Design of the Heat Exchanger for the Compressor After-Cooler of the Helium Plant

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DEPARTMENT OF MECHANICAL ENGINEERING

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Design of the Heat Exchanger for the Compressor After-Cooler of the Helium Plant

Major Project Report

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Submitted in partial fulfillment of the requirements

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(Thermal Engineering)

By

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Guided by

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May 2014

Declaration

This is to certify that

1. The thesis comprises of my original work towards the degree of Master of Technology in Mechanical Engineering (Thermal Engineering) at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgment has been made in the text to all other material used.

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Undertaking for Originality of the Work

I, Patel Amitkumar Rasikbhai, Roll. No. 12MMET17, give undertaking that the Major Project entitled "**DESIGN OF THE HEAT EXCHANGER FOR THE COMPRESSOR AFTER-COOLER OF THE HELIUM PLANT**" submitted by me, towards the partial fulfillment of the requirements for the degree of Master of Technology in Mechanical Engineering (Thermal) of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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Date: ________________

Place: Nirma University, Ahmedabad.

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This is to certify that the Major Project Report entitled "**Design of the Heat Exchanger for the Compressor After-Cooler of the Helium Plant**" submitted by **Mr Amit Patel(12MMET17)**, 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.

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Amit Patel

Abstract

Liquid Helium Plants (LHeP) are designed to produce liquid helium from any room temperature helium gas source with a minimum purity of 99%. The helium refrigerator/Liquefier (HRL) needs compression of helium gas, which is then, cooled and expanded to produce liquid helium at 4.5 K. To avoid high heating and to reduce the work requirement in the compression, oil of heavy molecule, which has a high specific heat, is mixed with helium before compression. Before using this helium gas for refrigeration, it is necessary to separate helium and oil from the mixture and cool it to about room temperature. For this cold water at 20 $^\circ$C temperature is used to cool helium and oil in separate heat exchangers. This project is about the design and analysis of the oil/water heat exchanger. Oil absorbs about 80% heat load produced due to compression process. It is planned to have a similar design concept and configuration as that of the existing one at Institute for Plasma Research (IPR) for the Helium Refrigerator Liquefier compressor system of Steady State Superconducting Toksmsk (SST-1) magnets. Heat duty requirement is about 900KW. The flow of helium in nominal case is at 14 bar pressure with temperature at 90 $^\circ$C at inlet and it needs to be cooled down to 35 $^\circ$C temperature. The cold water inlet is about 20 $^\circ$C temperature and it exits at 25 $^\circ$C temperature. To get the exact flow distribution, pressure drops and temperature distributions in the fluid, it is required to have a detailed analysis based on thermal design tools, and an optimized configuration of the heat exchanger to cool the oil.

In the present study, the results obtained analytically are validated with the thermal design tool HTRI software. The thermal design optimization was carried out by varying parameters like inside diameter of tube and inside diamter of the shell subsequently, which results in change of length of the heat exchanger, and pressure drop on tube side and shell side.

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Nomenclature

D_i	Shell inside diameter, m
Q	Heat duty, W
d_i	Tube inside diamter, m
d_o	Tube outside diameter, m
N_t	Number of tubes
Cp_h	Specific heat of hot fluid, J/kgK
Cp_c	Specific heat of cold fluid, J/kgK
Pr	Prandtl number
k_h	Thermal conductivity of hot fluid, W/mK
k_c	Thermal conductivity of cold fluid, W/mK
t	Thickness of the tube, m
m_h	Mass of hot fluid, kg
m_c	Mass of cold fluid, kg
F	Temperature correction factor
N_{pass}	Number of pass
Re	Reynold number
Nu	Nusselt number
h_i	Heat transfer coefficient inside tube, W/m^{2}K
h_s	Heat transfer coefficient for the shell side, W/m^{2}K
C	Clearance, m
p_t	Tube pitch, m
J_i	Colburn factor
D_{otl}	Diameter of outer tube limit, m
l_c	Baffle cut, m
N_b	Number of baffle
X_t	Transverse pitch, m
X_l	Longitudinal pitch, m
L	Length of the tube, m
$L_{b,i}$	Inlet baffle spacing, m
$L_{b,o}$	Outlet baffle spacing, m
B	Central baffle spacing, m

p_p	Tube pitch parallel to flow, m
V	Volume of the shell, m^3
h_{ideal}	Ideal heat transfer coefficient, $^{W}/_{m^2K}$
F_c	Fraction for total number of tubes in the cross flow section
F_w	Fraction of number of tubes in one window section
J_c	Correction factor for baffle cut and baffle spacing
$A_{o,sb}$	Shell to baffle leakage area, m^2
$A_{o,tb}$	Total tube to baffle leakage area, m^2
$A_{o,cr}$	Cross flow area at or near centreline for one cross flow section,m^2
J_l	Correction factor for baffle leakage effect
N_p	Number of pass partition lane
W_p	Width of lane, m
$A_{o,bp}$	Crossflow area for bypass flow, m^2
J_s	Correction factor for baffle spacing
A_m	Crossflow at or near centreline for one crossflow section, m^2
F_{sbp}	Fraction of cross flow area available for bypass flow
N_c	Number of tube rows crossed in one cross flow section
N_{cw}	Number of effective cross flow rows in each window
A_{wg}	Gross window area, m^2
A_{wt}	Window area occupied by tubes, m^2
I_1	Bessel function of first kind
K_1	Bessel function of first kind
r_1	Tube inside radius, m
N_f	Number of fins
k	Thermal conductivity of fin material, $^{W}/_{mK}$
R_{fi}	Fouling resistance for tube inside materail, $^{mK}/_{W}$
R_{fo}	Fouling resistance for tube outside material, $^{mK}/_{W}$
h_i	Heat transfer coefficient inside the tube, $^{W}/_{m^2K}$
h_o	Heat transfer coefficient for the shell side flow, $^{W}/_{m^2K}$
h_f	Height of fin, m
X_t	Transverse pitch, m
X_l	Longitudinal pitch, m
N_t	Number of tube
u_t	Velocity of the fluid inside the tube, $^{m}/_{s}$
D_e	Hydraulic diameter, m
Co	Capital cost of the heat exchanger, Rs
V_{tube}	Volume of the tube material, m^3
V_{shell}	Volume of the shell, m^3
d_f	Outside diameter of fin, m

N_{ss}	Number of sealing strips
A_w	Area for flow through window, m^2
R_b	Correction factor for bundle bypass used in pressure drop calculation
R_s	Correction factor for unequal baffle spacing used in pressure drop calculation
A_{finned}	Finned area, m^2
$A_{unfinned}$	Unfinned area, m^2
W	Width of the fin, m
A_c	Cross sectional area of the fin, m^2

Greek Symbols

μ_h	Dynamic viscosity of hot fluid, $^{Ns}/_{m^2}$
μ_c	Dynamic viscosity of cold fluid, $^{Ns}/_{m^2}$
ρ_h	Density of hot fluid, kg/m^3
ρ_c	density of cold fluid, kg/m^3
δ_{sb}	Diametral shell baffle clearance, m
δ_{tb}	Diametral clearance between tube and baffle, m
θ_{ctl}	Angle between baffle cut and two radiius of circle centres of outermost tubes, rad
θ_b	Angle between two radii intersected at inside shell with baffle cut, rad
η_f	Fin efficiency
η_i	Efficiency of the fin for the fin inside the tube
η_o	Efficiency of the fin for the fin outside the tube
η_{pump}	Efficiency of pump

Abbreviations

SST	Steady State Tokamak
ITER	International Thermonuclear Experimental Reactor
HRL	Helium Refrigerator Liquefier
ALCC	Annual Life Cycle Cost
CRF	Capital Recovery Factor
STHE	Shell and Tube Heat Exchanger
TEMA	Tubular Exchanger Manufacturers Association
CTP	Tube Count Calculation Constant
CL	Tube Layout Constant

Chapter 1

Introduction

1.1 Nuclear Fusion Reactor

Nuclear fusion is a process in which two nucleus fuse together to form large nucleus. In the fusion process, some of the mass of the original nucleus is lost and transformed to the energy in the form of high energy particles. Energy from fusion reactions is the most basic form of energy in the universe; our sun and all other stars produce energy through thermonuclear fusion reactions. Magnetic fields toroidal produced by external coils and the field of a flow the plasma cover the plasma.

International Thermonuclear Experimental Reactor (ITER) is the largest, multi-country, organization working to create a fusion reactor. And its projected completion Year is 2030.

1.1.1 Steady State Superconducting Tokamak

A Steady State Superconducting Tokamak SST-1 is under design and fabrication of the SST-1 at the Institute for Plasma Research, Gandhinagar. The objectives of SST-1 is to study out the physics of the plasma processes under steady state conditions and learning technologies related to the steady state operation of the tokamak. These studies are helpful in contributing to the tokamak physics database for very long pulse operations. The SST-1 tokamak is a large aspect ratio tokamak, configured mainly to run double null diverted plasmas with significant elongation and triangularity.

1.2 Liquid Helium Plant

Liquid Helium Plants (LHeP) are designed to produce liquid helium from any room temperature helium gas source with a minimum purity of 99%. The liquid helium (LHe) is produced and stored in the system's dewar. A standard liquid helium plant consists of a pulse tube cryorefrigerator with liquefaction heat exchangers, a liquid helium dewar as well as a liquid helium level sensor and controller.

The present project is to design the oil cooler. In the oil cooler, the oil is removed from the mixture of the oil and helium and after that helium cooler comes and in that cooler helium gets cooled down.

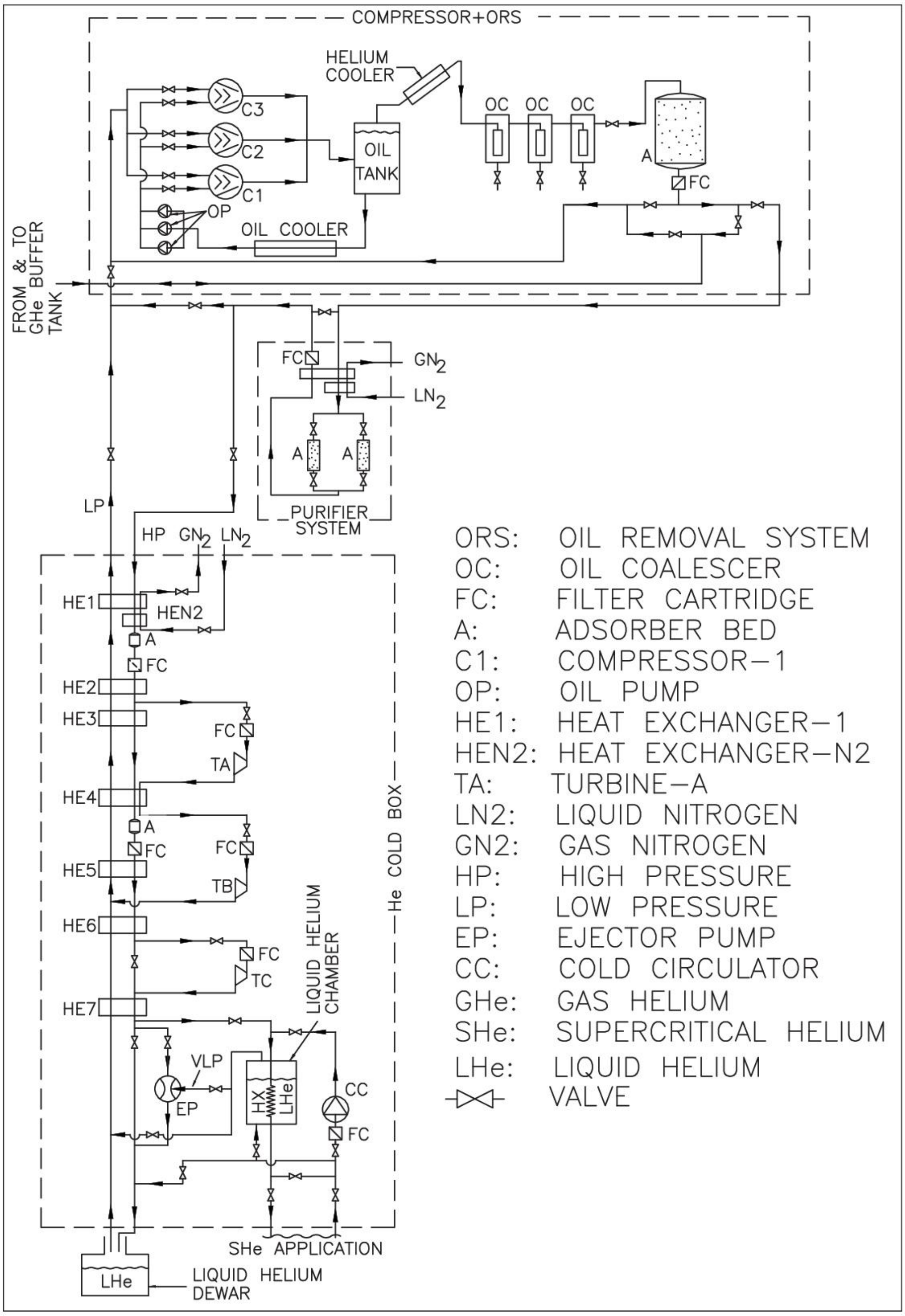

Figure 1.1: Schematic diagram of the liquid helium plant at IPR [Received from IPR]

1.3 Introduction to Heat Exchanger

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact.

In heat exchangers, there are usually no external heat and work interactions. Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single or multi component fluid streams. In other applications, the objective may be to recover or reject heat, or sterilize, pasteurize, fractionate, distil, concentrate, crystallize, or control a process fluid. In a few heat exchangers, the fluids exchanging heat are in direct contact.

Regular samples of heat exchangers are shell-and tube exchangers, auto radiators, condensers, evaporators, air preheaters, and cooling towers. There could be interior warm vitality sources in the exchangers, for example, in electric radiators and atomic fuel components. Burning and substance response may take put inside the exchanger, for example, in boilers, red warmers. Mechanical components may be utilized as a part of a few exchangers, for example, in scratched surface exchangers, and blended tank reactors. Convection process and also conduction process take place in the heat exchanger. As a rule, if the fluids are immiscible, the interface between the fluids replaces a high temperature exchange surface, as in an immediate contact heat exchanger.

The heat transfer surface is a surface of the heat exchanger core that is in direct contact with fluids and through which heat is transferred by conduction. That portion of the surface that is in direct contact with both the hot and cold fluids and transfers heat between them is referred to as the primary surfaces.

1.3.1 Parameters considered in the selection of heat exchanger

Following parameters have been considered in selection of heat exchangers: **[8]**

(i) Operating pressure and temperature

This is the main parameter which has to be considered in the selection of the heat exchanger.

(ii) Cost

Cost is a very important factor in the selection of the heat exchanger construction type. The cost per unit of heat transfer surface area is lower for a shell and tube heat exchanger than for a gasketed plate exchanger and any other type of heat exchanger.

(iii) Fouling and Clean ability

Fouling should also be taken into account. While designing the heat exchanger in the tube and also outside the tube that means in the shell side flow fouling resistance of the flowing fluid should be considered. And tube materials and also the shell material should be selected in such a way that the cleaning should be done easily.

(iv) Fluid leakage and contamination

In some applications, fluid leakage from one fluid side to the other fluid side is permissible within limits,while in other applications fluid leakage is absolutely not allowed.

(v) Fluids and materials compatibilities

Materials selection and compatibility between construction materials and working fluids are important issues, in particular with regard to corrosion and/or operation at elevated temperatures.

1.4 About Institute For Plasma Research

The institute for Plasma Research is a premier research organization in India, involved in research in various aspects of plasma science including basic plasma physics, magnetically confined hot plasmas and also plasma technologies for industrial application. This institute is currently working in the process of building a Steady State Superconducting Tokamak (SST-1) and is one of the partners in the international ITER project.

