellside of tube bundle from time to time. This requires the tube bundle of removable type. Accordingly, care has to be taken in selection of the heads to accommodate mechanical cleaning of the tube bundle.

2. Provision for differential thermal expansion: The temperature difference be-tween the tubeside fluid and the shellside fluid can induce thermal stresses. In order to accommodate the differential thermal expansion, suitable provision such as expansion bellows needs to be provided.

3. Operating temperatures of the fluids and the corresponding fluids pressures. Also the nature of the fluids on the tubeside and shellside.

2.2.4 Selection of the Exchanger Geometry

Tube Diameter and Thickness

For selecting the proper dimensions for the tubes that are to be used in the heat exchanger various considerations are to be followed such as:

1. Ease of cleaning
2. Tube to Tubesheet Joint
3. Pressure Drop
4. recommendations given in TEMA Standards for the nominal sizes of the tubes that can be used
5. Thickness governed by design pressure and temperature condition
6. For U tubes, e ect of thinning due to bending is to be considered

Tube Length

The length of the tube decides the heat transfer area of the heat exchanger and also roughly the actual length of the heat exchanger. So size of a heat exchanger is mainly dependent on this. The factors that needs to be considered for this purpose are :

1. For a given surface area, longer tube length results in a cheaper heat exchanger that would serve the purpose of :

- Smaller Shell Diameter
- Lower material and labor cost

2. Allowable pressure drop may restrict the length of the tubes as pressure drop across the tube is linearly proportional to the length of the tube.

3. Space constraint: Space constraint in the plant to accommodate aparticular heat exchanger is another factor that limits the length of the tubes

Number of Tubes

The selection of number of tubes inside the tube bundle is dependent on the factors as mentioned.

1. Tube Velocity: For a particular material of construction of the tubes there are permissible velocities which need to be considered. Lower velocity of the fluid in the tubes can result in higher rate of fouling of the tubes. Higher velocity of the fluid in the tubes can cause erosion of the tube material. Hence, an optimum velocity of fluid in the tubes needs to be selected.

2. Allowable pressure drop: The number of tubes is governed by the allowable pressure drop. Lower the number of tubes, higher will be the velocity and higher the tubeside pressure drop.

Tube Layout

Selection of the tube layout pattern or the arrangement of the tubes inside the tube bundle plays an important role. Various factors that needs to be considered for selection of the tube layout pattern are:

1. Mechanical Cleaning of the shellside of the tube bundle
2. Heat Transfer and allowable Pressure drop
3. Size of the shell
4. Fluid circulated on the shellside
5. Tube to Tubesheet joints

Tube Pitch

It is the shortest distance between two adjacent tubes. For a triangular tube layout pattern, TEMA Standards specifies a minimum tube pitch of 1.25 times the tube OD. It is a practice to design for minimum tube pitch as per Standards as it will reduce the size of the shell. However, the tube pitch may be increased to a higher value in order to either reduce shellside pressure drop or when mechanical cleaning is required for the shellside of the tube bundle.

Number of Tube Passes

The tube passes indicate the number of turns the fluid will make while flowing in the tubes. The number of passes directly affects the shell diameter and the size of heat exchanger. The parameters that needs to be considered are:

1. Maximum allowable pressure drop
2. Temperature profile of the fluids inside the Heat Exchanger
3. Fluid phase
4. Minimum velocity
5. Length constraint

Baffles

Baffles form an integral part in the heat exchanger design. It not only supports the tubes, also it guides the flow of the shellside fluid inside the heat exchanger. The baffles have a major effect on the shellside heat transfer coefficient and the performance of the heat exchanger. Parameters for the baffle design depend on:

1. Type
2. Spacing
3. Cut, Orientation

Selection Criteria on which the parameters depend :

1. Flow of Stream on the shellside
2. Shellside pressure drop
3. Flow induced Vibration
4. TEMA Standards constraints on maximum unsupported span of the tubes

Nozzle and Impingement Protection

Nozzles for the tubeside flow and shellside flow also needs to be designed carefully as they also can affect the performance of the heat exchanger. The impingement plate is a plate that protects the tubes from erosion by the direct impingement of fluid coming at high velocity from the shellside inlet nozzle. The factors to be considered for selection of nozzle sizes and impingement plate are as follows:

1. Pressure loss in the nozzles (typically, the pressure loss in the nozzles should be restricted to less than 15% of the total pressure drop)
2. Shell and bundle entrances ρ V^2 values
3. TEMA Standards requirements
4. Maximum allowable velocity
5. Cost

2.3 Thermal and Hydraulic Analysis for STHE

2.3.1 Kern Method

The Kern Method [10], the first method developed for calculation of shellside heat transfer coefficients and the pressure drops. The calculations and correlations developed were based on the series of experimental data's for some typical heat exchangers. The correlations developed were similar to the equations developed for flow in the tubes. However the correlations developed by him were based on certain fixed parameters such as 25 % baffle cut, hence these does give any idea about the flow of fluid on the shellside.

2.3.2 Bell Delaware Method

As the Kern Method does not give any idea on the flow of fluid on the shellside, Professor Kenneth J. Bell conducted a study on the STHE, and derived and calculated correlations for shellside heat transfer coefficients and pressure drops [3] and also estimated the flow of fluids on the shellside.

The Delaware method [11] uses empirical correlation for the calculation of heat transfer coefficient and pressure drop in flow perpendicular to banks of tubes; which are referred as ideal tube bank correlations. The deviations from the ideal tube bank correlations are accounted for by a set of empirical correction factors for heat transfer and pressure drop.

2.3.3 Shellside Stream Analysis

Flow of the fluid across the shell can be identified in terms of five streams. These flow streams define the mechanism of the fluid flow inside the shell. Also it is very important to study the flow in the shell as it impacts the shellside heat transfer coefficient and pressure drop. The streams are categorized as :

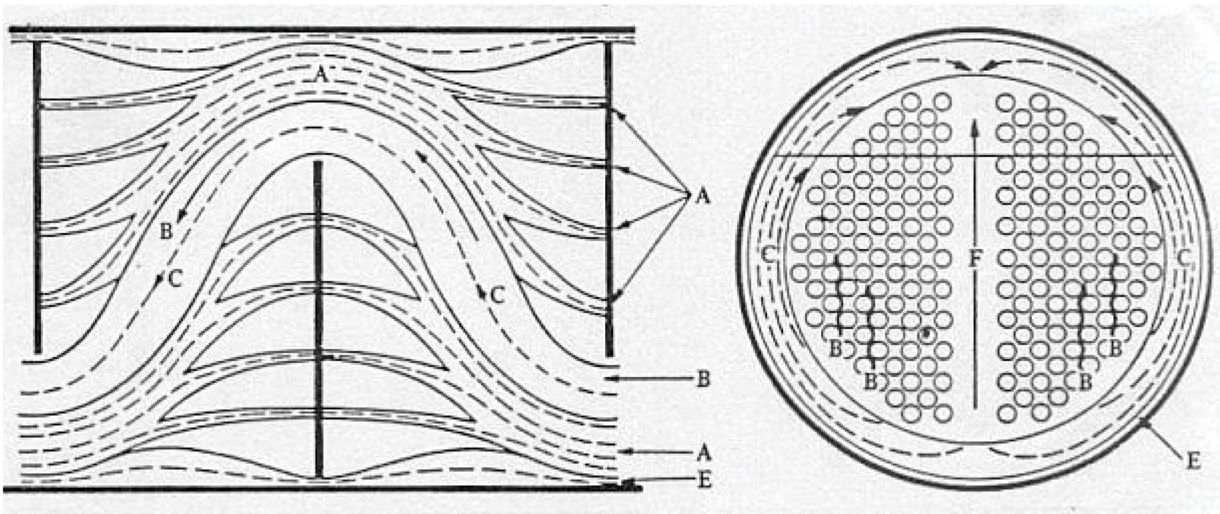

Figure 2.1: Shellside Flow Path Analysis [3]

1. A stream: Tube hole to Baffle Leakage Stream. In this stream the fluid on the shellside leaks through the diametral clearance present between the tube and the tube holes on the baffles.

2. B Stream: The Main Cross Flow Stream. This is the main idealized stream in which the fluid flows across the window section from one baffle compartment to the other. The B stream is the main flow that enhances the performance of the heat exchanger.

3. C Stream: Bundle Bypass Stream As the fluid flows from the nozzle to the shell, some of the fluids tends to bypass the outermost part of the tube bundle. The fluid flows in the clearance between the inside of the shell and the outermost part of the tube bundle.

4. E Stream: Shell to Baffle Bypass Stream In order to facilitate insertion of the tube bundle inside the shell, certain gaps or clearances need to be maintained between the inside diameter of the shell and the diameter of the baffle. The shellside fluid flows inside

the clearance between the shell and baffle.

5. F stream: Pass Partition Bypass Stream This stream exists only in tube bundles having two or more tube pass configurations, where the fluid flows through the inline pass partition lane in the tube bundle.

2.4 Bayonet Tube Heat Exchanger

With the growing need of the optimisation of the Shell and Tube Heat Exchanger (STHE), Bayonet Tube Heat Exchanger is the type of STHE that can eliminate various limitations of the conventional STHE.

Bayonet Tube Heat Exchanger [12], similar to the STHE it consists of a shell, with plurality of bayonets tubes inside the tube bundle. The Bayonet Tubes are fixed to one end of the tubesheet while the other end is floating inside the shell.

2.4.1 Construction

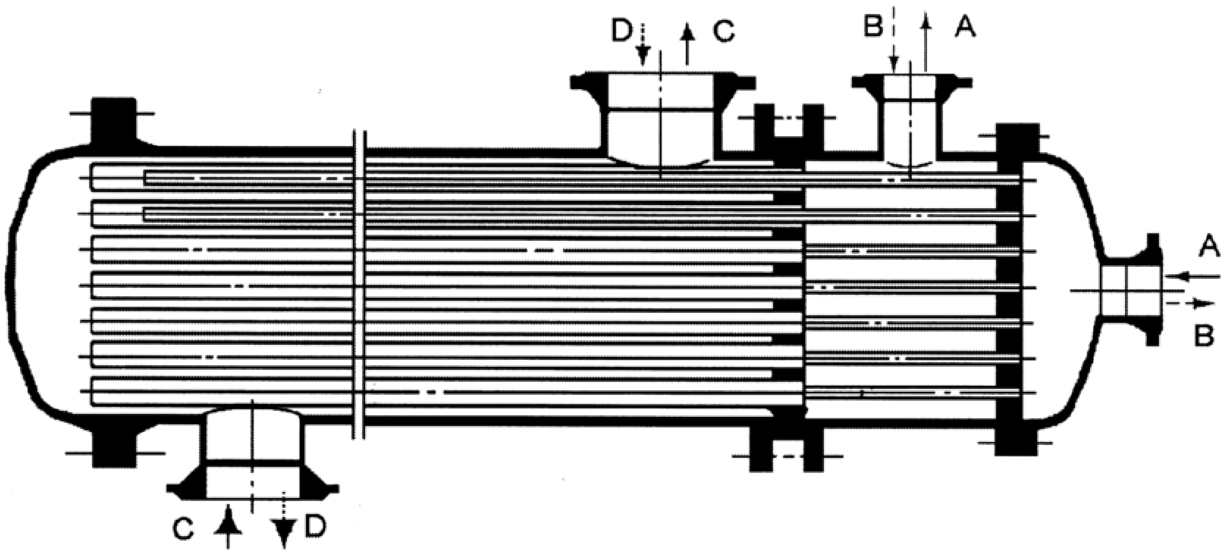

Figure 2.2: Bayonet Tube Heat Exchanger [4]

The bayonet tube consists of an inner tube and outer tube. There are two fixed tubesheets as compared to single tubesheet in the STHE, one for the inner tube and the other for the outer tube respectively where the tubes are being fixed. One end of each of the outer tubes is secure*d to the primary tubesheet and the other end is closed with a cap. One end of the inner tube is secured to secondary tubesheet, whilethe other end is free in form of a cantilever beam. The free end of the inner tube is open and communicates with the surrounding outer tube directly. The other components in the Bayonet Tube Heat Exchanger are similar to the STHE.

2.4.2 Principle of Operation

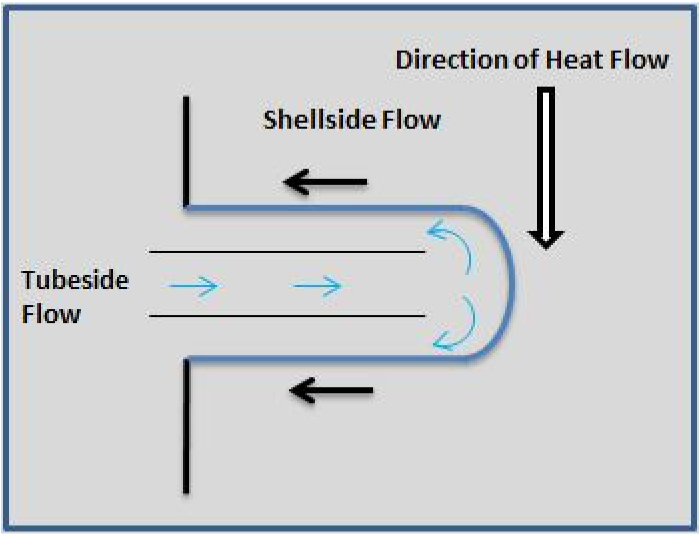

Figure 2.3: Principle of Operation

In the Bayonet Tube Heat Exchanger , the tubeside fluid flows through the inner tubes, turns around at the open end of the inner tubes and flows through the annulus between the inner and outer tubes. In this type of heat exchanger, simultaneous heat transfer occurs between the fluid flowing through the annulus and the shellside fluid as well as the fluid flowing through the inner tubes.