The research areas are:

Basic tests in plasma physical science including free electron laser, dusty plasmas also other nonlinear phenomena industrial plasma handling and requisition. In current circumstance the work completed is on high temperature attractively concerned plasmas is continuously led in tokamak.

The plasma is framed by an electrical breakdown in a ultra high vacuum toroidal vessel and a current is drivening in the plasma. It is important to utilize helper warming plans, since the efficiency to high temperature plasmas drops as the plasma temperature climbs. Thomson diffusing for electron temperature estimation, delicate X-beam Polaroid and laser blow, are completed on Aditya. An unfaltering state tokamak which is the first of this kind in India, is constantly created and situated up, to study issues identified with vitality, molecule commencement throughout enduring state operation. Plasma interruptions and vertical uprooting scenes will be contemplated in this set up. Non-inductive current drive might manage the plasma current. The model creation of the greater part of the parts of the different subsystems of the tokamak has been finished, and is constantly tried, before final combination. Essential tests, including generally cooler plasmas are continuously completed to comprehend the different features of plasma that are difficult to study in greater systems. The study identified with the steadiness and harmony of toroidal plasma in the vicinity of radio recurrence waves and new present drive system with these waves is, no doubt mulled over. The effects identified with excitation, and direct, nonlinear cooperation of whistler and helicon waves are constantly examined in an extensive volume plasma gadget. Likewise the study identified with dusty plasma is constantly completed.

A multi disciplinary group of physicists, designers and material researchers are meeting expectations together to produce propelled material preparing innovations in this unit.

For instance, Business model of medicinal waste pyrolisis framework is a percentage of the real exercises finished up by the gathering at the Facilitation Center for Industrial Plasma Technologies focus of IPR

1.5 Motivation and Objective of Project Study

The Helium Refrigerator/Liquefier (HRL) needs compression of helium gas, which is then, cooled and expanded to produce liquid helium at 4.5 K. To avoid high heating and to reduce the work requirement in the compression, oil of heavy molecule, which has high specific heat, is mixed with helium before compression. Before using this helium gas for refrigeration, it is necessary to separate helium and oil from the mixture and cool it to about room temperature. The heat duty requirement is about 900 KW. The flow to the helium plant in nominal case is at 14 bar pressure with a temperature of 90°C at inlet and it needs to be cooled down to 35°C temperature. The cold water inlet is at 20°C temperature and it comes out at 25°C temperature. The present project aims to design and analysis of the oil/water heat exchanger.

As oil absorbs nearly 80% heat load produced due to compression process, it is beneficial to design the heat exchanger efficiently. Hence the aim of the present project is to design an effective heat exchanger for the Compressor After-Cooler of the Helium Plant for cooling oil in the oil cooler in an economical manner so to reduce the cost by decreasing the volume of heat exchanger.

Basically, the study is concerned with the,

(i) Heat transfer and pressure drop inside tube of the shell and tube heat exchanger

(ii) Hear transfer and pressure drop anlysis inside the shell of the heat exchanger

(iii) Optimization of the heat exchanger

Chapter 2

Literature review

2.1 Introduction to Heat Exchanger

A heat exchanger is an equipment that is utilized to exchange warm vitality (enthalpy) between two or more fluids, between a strong surface and a fluid, or between robust particulates and a fluid, at different temperatures and in warm contact.

In heat exchangers, there are generally no outer hotness and work associations. Normal requisitions include warming or cooling of a fluid stream of concern and vanishing or buildup of single or multi part fluid streams. In different provisions, the target may be to recuperate or reject hotness, or, purify, fractionate, distil, concentrate, solidify, or control a procedure fluid. In a couple of high temperature exchangers, the fluids trading hotness are in immediate contact.

In most heat exchangers, heat exchange between fluids happens in a transient way.

Regular illustrations of heat exchangers are shell-and tube exchangers, vehicles radiators, condensers, evaporators, air preheaters, and cooling towers. On the off chance that no stage change happens in any of the fluids in the exchanger, it is now and then alluded to as a sensible heat exchanger. There could be inside warm vitality sources in the exchangers, for example, in electric warmers and atomic fuel components. Burning and compound response may take put inside the exchanger, for example, in boilers, red warmers, and uidized bunk exchangers. Mechanical components may be utilized as a part of a few exchangers, for example, in scratched surface exchanger, unsettled vessels, and blended tank reactors.

2.2 Application of Heat Exchanger

Most common heat exchangers are two-fluid heat exchangers. Three-fluid heat exchangers are widely used in cryogenics. They are used in various heat exchangers that are used in chemical and process industries, such as air separation systems, purification and liquefaction of hydrogen, and ammonia gas synthesis. Three and multi component heat exchangers are very complex to design. They may also include multi component two-phase convection as in condensation of mixed vapours in the distillation of hydrocarbon.

Heat exchangers are used in wide variety of application as in the process, power, air conditioning, refrigeration, cryogenics, heat recovery and manufacturing industries. In the power industry, various kinds of fossil boilers, nuclear steam generators, steam condenser, regenerator, and cooling towers are used. In the process industry, two phase flow heat exchangers are used for vaporising, condensing, and freezing, in crystallization, and for fluidized beds with catalytic reaction. The air-conditioning and refrigeration industries need condenser and evaporators.

Energy can be saved by direct contact condensation. By direct contact condensation of a vapour in liquid of the same substance under high pressure, thermal energy can be stored in the storage tank. When the energy is needed again, the liquid is depressurized and flashing occurs that results in producing vapours. The vapour can then be used for heating or as a working fluid for an engine.

There have been abrupt developments in a heat exchanger application. One of the main steps for the early development of boiler was the introduction of the water-tube boilers. The demand for more powerful engines created a need for boiler that operated at higher pressure, and as a result, individual were built larger and larger.

2.3 Classification of Heat Exchanger

1. **According to transfer process**
   - Indirect contact
   - Direct contact
2. **According to number of fluids**
   - Two-fluid
   - Three-fluid
   - N-fluids (N>3)
3. **According to constructional features**
   - Tubular heat exchanger
     - Shell and tube heat exchanger
     - Double-pipe heat exchanger
     - Spiral tube heat exchanger

- Plate-type heat exchanger
  - Gasketed plate heat exchangers
  - Spiral plate heat exchanger
  - Lamella heat exchanger
- Extended surface heat exchanger
  - Plate-fin heat exchanger
  - Tube-fin heat exchanger

4. **According to flow arrangements**
   - Single-Pass Exchanger
     - Counter flow Exchanger
     - Parallel flow Exchanger
     - Cross flow Exchanger
     - Split-Flow Exchanger
     - Divided-Flow Exchanger
   - Multi pass Exchanger
     - Multi pass Cross flow exchanger
     - Multi pass Shell and tube Exchanger
     - Multi pass Plate Exchanger

5. **According to the heat transfer mechanism**
   - Single phase convection on both sides
   - Single phase convection on one side, two phase convection on other side
   - Two phase convection on both sides
   - Combined convection and radiative heat transfer

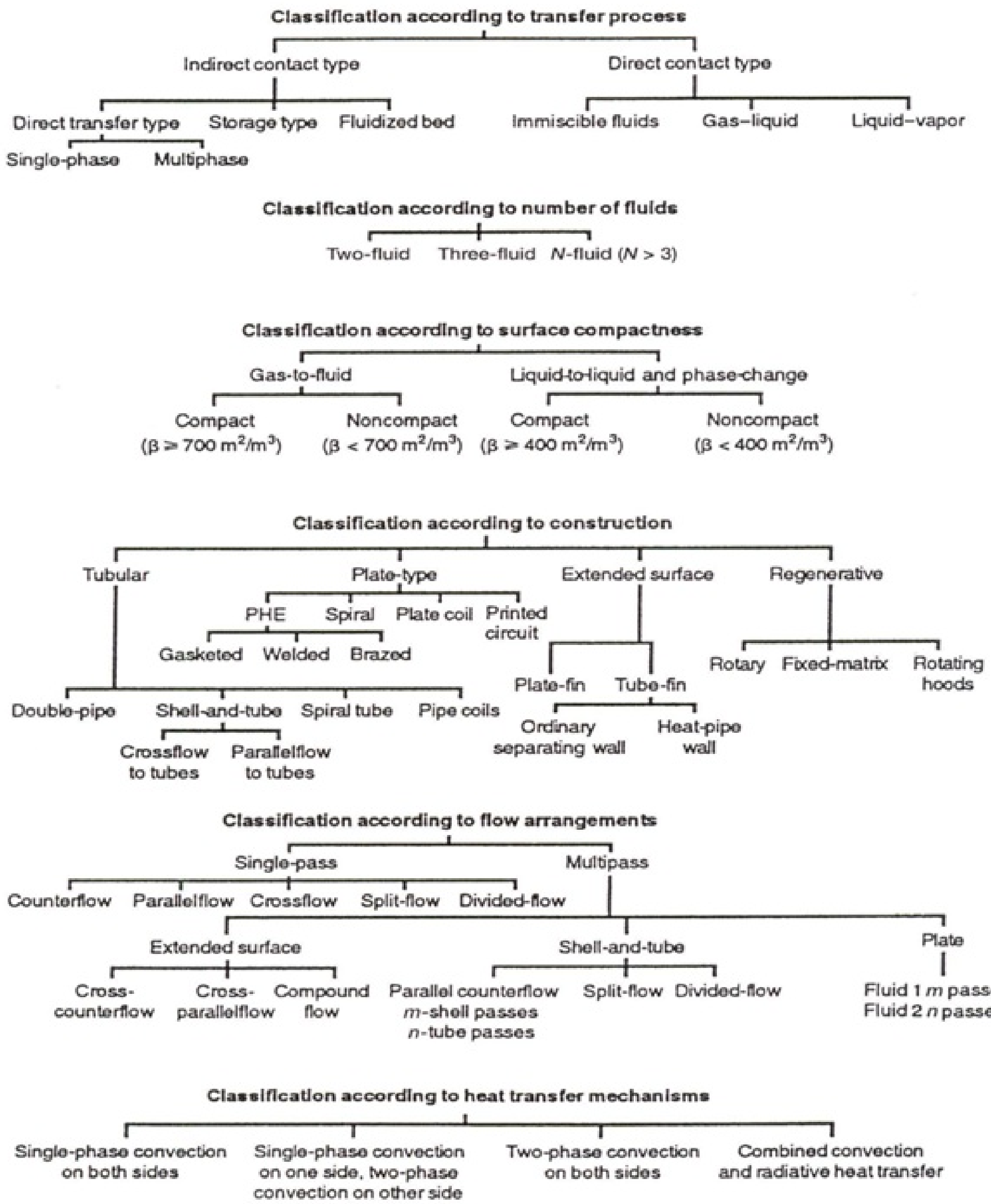

Figure 2.1: Classification of heat exchanger[Shah 1981][8]

Fig. 2.2 shows commonly used heat exchanger in idustries.

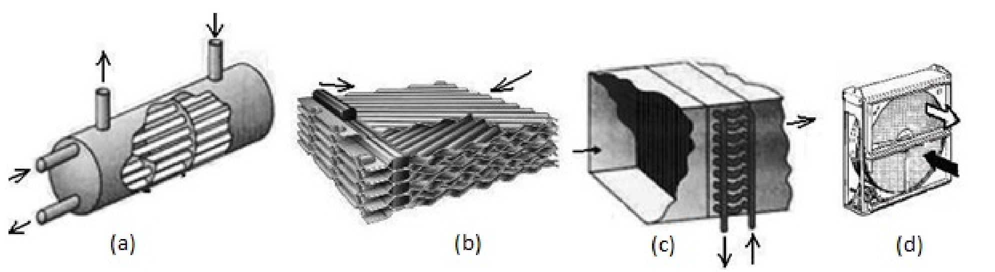

Figure 2.2: Type of heat exchangers: (a) Shell and tube, (b) Plate, (c) Open flow, (d) Rotating wheel [8]

2.4 Shell and Tube Heat Exchanger

Shell and tube heat exchangers are more versatile type of heat exchanger. They are having wide variety of applications. They are used in process industries, in conventional and nuclear power station as condenser, in steam generators, in pressurised water reactor power plants, in feed water heaters and some air conditioning and refrigeration systems. Shell and tube heat exchanger provide relatively large ratio of heat transfer area to volume and weight and they can be easily cleaned. Shell and tube heat exchanger offer great flexibility to meet almost any service requirement. Shell and tube heat exchanger can be designed for high pressure relative to the environment and high pressure difference between the fluid streams [1].

The most broadly utilized heat exchanger within industry is the shell-and-tube heat exchanger, on the grounds that of its generally straight forward development and multi-reason provision conceivable outcomes for vaporous what's more fluid media in an extensive temperature and weight range. For a long time, different sorts of tries have been utilized within shell-and-tube heat exchangers to enhance high temperature exchange while keeping up a sensible weight drop over the exchanger. This enhances heat exchange by improving turbulence or neighborhood blending on the shell side of the exchanger, however at the expense of a high pressure drop.[7].

2.4.1 Material properties[14]

Heating and cooling are the processes which are widely used in the industries.

Industries use large amount of stainless steel for the manufacturing of the tube and also for the shell to gain high efficiency. And it is also helpful to reduce the cost in the operation. Stainless steel is economical in the capital cost also.

Following are advantages for use of stainless steel in the shell and tube heat exchanger.

1. Resistance to corrosion in virtually all cooling water and many chemical environment
2. High temperature resistance to oxidation and scaling
3. Good strength characteristic in low and high temperature service
4. Maintains excellent heat transfer properties in service
5. Fouling resistance and corrosion resistance.
6. No contamination of product and process by corrosion
7. Easy to clean mechanical, chemical
8. Fabric ability
9. Availability in variety of composition as seamless and welded tubing for shell and tube heat exchanger, thin flat rolled sheet and strip for plate heat exchanger, plate for tube sheets, bar and wire for mechanical fastener
10. Economical in terms of first cost and long term services

The choice of the tube material depends upon the parrameters like corrosion resistance, contamination, design versatility, etc. Stainless steel is also economical as compared to the other material which are to be used in the making of the shell and tube heat exchanger.

At final it is concluded that stainless steels are economical.

2.4.2 Literature review related to the design of the shell and tube heat exchanger

Su Thet Mon Than, Khin Aung Lin, Mi Sandar Mon [15] had evaluated for heat transfer area and pressure drop and checked whether the assumed design satisfies all requirement or not. The primary aim of this design is to obtain a high heat transfer rate without exceeding the allowable pressure drop.

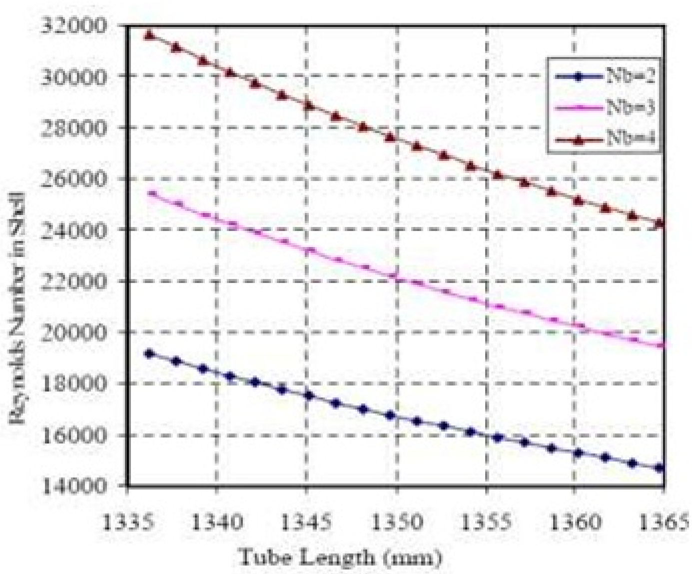

Figure 2.3: Reynolds Number on Number of Baffles and Length of Tube[1]

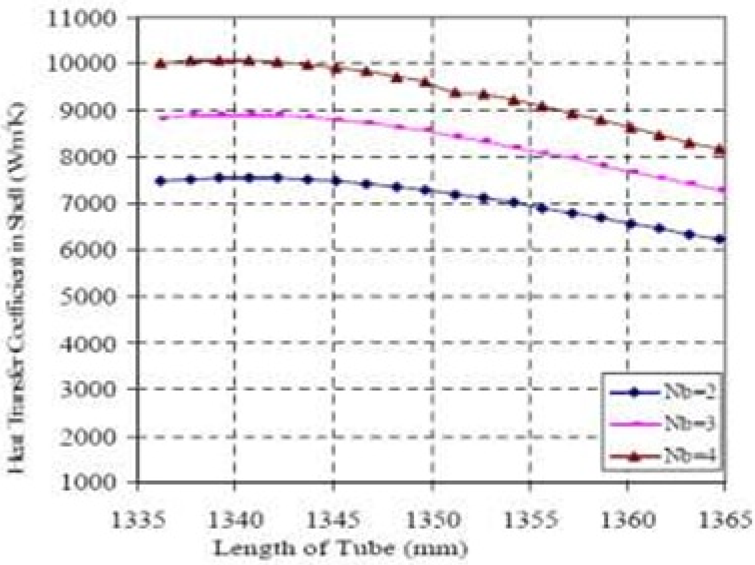

Figure 2.4: Heat transfer coefficient on number of baffles and length of tube[1]

The decreasing pattern of curves of Reynolds Number and heat transfer coefficient shown in Fig.2.3 and as shown in Fig.2.4 that the Re and h are gradually decreases corresponding as high as tube effective length. Gradual decrease in Reynolds Number means there is significant decrease in pressure drop respectively.

Rajiv Mukherjee[8]explains the basics of exchanger thermal design, covering such topics as shell and tube heat exchanger components; classification of STHEs according to construction and according to service; data needed for thermal design; tube side design; shell side design, including tube layout, baffling, and shell side pressure drop; and mean temperature difference. The basic equations for tube side and shell side heat transfer and pressure drop. Correlations for optimal condition are also focused and explained with some tabulated data. The optimized thermal design can be done by sophisticated computer software however a good understanding of the under-lying principles of exchanger designs needed to use this software effectively.

Yusuf Ali Kara, Ozbilen Guraras[21] prepared a computer based design model for preliminary design of shell and tube heat exchangers with single phase fluid flow both on shell and tube side. The program determines the overall dimensions of the shell, the tube bundle, and optimum heat transfer surface area required to meet the specified heat transfer duty by calculating minimum or allowable shell side pressure drop. He has concluded that circulating cold fluid in shell side has some advantages on hot fluid as shell stream since the former causes lower shell side pressure drop and requires smaller heat transfer area than the latter and thus it is better to put the stream with lower mass flow rate on the shell side because of the baffled space.

M.Serna and A.Jimenez[19] have presented a compact formulation to relate the shell side pressure drop with the exchanger area and the film coefficient based on the full Bell

Delaware method. In addition to the derivation of the shell side compact expression, they have developed a compact pressure drop equation for the tube-side stream, which accounts for both straight pressure drops and return losses. They have shown how the compact formulations can be used within an efficient design algorithm. They have found a satisfactory performance of the proposed algorithms over the entire geometry range of single phase, shell and tube heat exchangers.

Andre L.H. Costa, Eduardo M. Queiroz[18] studied that techniques were employed according to distinct problem formulations in relation to: (i) heat transfer area or total annualized costs, (ii) constraints: heat transfer and fluid flow equations, pressure drop and velocity bound; and (iii) decision variable: selection of different search variables and its characterization as integer or continuous.

M. M. El-Fawal, A. A. Fahmy and B. M. Taher[20] had established a computer program for economical design of shell and tube heat exchanger using specified pressure drop to minimize the cost of the equipment. The design procedure depends on using the acceptable pressure drops in order to minimize the thermal surface area for a certain service, involving discrete decision variables. Also the proposed method takes into account several geometric and operational constraints typically recommended by design codes, and provides global optimum solutions as opposed to local optimum solutions that are typically obtained with many other optimization methods.

2.4.3 Literature review related to experimental method for evaluating shell side heat transfer coefficient

Zahid H. Ayub[16] presented new chart method to calculate single-phase shell side heat transfer coefficient in a typical TEMA style single segmental shell and tube heat exchanger. A case study of rating water-to-water exchanger has been carried out. And finally it has been conclued that this new method is reliable and comparable to the most widely known HTRI software.

R. Hosseini, A. Hosseini-Ghaffar, M. Soltani[17]experimentally obtained the heat transfer coefficient and pressure drop on the shell side of a shell and tube heat exchanger for three different types of copper tubes (smooth, corrugated and with micro-fins). Also, experimental data has been compared with theoretical data available. Experimental work shows higher Nusselt number and pressure drop with respect to theoretical correlation based on Bell's method. The optimum condition for flow rate (for the lowest increase of pressure drop) in replacing the existing smooth tube with similar micro-finned tube bundle was obtained for the oil cooler of the transformer under investigation.

At last there is conclusion like an increase in pressure drop with increase fluid flow rate in shell and tube heat exchanger which increase the pumping power.

2.4.4 General guideline for the basic components of shell and tube heat exchanger

Shell and tube heat exchanger are built of round tubes mounted in a cylindrical shell with the tubes parallel to the shell. One fluid flow inside the tubes, while the other fluid flows across and along the axis of the exchanger, the major components of this exchanger are tubes (tube bundles), shell, front end head, rear end head, baffles and tube sheets.

TEMA Standards

TEMA is a The standard of the Tubular Exchanger Manufacturers Association[8] describe various components in detail of shell and tube heat exchanger (STHE). STHE is divided into three parts namely, the front head, the shell and the rear head. Each part has different construction and specific function. The construction of front and rear head as well as flow patterns in the shell are defined by the TEMA standards. For example, a BFL exchanger has a bonnet cover, a two-pass shell with a longitudinal baffle and a fixed tube sheet rear head.

There are various types of STHE, but From industrial point of view it is necessary to operate shell and tube heat exchanger at optimal condition thus it reduces an operating and maintenance cost.

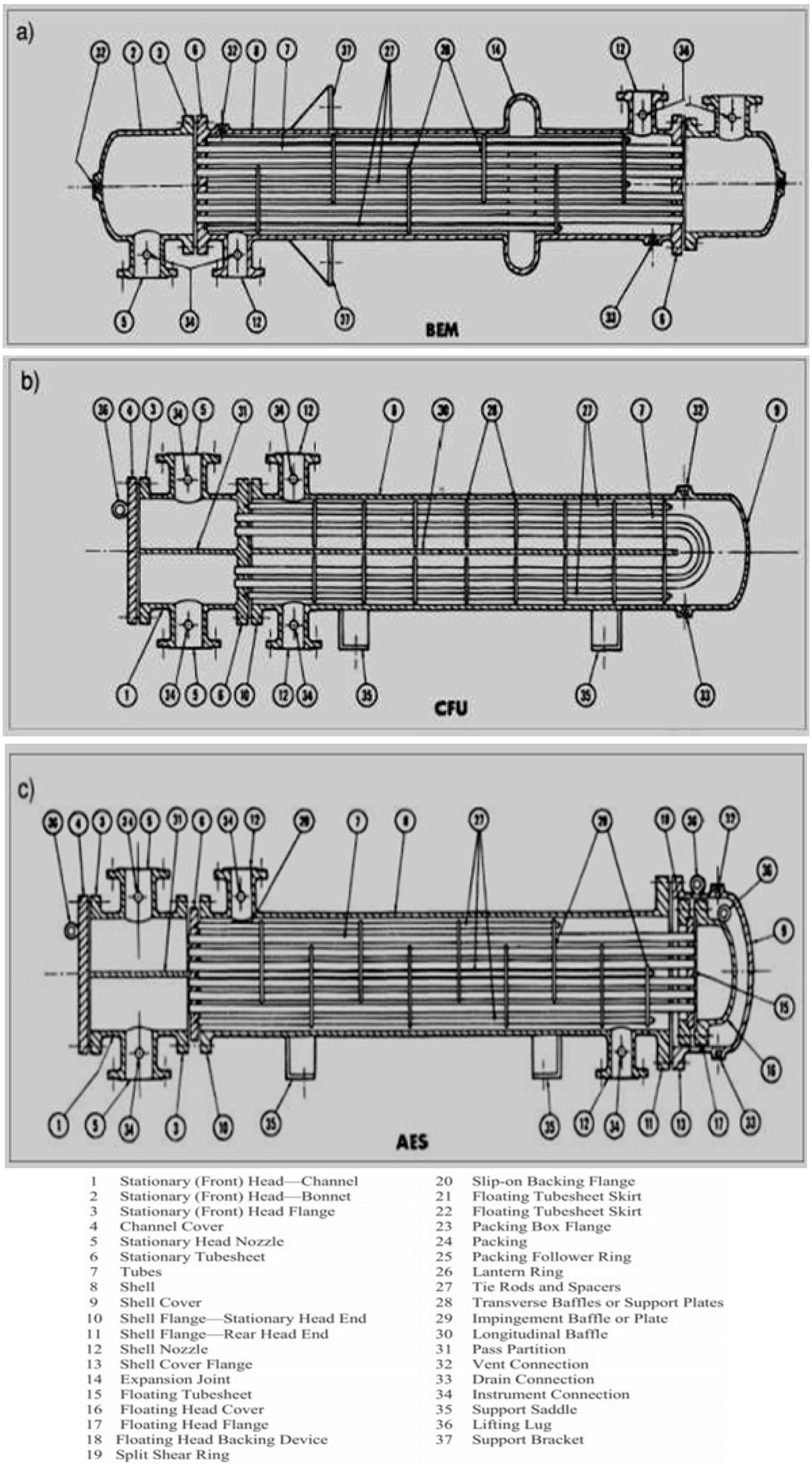

Figure 2.5: Construction parts and connection of heat exchanger [8]

Mass velocity strongly influences the heat transfer coefficient. For turbulent flow, the tube side heat transfer coefficient varies to the 0.8 power of tube side mass velocity, whereas tube side pressure drop varies to the square of mass velocity. Thus, with increasing mass velocity, pressure drop increases more rapidly than does the heat-transfer coefficient. Consequently, there will be an optimum mass velocity above which it will be wasteful to increase mass velocity further. The construction geometry and thermal parameters such as mass flow rate, heat transfer coefficient etc. are strongly influenced by each other.

Tubes[8]

Number of tube in the shell and tube heat exchanger mainly depends upon the pressure drop allowable in the heat exchanger. Tube material should also be taken into account for the design saftey point of view.

The number of tube passes depends on the available pressure drop. Higher velocities in the tube result in higher heat transfer coefficients, at the expense of increased pressure drop. Therefore, if a higher pressure drop is acceptable, it is desirable to have fewer but longer tubes that means reduced flow area and increased flow length. Long tubes are accommodated in a short shell exchanger by multiple tube passes. The number of tube passes in a shell generally range from 1 to 10. The standard design has one, two, or four tube passes. An odd number of passes is uncommon and may result in mechanical and thermal problems in fabrication and operation.

Tube Pitch and Layout

Tube pitch is the distance between two consequtive tube. And it is different for the different tube layout material. Fig.2.6shows the different layout for the tube arrangement.

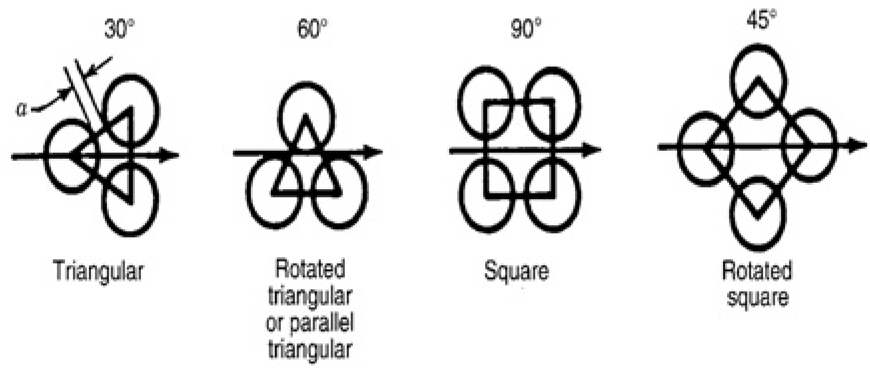

Figure 2.6: Tube layout arrangement[8]

Tubes and Tube Passes

Only E-shell with one tube pass and a F-shell with two tube passes result in nominal counterflow. All other multiple tube passes require a temperature profile correction (factor F), or, in some cases, simply cannot deliver the desired temperatures because of temperature cross. The next resort is to use multiple units in series. Generally, a large number of tube passes are used to increase tube side fluid velocity and the heat transfer coefficient (within the available pressure drop) and to minimize fouling. Tube metal is usually low carbon steel, low alloy steel, stainless steel, copper, admiralty. Cupro nickel, inconel, aluminium (in the form of alloys), or titanium. Other materials can also be selected for specific applications. The wall thickness of heat exchanger tubes is standardized in terms of the Birmingham Wire Gage (BWG) of the tube. Small tube diameters (8-15 mm) are preferred for greater area/volume density but are limited, for purposes of in tube cleaning, to 20 mm (3/4 in.). Larger tube diameters are often required for condensers and boilers. Tube length affects the cost and operation of heat exchangers. Basically, the longer the tube for any given total surface, the fewer tubes are needed, fewer holes are drilled, and the shell diameter decreases, resulting in lower cost. There are, of course, several limits to this general rule, best expressed by the shell-diameter-to-tube-length ratio, which should be within limits of about 1/5 to 1/15, Maximum tube length is sometimes dictated by architectural1ayouts and, ultimately, by transportation to about 30 m.

Baffle type and Geometry

Baffles

Baffles are used to support tubes enable a desirable velocity to be maintained for the shell side fluid, and prevent failure of tubes due to flow-induced vibration.

Baffles must overlap at least one tube row in order to provide adequate tube support. They are spaced throughout the tube bundle and somewhat to provide even fluid velocity and pressure drop at each baffled tube section. Single segmental baffles force the fluid or gas across the entire tube count, where is changes direction as dictated by the baffle cut and spacing. This can result in excessive pressure loss in high velocity gases. In order to affect Heat Transfer, yet reduce the pressure drop, double-segmental baffles can be used. This approach retains the structural effectiveness of the tube bundle, yet allows the gas to flow between alternating sections of tube in a straighter overall direction, thereby reducing the effect of numerous changes of direction.

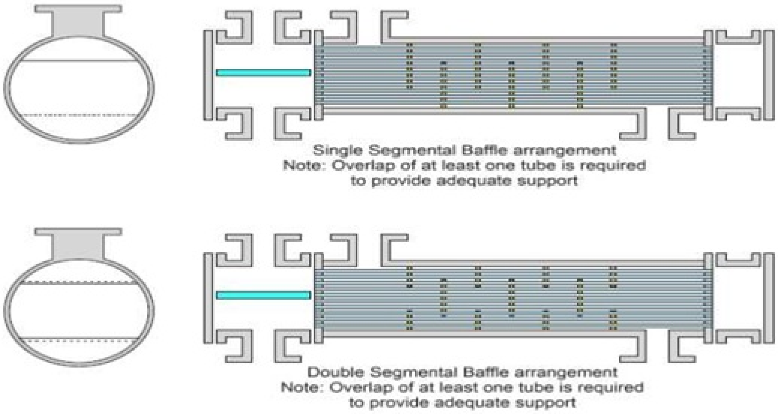

Figure 2.7: Types of Baffle[1]

Plate baffles

Two types of plate baffles are segmental, disk and doughnut. Single and double segmental baffles are used most frequently.

Rod baffles

Rod baffles are used to eliminate flow-induced vibration problems.