2.4.3 Advantages over STHE

In the STHE the large temperature difference between the fluids can cause thermal expansion between the shell and tube, and as a result can lead to thermal stresses. The thermal stresses are present between the tubes and the tubesheets and also between the shell and the tubesheets. Also thermal stresses are induced between the inside and the outer of the tubesheets. In order to take care of the thermal stresses expansion joints or bellows are provided. Generally, in the U-tube kind of construction there is no problem of thermal expansion between the shell and the U tubes, but as the chambers are separated into two volumes: hot chamber and a cold chamber with a pass partition in between, there is a large difference of temperature along tubesheet that induces thermal stresses.

In the Bayonet Tube Heat Exchanger, the construction of this heat exchanger permits free expansion of the bayonet tubes inside the shell caused by the temperature difference between the bayonet tubes and the shell, and the thick tubesheet in contact with the shell is not exposed at high temperature and there is a uniform temperature distribution along the tubesheet making thermal design and selection of materials simple. Also reducing the thermal stresses provides a safe and low cost design.

The Bayonet tube Heat Exchanger [5] can be used for the purposes where there exists a large temperature difference between the fluids on the shellside and the tubeside. Its application can be found in waste heat recovery system as well where the waste gas that could be corrosive and is at very high temperature and a large temperature drop is to be achieved for the gas.

2.5 Triple Concentric Tube Heat Exchangers

Triple Concentric Tube heat Exchangers are the modification of the Tube in tube heat exchangers. It consists of a concentric tube over the double pipe. In addition to the two fluids flowing inside the double pipe, there is another fluid flowing through the third concentric tube.

The case study was done by Carlos A. Zuritch for the design of Triple Concentric

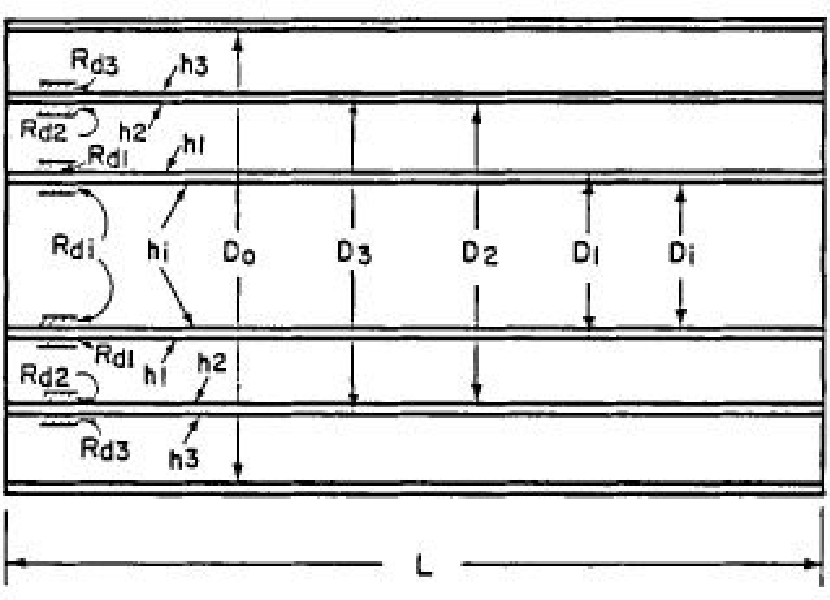

Figure 2.4: Triple Concentric Tube Heat Exchangers[5]

Tube Heat Exchanger [5]. The objective was to develop analytical correlations for Triple Concentric Tube Heat Exchanger. The relations were based for calculation of heat transfer coefficients for the flow through tubes and annulus. A case study and computational procedure was been carried out and presented.

With the computational procedure the results obtained were compared with the analytical equation's and it was seen that the results were in excellent agreement with each other. Hence the conclusion of the paper was that the correlations developed can be used for design purposes.

When the fluid is flowing through the annulus, the heat transfer coefficient of the fluid is affected by simultaneous heating or cooling of the annulus because of the heat transfer both from the inner tube and the outer tube. Hence for Reynolds number calculation equivalent diameter is to be used. Due to this there will be two values of Reynolds number obtained, one for the outer surface of inner tube and other for the inner surface of the annulus. Also wetted perimeter is to be used for calculation of pressure drop for the annulus.

2.6 Design of Kettle Reboilers

Kettle Reboilers consists of an enlarged shell [6], of which the lower part is occupied by the tube bundles in which fluid is flowing and the upper part is provided for vapour-liquid disengagement. Also there is a weir which maintains level of liquid such that the tube bundle remains submerged inside the liquid level.

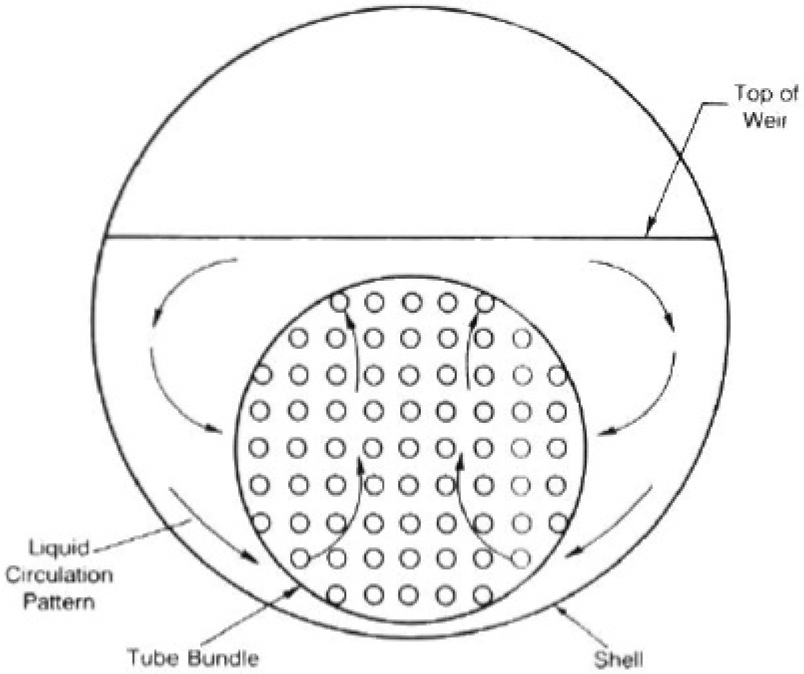

Figure 2.5: Liquid Circulation in Kettle Reboiler [6]

The liquid inside the reboiler completely surrounds the tube bundle to form a "large pool", also the amount of liquid present in this type of reboiler is large. The liquid inside the reboiler is not stagnant; hence the boiling taking cannot be considered as pool boiling. Density differences arising due to vaporization of the liquid cause's strong up flow throughout the bundle. Hence for this reason, the heat transfer coefficient calculated will always be higher than the heat transfer coefficient of pure pool boiling

The boiling of liquid around the tube bundles produces vapour in the bundle and hence leads to large natural circulation of the liquid up to the top of the tube bundle and then recirculates to the bottom of the tube bundle. This recirculation of the liquid enhances the nucleate boiling with convection effects throughout the bundle

For calculation of heat transfer coefficient [13] (α_b) it involves nucleate boiling heat transfer as well as convective heat transfer (α_{cb}) .

1. Calculation of the pool boiling heat transfer coefficient depends on physical properties of the liquid as well as the temperature profile at the wall or the heat flux.

2. Calculation of convective heat transfer coefficient is the function of liquid heat transfer coefficient (α_l) and the correction factor (F).

Chapter 3

Shell and Tube Heat Exchanger

3.1 Design of STHE

The design of STHEs comprises two distinct activities: thermal design and mechanical design. In the thermal design, the heat exchanger is sized, which means that all the principal construction parameters such as shell type and diameter, number of tubes, diameter of tubes and thickness, tube length, tube pitch, number of tube passes, baffle spacing and cut, and nozzle sizes are determined. In mechanical design, detailed calculations are carried out to determine the dimensions of various components such as tubesheets, flanges, shell, etc. and a complete bill of materials and engineering drawings such as bundle assembly and setting plan drawings are generated.

For the proper design of the STHE [14], the designer needs to follow a certain steps for carrying out optimum performance of the heat exchanger. The steps that need to be followed are indicated in the figure below.

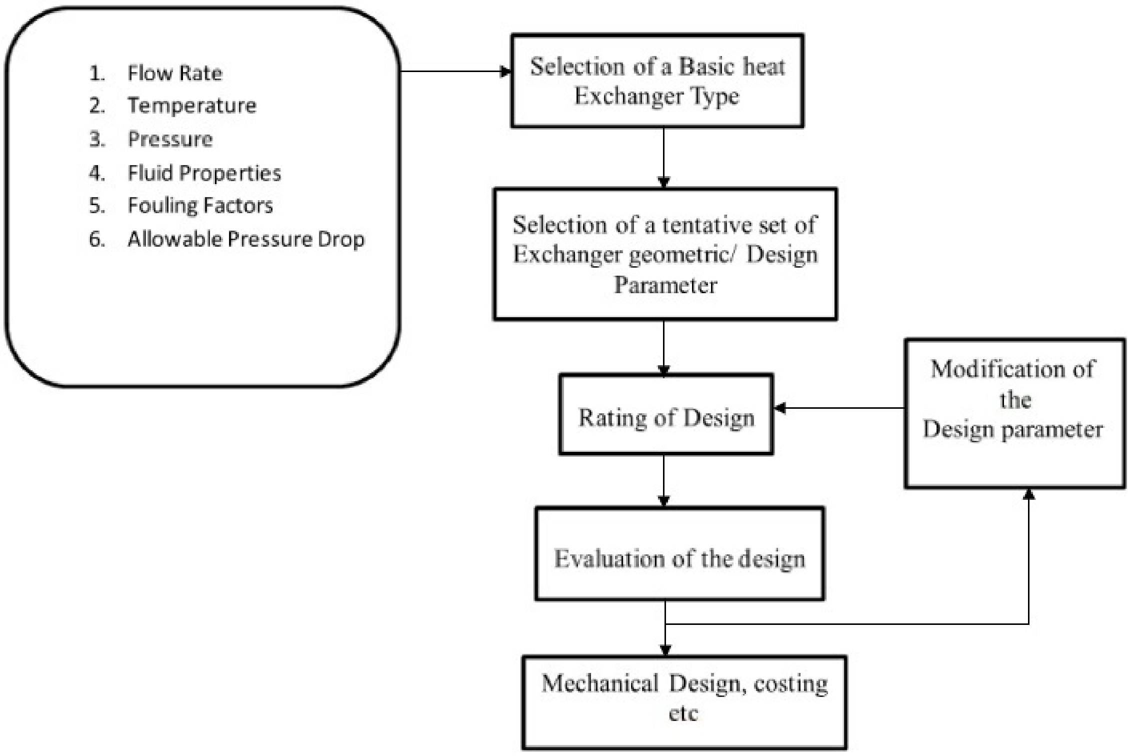

Figure 3.1: Design Procedure

3.2 Rating of Heat Exchanger Design

Rating of a heat exchanger [7] means evaluating the thermal and hydraulic performance of a fully specified heat exchanger. The input parameters in the rating consists of heat exchanger geometrical parameters (constructional parameters), process conditions (temperature, pressure, flow rates), and fluid and material properties (density, specific heat, thermal conductivity, etc.).

The output results from the rating problem is the overdesign available on the surface area provided. Another output is the pressure drop results for the shellside and the tubeside. For obtaining a proper performance from the output results available after rating of a heat exchanger, modifications in the exchanger configuration must be made appropriately to obtain the desired optimum solution.

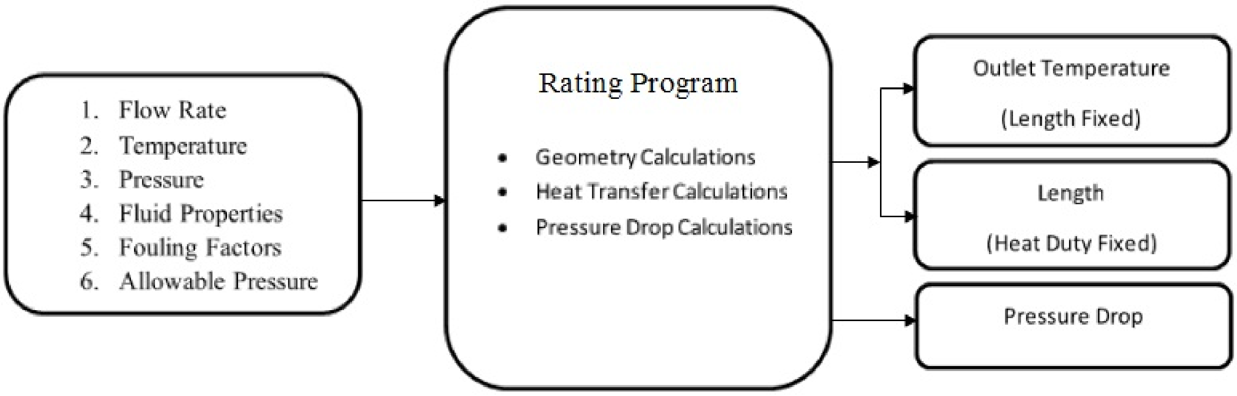

Figure 3.2: Rating of Heat Exchanger [7]

3.3 Case Study I

3.3.1 Process Input Data Sheet

Table 3.1: Construction Data

Sr. No.	Description	Units	Parameters
1.	Tube O.D	mm	25
2.	Tube Thickness	mm	2
3.	Tube Pitch	mm	32
4.	Tube Layout		30 °
5.	Tube Material		SS321

Table 3.2: Process Data Sheet

Sr. No	Description	Units	Operating Conditions			
			Shellside		Tubeside	
	Fluid		Recycle Gas		Reactor Effluent	
			In	Out	In	Out
1	Flow Rate	Kg/s	5.7884		57.597	
2	Operating Temperature	°C	227.4	368.7	397.5	367.8
3	Operating pressure	KPa(G)	12772		11281	
4	Properties of Fluid					
	Density	kg/m3	11.77	9.216	39.44	41.06
	Viscosity	cP	0.014	0.018	0.029	
	Specific Heat	KJ/Kg.K	8.0386	8.3736	3.9565	3.881
	Thermal Conductivity	W/m.K	0.221	0.264	0.2024	0.1954
5	Allowable Pressure Drop	KPa	19.613		58.84	
6	Fouling Resistance	m^2K/W	0.000172		0.000344	
7	Heat Duty	MW	6.1			
8	Heat Duty Multiplier		1.1			

3.3.2 Design Steps Involved

Based on the input data sheet, , first step is the selection of type of heat exchanger according to the TEMA Standards [2] :

1. The heat duty (Q) is calculated for the shellside and the tubeside.

$$Q = m_h.C_{ph}.\triangle T_h = m_c.C_{pc}.\triangle T_c \tag{3.1}$$

2. Calculate the Log Mean Temperature Difference for the process parameters and the correction factor for Log Mean Temperature Difference (LMTD) for multi-pass heat exchangers.

$$LMTD = (\theta_{in} - \theta_{out}/ln(\theta_{in}/\theta_{out}) \tag{3.2}$$

3. Assume the permissible velocity of fluid flowing along the tubeside and Calculate the heat transfer coefficient (α_i) for the tubeside [15], wall thermal resistances and the fouling resistances associated with tubeside.