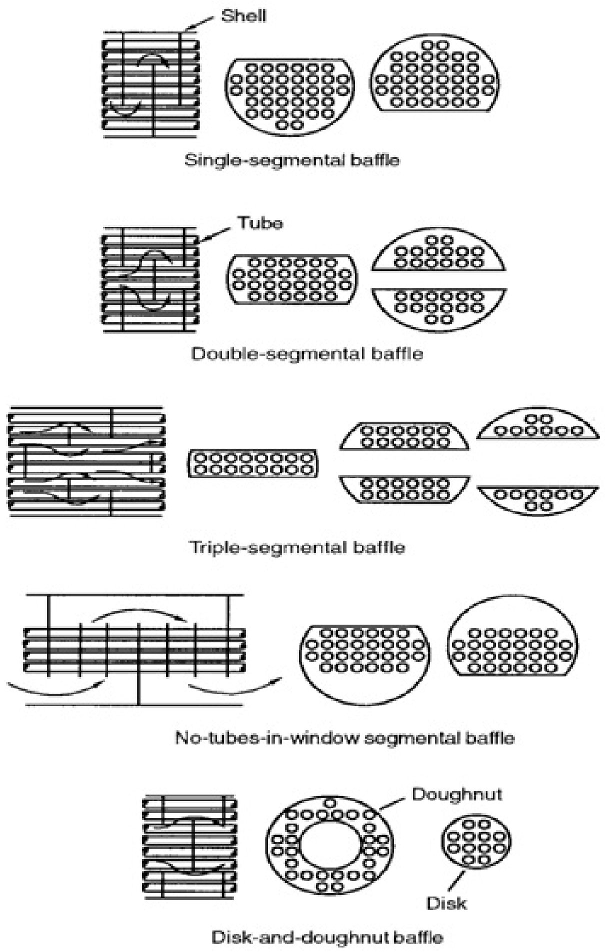

Figure 2.8: Plate baffle types, Modified from Mueller (1973) [8]

Impingement baffles

Impingement baffles are mostly used in the shell side just below the inlet nozzle.

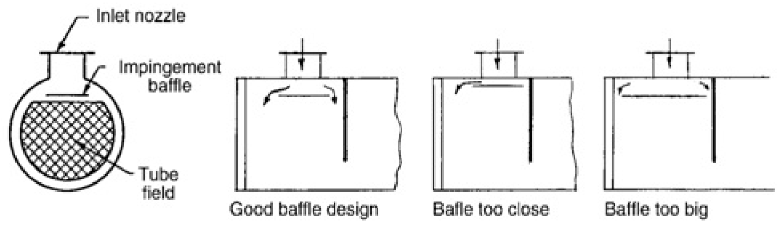

Figure 2.9: Impingement baffle at the shell-side inlet nozzle(from Bell,1998)[8]

Baffle Geometry

Dogan Eryener[5]studied the different effect like pressure drop and the heat transfer coefficient on the fluid and had studied the thermo economic optimization of baffle spacing for shell and tube heat exchanger. Despite the importance of thermo economic analysis in shell and tube heat exchanger design, the determination of the optimum baffle spacing by using the thermo economic analysis is usually neglected. On the other hand, baffle spacing is one of the most important parameters used in the design of shell and tube heat exchangers, and there is no precise criterion for the determination of baffle spacing in the literature. He studied thermo economic analysis to determine the optimum baffle spacing, accompanied by an example of such an optimization of baffle spacing for a shell and tube heat exchanger**[7]**

The results of this example are then used to demonstrate how the optimum ratio of baffle spacing to shell diameter is determined precisely and affected by the varying values of the geometrical parameters. Finally, the results are compared to those obtained by classical approaches. One of the most effective methods of enhancing the heat transfer coefficient for shell and tube heat exchangers is the use of a baffle arrangement. Besides, baffles support the tubes during operation and help to prevent vibration from flow induced eddies. Although heat transfer is increased through the baffle arrangement, the pressure drop of the shell side fluid is also increased due to the decreased flow area, leakage and bypass effects. From this point of view, baffle spacing is among the most important parameters used in the design of shell and tube heat exchangers. Closer spacing causes higher heat transfer, but this leads to poor stream distribution and higher pressure drop. On the other hand, higher baffle spacing reduces the pressure drop, but this will allow more longitudinal flow, which decreases the heat transfer. It is, thus, difficult to realize the advantage of baffle arrangements.

H. Li, V. Kottke[7] studied the shell side heat and mass transfer in the shell-and-tube heat exchanger with disc-and-doughnut baffles. A mass transfer measuring technique is used to visualize and determine the shell side local heat transfer coefficients at each tube in two representative baffle compartments of a shell-and-tube heat exchanger with disc-and-doughnut baffles. The fluid flow adjacent to the tube is analysed and the heat transfer in the zones of separated flow. The shell side flow distribution is determined through the measurements of the local pressure drop in the baffle-tube and baffle-shell clearances. Compared to the single-segmental baffle, the disc-and-doughnut baffles have a higher effectiveness of heat transfer to pressure drop. This investigation presents also per tube and per compartment averaged heat transfer coefficients. The most commonly used baffle, the segmental baffle, causes the shell side fluid to flow across the tube bundle. This improves heat transfer by enhancing turbulence or local mixing on the shell side of the exchanger, but at the cost of a high pressure drop. This is caused by flow separation at the edge of the baffles with subsequent flow contraction and expansion. Due to its broad application, a lot of work has been reported. Compared to segmental baffles, disc-and-doughnut baffles have not achieved similar popularity, mainly because of manufacturing problems and the absence of comparable information on heat transfer and pressure loss. They have higher effectiveness of heat transfer to pressure drop than segmental baffles which is due to the radial flow between the bundle centre and periphery, which eliminates bundle bypass, and uses much lower cross-flow mass velocity than segmental baffles. In this investigation, the shell side local heat transfer coefficients of a shell-and-tube heat exchanger with disc-and-doughnut baffles are measured by applying a mass transfer measuring technique which is based on absorption, chemical and colour giving reaction. The visualization of the local shell side heat transfer coefficient by mass transfer at the individual tubes gives information on the local fluid flow adjacent to the surface of each tube. Through measurements of pressure drop in the baffle-shell and the baffle-tube clearances, local shell side flow distributions are calculated for different inlet flow rates. The results in this paper allow the formulation and appraisal of prediction methods and, as a result, improve current design of the shell-and-tube heat exchangers with disc-and doughnut baffles. The development of the local heat transfer coefficients in the stagnation region important for the improvement of the integral heat exchanger and for reduction of fouling in these regions.

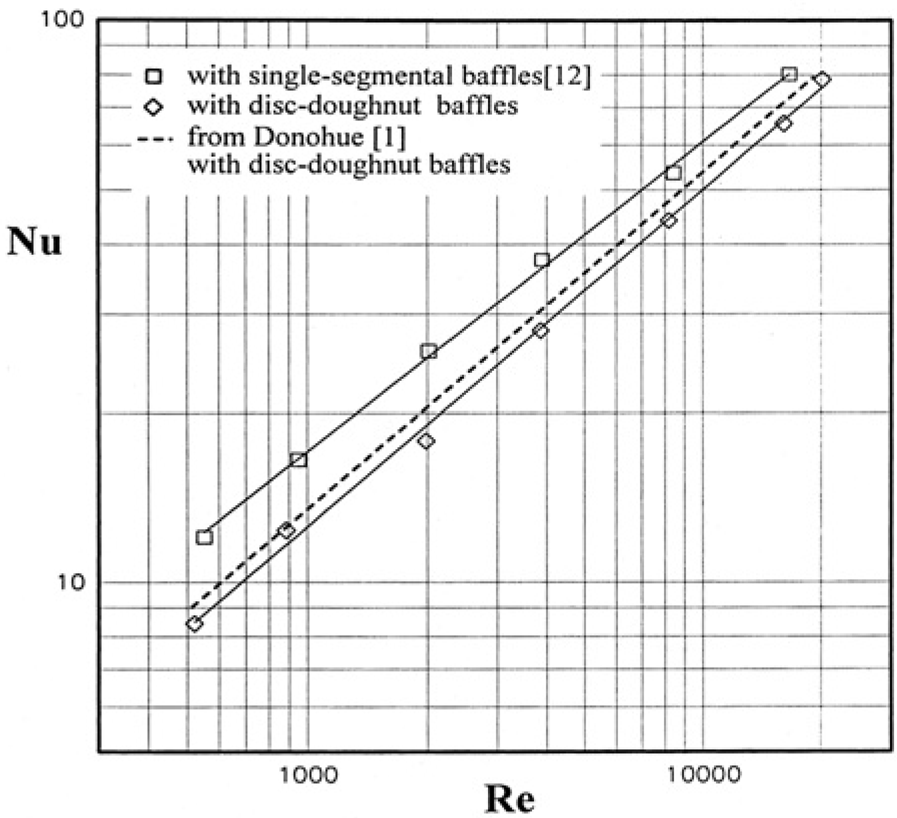

Figure 2.10: Averaged Nusselt number in the test baffle compartment[7]

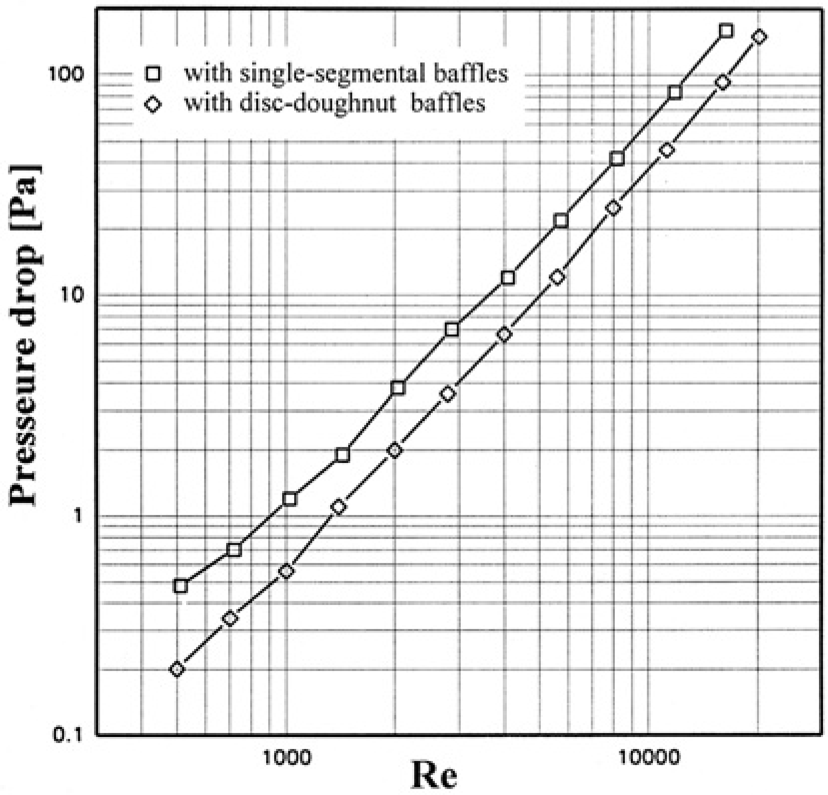

Figure 2.11: Pressure drop in the two baffle compartment[7]

A mass transfer measuring technique is used in this investigation to visualize and to determine the shell side distribution of the heat transfer coefficients at each tube surface in two representative baffle compartments of a shell-and-tube heat exchanger with disc-and dough-nut baffles, which reveals the flow field, particularly the size of the stagnation behind the baffles. The low heat transfer in this stagnation region can be considerably improved through the recirculating flow for high Reynolds number. Longitudinal vortices are observed in the centre. The disc-and-doughnut baffles have a higher effectiveness of heat transfer to pressure drop than single-segmental baffles because of partly a lower percentage of baffle-shell leakage. The detailed knowledge about the shell side local heat transfer coefficients and about the flow distribution provided in this investigation allows a further improvement of the shell-and-tube heat exchangers[7].

Muhammad Mahmood Aslam Bhutta et.al [3]studied the CFD application on heat exchanger and he has found that CFD has been employed for the following areas of study in various types of heat exchangers: fluid flow application, fouling, pressure drop and thermal analysis in the design and optimization phase.

Conventional methods used for the design and development of heat exchangers are largely tedious and expensive in today's competitive market. CFD has emerged as a cost effective alternative and it provides speedy solution to heat exchanger design and optimization. Easily accessible general purpose CFD commercial software can fulfill the requirements of CFD analysis of various types of heat exchangers including but not limited to Plate, Shell and Tube, Vertical Mantle, Compact and Printed Circuit Board Exchangers. These are flexible enough to accommodate any kind of analysis requirement ranging from prediction of fluid flow behaviour to complete heat exchanger design and optimization involving a wide range of turbulence models and integrating schemes available in CFD softwares. Of these, the k-ε turbulence model has been most widely employed for heat exchanger design optimization. The simulations generally yield results within good agreement with the experimental studies ranging from 2% to 10% while in some exceptional cases, vary up to 36%. In case of large deviations, user defined sub routines specific to the design problem may become necessary. Dependability and reliability of CFD results has reached a point making it an integral part of all design processes, leading towards eliminating the need of prototyping.

Shell type

Various front and rear head types and shell types have been standardized by Tubular Exchanger Manufacturer Association (TEMA).

The most common shell types as condensers. The E-shell is the most common due to its cheapness and simplicity. In this shell, the shell fluid enters at one end of the shell and leaves at the other end; that is, there is one pass on the shell side. The tubes may have a single or multiple passes and are supported by transverse baffles. This shell is the most common for single-phase shell fluid applications. With a single tube pass, a nominal counterflow can be obtained.

To increase the effective temperature differences and, hence, exchanger effectlveness, a pure counterflow arrangement is desirable for a two tube pass exchanger. This is achieved by the use of an F-shell with a longitudinal baffle and resulting in two shell passes, It is used when units in series are required, with each shell pass representing one unit. The pressure drop is much higher than the pressure drop of a comparable E-shell. Other important shells are the J-shell and X-shell. In the divided flow J- shell, fluid entry is centrally located and split into two parts.

The X-shell has a centrally located fluid entry and outlet, usually with a distributor dome. The two fluids are over the entire length of the tubes and are in crossflow arrangement. No baffles are used in this shell tube. Consequently, the pressure drop is extremely low. It is used for vacuum condensers and low pressure gases. The split flow shells such as the G-shell and H-shell are used for specific applications. The G-shell can be used for single phase flows but is very often used as a horizontal thermo siphon reboiler. In this case, the longitudinal baffle serves to prevent flashing out of the lighter components of the shell fluids and provides increased mixing.

The K-shell is a kettle reboiler with the tube bundle in the bottom of the shell covering about 60% of the shell diameter. The liquid covers the tube bundle and the vapour occupies the upper space without tubes. This shell is used when a portion of a stream needs to be vaporized, typically to a distillation column. The feed liquid enters the shell at the nozzle near the tube sheet, the nearly dry vapour exits out the top nozzle, and non-vaporized liquid overflows the end weir and exits through the right-hand nozzle. The tube bundle is commonly a U-tube configuration [9].

2.5 Basic Design Procedure of a Shell and Tube Heat Exchanger

A preliminary design of the heat exchanger should be done. Then the design should be rated. For that first of all we have to calculate the individual heat transfer coefficients with fouling factor. Tables for that are available in various hand books. Overall heat transfer coefficients based on the outer diameter of tubes can be estimated from the estimated value of the individual heat transfer coefficient, the wall and fouling resistance, and the overall surface efficiency. Then we have to calculate the LMTD for the flow from the given inlet and outlet temperatures. From the heat balance equation we can find the unknown value for the required calculation. The problem is now to convert the area to a reasonable dimension of the first trail. The objective is to find the number of tubes of diameter, and shell diameter.

2.5.1 Shell side heat transfer and pressure drop analysis

Shell side heat transfer and pressure drop analysis is not as simple as tube side heat transfer and pressure drop analysis. For the same many researchers have been carried out work for the same and different simplified correlations have been given by them. Edward S. Gaddis, Volker Gnielinski has studied the Pressure drop on the shell side of shell-and-tube heat exchangers with segmental baffles. He has studied the pressure drop for the nozzle inlet and nozzle outlet, pressure drop for in an end cross flow section, in a window section etc. And after that he has compared the results of the theoretical values with the experimental measurements and after that he has concluded that this method for the analysis for the shell side heat transfer and pressure drop is not valid and gives the variation was very large.

2.5.1.1 Shell side flow pattern

The total shell-side flow distributes itself into a number of distinct partial streams due to varying flow resistances, as shown in Fig.2.12 This flow model was originally proposed by Tinker (1951) and later modified by Palen and Taborek (1969) for a segmental baffle exchanger. Various streams in order of decreasing influence on thermal effectiveness are as follows[8]:

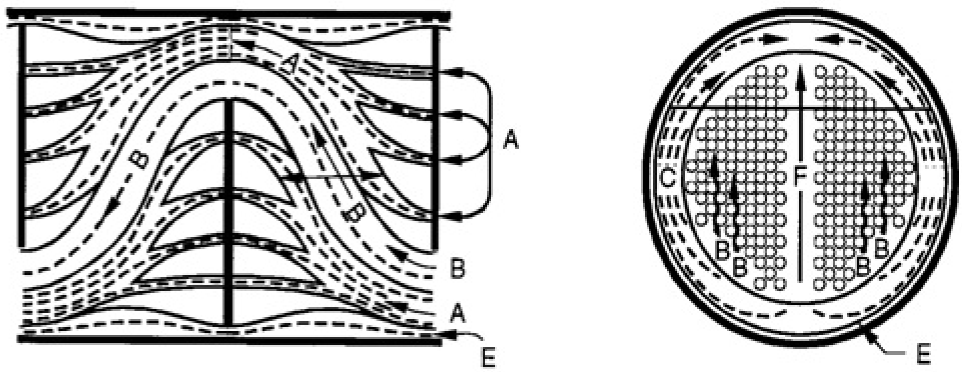

Figure 2.12: Shell side flow distributaion and identification of various streams[8]

B stream

Crossflow stream flowing over the tubes and fins between successive windows. This stream is the ‘‘desired’’ stream and is considered fully effective for both heat transfer and pressure drop.

A stream

Tube-to-baffle hole leakage stream through the annular clearance between the tubes and baffle holes of a baffle. This stream is created by the pressure difference on the two sides of the baffle. As heat transfer coefficients are very high in the annular spaces, this stream is considered fully effective.

C stream

Bundle-to-shell bypass stream through the annular spaces that means clearances between the tube bundle and shell. This bypass flow area exists because the tube holes cannot be punched close to the tube sheet edge, due to the structural strength requirement. The C stream flows between successive baffle windows. This stream is only partially effective for heat transfer, as it contacts only those tubes near the circumference.

E stream

Shell-to-baffle leakage stream through the clearance between the edge of a baffle and the shell. This stream is least effective for heat transfer, particularly in laminar flow, because it may not come in contact with any tube.

Flow Fractions for Each Shell-Side Stream

Each of the streams has a certain flow fraction F of the total flow such that the total pressure drop is the same for each stream from the entrance to the exit of the exchanger. Each stream undergoes different acceleration or deceleration and frictional processes and influences heat transfer in different ways. The design of the plate-baffled shell-and-tube exchanger should be such that most of the flow represents the crossflow B stream. However, this is rarely achieved in practice. The narrow baffle spacing results in a higher Δp for the B stream and forces more flow into the A, C, and E streams. If the computed values of the B stream are lower than those indicated, the baffle geometry and various clearances should be checked.

Since the A stream is effective from a heat transfer point of view, it is not of great concern if its flow fraction is large, as in the case of narrow baffle spacing. If the tube-to baffle hole clearance is plugged due to fouling, the shell-side pressure drop generally increases. The flow fraction of the A stream generally decreases for increasing values of multiple-segmental baffles.

Since C and F streams are only partially effective, the design of the tube bundle should be such that it minimizes the flow fraction for each of these streams to below 10%. Sealing devices are used for this purpose.

The E stream does not contact the heat transfer area and is ineffective from the heat transfer viewpoint. It mixes only poorly with other streams. Since the baffle-to-shell clearances are dictated by TEMA standards, if the flow fraction computed for the E stream is found to be excessive (15% or more); the designer should consider multiple segmental baffles instead of a single-segmental baffle. This is because the total shell-side pressure drop is lower for the multiple-segmental baffle case, thus forcing more flow to the B, A, and C streams.

Based on extensive test data, Palen and Taborek (1969) arrived at the flow fractions for various streams is shown in table. It should be noted from this table that the B stream may represent only 10% of the total flow for some exchangers. Even for a good design, it represents only 65% of the total flow in turbulent flow. Hence the performance predicted based on the conventional MTD method will not be accurate in general. As a result, there is no need to compute very accurate values of the Mean Temperature Correction Factor F for various exchanger configurations.

Table 2.1: Flow fraction(%) for various shell side flow streams[Palen and Taborek][8]

Flow streams	Turbulent flow	Laminar flow
Cross flow streams	30-65	10-50
Tube-to-baffle leakage streams A	9-23	0-10
Bundle-to-shell bypass streams C	15-33	30-80
Bundle-to-shell leakage streams E	6-21	6-48

2.5.1.2 The Bell-Delaware method[22]

From 1947 to 1963, the Department of Chemical Engineering at the University of Delaware carried out a comprehensive research program on shell side fluid flow and heat transfer in shell and tube heat exchanger. From 1947 to 1959, the experimental program was carried on, beginning with measurements of heat transfer and pressure drop during flow across ideal tube bank. These efforts were successfully extended to introduce the various design features characteristics of shell and tube heat exchangers in commercial use. Sequentially, various baffle cut and spacing configurations were investigated inside a cylindrical shell with Baffle leakage and minimal bypass clearance. Baffle leakage between and the shell and between the tubes and baffles were added later.

Shell side heat transfer coefficient[22]

In the Bell–Delaware method, the shell side heat transfer coefficient h is determined using equation:

$$h_{shell} = h_{ideal} J_c J_l J_b J_s J_r \tag{2.1}$$

By correcting the ideal heat transfer coefficient for various leakage and bypass flow streams in a segmentally baffled shell-and-tube exchanger. The h_{ideal} is determined for pure crossflow in a rectangular tube bank assuming that the entire shell-side stream flow across the tube bank is at or near the centreline of the shell.

Where, J_c = Correction factor for baffle configuration (baffle cut and spacing). It takes into account heat transfer in the window and leads to the average for the entire heat exchanger. It is dependent on the fraction of the total number of tubes in crossflow between baffle tips. Its value is 1.0 for exchanger with no tubes in the windows and increases to 1.15 for small baffle cuts and decreases to 0.65 for large baffle cuts. For a typical well designed heat exchanger, its value is near 1.0.

J_l= Correction factor for baffle leakage effects, including both tube-to-baffle and baffle to-shell leakages (A and E streams) with heavy weight given to the latter and credit given to tighter constructions. It is a function of the ratio of the total leakage area per baffle to the crossflow area between adjacent baffles, and also of the ratio of the shell-to-baffle leakage area to tube-to-baffle leakage area. If the baffles are too close, it will be lower, due to higher leakage streams. A typical value of its is in the range 0.7 to 0.8.

J_b = Correction factor for bundle and pass partition bypass (C and F) streams. It varies from 0.9 for a relatively small clearance between the outermost tubes and the shell for fixed tube sheet construction to 0.7 for large clearances in pull-though floating head construction. It can be increased from about 0.7 to 0.9 by proper use of the sealing strips in a pull-through bundle.

J_s = Correction factor for larger baffle spacing at the inlet and outlet sections compared to the central baffle spacing. The nozzle locations result in larger end baffle spacing and lower velocities and thus lower heat transfer coefficients. It is usually varies from 0.85 to 1.0.

J_r = Correction factor for any adverse temperature gradient build up in laminar flows. This correction applies only for shell-side Reynolds numbers below 100 and fully effective for $Re_s < 20$; otherwise, it is equal to 1.

Shell side pressure drop

Similar to shell-side heat transfer, the shell-side pressure drop is also affected by various leakage and bypass streams in a segmentally baffled exchanger. The shell-side pressure drop has three components: (1) pressure drop in the central (crossflow) section, Δp_{cr}; (2) pressure drop in the window area, Δp_w; and (3) pressure drop in the shell-side inlet and outlet sections, $\Delta p_{inlet-outlet}$. It is assumed that each of the three components is based on the total flow rate, and that it can be calculated correcting the corresponding ideal pressure drops. From very extensive tests at the University of Delaware in which the geometrical variables were varied systematically and independently over the practical range of construction.

$$\triangle P_{shell} = \left[(N_b - 1)\left(\triangle P_{b,i}\right) R_b + N_b \triangle P_{w,i}\right] R_l + 2\triangle P_{b,i} R_b \left(1 + \frac{N_{cw}}{N_c}\right) R_s \tag{2.2}$$

Where N_b is the number of baffles which are used to enhance the heat transfer and also to support the tube in the tube bank.

Pressure drop for an ideal cross flow section, $\Delta P_{b,i}$is given by following expression:

$$\triangle P_{b,i} = \frac{4 f_i m_s^2 N_c}{2 \rho_s g A_m^2} \left(\frac{\mu_{s,w}}{\mu_s}\right)^{0.14} \tag{2.3}$$

and the pressure drop for an ideal window section is given by equation:

$$\triangle P_{w,i} = \frac{m_s^2 \left(2 + 0.6 N_{cw}\right)}{2 g A_m A_w \rho_s} \tag{2.4}$$

And the correction factor for the effect of baffle leakage on pressure drop R_l could be read from the graph from the literature provided by Bell-Delaware method. And same is done for the correction factor for the bundle bypass, R_b.

Tube with circular fin

For the calculation of the heat transfer coefficient for the tube side flow remains the same for the shell and tube heat exchanger of the tube having cirular fin outside the tube, i.e., tube with external fins.

In caclulation of the heat transfer coefficient for the shell side flow using Bell Delaware method, the free flow area for the shell side flow and crossflow area at or near the centre line of the cross-section would be changed.

According to the Bell delaware, **[22],** the free flow area for the shell side flow would be

$$A_{freeflow} = B\left[(D_i - D_{otl}) + \left(\frac{D_{otl} - d_r}{p_t}\right)(p_t - d_r) - 2N_f h_f t_f\right] \tag{2.5}$$

And the crossflow area at of near the centreline for one crossflow section

$$A_{o,cr} = \left\{D_i - D_{otl} + \frac{D_{ctl}}{X_t}\left[(X_t - d_o) - (d_f - d_o)\, t_f N_f\right]\right\} B \tag{2.6}$$

So from the above equation it is obvious that the correction factor for the baffle leakage effect would be changed as compared to the tube without fin and also the ideal heat transfer coefficient would be changed.

Pressure drop for the tube remains unchanged by implementing the circular of radial fin outside the tube of the tube with external radial fins but the pressure drop for the shell side flow would be affected.

Pressure drop for an ideal crossflow section and pressure drop for an ideal window section would be affected, also the fraction of crossflow area available for bypass flow and friction factor for an ideal tube bank would be changed. So finally pressure drop for the shell side flow would be changed. Pressure drop formula for the shell side flow remains the same as for the plain tube without fins but the fact is that the parameters used in the formula remains changed and it depends upon the freeflow area available for the shell side flow after implementing the fin outside the tube.

Fin efficiency for the radial fin is ;[23]

$$\eta_f = c_2 \frac{K_1(mr_1)\, I_1(mr_{2c}) - I_1(mr_1)\, K_1(mr_{2c})}{I_0(mr_1)\, K_1(mr_{2c}) + K_0(mr_1)\, I_1(mr_{2c})} \tag{2.7}$$

, where subscript 0 and 1 represent the Bessel function of first kind and second kind respectively.

Tube with longitudinal fin

Heat transfer coefficient calculation for the tube side flow remains the same as for the tube without fin but it is changed with respect to the change in the length. And the heat transfer coefficient for the shell side flow would be based on the free flow area available for the shell side flow because of the fin outside the tubes. Pressure drop inside the tube remains the same but it is changed with respect to the change in the langth of the tube. Pressure drop for the shell side flow would be changed as per the free flow area for the shell side flow and as per the crossflow area at or near the centreline of one crossflow section.

2.6 Summary: Literature Review

Sr. No.	Auther	Remark
1	Su Thet Mon Than et.al [15]	As the tube length increases$h_i$and $R_e$decreases gradually
2	Yusuf Ali Kara et.al[21]	Circulating cold fluid in shell-side has advantage on hot fluid as shell stream since it causes lower shell side pressure drop and requires smaller heat transfer area than the latter
3	M.Serna[19]	Presented a compact formulation to relate the shell-side pressure drop with the exchanger area and film coefficient based on Bell-Delaware method
4	Zahid H.Ayub[16]	New chart method is presented and the results are comparable to HTRI software
5	H.Li, V.Kottke[7]	compared to the single-segmental baffle, the disc-and-doughnut baffles have a higher effectiveness of heat transfer to pressure drop

Finally it should be emphasized that the window section contributes high pressure drop with insignificant contribution to heat transfer. This results in an overall lower heat transfer rate to pressure drop ratio for the segmental baffle exchanger than that for grid baffle and most new shell-and-tube heat exchanger designs.

Chapter 3

Thermal Design of a Heat Exchanger

3.1 Design Requirement

3.1.1 Operating environment

Here the most important consideration is the size, because it is to placed in the compressor chamber, so in any case the dimension of the heat exchanger are restricted as to diameter of 1m and length of 6 m. Operating pressure for the oil is 14 bar and the inlet pressure for the cooling water is 1 bar. It is required to meet the heat duty requirement of $900KW$ for the three compressor load. And it should be compact one with the economical point of view.

3.1.2 Operating process parameters

The hot oil enters in the heat exchanger at temperature of $90^\circ C$ and it exit at temperature of $35^\circ C$. Cooling water enters at temperature of $20^\circ C$ temperature and it is required to exit at temperature of $25^\circ C$. Oil absorb about 80% heat load produced due to compression process. In nominal case the operating pressure of the oil is about 14 bar. And the heat duty requirement for the whole system is about $900KW$. And there is a separate system for the cooling water for providing the temperature of $20^\circ C$.

3.2 Type of Heat Exchanger Considered

Here shell and tube type of heat exchanger is considered according to the restriction of the dimension and also to the cost. And it is also a simple in construction as compared to the other type of the heat exchanger. Since it has higher capacity of the heat transfer and also has a lower pressure drop as compared to the other type of heat exchangers.

3.3 Design Methodology for the Shell and Tube Heat Exchanger

Material properties for the oil and water which are to be used in the shell and tube heat exchanger can be found out from the thermal physical properties of the various materials. The properties like density, dynamic viscosity, specific heat, thermal conductivity etc., for the water could be found out using **[9]** The thermal property for the oil which is compressor oil and its chemical name is poly alkaline glycol is found out from the [13]

Initially choosing the shell diameter for the required heat duty one can able to get the number of tube count for the given shell and tube heat exchanger. This step can be done by using the standard table for the shell inner diameter, tube inside diameter, tube outside diameter and also tube thickness. So that one can have idea to chose the first value of the shell diameter and also have the tube dimension for the basic design[9]. Then it is possible to get the final design for the required heat exchanger and after that we have to iterate it to meet the required specification and also have to meet the allowable pressure drop.

Number of tube can be calculated by following equation**[9]**

$$N_t = 0.785\left(\frac{CTP}{CL}\right)\frac{D_i^2}{(PR)^2\, d_o^2} \tag{3.1}$$

Where CTP is the tube count calculation constant that accounts for the incomplete coverage of the shell diameter by the tube, due to necessary clearance between the shell and the outer tube circle and tube omissions due to tube pass lanes for multiple pass design.