By using Dittus Boelter's Equation;

$$Nu_i = 0.023.Re^{0.8}.Pr^{0.4} \tag{3.3}$$

$$\alpha_i = Nu_i.k/d_e \tag{3.4}$$

4. Assume the value for shellside heat transfer coefficient (α_o) for the fluid flowing on the shellside according to the TEMA standards for the fluid and calculate the Overall Heat Transfer Coefficients (U).

$$(1/U) = (1/\alpha_o) + ((1/\alpha_i) \times (d_o/d_i)) + (d_o \times ln(d_o/d_i)/2k_t) \tag{3.5}$$

5. With the value of the Overall heat Transfer coefficient calculate the Heat Transfer Surface area (A_s).

$$Q = U.A_s.LMTD \tag{3.6}$$

6. Assume the effective length of the tubes (L_{eff} and calculate the estimate of the number of tubes (N_t) required for the heat exchanger.

$$A_s = \pi d_o.L_{eff}.N_t \tag{3.7}$$

7. Based on the data calculated, calculate the inside diameter of the shell and assume the baffle cut and spacing based on TEMA standards.

8. Correct the shellside heat transfer coefficients based on the empirical correlations estimated by the Bell Dalaware Method[3].

9. Calculate the pressure drop on the shellside based on the Bell Dalaware method

10. With the corrected Shellside heat transfer coefficient, calculate the Overall Heat Transfer Coefficient again and the heat transfer surface area.

11. After providing some percentage of overdesign/ underdesign, the Overall heat transfer coefficients is then again recalculated and results are obtained.

$$Overdesign\% = (A_{reqd} - A_{prov})/A_{prov} \tag{3.8}$$

12. The pressure drop ($\triangle p$) on the tubeside is then calculated and checked if the values are less than the allowable pressure drop range.

$$\triangle p = 4fL_{eff}v_{tf}^2 \times \rho_f/2d_i \tag{3.9}$$

13. The parameters such as velocity of fluid (v_{tf}) on the tubeside, length, baffle cut and baffle spacing of the tubes are then further varied for obtaining optimum result.

3.3.3 Output Results

Table 3.3: Output Results

Sr. No.	Description	Units	Operating Conditions	
			shellside	Tubeside
1.	TEMA Type		DEU	
2.	Mean Temperature Difference	°C	53.3	
3.	Heat Transfer Coefficient	W/m^2K	2327.1	5526.5
4.	Pressure Drop	KPa	17.104	55.949
5.	Overall Heat Transfer Coefficient	W/m^2K	725.08	
6.	Effective Heat Transfer Area	m^2	173.51	
7.	Total Heat Duty	MW	6.71	
8.	Shell I.D	mm	880	
9.	Overdesign	%	2.91	
10.	Effective Length of Tube	m	4.3	
11.	Tube Count		544	
12.	Baffle Type		Single Segmental	
13.	No. of Baffles		10	
14.	Baffle Cut	%	30	
15.	Central Spacing	mm	295.68	

3.4 RESULTS and DISCUSSIONS

Parametric study was done on the design of heat exchanger to select an optimum design. The parametric study was based on the baffle cut and baffle spacing. As we know that the baffle cut and baffle spacings play an important part for the flow of fluid on the shellside. The flow of fluid on the shellside must be well defined and the number of dead zones needs to be minimized to increase the performance of the heat exchange of the fluid on the shellside.

3.4.1 Comparison with HTRI

Table 3.4: Comparison with HTRI

Sr. No	Description	Units	Calculated	HTRI
1	Log Mean Temperature Difference	C	53.3	52.7
2	Shellside Heat Transfer Coefficient	W/m.K	2327.1	2444.4
3	Tubeside Heat Transfer Coefficient	W/m.K	5526.5	5526.3
4	Overall Heat Transfer Coefficient	W/m.K	725.08	757.83
5	Shellside Pressure Drop	KPa	17.104	16.35
6	Shellside Pressure Drop	KPa	55.94	55.98
7	Overdesign	%	2.91	3.3