Based on the fixed tube sheet the following values are suggested:

One tube pass: CTP = 0.93

Two tube pass: CTP = 0.9

Three tube pass: CTP = 0.85

CL is the tube layout constant:

CL = 1.0 for 90° and 45°

CL = 0.87 for 30° and 60°

PR is the pitch ratio.

LMTD can be found out by using following formula:**[9]**

$$\triangle T_{lmtd} = \frac{(T_{hi} - T_{ho}) - (T_{ho} - T_{ci})}{ln\frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}} \tag{3.2}$$

For counter flow we have to include the temperature correction factor, F and is given by the graph plotted for the shell and tube heat exchanger for the one shell pass and two of more pass for the tube side and it is from the standard of Tubular Exchanger

Manufacturer Association [9]. Then LMTD for the counter flow heat exchanger can be calculate by multiplying the LMTD to the temperature correction factor, which we have to find it out from the graph.

3.3.1 Tube side calculation

3.3.1.1 Heat transfer calculation

Number of passed should be considered for the better heat transfer among the tube side and also for the shell side. Increasing the number of passed one can able to get the higher heat transfer coefficient for the tube and shell side.

Flow area through tube can be found out using the cross-section area for the tube and by considering the total number of tubes and also the number of passes and is expressed as:

$$A_{tp} = \frac{\pi}{4} d_i^2 \left(\frac{N_t}{N_{pass}} \right) \tag{3.3}$$

Velocity of the fluid in the tube can be found by mass conservation equation,

$$u_{mt} = \frac{m_t}{\rho_t A_{tp}} \tag{3.4}$$

Then we have to check for the turbulence and for the laminar of the flow, for that one can calculate the Reynold number for the tube side flow and it is give by following equation:

$$R_e = \frac{\rho_t u_{mt} d_i}{\mu_t} \tag{3.5}$$

If R_e is greater than the 10^4, the flow is turbulent and if it is less than that value, the flow is laminar. And by using the correlation for the Nusselt number for the laminar and turbulence, we have to find out the Nusselt number and using this value now we are in position to get the value for the heat transfer coefficient for tube side flow of the fluid.

Correlations for the fully developed turbulent forced convection through a circular duct with constant properties is taken from the table [9], properties are evaluated at the mean bulk temperature. Various correlations for the laminar flow is also available in the literature.

Then the heat transfer coefficient for the tube side flow is calculated by using the following expression:

$$h_i = \frac{Nu_t k_c}{d_i} \tag{3.6}$$

Where d_i is the tube inside diameter and the other parameters have the usual meaning.

3.3.1.2 Pressure drop analysis

The pressure drop encountered by the fluid making the number of pass N_{pass} through the heat exchanger is a multiple of kinetic energy of the flow. Therefore, the tube side pressure drop is calculated by [9]

$$\triangle P_{tube} = \left[\frac{4fLN_{pass}}{d_i} + 4N_{pass}\right]\frac{\rho_t u_m^2}{2} \tag{3.7}$$

At this stage we have to check it for the meeting of the allowable pressure drop for the tube side flow, if it is found to be in a limiting value, one can go for the design and if it is not in range then we have to modified it to get the required range for the tube side pressure drop.

3.3.2 Shell-side heat transfer and pressure drop calculation

3.3.2.1 Heat transfer calculation

To predict the overall heat transfer coefficient, we should calculate the tube side and shell side heat transfer coefficient from available correlations. For tubes in a shell and tube exchanger, the correlation from the literature one can find out by applying depending upon flow condition. For the shell side heat transfer and pressure drop calculation is done by large number of researchers and also has been presented the design procedures. Kern has also represented a calculation for the same. But the calculation for heat transfer and pressure drop analysis for the shell and tube heat exchanger using Kern method is a simplified method. The shell side analysis is not a straight forward as the tube side analysis, this is because the shell flow is complex, combining cross flow and baffle window flow, as well as baffle-shell bypass streams and complex flow patterns.

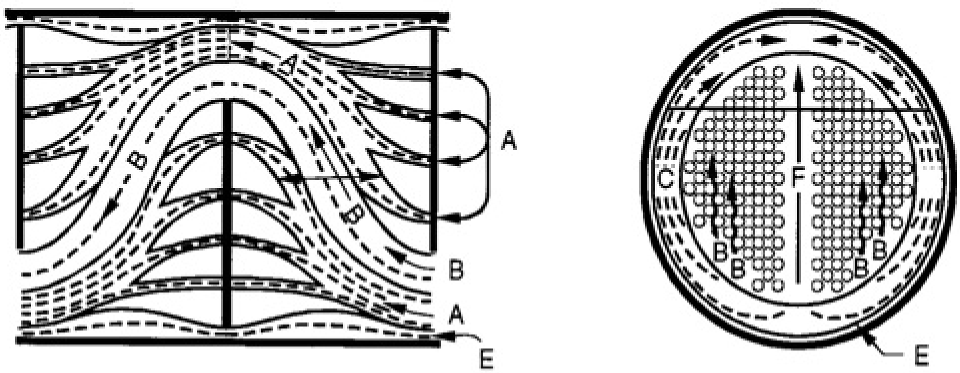

Figure 3.1: Allocatiton of flow streams[8]

The calculation for the heat transfer coefficient for the shell side fluid flow is given by the following formula:

$h_{shell} = h_{ideal}J_cJ_lJ_bJ_sJ_r$

Where h_{ideal} is found out using the formula given by,

$$h_{ideal} = J_i C_{ps} \left(\frac{m_s}{A_s}\right) \left(\frac{k_s}{C_{ps}\mu_s}\right)^{\frac{2}{3}} \left(\frac{\mu_s}{\mu_{s,wall}}\right)^{0.14} \tag{3.8}$$

,$J_i$is the Colburn factor for an ideal tube bank.

At this stage we should have to define the central baffle spacing. From the literature it is taken ranging from the 25 to 50% of the shell inner diameter. And also have to fix the pitch ratio for the arrangement of the staggered type; it is usually taken as 1 to 1.33. In most of the shell and tube heat exchanger it is taken as 1.25. Clearance is the linear difference between the tube pitch and the tube outside diameter.

The variables that affect the velocity are the shell inside diameter Ds, clearance C, between adjacent tubes; the pitch size $p_t$and the baffle spacing B. The width of the flow area at the tubes located at the centre of the shell is $(D_s C/p_t)$ and the length of the flow area is taken as the baffle spacing, B. Therefore the bundle cross flow area, A_s at the centre of the shell is given by

$$A_s = \frac{D_i CB}{P_t} \tag{3.9}$$

Reynold number for the shell side flow is calculated using the formula,

$$Re_{shell} = \frac{d_o m_s}{\mu_s A_s} \tag{3.10}$$

Depending upon the shell side constructional parameters, the correction factors are calculated.

Although the ideal values for the Colburn factor can be obtained from the following form:

$J_i = a_1 \left(\frac{1.33}{\frac{p_t}{d_o}}\right)^a (Re_s)^{a_2}$, where $a = \frac{a_3}{1 + 0.14\,(Re_s)^{a_4}}$, a_1, a_2, a_3, a_4 are different coefficient for the coefficient factor and it could be taken from the table.

Table 3.1: Correlation coefficient for the correction factor,j_i [7]

Layout angle	R_e	a_1	a_2	a_3	a_4	b_1	b_2	b_3	b_4
	105-104	0.321	-0.388			0.372	-0.123		
	104-103	0.321	-0.388			0.486	-0.152		
30 °	103-102	0.593	-0.477	1.450	0.519	4.570	-0.476	7.00	0.500
	102-10	1.360	-0.657			45.100	-0.973		
	<10	1.400	-0.396			48.000	-1.000		
	105-104	0.370	-0.396			0.303	-0.126		
	104-103	0.370	-0.396			0.333	-0.136		
45 °	103-102	0.730	-0.500	1.930	0.500	3.500	-0.476	6.59	0.520
	102-10	0.498	-0.656			26.300	-0.913		
	<10	1.550	-0.667			32.00	-1.000		
	105-104	0.370	-0.395			0.391	-0.148		
	104-103	0.107	-0.266			0.0815	+0.022		
	103-102	0.408	-0.460	1.187	0.370	6.090	-0.602	6.30	0.378
	102-10	0.900	-0.631			32.1000	-0.963		
	<10	0.970	-0.677			35.0000	-1.000		

By using the above value for the different layout for the tube layout angle, we are in position to calculate the ideal heat transfer coefficient for the shell side. After that we have to find out the actual heat transfer coefficient for this method.

For a fixed tube sheet design, the outermost tubes can be close to the shell inside diameter, or the diameter of the outer tube limit and it is denoted as D_{otl}, can be the largest value followed by that for a split-ring floating head S, andD_{otl} that is the diameter of outer most tube in the tube bundle being the smallest for a pull-through head T.

For a specified diameter of the circle through the centres of the outermost tubes, D_{ctl} is given by $D_{ctl} = D_{otl} - d_o$

Baffle cut is generally taken as 25 to 50% of the shell diameter.

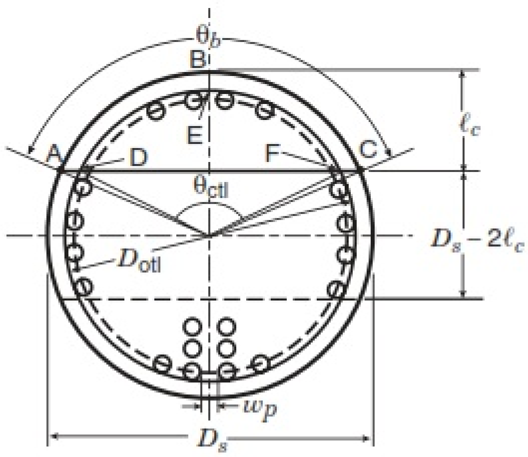

Figure 3.2: Nomenclature for basic geometry relations for a single segmental exchanger (from Taborek, 1998)[8]

As shown in figure, angle between the baffle cut and two radius of circle through centre of outermost tubes,

$$\theta_{ctl} = 2cos^{-}\left(\frac{D_s - 2l_c}{D_{ctl}}\right) \tag{3.11}$$

For the window section, we have to consider the fraction of the number of tubes in one win-dow section encircled by centreline of the outer tube row is

$$F_w = \frac{\theta_{ctl}}{2\pi} - \frac{sin\theta_{ctl}}{2\pi} \tag{3.12}$$

Cross flow section

The fraction $F_c$of the total number of tubes in the cross flow section is found from

$$F_c = 1 - 2F_w \tag{3.13}$$

Then the correction factor for baffle cut and spacing is given by **[9]**

$$J_c = 0.55 + 0.72F_c \tag{3.14}$$

Angle between two radiius intersected at the inside shell wall and with the baffle cut and is given by,

$$\theta_b = 2cos^{-}\left(1 - \frac{2l_c}{D_s}\right) \tag{3.15}$$

Diametral shell to baffle clearance $\delta_{sb}$is could be taken from the table of the TEMA standard. Shell to baffle leakage area is given by,

$$A_{o,sb} = \pi D_s\left(\frac{\delta_{sb}}{2}\right)\left(1 - \frac{\theta_b}{2\pi}\right) \tag{3.16}$$

Diametral clearance between the tube and baffle $\delta_{tb}$is taken from the TEMA standard. And the total tube to baffle leakage area is given by,

$$A_{o,tb} = \frac{\pi D_s \delta_{tb} N_t\left(1 - F_w\right)}{2} \tag{3.17}$$

Transverse and longitudinal pitch for the tube is taken for the respective tube layout. Then cross flow area at or near the centre line for one cross flow section,

$$A_{o,cr} = \left[D_s - D_{otl} + \frac{D_{ctl}}{X_t}\left(X_t - d_o\right)\right] B_c \tag{3.18}$$

Correction factor for baffle leakage effects, including both tube-to-baffle and baffle-to-shell leakages (A and E streams) with heavy weight given to the latter and credit given to tighter constructions. It is a function of the ratio of the total leakage area per baffle to the cross flow area between adjacent baffles, and also of the ratio of the shell-to-baffle leakage area to tube to-baffle leakage area. If the baffles are too close, it will be lower, due to higher leakage streams. It is given by,

$$J_l = 0.44\,(1 - r_s) + [1 - 0.44\,(1 - r_s)]\,e^{-2.2 r_{lm}} \tag{3.19}$$

where non-dimensional parameters are given by,

$$r_{lm} = \frac{A_{o,sb} + A_{o,tb}}{A_{o,cr}}$$

$$r_s = \frac{A_{o,sb}}{A_{o,sb} + A_{o,tb}}$$

To prevent the leakage between the shell to bundle, a sealing strip is provided.

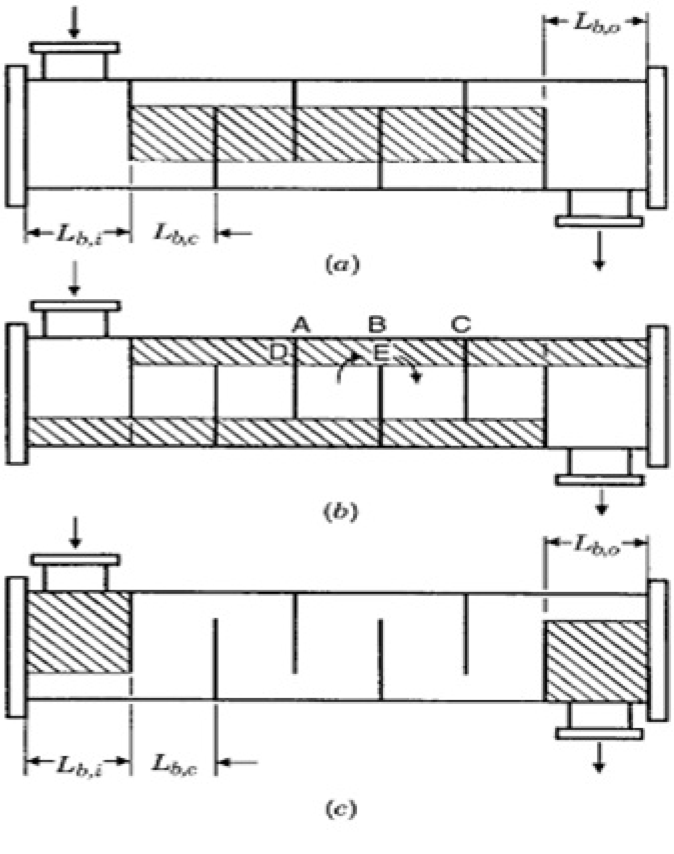

Figure 3.3: TEMA E Shell heat exchanger: (a) internal cross flow section, (b) window section, (c) Entrance and exit section.(from Taborek, 1998)[8]

Cross flow area for bypass flow is given by,

$$A_{o,bp} = B(D_s - D_{otl} + 0.5 N_p W_p) \tag{3.20}$$

Correction factor for the tube bundle and pass partition bypass (C and F) streams, and it is varies from 0.9 for a relatively small clearance between the outer most tubes and shell for a fixed tube sheet construction to 0.7 for a large clearance in pull-through floating and construction.

It is alos given by following expression,

$$J_b = e^{-Cr_b\left[1-\left(2N_{ss+}\right)\frac{1}{3}\right]} \tag{3.21}$$

where constant C and non-dimensional parameters $r_b$is given by, $r_b = \dfrac{A_{o,bp}}{A_{o,cr}}$and constant $C = -Cr_b\left[1 - (2N_{ss+})^{\frac{1}{3}}\right]$,

Number of baffles could be calculated using the following formula,

$$N_b = \frac{L}{B} - 1 \tag{3.22}$$

inlet and outlet baffle spacing is shown in figure would be taken for the design calculation and it is desire to take the inlet and outlet baffle spacing be the same. Correction factor for the large baffle spacing at the inlet and outlet should be taken as following way:

$$J_s = \frac{N_b - 1 + \left(L_i^+\right)^{(1-n)} + (L_o^+)^{(1-n)}}{N_{_b} - 1 + L_{ii}^+ + L_o^+} \tag{3.23}$$

where constant n is 0.6 for the turbulent flow and is 0.33 for the laminar flow.

Other parameters will be taken as:

$$L_i^+ = \frac{L_{b,i}}{L_{b,c}} \tag{3.24}$$

,

$$L_o^+ = \frac{L_{b,o}}{L_{b,c}} \tag{3.25}$$

Correction factor for any adverse temperature gradient build up in flows J_r is taken as 1 for turbulent flow.

Finally one can get the shell side heat transfer coefficient by using this method and by considering the leakage effect that are occurs in the flow phenomenon by multiplying all this correction factors to the ideal heat transfer coefficient and is given by,

$h_{shell} = h_{ideal} J_c J_l J_b J_s J_r$

3.3.2.2 Pressure drop calculation

Similarly shell side heat transfer, shell side pressure drop is also affected by various leakage and bypass streams in a segmentally baffled exchanger.

The shell side pressure drop has three components:

(1) Pressure drop in the central section,

(2) Pressure drop in the window section,

(3) Pressure drop in the shell side inlet and outlet section.

Here it is assumed that each of three component is based on the total flow rate, and that it can be calculated correcting the corresponding the ideal pressure drop.

Pressure drop for an ideal tube cross flow section

Find out the friction factor f_i from the ideal tube bank friction curve for the given tube layout at the calculated value of the Reynold number[22]. Cross flow area at or near the centre line for one cross flow section is given by,

$$A_m = B\left[D_i - D_{otl} + \left(\frac{D_{otl} - d_o}{P_t}\right)(p_t - d_o)\right] \tag{3.26}$$

Fraction of cross flow area available for bypass flow is given by,

$$F_{sbp} = \frac{\left[D_i - D_{otl} + \frac{1}{2}N_p W_p\right]B}{A_m} \tag{3.27}$$

Tube pitches parallel and normal to flow is taken from the table[22].

Number of tube rows crossed in one cross flow section (between baffle tips), N_c is estimated from:

$$N_c = \frac{D_i\left[1 - 2\left(\frac{l_c}{D_i}\right)\right]}{p_p} \tag{3.28}$$

then the pressure drop for the ideal cross section is:

$$\Delta P_{b,i} = \frac{4 f_i m_s^2 N_c}{2\rho_s g A_m^2}\left(\frac{\mu_{s,w}}{\mu_s}\right)^{0.14} \tag{3.29}$$

3.3.2.3 Pressure drop for the window section

Number of effective cross flow rows in each window,

$$N_{cw} = \frac{0.8 l_c}{p_p} \tag{3.30}$$

Gross window area is given by,

$$A_{wg} = \frac{D_i^2}{4}\left\{\frac{\theta}{2} - \left[1 - 2\left(\frac{l_c}{D_i}\right)\right] sin\left(\frac{\theta}{2}\right)\right\} \tag{3.31}$$

Window area occupied by tubes is

$$A_{wt} = \frac{N_t}{8}\left(1 - F_c\right)\pi d_o^2 \tag{3.32}$$

Then the area for the flow through the window section is the difference between the gross area and the area occupied by the tubes in the window section.

Then the pressure drop for the ideal window section is given by,

$$\triangle P_{w,i} = \frac{m_s^2 (2 + 0.6 N_{cw})}{2g A_m A_w \rho_s} \tag{3.33}$$

Calculate the correction factor for effect of baffle leakage on pressure drop, R_l and it is read from the graph as a function of the other parameters like

$$\frac{A_{o,sb} + A_{o,tb}}{A_m}$$

and

$$\frac{A_{o,sb}}{A_{o,sb} + A_{o,tb}}$$

Then the fraction of the cross flow area available for bypass flow is given by

$$F_{sbp} = \frac{\left[D_i - D_{otl} + \frac{1}{2}(N_p W_p)\right] B}{A_m} \tag{3.34}$$

Then the calculation for the correction factor for the bundle bypass, R_b is to be done and is read from the graph as function of fraction of the cross flow area available for the bypass and $\frac{N_{ss}}{N_c}$

Correction factor for the unequal baffle spacing, R_s. The equation is

$$R_s = \frac{1}{2}\left[\left(l_{s,i}^*\right)^{-n} + \left(l_{s,o}^*\right)^{-n}\right] \tag{3.35}$$

Where $l_{s,i}^* = \frac{l_{b,i}}{l_b}$ and $l_{s,o}^* = \frac{l_{b,o}}{l_b}$, and $n'$is the constant and is 1 for the laminat flow and 1.6 for the turbulent flow.

3.3.2.4 Total pressure drop for the shell side flow is

$$\triangle P_{shell} = \left[(N_b - 1)(\triangle P_{b,i}) R_b + N_b \triangle P_{w,i}\right] R_l + 2\triangle P_{b,i} R_b \left(1 + \frac{N_{cw}}{N_c}\right) R_s \tag{3.36}$$

3.4 Overall Heat Transfer Coefficient

It can be found with the help of the inside or outside diameter of the tube. For that internal heat transfer area is $A_i = \pi d_i L N_t$and external heat transfer area is, $A_o = \pi d_o L N_t$and it needed to calculate the surface area for the tube material and it is generally taken as the average of the internal and external area of the tube. So it can be expressed as, $A = \frac{1}{2}(A_i + A_o)$then overall heat transfer coefficient is given by following equation:

$$U_iA_i = \frac{1}{\frac{1}{h_iA_i} + \frac{ln\left(\frac{r_o}{r_i}\right)}{2\pi kLN_t} + \frac{R_{fi}}{A_i} + \frac{R_{fo}}{A_o} + \frac{1}{A_oh_o}} \tag{3.37}$$

3.5 Length of the Tube

Length of the tube is given as a ratio of the UA_{total} to the $UA_{per\,meter}$

$$L = \frac{UA_{total}}{UA_{per\,meter}} \tag{3.38}$$

where,

$$UA_{total} = \frac{Q}{\triangle T_{lmtd}} \tag{3.39}$$

3.6 Circular Fin outside the Tube

Fin efficiency is caclulated from the equation[23]

$$\eta_f = c_2 \frac{K_1(mr_1)\,I_1(mr_{2c}) - I_1(mr_1)\,K_1(mr_{2c})}{I_0(mr_1)\,K_1(mr_{2c}) + K_0(mr_1)\,I_1(mr_{2c})} \tag{3.40}$$

where $I_1$is the Bessel function of the first kind and $K_1$is the Bessel function of the second kind. $r_1$is the tube inside radius and $r_{2c} = r_2 + \frac{t}{2}$, where t is the fin thickness and $r_2$is the outside radius of the tube.

Fin area is calculated from the equation:

$$A_{finned} = 2\pi\left(r_{2c}^2 - r_1^2\right) \tag{3.41}$$

Unfinned area for the single tube with radial fin outside the tube could be

$$A_{unfinned} = 2\pi r_1 (H - N_f t) \tag{3.42}$$

Where N_f is the total number of fin per tube.

$$m = \sqrt{\frac{2h}{kt}} \tag{3.43}$$

Where h is the heat transfer coefficeint for the shell side flow and t is the thickness of the fin and k is the thermal conductivity of the tube material.

then the effective surface area for the heat transfer is

$$A_{effective} = A_{finned}\eta_f + A_{unfinned} \tag{3.44}$$

Then overall surface efficiency,

$$\eta_o = \left[1 - (1 - \eta_f)\,\frac{A_f}{A}\right] \tag{3.45}$$

A is the total surface area and is the summation of the finned area and unfinned area of the tube.

Then also $U_iA_i$per meter would also be changed.

$$\frac{1}{U_iA_i} = \frac{1}{\eta_i h_i A_i} + \frac{R_{fi}}{\eta_i A_i} + \frac{ln\left(\frac{r_o}{r_i}\right)}{2\pi LkN_t} + \frac{R_{fo}}{\eta_o A_o} + \frac{1}{\eta_o h_o A_o} \tag{3.46}$$

Free flow area for the shell side flow

$$A_{free\,flow} = B\left[(D_i - D_{otl}) + \left(\frac{D_{otl} - d_r}{p_t}\right)(p_t - d_r) - 2N_f h_f t_f\right] \tag{3.47}$$

And the crossflow area at or near the centreline of one crossflow section would be

$$A_{o,cr} = \left\{D_i - D_{otl} + \frac{D_{ctl}}{X_t}\left[(X_t - d_o) - (d_f - d_o)\,t_f N_f\right]\right\} B \tag{3.48}$$

Ideal heat transfer coefficient for the shell side flow would be change according to the free flow area for the shell side flow. friction factor for the ideal tube bank is taken as the half of the friction factor for the tube without tube **[22]**.

Pressure drop for the tube would be changed according to the length of the tube and pressure drop for the shell would be changed due to the change in the free flow area for the shell side flow.

3.7 Longitudinal Fin outside the Tube

Design calculation for the longitudinal fin would be the same as that of the radial or circular fin outside the fin, only the difference is for the free flow area for the shell side flow and the crossflow area at or near one crossflow section. The calculation for the fin efficiency, heat transfer coefficient for the tube side flow and for the shell side flow would be the same as that of the circular fin design, only the difference is of the paramaters like free flow area and it is because of the dimension of the fin.

3.8 Design Calculations

At the initial stage hot fluid could be taken in the tube and cold fluid could be taken at shell side.

Input data:

- Heat duty-900KW
- Inlet temperature of hot fluid- 90$^\circ C$
- Outlet temperature of hot fluid- 35$^\circ C$
- Inlet temperature of cold fluid- 20$^\circ C$
- Outlet temperature of cold fluid-25$^\circ C$
- Hot fluid- Breox oil (oil) (Tube side)
- Cold fluid- Water (Shell side)
- Heat exchanger type- Shell and tube heat exchanger

Other parameters could be taken as:

$T_{hi} = 90^\circ C$
$T_{ho} = 35^\circ C$
$T_{ci} = 20^\circ C$
$T_{co} = 25^\circ C$
$Cp_h = 2006\,{}^{\mathrm{J}}/_{\mathrm{kgK}}$
$Cp_c = 4182\,{}^{\mathrm{J}}/_{\mathrm{kgK}}$
$\rho_h = 925\,{}^{\mathrm{kg}}/_{\mathrm{m^3}}$
$\rho_c = 1000\,{}^{\mathrm{kg}}/_{\mathrm{m^3}}$
$k_h = 0.146\,{}^{\mathrm{W}}/_{\mathrm{mK}}$
$k_c = 0.6\,{}^{\mathrm{W}}/_{\mathrm{mK}}$
$Pr_h = 88.96$
$Pr_c = 6.97$
$\mu_h = 0.00647\,{}^{\mathrm{Ns}}/_{\mathrm{m^2}}$
$\mu_c = 0.001\,{}^{\mathrm{Ns}}/_{\mathrm{m^2}}$

Now let us, take shell inside diameter,$D_i = 0.635\,m$, Tube inside diameter,$d_i = 0.014224\,m$ and Thickness of the tube $t = 0.002413\,m$

Tube layout constant, $CL = 0.87$ for 30°arrangement

Tube count calculation constant, $CTP = 0.9$

Pitch ratio $P_t/do = 1.25$

So $P_t = 1.25 * d_o = 1.25 * 0.01905 = 0.02381\,m$

3.8.1 Calculation for the heat transfer coefficient within the tube

Number of tube is given by,

$$N_T = 0.785\left(\frac{CTP}{CL}\right)\frac{D_i^2}{(PR)^2\,d_o^2}$$

$$= 0.785\left(\frac{0.9}{0.87}\right)\frac{0.635^2}{(1.25)^2\,0.01905^2}$$
$$= 577.47$$

And surface area for the tube is calculated using,

$$A_{tp} = \frac{\pi d_i^2}{4}\frac{N_t}{2}$$
$$= \frac{\pi * 0.014224^2 * 577.47}{4 * 2}$$
$$= 0.045857\,m^2$$

Fluid velocity of the fluid flowing inside the tube,

$$u_t = \frac{m_t}{\rho_t A_{tp}}$$
$$= \frac{8.1573}{925 * 0.045857}$$
$$= 0.1923\,\mathrm{m/s}$$

Reynold number,

$$Re = \frac{\rho_t u_t d_i}{\mu_t}$$
$$= \frac{925 * 0.1923 * 0.014224}{0.006475}$$
$$= 390$$

So the flow is laminar, and for the Heat transfer coefficient for the tube side fluid can be calculated by using the Sieder and Tate correlation[8] ,

$$Nu_t = 1.86\,(RePr)^{0.33}\left(\frac{d_i}{L}\right)^{0.33}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$
$$= 1.86\,(390 * 88.96)^{0.33}\left(\tfrac{0.014224}{4}\right)^{0.33}\left(\tfrac{0.006475}{0.00925}\right)^{0.14}$$
$$= 8.80$$

So heat transfer coefficient for the tube side fluid flow,

$$h_i = \frac{Nu_t k_t}{d_i}$$
$$= \frac{8.80 * 0.146}{0.014224}$$
$$= 90\,\mathrm{W/m^2K}$$

3.8.2 Heat transfer coefficient for the shell side fluid flow

Clearance, $C = p_t - d_o = 0.02381 - 0.01905 = 0.00476\,m$

Chossing Baffle spacing as 0.5 of the shell diameter

So, $B = 0.5 * 0.635 = 0.3175\,m$

Hydraulic diameter,

$$D_e = \frac{4\left(\frac{Pt^2\sqrt{3}}{4} - \frac{\pi d_o^2}{8}\right)}{\frac{\pi d_o}{2}}$$
$$= 0.01371\,m$$

Cross flow area across the tube bundle,

$$A_s = \frac{D_s C B}{p_t}$$
$$= \tfrac{0.635*0.00476*0.3175}{0.02381}$$

$= 0.040m^2$

Mass velocity, $G_s = \frac{m}{A_s} = \frac{43.01}{0.040} = 1067\,{}^{\mathrm{kg}}\!/_{\mathrm{sm^2}}$

Reynold number for the shell side fluid flow is given by,

$$Re_s = \frac{G_s D_e}{\mu_s}$$
$$= \frac{1067*0.01371}{0.001}$$
$$= 14629.91$$

By using Kern suggested correlation for the heat transfer coefficient,

$$\frac{h_o D_e}{k} = 0.36\left(\frac{D_e G_s}{\mu}\right)^{0.55}\left(\frac{\mu Cp}{k}\right)^{0.33}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$
$$= 0.36\left(\frac{0.01371*1067}{0.001}\right)^{0.55}\left(\frac{0.001*4182}{0.6}\right)^{0.33}\left(\frac{0.001}{0.00925}\right)^{0.14}$$
$$h_o = 98.382*\frac{0.6}{0.01371}$$
$$h_o = 4305\,{}^{\mathrm{W}}\!/_{\mathrm{m^2K}}$$

So from the initial design, here heat transfer coefficient for the shell side flow is higher than the heat transfer coefficient for the tube side flow, so it is desire to design the shell and tube heat exchanger by taking the water(cold fluid) inside the tube and oil (hot fluid) outside the tube that means in shell side.

Outside heat trasnfer coefficient is higher than the heat transfer coefficient inside the tube. So it is difficult to implement the fin inside the tube due to the fabrication limit and it is easy to implement the fins outside the tube. So it is desire to keep the hot fluid outside the tube that means on the shell side and th cold fluid (water) inside the tube to get the higher heat transfer coefficient within the tube and also in the shell side flow.

Sample calculations of the "Design of the Heat Exchanger for the Compressor After-Cooler of the Helium Plant" is shown in Appendix A

Sample caclulations of the "Design of the Heat Exchanger for the Compressor After-Cooler of the Helium Plant with Longitudinal Fin" is shown in Appendix B

Sample caclulations of the "Design of the Heat Exchanger for the Compressor After-Cooler of the Helium Plant with Circular Fin" is shown in Appendix C.