3.4.2 Variable Baffle Cut and constant Baffle Spacing

Stream flow Analysis

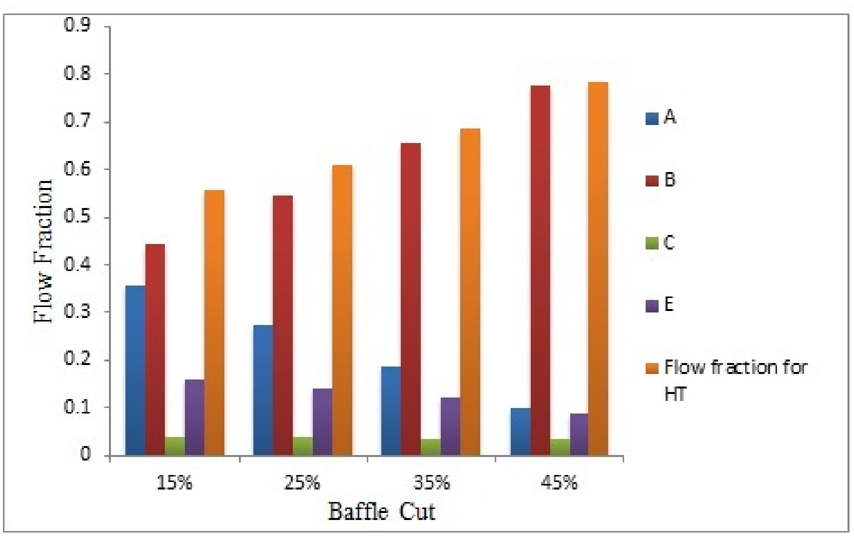

Figure 3.3: Stream Analysis

Crossflow velocity and Window Velocity on the shellside fluid

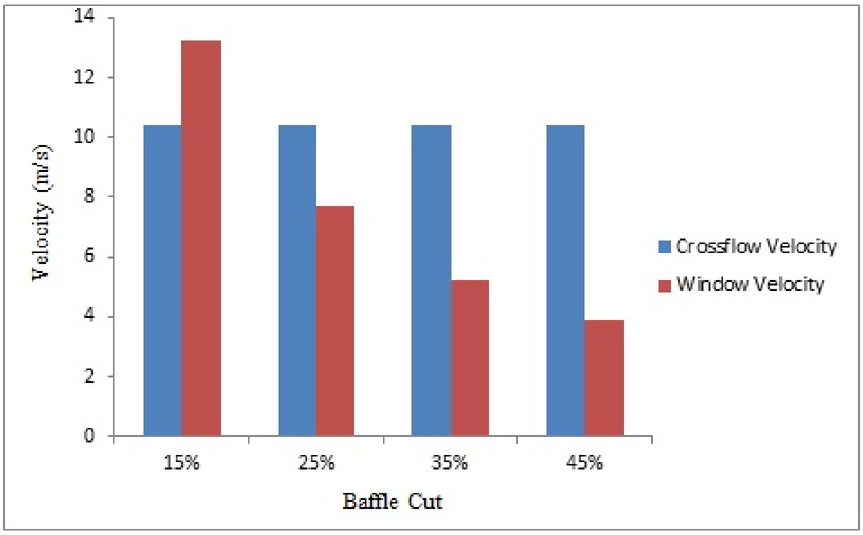

Figure 3.4: Crossflow and Window Flow

Heat Transfer Coefficient

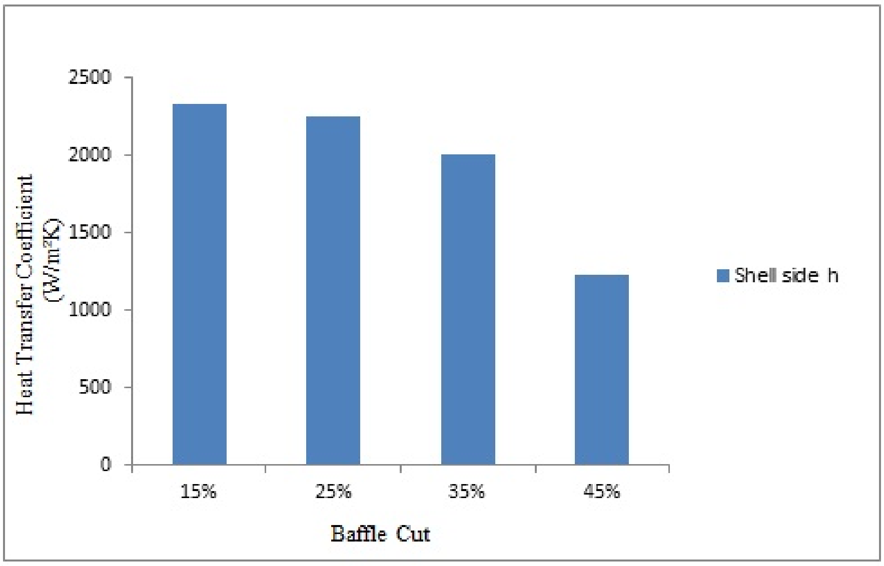

Figure 3.5: Heat Transfer Coefficient

Pressure Drop

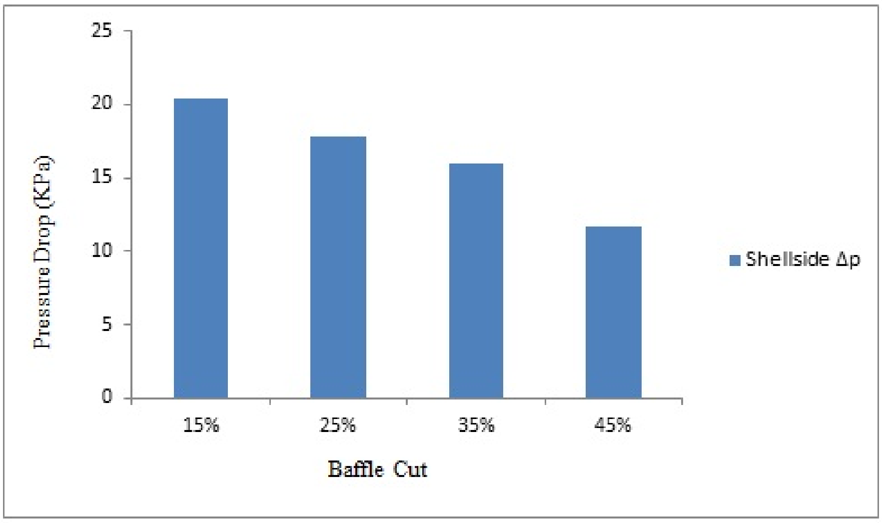

Figure 3.6: Pressure Drop

3.4.3 Variable Baffle Spacing and Constant Baffle Cut

Stream flow Analysis

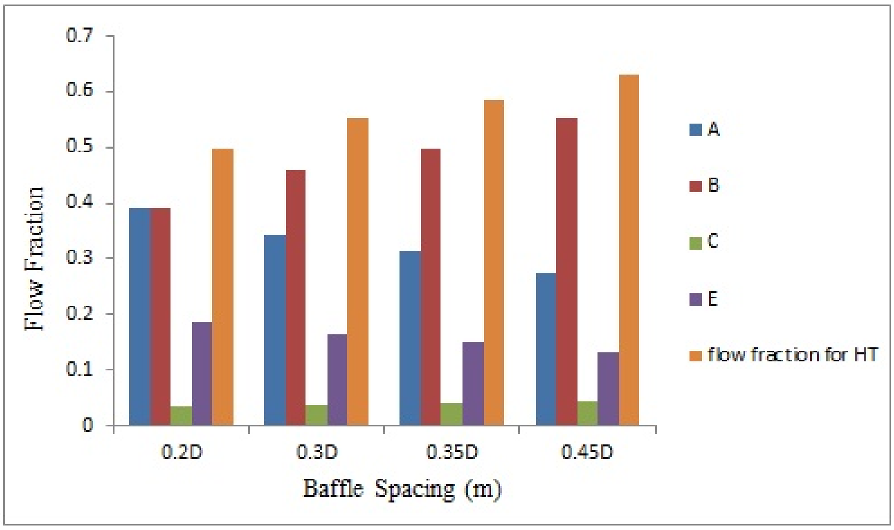

Figure 3.7: Stream Analysis

Crossflow velocity & Window Velocity on the shellside fluid

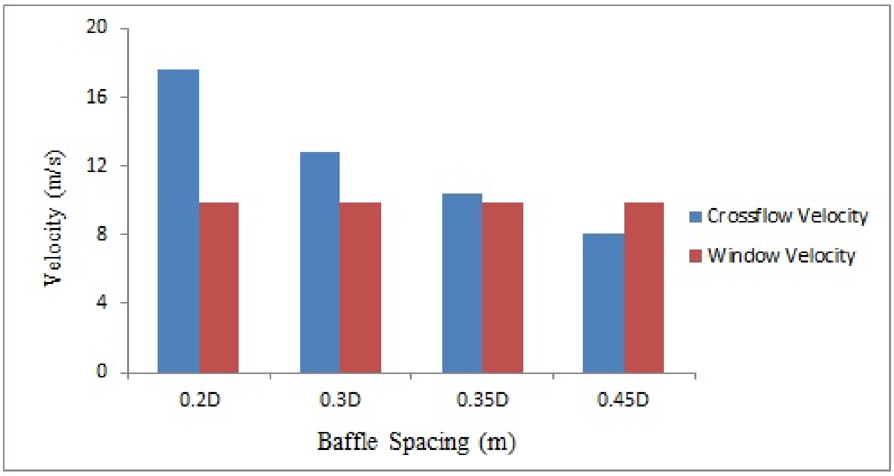

Figure 3.8: Crossflow and Window Flow

Heat Transfer Coefficient

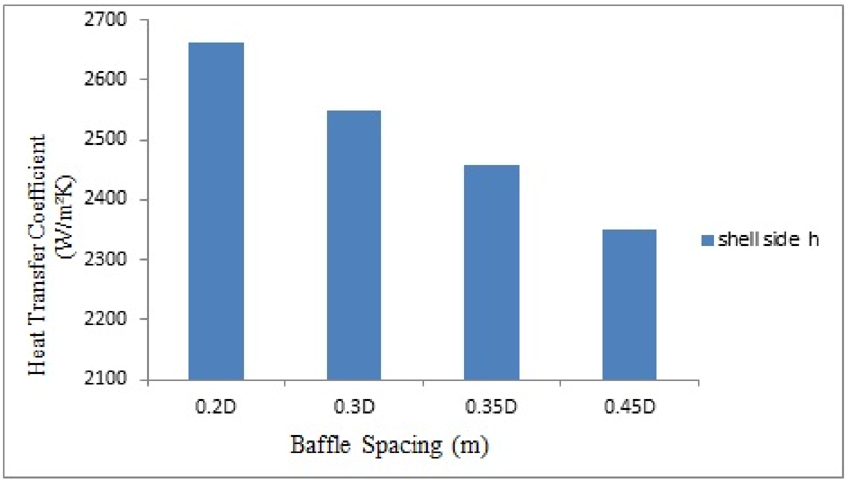

Figure 3.9: Heat Transfer Coefficient

Pressure Drop

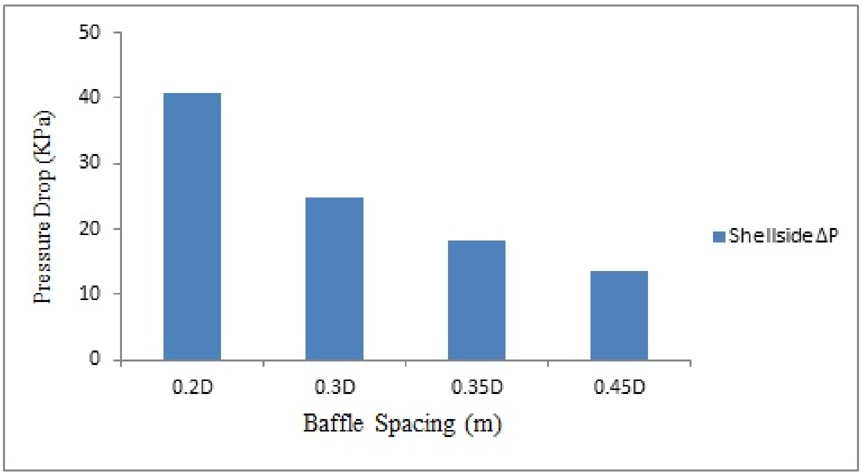

Figure 3.10: Pressure Drop

3.5 Conclusion

From the above parametric study, certain conclusions were drawn:

1. The baffle cut and baffle spacing should be selected in such a way that the difference between the crossflow velocity and the window velocity on the shellside should be as small as possible. The lesser the difference between them, better will be the flow distribution and lesser will be the number of dead zones on the shellside.

2. Increase in the baffle cut or the baffle spacing will improve the flow fraction on the shellside. But at the same time, it will reduce the heat transfer coefficient on the shellside, and reduce the pressure drop. Hence, baffle cut and baffle spacing should be optimally selected to improve the heat transfer coeffi cient, while utilizing the shellside pressure drop.

3. Increasing the baffle spacing increases the cross flow area and decreases the cross flow velocity of fluid on the shellside., Since pressure drop is directly proportional to the square of velocity, shellside pressure drop also decreases.

4. Increase in the baffle cut reduces length of crossflow i.e. the number of tube rows

crossed, and since pressure drop is directly proportional to the length of flow, pressure drop in crossflow also decreases. Further, increasing the baffle cut results in reduction of window flow velocity and consequently, window pressure drop.

5. For the tubeside, velocity of the fluid in the tubes and the length of the tubes is an important parameter, as for the pressure drop, it is directly proportional to the square of velocity, and also linearly proportional to length. Increasing the velocity would increase the heat transfer coefficient on the tubeside, but at the cost of increase in the pressure drop, so the trade-off has to be considered properly.

Chapter 4

Bayonet Tube Heat Exchanger

4.1 Design of Bayonet Tube Heat Exchanger

Referring from the literature [16] several researchers have done experiments and analysis on the Design of Bayonet Tube Heat Exchanger and their applications. To account for the study on Three fluid shell and tube heat exchanger, the study and design of Bayonet Tube Heat Exchanger becomes very important as this also consists of two thermally conducting walls exchanging heat simultaneously, but with the use of only two fluids. Also the first thermal wall lies between the outer tube of the bayonet tube and the shell. The second thermal wall lays between the outer tube and the inner tube. Hence there are two individual thermal resistances acting simultaneously.

4.2 CASE Study II

4.2.1 Process Data Sheet

Table 4.1: Input Process Data [4]

Sr.No.	Description	Units	Operating Conditions			
			Shellside		Tubeside	
			Natural gas + Air		Air	
			In	Out	In	Out
1.	Flowrate	Kg/s	0.082		0.65	
2.	Operating Temperature	°C	1400	358	42.47	188
3.	Operating pressure	KPa	101.325		101.325	
4.	Properties of Fluid					
	Density	Kg/m^3	0.2002	0.5029	1.1180	0.7648
	Viscosity	$mN.s/m^2$	0.069	0.0304	0.0196	0.0258
	Specific Heat	KJ/Kg.K	1.2533	0.9813	0.9955	1.023
	Thermal Conductivity	W/m.K	0.001159	0.049	0.02589	0.03502

Also with the Process condition there were constructional parameters that were also considered in the experimental analysis. So also, this parameters have been tabulated for the thermal design purpose.

Table 4.2: Construction Data Input

Sr. No	Description	Units	Operating Parameters
1.	Inner Tube O.D	mm	50.8
2.	Inner Tube Thickness	mm	2
3.	Length of Tubes	mm	2295
4.	Outer Tube OD	mm	102.26
5.	Outer Tube Thickness	mm	6.02
6.	No. of Tubes		7
7.	Length of Outer Tubes	mm	1770
8.	Tube Pitch	mm	181.74
9.	Tube Layout		30 °
10.	Tube Material		Mild Steel
11.	Shell ID	mm	600
12.	Length of Shell	mm	2000
13.	Baffle Type		Single Segmental
14.	No. of Baffles		3
15.	Baffle Spacing	mm	460
16.	Baffle Cut	%	30
17.	Shell Material		Carbon Steel

4.2.2 Design Steps Involved

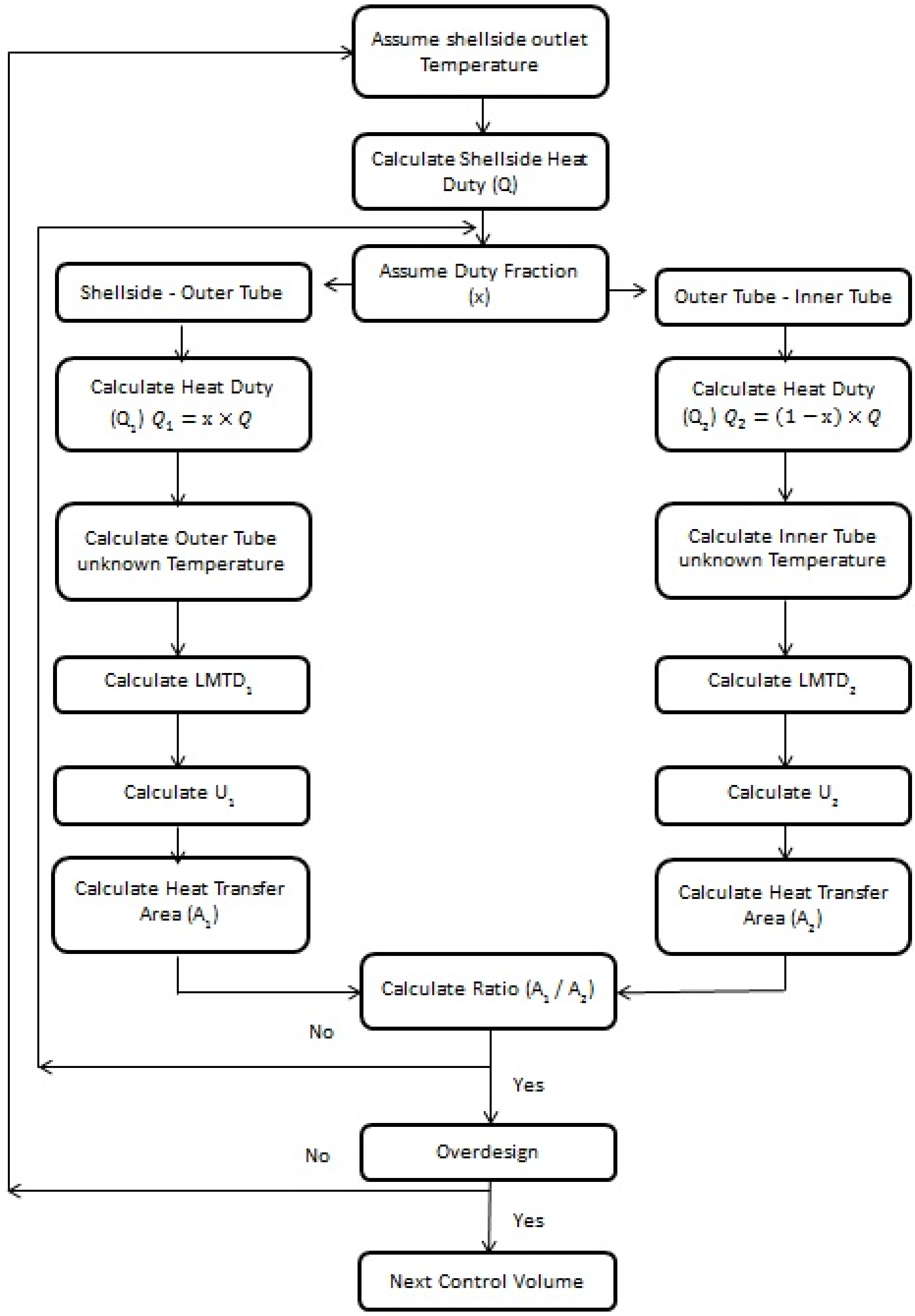

Figure 4.1: Design Flow Chart

In the design of Bayonet heat exchanger, the design steps are more complex than the conventional shell and tube heat exchanger. For the case whatever the Heat Duty is transferred from the hot fluid on the shellside is further distributed in proportions to the fluid on the outer tube and subsequently to the inner tube. Hence the proportion of heat duty distribution becomes a governing factor that needs to be analysed for the design. Moreover the design procedure is in such a way as if we are designing two STHE simultaneously which also adds complex to the problem. The following are the steps which were followed for the design of the same:

1. Calculate the Shellside heat duty (Q_s) and overall heat duty (Q) of the heat exchanger based on the process conditions.

$$Q_s = Q = m_s.Cps.\triangle T_s \tag{4.1}$$

2. Now as we know that there will be two heat duty corresponding to the shell-outer tube(Q_1) and outer tube-inner tube(Q_2) that will sum up to the total Heat Duty. The proportion is to be given to the corresponding heat duties in a way that it fulfills the design.

$$Q = Q_1 + Q_2 \tag{4.2}$$

3. Initially the proportion (x) needs to be to be assumed and the heat duties corresponding to the shell-outer tube and the outer tube-inner tube needs to be evaluated.

$$Q_1 = x \times Q; \tag{4.3}$$

$$Q_2 = (1 - x) \times Q \tag{4.4}$$

4. With the Heat duty known from the previous step calculate the unknown temperature for the outer tube through energy balance and similarly calculate the temperature for the inner tube.

$$Q_1 = m_{otf}.C_potf.\triangle T_{ot} \tag{4.5}$$

$$Q_2 = m_{itf}.C_pitf.\triangle T_{it} \tag{4.6}$$

5. For the Shell to the outer tube calculate the log mean temperature difference (LMTD_1), and similarly calculate the log mean temperature difference for the outer tube to the inner tube (LMTD_2).

6. Calculate the Heat transfer coefficient (α_s) for the fluid on the shellside using Bell Dalaware Method; inside of the outer tubes (α_1), conductance resistances and the fouling resistances on the outer and inner of the outer tube, and calculate the Overall Heat Transfer coefficient (U_1)for the same.

$$Re = \rho.v_{ot}.d_h \tag{4.7}$$

$$d_h = D_i - d_o \tag{4.8}$$

$$Nu_{ot} = 0.023.Re^{0.8}.Pr^{0.4} \tag{4.9}$$

$$\alpha_1 = Nu_{ot}.k/d_{e1} \tag{4.10}$$

$$(1/U_1) = (1/\alpha_s) + ((1/\alpha_1) \times (D_o/D_i)) + (D_o \times ln(D_o/D_i)/2k_{ot}) \tag{4.11}$$

7. Calculate the Heat transfer coefficient for the fluid on the inside of the inner tube (α_i) and also on the outside of the inner tube (α_2) [17], conductance resistance between the inner tube and the outer tube and the fouling resistances on the outer and inner of the inner tube, and calculate the Overall Heat Transfer coefficient (U_2) for the same.

$$\alpha_2 = Nu_{ot}.k/d_{e2} \tag{4.12}$$

$$Nu_i = 0.023.Re_i^{0.8}.Pr^{0.4} \tag{4.13}$$

$$\alpha_i = Nu_i.k/d_i \tag{4.14}$$

$$(1/U_2) = (1/\alpha_1) + ((1/\alpha_2) \times (d_o/d_i)) + (d_o \times ln(d_o/d_i)/2k_t) \tag{4.15}$$

8. Based on the Log Mean Temperature Difference (LMTD) , Overall Heat Transfer Coefficients, calculate the Heat Transfer Area for the shell-outer tube (A_{s1}) and the outer tube-inner tube (A_{s2}) respectively.

$$Q_1 = U_1.A_{s1}.LMTD_1 \tag{4.16}$$

$$Q_2 = U_2.A_{s2}.LMTD_2 \tag{4.17}$$

9. As we know that the effective length of heat transfer for the inner tube and outer tube has to be the same, the fraction of heat duty is iterated in the third step again till the till the difference in length for the outer tube and inner tube converges.

10. Based on the overdesign or under-design that needs to be given, the design is iterated in the first step till the overdesign value converges with the value desired.

4.2.3 Output Data Sheet

Table 4.3: Output Results

Sr. No	Description	Units	Operating Conditions	
			Shell - Outer Tube	Outer Tube - Inner Tube
1	Total Heat Duty	KW	95.463	
2	Heat Duty Distribution	KW	80.633	14.83
3	Log Mean Temperature Difference	°C	649.63	49.36
4	Overall Heat Transfer Coefficient	$W/m^2.K$	11.271	59.34
5	Heat Transfer Area	m^2	11.468	5.097
6	Fraction of Heat Duty		0.845	0.155
7	Overdesign	%	-60	

4.2.4 Results

Shellside Temperature Profile along the length

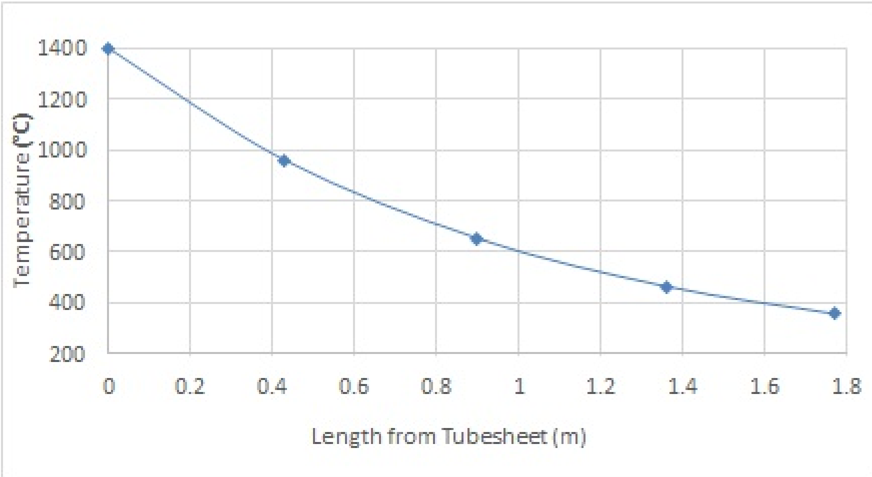

Figure 4.2: Shellside Temperature Profile Along Length

Tubeside Temperature Profile along the length

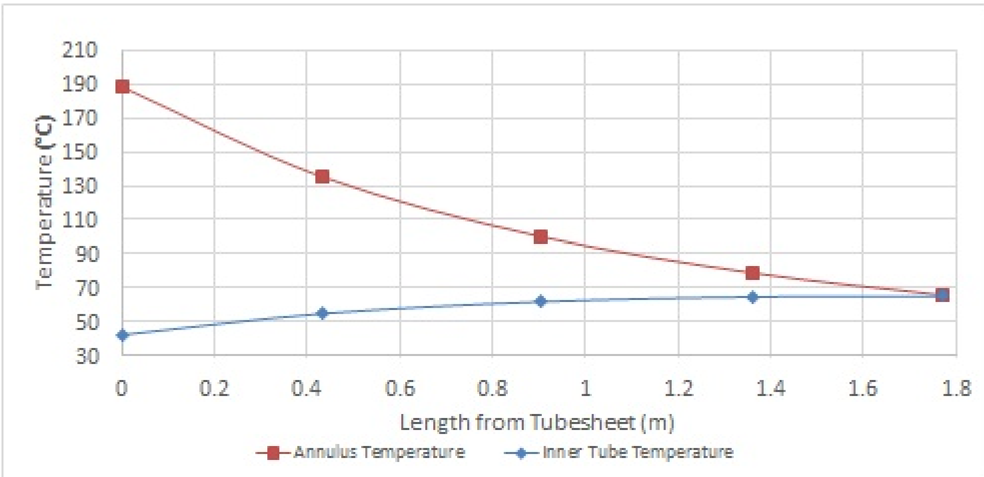

Figure 4.3: Tubeside Temperature Profile Along Length

Fraction of heat Duty

The fraction indicates the amount of heat duty transferred to the outer tube from the total heat duty achieved on the shellside. The fraction of heat duty is being plotted along the length of the heat exchanger.

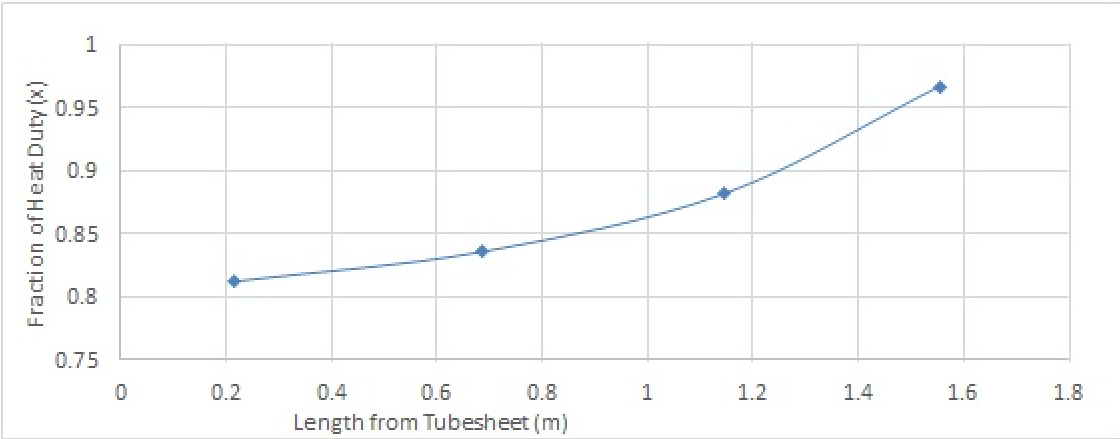

Figure 4.4: Fraction of Heat Duty to the Outer Tube Along Length

4.3 Conclusions

From the study on Bayonet Tube Heat Exchanger [4], there are certain conclusions that were carried out:

1. The Overall Heat Transfer coefficients and the LMTD plays an important role for the fraction distribution on the shell-outer tube and the outer tube-inner tube as both are acting simultaneously on the Heat Exchanger.

2. The inner tube is open at one of its end to the outer tube, so the temperature at the opening for the inner tube and the outer tube must be equal.

3. Since there is gas flowing on the shellside, the value of heat transfer coefficient is very less and due to this overall heat transfer coefficient for shell-outer tube is dominated by the shellside fluid. But because of large LMTD fraction of heat transfer for the shell-outer tube is more compared to the outer tube-inner tube.

4. Also it was observed that with this type of heat exchanger, it can be used for high temperature drop applications with the use of limited constraints.

5. With the comparison with the paper, it was observed that with the calculations, underdesign value was coming to be around 60 % whereas there should not have been any underdesign or overdesign. The difference in the value were because the heat exchanger shellside is at very high temperature, and because of this radiation would be the dominating mode of heat transfer compared to convection.

6. For this purpose the values of Overall Heat transfer coefficient on the shellside would have been much higher compared to the values obtained through the calculation.

tu

Chapter 5

Three Fluid Shell and Tube Heat Exchanger

5.1 Performance Analysis of a Shell and Double Concentric Tube Heat Exchanger using CFD[1]

In the literature , study was carried out for analysis of performance of the shell and double concentric tube heat exchanger using CFD software[1]. During the analysis the Shell and Tube Heat Exchanger is compared with the shell and double concentric tube heat exchanger. First the analysis was done considering different mass flow rates, and different diameters of the inner tube and the performance of the shell and double concentric tube heat exchanger was compared. For the purpose of comparison, the mass flow rate for the shellside fluid for the Shell and Tube Heat Exchanger was split into two. Half the mass flow rate for the shellside and other half to the inner tube. The results obtained from the literature were compared and validated with the design methodology developed for three fluid shell and tube heat exchanger

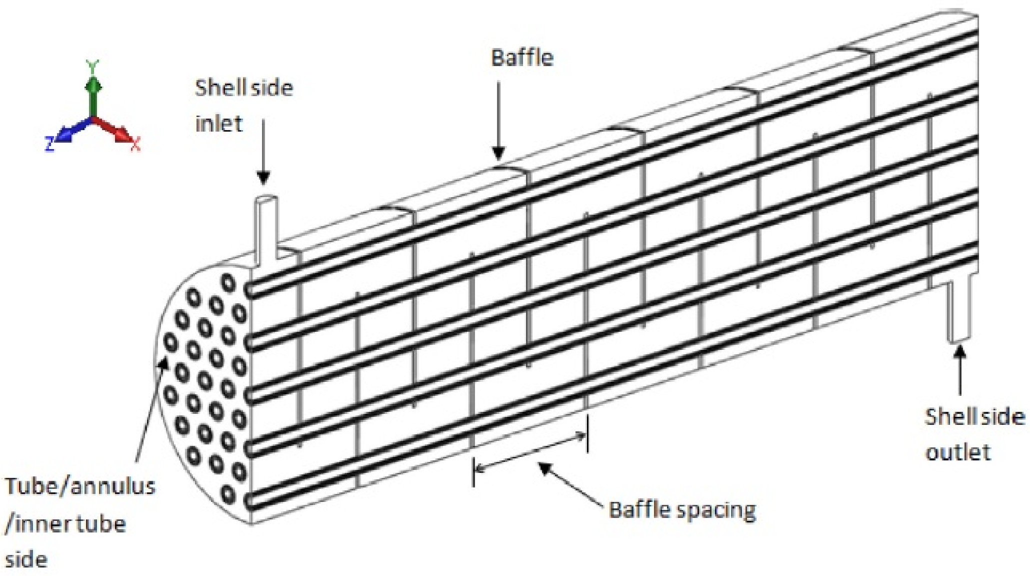

Figure 5.1: Isometric View of STHE [1]

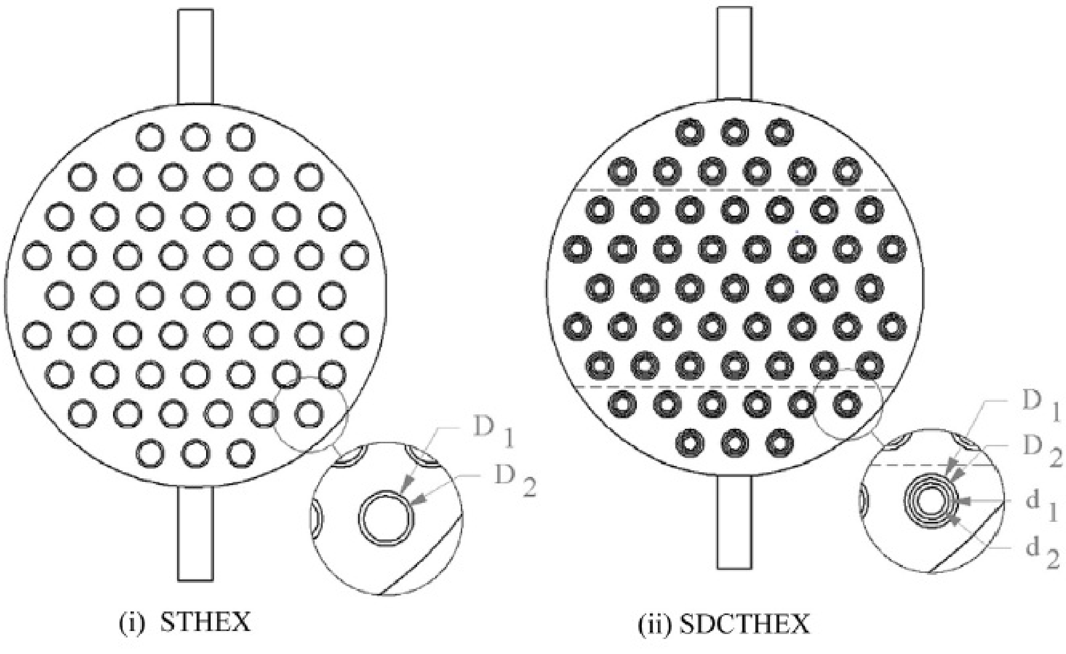

Figure 5.2: Tubes Arrangement inside Heat Exchanger[1]

5.2 CASE STUDY III

Table 5.1: Constructional Parameters

Sr. No	Description	Units	Operating Parameters
1.	Inner Tube O.D	mm	12
2.	Inner Tube Thickness	mm	2
3.	Effective Tube Length	mm	1270
4.	Outer Tube OD	mm	24
5.	Outer Tube Thickness	mm	2
6.	No. of Tubes		55
7.	Pitch Ratio		1.667
8.	Tube Layout		30 °
9.	Shell ID	mm	337
10.	Baffle Type		Single Segmental
11.	No. of Baffles		12
12.	Baffle Spacing	mm	100
13.	Baffle Cut	%	30
14.	Tube Material		Mild Steel
15.	Shell Material		Carbon Steel

Table 5.2: Process Input Data

Sr. No	Description	STHE		SDCTHE		
		Shell	Tube	Shell	Annulus	Inner Tube
1	Fluid Name	Water	Engine Oil	Water	Engine Oil	Water
2	Mass Flow Rate (kg/s)	40	10.14	20	10.14	20
3	Inlet Temperature (°C)	120	20	120	20	120
4	Outlet Temperature(°C)	114.92	31.05	111.87	35.18	114.16

5.3 Output Results

5.3.1 STHE

Table 5.3: STHE Output Results

Sr. No	Description	Units	Operating Conditions	
			Shellside	Tubeside
1	Total Heat Duty	KW	467.31	
2	Log Mean Temperature Difference	C	91.904	
3	Overall Heat Transfer Coefficient	W/m.K	876.5	
4	Heat Transfer Area	m2	5.267	
5	Pressure Drop	KPa	144.83	0.37
6	Overdesign	%	-9.2	

5.3.2 SDCTHE

Table 5.4: SDCTHE Output Results

Sr. No	Description	Units	Operating Conditions	
			Shell - Annulus	Annulus - Inner Tube
1	Total Heat Duty	KW	643.21	
2	Heat Duty Distribution	KW	373.73	269.48
3	Log Mean Temperature Difference	°C	88.33	89.41
4	Overall Heat Transfer Coefficient	$W/m^2.K$	729.49	1039.07
5	Heat Transfer Area	m^2	5.267	2.634
6	Fraction of Heat Duty		0.581	0.519
7	Overdesign	%	-9.2	

5.4 Results and Discussions

5.4.1 Comparison with Literature

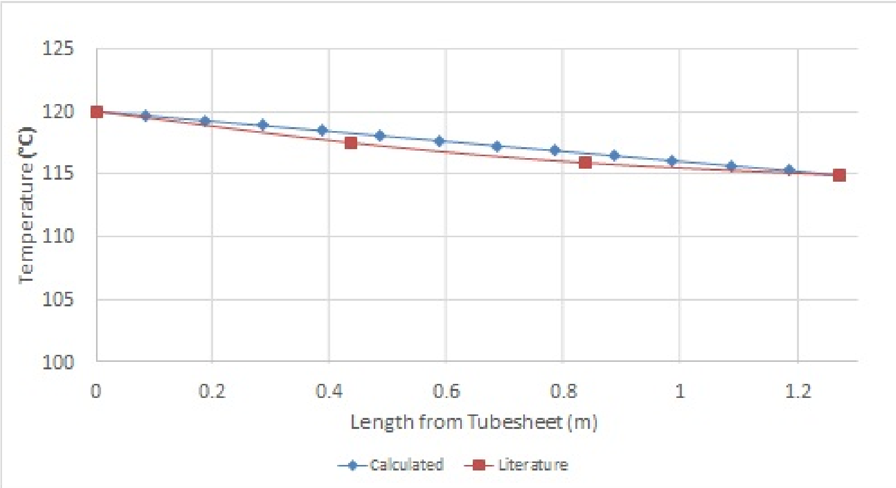

Figure 5.3: STHE Shellside Temperature Profile

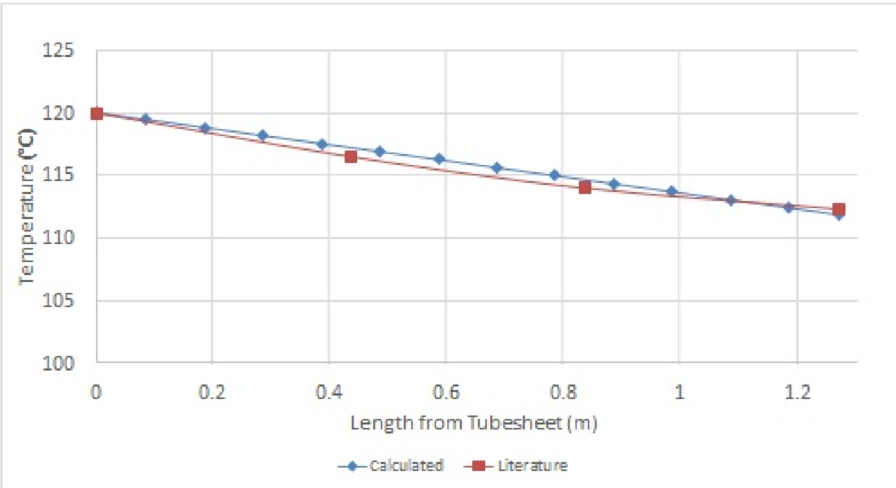

Figure 5.4: SDCTHE Shellside Temperature Profile

5.4.2 SDCTHE vs STHE Comparison

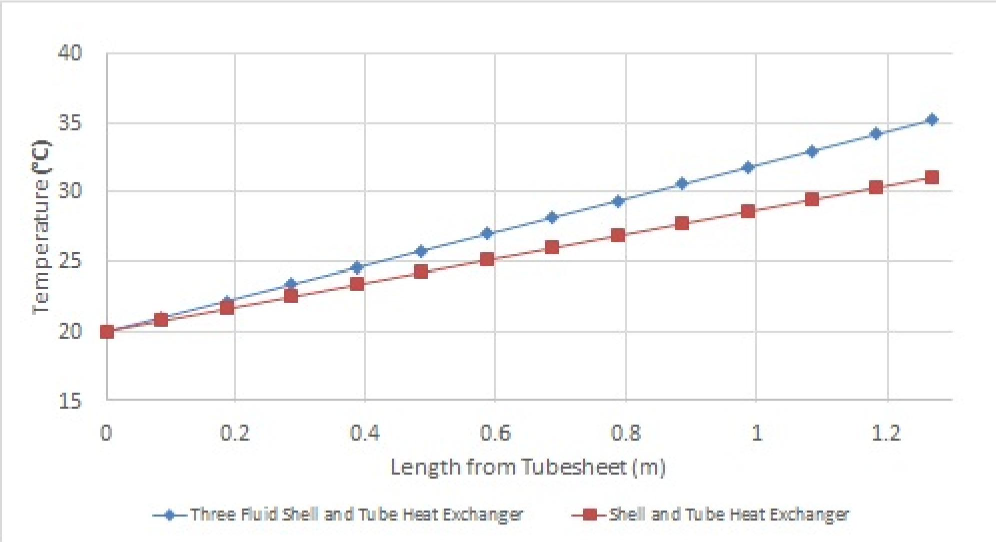

Figure 5.5: SDCTHE vs STHE Temperature Profile

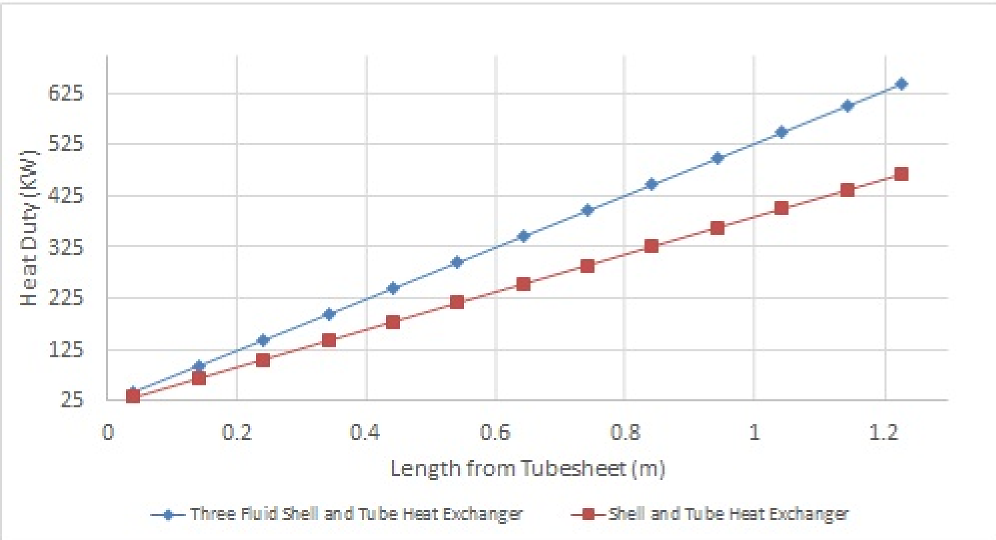

Figure 5.6: SDCTHE vs STHE Heat Duty

5.4.3 Fraction of Heat Duty

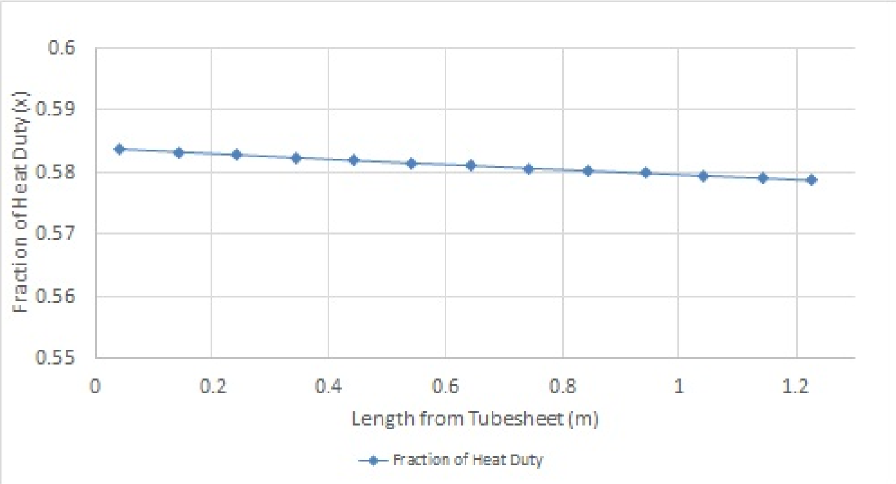

Figure 5.7: Fraction of Heat Duty to Shellside

5.5 Conclusion

The calculations done using analytical method was compared with the case study from the literature. Certain conclusions were made which are as follows.

1. The calculations carried out for the comparison between Shell and Tube Heat Exchanger and Shell and Double Concentric Tube Heat Exchanger was found to be in good agreement with CFD analysis done in the literature

2. The overall heat transfer rate for SDCTHE for the same mass flow rate was greater compared to the STHE. The heat transfer rate was 37.67 % greater than STHE.

3. The heat transfer rate per unit pressure drop for SDCTHE was higher than STHE. Hence the pressure drop for SDCTHE is less compared to STHE for the same heat duty.

4. The effect of inner tube increases the heat transfer coefficients which enhances the performance of the heat exchanger

5.6 Implementation on Project

5.6.1 Three Fluid Shell and Tube Heat Exchanger

Three Fluid Shell and Tube Heat Exchanger consists of a kettle reboiler with saturated water that is boiling, in the outer tube there is flue gas that is flowing at a very high temperature, and for the inner tube there is feed water that is being heated due to the high temperature of the flue gas.

The application of this heat exchanger is in the waste heat recovery system, wherein the flue gas which is at a very high temperature contains high heat potential, so the heat is transferred for production of steam as well as for preheating of feed water.

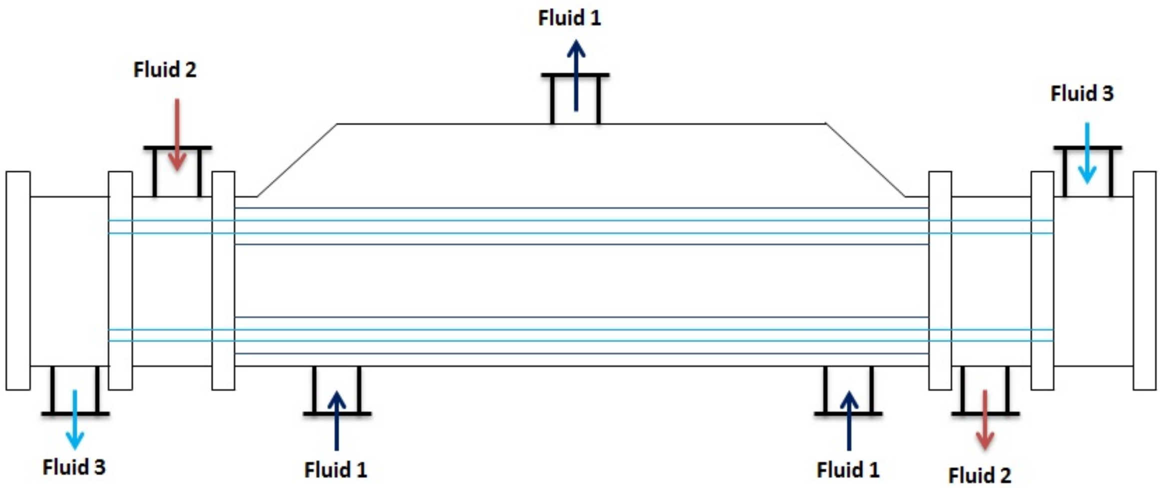

Figure 5.8: Three Fluid Shell and Tube Heat Exchanger

5.6.2 Constructional Layout and Analysis

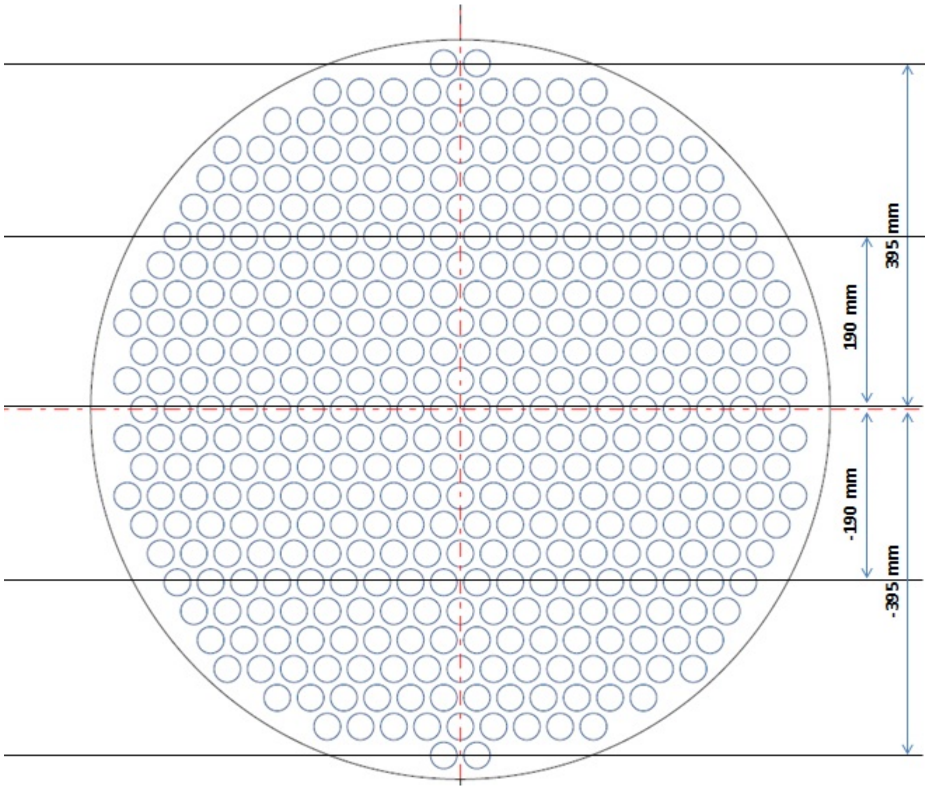

Figure 5.9: Tube Layout and Analysis Along Height

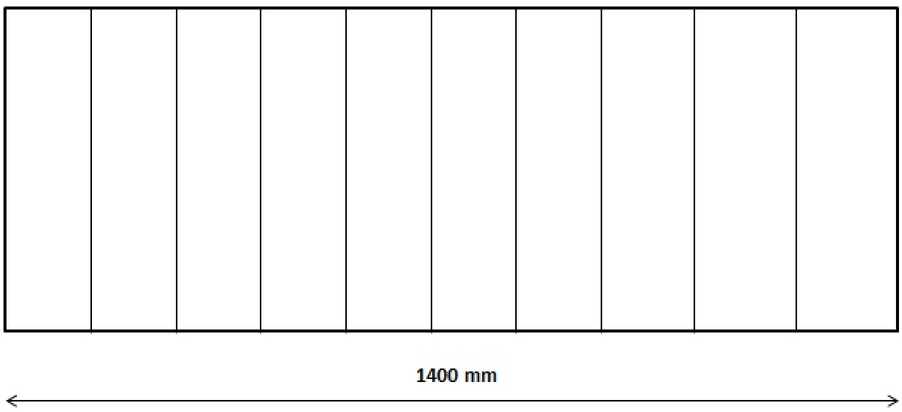

Figure 5.10: Analysis Along Length

5.6.3 Process Data Sheet

Table 5.5: Constructional Parameters

Sr. No	Description	Units	Operating Parameters
1	TEMA Type		AKL
2	Kettle Inside Diameter	mm	1600
3	Port Inside Diameter	mm	880
4	Outer Tube Outside Diamater	mm	31.75
5	Wall Thickness	mm	2.77
6	Inner Tube Outside Diameter	mm	19.05
7	Wall Thickness	mm	1.651
8	Tube Pitch	mm	39.62
9	Total No. of Tubes		400
10	Tube Layout Pattern		30°
11	Effective Tube Length	mm	1400

Table 5.6: Process Data

Sr. No	Description	Units	Annulus		Inner Tube	
			In	Out	In	Out
1	Fluid Name		Flue Gas		Feed Water	
2	Mass Flow Rate	Kg/s	3.5		30	
3	Temperature	°C	800	263.5	190	181.3
4	Inlet Pressure	KPa	101.325		1800	
5	Fouling Resistances	m^2.K/W	0.000081		0.000176	

			Shellside
1	Fluid Name		Steam
2	Mass Flow Rate	Kg/s	0.5
3	Operating Temperature	°C	250.36
4	Operating Pressure	KPa	4000
5	Fouling Resistances	m^2.K/W	0.00088

5.6.4 Design Steps Involved

For the analysis of the heat exchanger the following steps were carried out:

1. Starting the analysis from one particular direction, divide the heat exchanger control volumes, such that it is divided along the length and along the height as well.

2. Assume the inlet temperature of the fluid in the inner tube.

3. With the inlet temperature known, assume the outlet temperature of the outer tube fluid.

4. Calculate the annulus heat duty (Q_a), and hence the overall heat duty that can be achieved.

$$Q_a = Q = m_a.\triangle h \tag{5.1}$$

5. Calculate the heat transfer coefficients [5] for the annulus . There will be two values of heat transfer coefficient, one for the inside of the outer tube (α_1) and other for the outside of the inner tube (α_2). Also calculate the fouling resistances for outside of the inner tube and inside of the outer tube.

$$Re_1 = \rho.v_{ot}.d_{e1} \tag{5.2}$$

$$Re_2 = \rho.v_{ot}.d_{e2} \tag{5.3}$$

$$d_{e1} = (D_i^2 - d_o^2)/D_i \tag{5.4}$$

$$d_{e2} = (D_i^2 - d_o^2)/d_o \tag{5.5}$$

By using Monrad and Pelton's Correlation

$$Nu_1 = 0.02.Re_1^{0.8}.Pr^{1/3}.(D_i/d_o)^{0.53} \tag{5.6}$$

$$Nu_2 = 0.02.Re_2^{0.8}.Pr^{1/3}.(D_i/d_o)^{0.53} \tag{5.7}$$

$$\alpha_1 = Nu_1.k_{ot}/d_{e1} \tag{5.8}$$

$$\alpha_2 = Nu_2.k_t/d_{e2} \tag{5.9}$$

6. Assume heat duty fraction (x), that ranges from the value 0 to 1. The heat duty fraction is the fraction of heat to the shellside (Q_1) and the inner tube (Q_2)

$$Q = Q_1 + Q_2 \tag{5.10}$$

7. With the fraction of heat duty calculate the heat duty to the shellside and to the inner tube

$$Q_1 = x \times Q; \tag{5.11}$$

$$Q_2 = (1 - x) \times Q \tag{5.12}$$

8. For the inner tube, with energy balance equation, calculate the unknown temperature.

$$Q_2 m_{it}.C_p it.\triangle T_{it} \tag{5.13}$$

9. Calculate heat transfer coefficient of the fluid flowing in the inner tube, also calculate the fouling resistances. By using Dittus Boelter's Correlation

$$Nu_i = 0.023.Re_i^{0.8}.Pr^{1/3} \tag{5.14}$$

$$\alpha_i = Nu_i.k/d_i \tag{5.15}$$

10. For the shellside, calculate the heat transfer coefficient (α_b)[13]. The calculation of heat transfer coefficient is little complex. To calculate heat transfer coefficient, nucleate boiling coefficient (α_{nb}) and convective heat transfer coefficient (α_{cb}) needs to be evaluated.

$$\alpha_b = (\alpha_{nb}^2 + \alpha_{cb}^2)^{0.5} \tag{5.16}$$

$$\alpha_n b = 0.23(k_l/d_b).\chi_1^{0.674}\chi_5^{0.297}\chi_4^{0.371}\chi_{13}^{-1.73}\chi_2^{0.35} \tag{5.17}$$

where;

χ Parameter Coefficients

d_b bubble diameter

$$\alpha_{cb} = F.\alpha_l \tag{5.18}$$

$$\alpha_l = a.(k_l/d_o).Re^m.Pr^{0.34}.\emptyset.(1 - x_g)^m \tag{5.19}$$

where,

a,m are constant coefficients

F correction Factor

α_l Liquid Heat Transfer Coefficient

k_l Thermal conductivity of liquid

x_g vapor fraction of the fluid

11. As all the temperatures are known, Log Mean Temperature Difference can be calculated for Annulus – Shell ($LMTD_1$) and Annulus - Inner Tube ($LMTD_2$) respectively.

12. Based on the heat transfer coefficient for the shellside, tubeside and outer tube fluids and fouling resistances, Overall Heat Transfer Coefficient is evaluated for Annulus – Shell (U_1) and Annulus - Inner Tube (U_2)

$$(1/U_1) = (1/\alpha_b) + ((1/\alpha_1) \times (D_o/D_i)) + (D_o \times ln(D_o/D_i)/2k_{ot}) + R_{fs} + R_{fa} \times (D_i/D_o) \tag{5.20}$$

$$(1/U_2) = (1/\alpha_1) + ((1/\alpha_2) \times (d_o/d_i)) + (D_o \times ln(d_o/d_i)/2k_{it}) + R_{fa} + R_{fit} \times (d_i/d_o) \tag{5.21}$$

13. Based on the Heat Duty, Log Mean Temperature Difference, Overall Heat Transfer Coefficients for Annulus – Shell and Annulus - Inner Tube respectively, calculate the Heat Transfer Area for Annulus – Shell (A_1) and Annulus - Inner Tube (A_2) respectively.

$$Q_1 = U_1.A_1.LMTD_1 \tag{5.22}$$

$$Q_2 = U_2.A_2.LMTD_2 \tag{5.23}$$

14. As we know that the effective length of heat transfer for the inner tube and outer tube has to be the same, hence the ratios of heat transfer area (A_1/A_2) remains constant,

the fraction of heat duty is varied unless the ratio obtained converges to the constant value.

15. Once the ratio is converged then overdesign is checked. If the overdesign value does not converge with the desired value then the Annulus outlet temperature is iterated till the overdesign values are converged.

$$Overdesign_1\% = (A_1 - A_{1prov})/A_{1prov} \tag{5.24}$$

$$Overdesign_2\% = (A_2 - A_{2prov})/A_{2prov} \tag{5.25}$$

16. Then the similar process is carried forward for the next control volume

17. If the inner tube outlet temperature does not match with the desired value, then inlet temperature of inner tube is changed, till the desired outlet temperature is obtained.

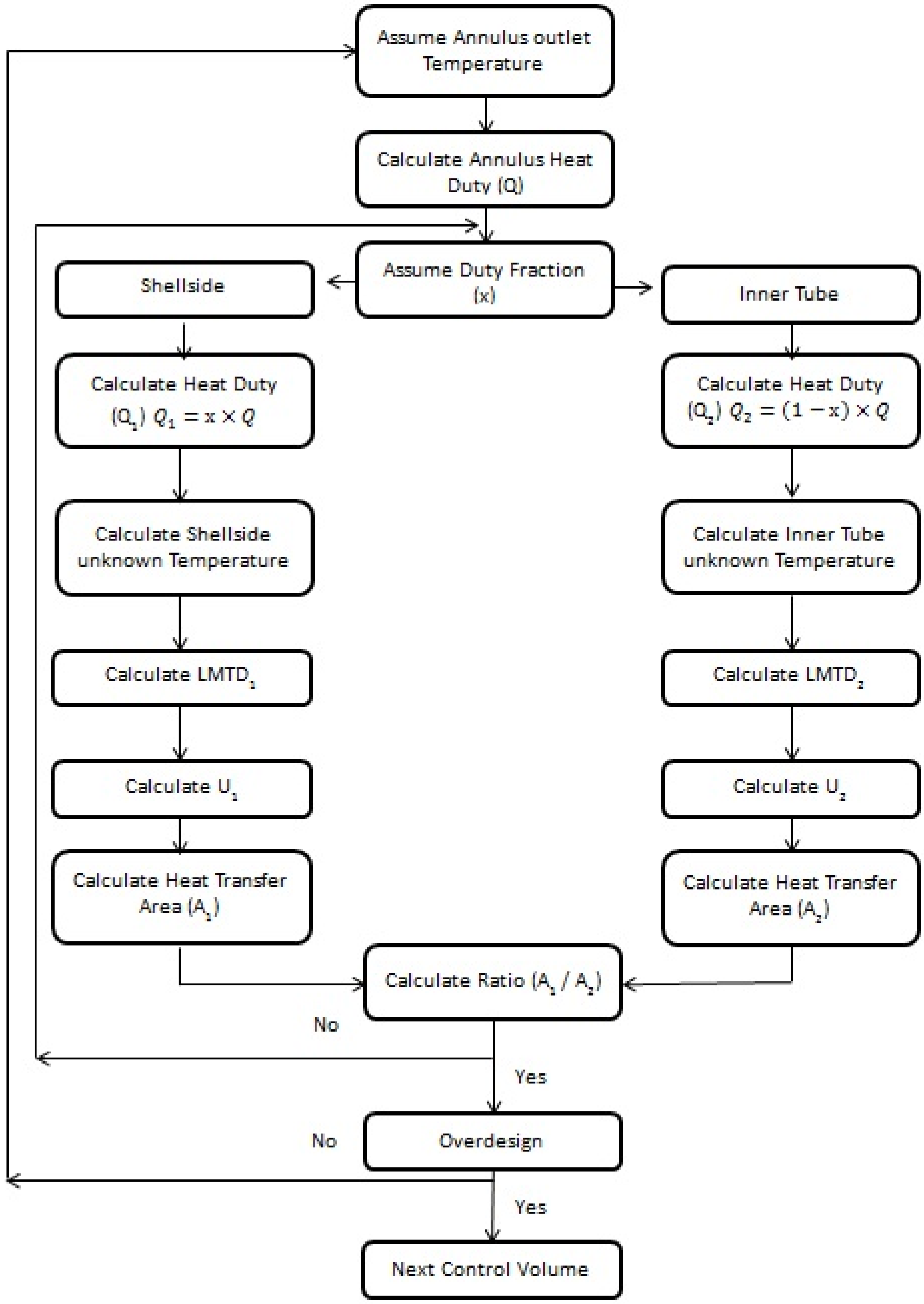

Figure 5.11: TFSTHE Design Flow Chart

5.6.5 Overall Summary

Table 5.7: Output Results

Sr. No	Description	Units	Operating Conditions			
			Shell	Annulus		Inner Tube
1	Total Heat Duty	KW	2204.99			
2	Heat Duty Distribution	KW	1047.18		1157.81	
3	Log Mean Temperature Difference	°C	180.95		247.191	
4	Heat Transfer Coefficient	$W/m^2.K$	11343.5	123.1	176.59	3984.75
5	Overall Heat Transfer Coefficient	W/m.K	106.118		142.02	
6	Heat Transfer Area	m^2	55.86		33.514	
7	Fraction of Heat Duty		0.475		0.525	
8	Pressure Drop	KPa		9.14		0.2
9	Overdesign	%	0		0	

5.6.6 Validation

Table 5.8: Comparison with HTFS

Sr. No	Description	Units	Comparison			
			Calculations		HTFS	
			In	Out	In	Out
1	Annulus Fluid Velocity	m/s	111.15	56.3	112.32	54.2
2	Inner Tube Fluid Velocity	m/s	0.44	0.44	0.43	0.44
3	Shellside Heat Transfer Coefficient	$W/m^2.K$	11343.49		9929	
4	Annulus Heat Transfer Coefficient	$W/m^2.K$	123.09	176.59	175.9	
5	Inner Tube Heat Transfer Coefficient	$W/m^2.K$	3984.757		3719.4	

5.7 Results and Discussions

5.7.1 Temperature Profile

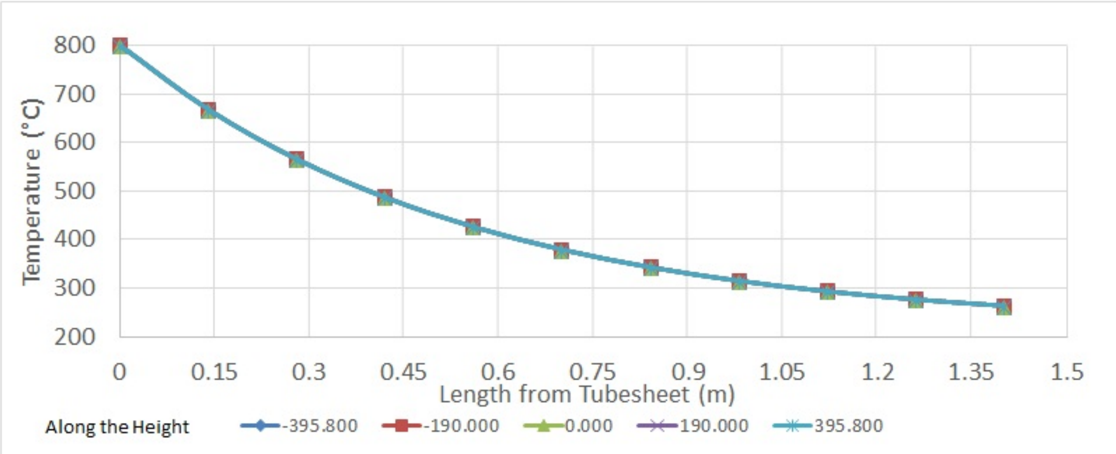

Figure 5.12: Annulus Temperature Profile

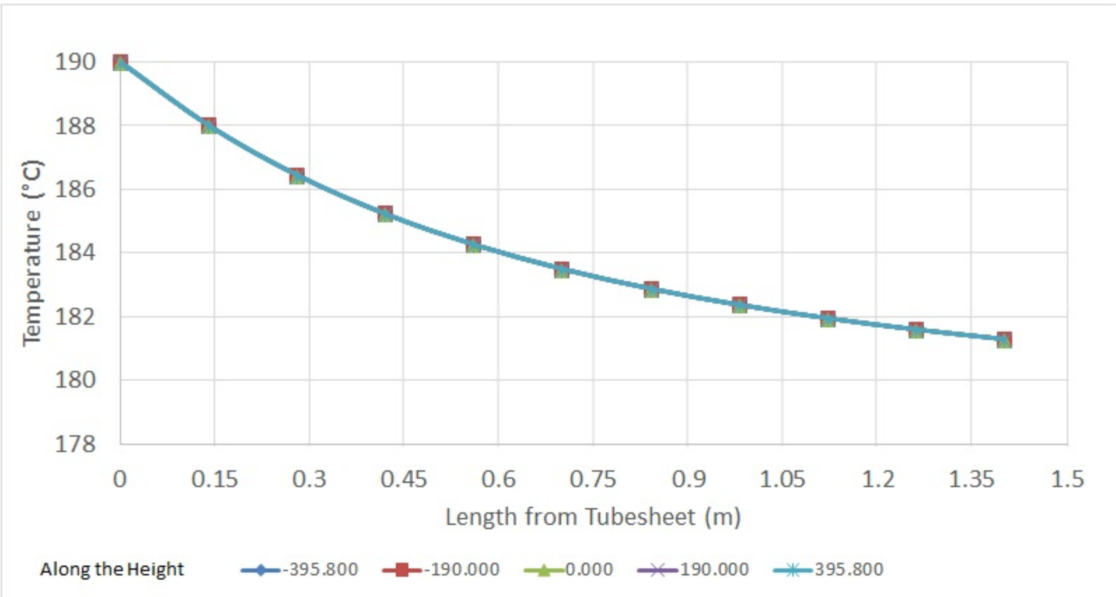

Figure 5.13: Inner Tube Temperature Profile

5.7.2 Heat Duty & Heat Transfer Coefficient

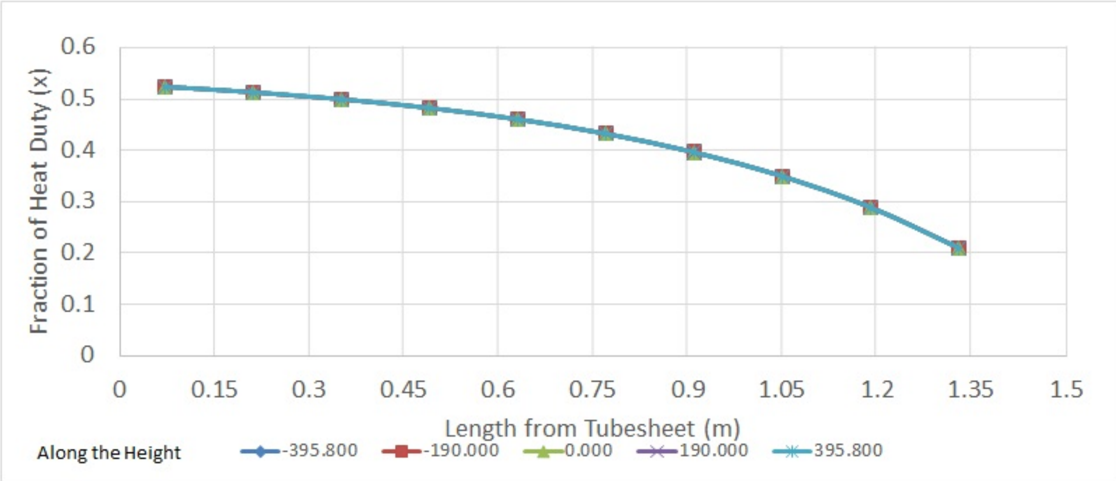

Figure 5.14: Fraction of Heat Duty to Shellside

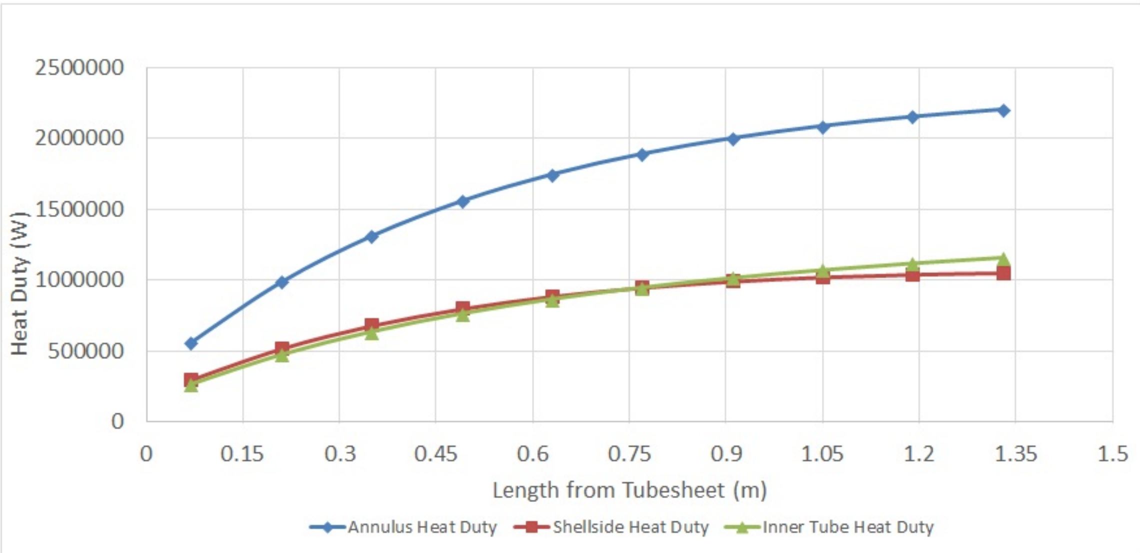

Figure 5.15: Heat Duty Distribution

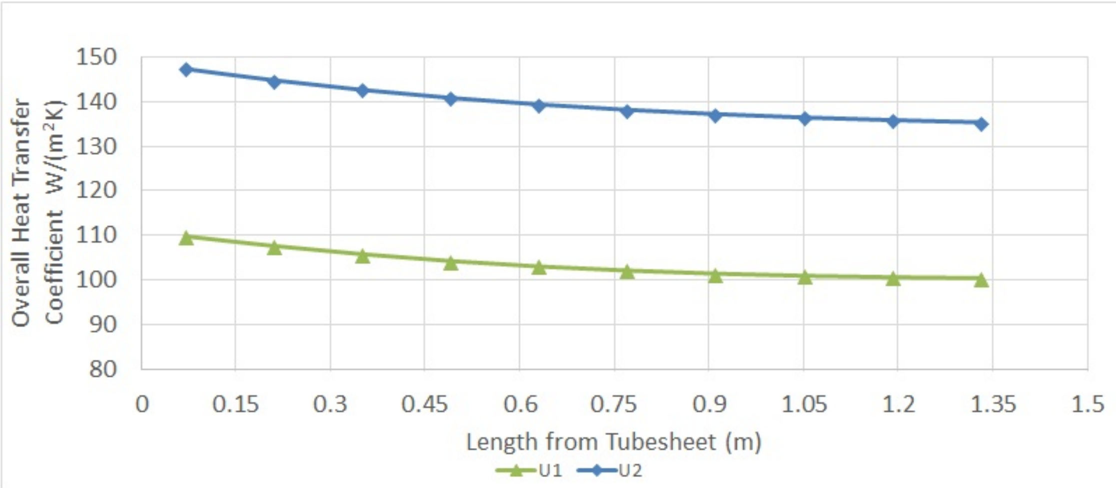

Figure 5.16: Overall Heat Transfer Coefficient

5.8 Conclusion

After the analysis on Three Fluid Shell and Tube Heat Exchanger several conclusions were carried out:

1. The calculation parameters such as velocity and heat transfer coefficient for shell-side, annulus and inner tube were validated with the HTFS software, and it was found that the values were in close adherence to each other.‘

2. The flow of flue gas in the annulus increases its heat transfer coefficient, which also leads to an increase in the temperature drop and total heat duty for the same length

3. Since the annulus fluid's heat transfer coefficient was governing, if the annulus heat transfer coefficient is increased it would lead to an increase in the Overall Heat Transfer coefficient

4. Also, the use of Three Fluid Shell and Tube Heat Exchanger will be beneficial if the annulus fluid is a governing parameter, otherwise it would lead to an increase in the cost with minimal increase in heat duty

Chapter 6

CONCLUSION

Three Fluid Shell and Tube heat exchanger is an innovative heat transfer equipment in the category of STHE. Compared to conventional STHE it has various advantages such as increased heat duty, reduced foot print area, higher heat transfer coefficient, less pressure drop, reduction in total cost of the heat exchanger, etc.

Since the Design of Three Fluid Shell and Tube Heat Exchanger, being complex, the design methodology has to be derived starting from the basic fundamentals. Hence for this purpose several cases were studied. First the parametric studies were done of STHE.Then from the literature, study was carried out on Bayonet Tube heat exchanger since it can be comparable to Three Fluid STHE, though it involves two fluids exchanging heat with each other. The design methodology was developed for Bayonet Tube heat exchanger and was validated and compared with the literature. Then from literature, case study on Shell and Double Concentric Tube Heat Exchanger was compared and validated with the design methodology established.

Finally with the design methodology established, it was used for design of Three Fluid STHE. Since there is no software existing for the Three Fluid STHE, certain parameters such as velocity, heat transfer coe cients were validated using HTFS software. After validation the parameters were used for further calculation

During the analysis certain conclusions were obtained

1. Heat Duty per unit length for Three Fluid STHE is more compared to conventional

STHE. Hence for same heat duty compact geometry can be used.

2. Better heat transfer coefficients with increased heat duty per unit pressure drop can be obtained

3. Reduction in operating cost and capital cost due to less number of piping arrangements and reduced foot print area

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