Chapter 4

Optimization of the Thermal Design

Optimization of the thermal design is carried out using Excel and it is done by varying the parameters like tube inside diameter, shell inside diameter and then by analysing the effect of this variables on the tube length, pressure drop for the tube, for the shell and also a volume of the shell.

Optimization of the design of the shell and tube heat exchanger designed above, are made depending above the four main basis, as described below:

1.Size

2. Pressure drop

3. Weight

4. Cost

The main aim of the optimization of the heat exchanger here is to obtain the required pressure drop with minimum size possible. So, in any case, the length of the heat exchanger is restricted to 6 meter, and the shell diameter of the shell is also not to go beyond 1 meter. It is thus obvious that if the size of the heat exchanger is reduced its weight and cost also will be reduced automatically. Hence, the optimization was to be done for obtaining the minimum size required for the pressure drop required i.e. 1 bar.

4.1 Variation of different parameters of the heat exchanger with changing inside diameter of the tube

4.1.1 Variation of tube side pressure drop

Here variation of the parameters of the heat exchanger is measured w.r.t varying the inside diameter of the tube for achieving the required pressure drop and to achieving the required length of the tube. So that overall volume and hence the total cost for manufacturing would be low. This minimum diameter of the tube will then be used for deciding the number of tubes for the heat exchanger will then be required for the required pressure drop and for the required length of the tube with minimum volume and dimensions of the heat exchanger within the limitations.

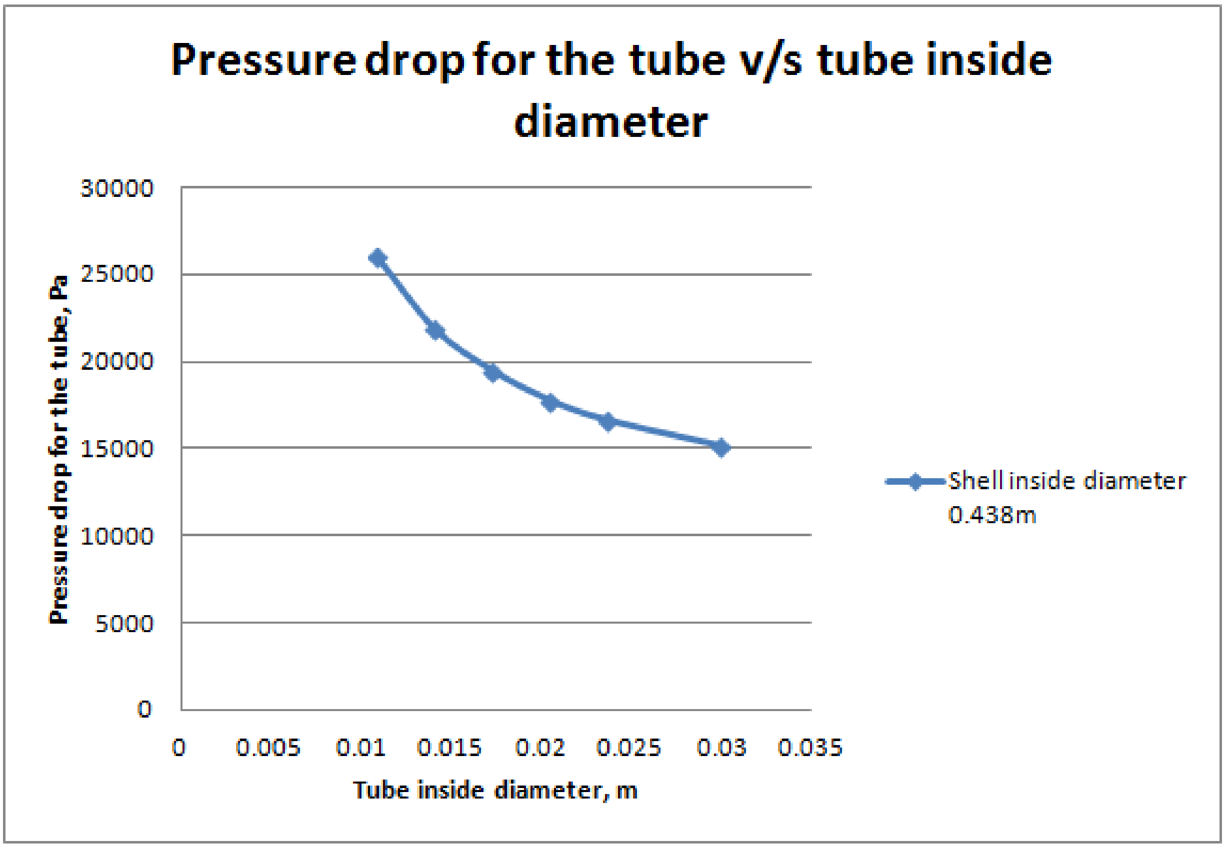

Figure 4.1: Pressure drop for the tube vs tube inside diameter

Here the pressure drop is decreases as the tube inside diameter is increasing. It should be noted that the tube outside diameter and also the tube thickness is also varying with the variation of the tube inside diameter and it increases as the tube inside diameter is increases. For this task we should have chose the standard value for the tube inside diameter, tube outside diameter, and also a tube thickness[9]. As shown in the Fig.4.1, tube side pressure drop is decreasing as the tube inside diameter is increasing. This is due to the fact that, if the tube inside diameter is increasing, the velocity of the fluid flowing through the tube is also increases as the number of tube decreases. And also from the Bernoulli's equation to meet the required increase in the pressure heat, there must be a decrease in the tube side pressure drop. And this is true for the incompressible fluid.

4.1.1.1 Variation of the pressure drop for shell side flow

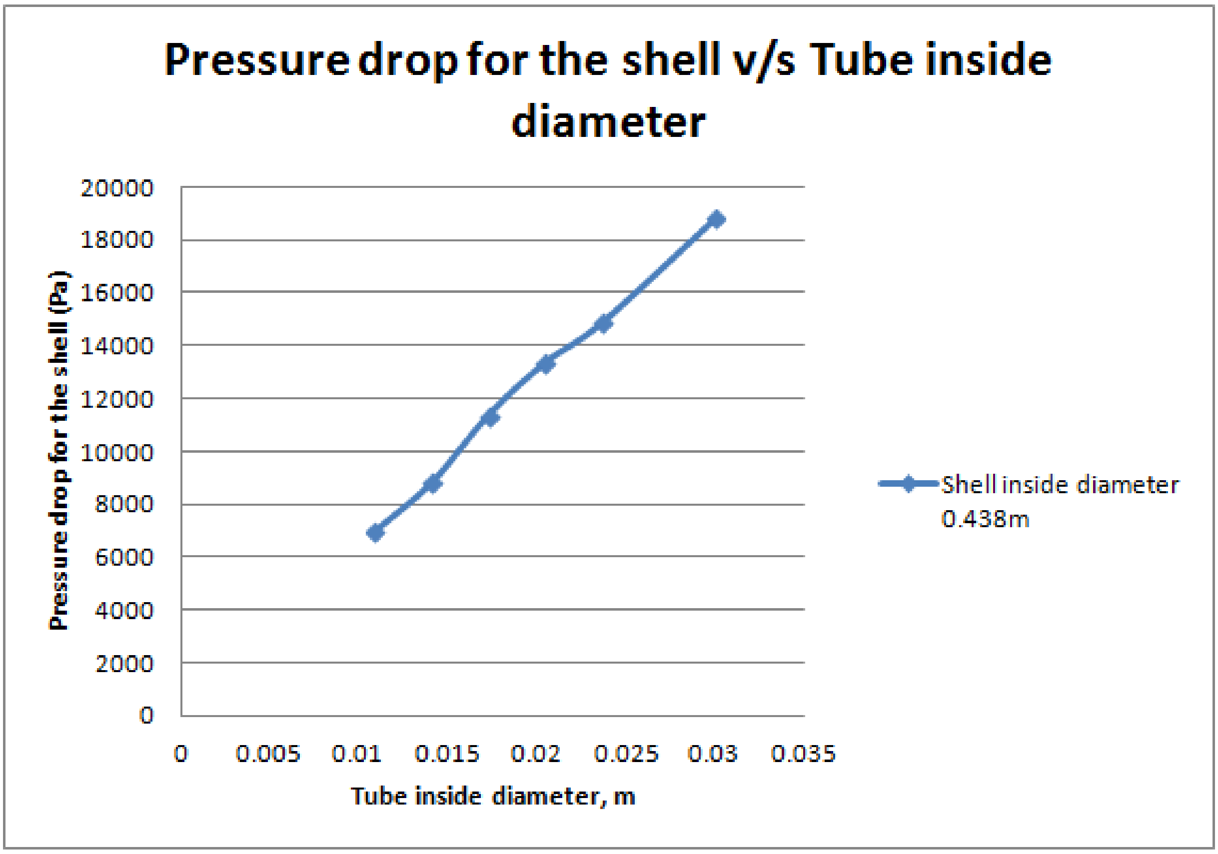

Figure 4.2: Pressure drop for the shell vs Tube inside diameter

As shown in Fig.4.2, as the tube inside diameter is increasing, the pressure drop for the shell side flow is increasing. This happens due to the fact that as the tube inside diameter increasing, tue number of tubes for the constant shell inside diameter decreasing, and because of that length of the tube increasing, so that number of baffles for the same mass flow rate increasing and therefore the free flow area for the shell side flow decreasing, and because of that frictional pressure drop is increased.

4.1.1.2 Variation of the length of the tube with changing tube inside diameter

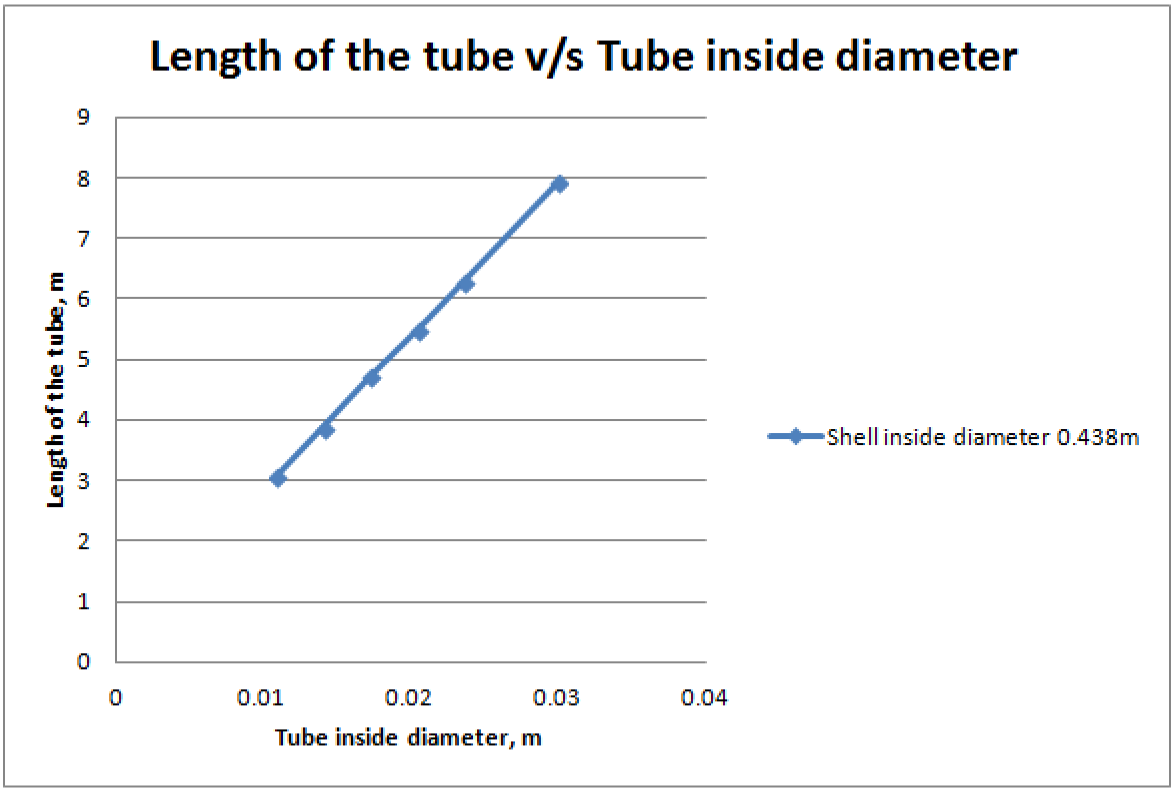

Figure 4.3: Length of the tube vs Tube inside diameter

As shown in the Fig.4.3, as the tube inside diameter increases, the length of the tube increases. This is because with the increase in the tube inside diameter, the number of tube also has to change. Number of tube decreases as the tube inside diameter of the tube increases. So for the same hear duty the tube length has to be increase for a given shell diameter of the shell.

4.1.1.3 Variation of the volume of the shell

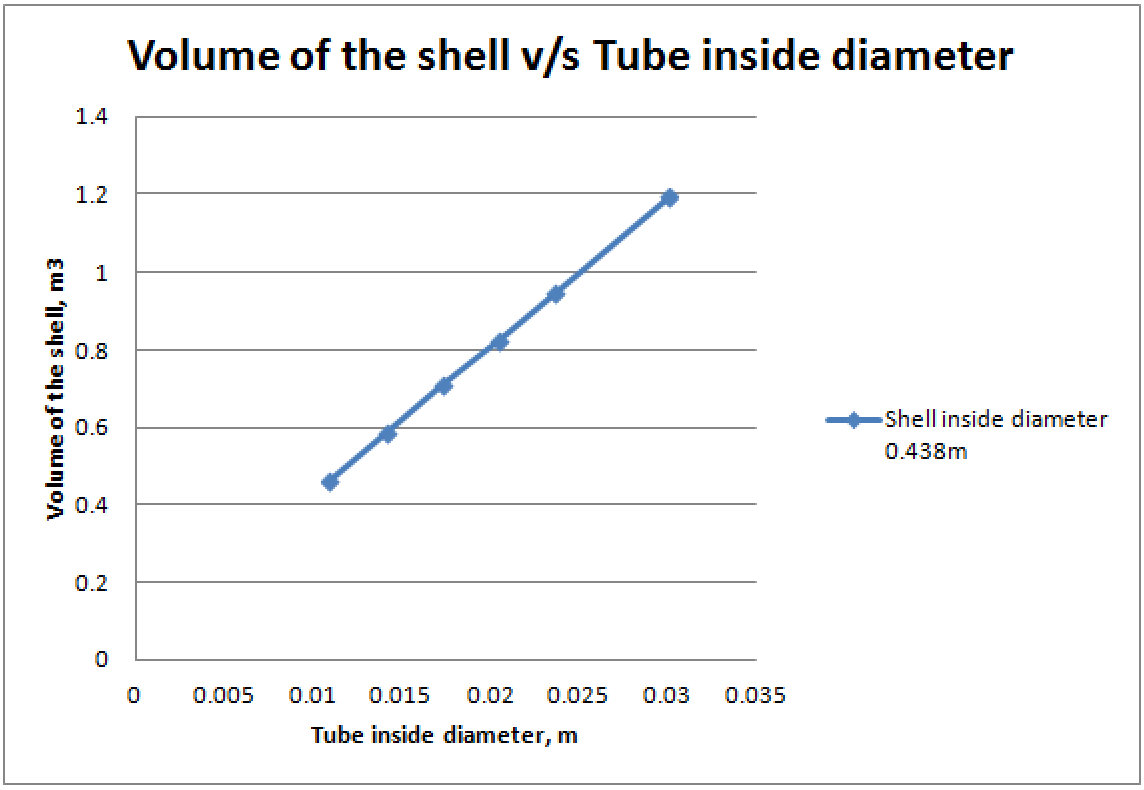

Figure 4.4: Volume of the shell vs Tube inside diameter

From the Fig.4.4we can conclude that with increasing the tube inside diameter, there is increase in the volume of the shell. This due to that volume is directly proportional to the length of the tube and also to the outside diameter of the tube.

4.1.1.4 variation of the number of tube

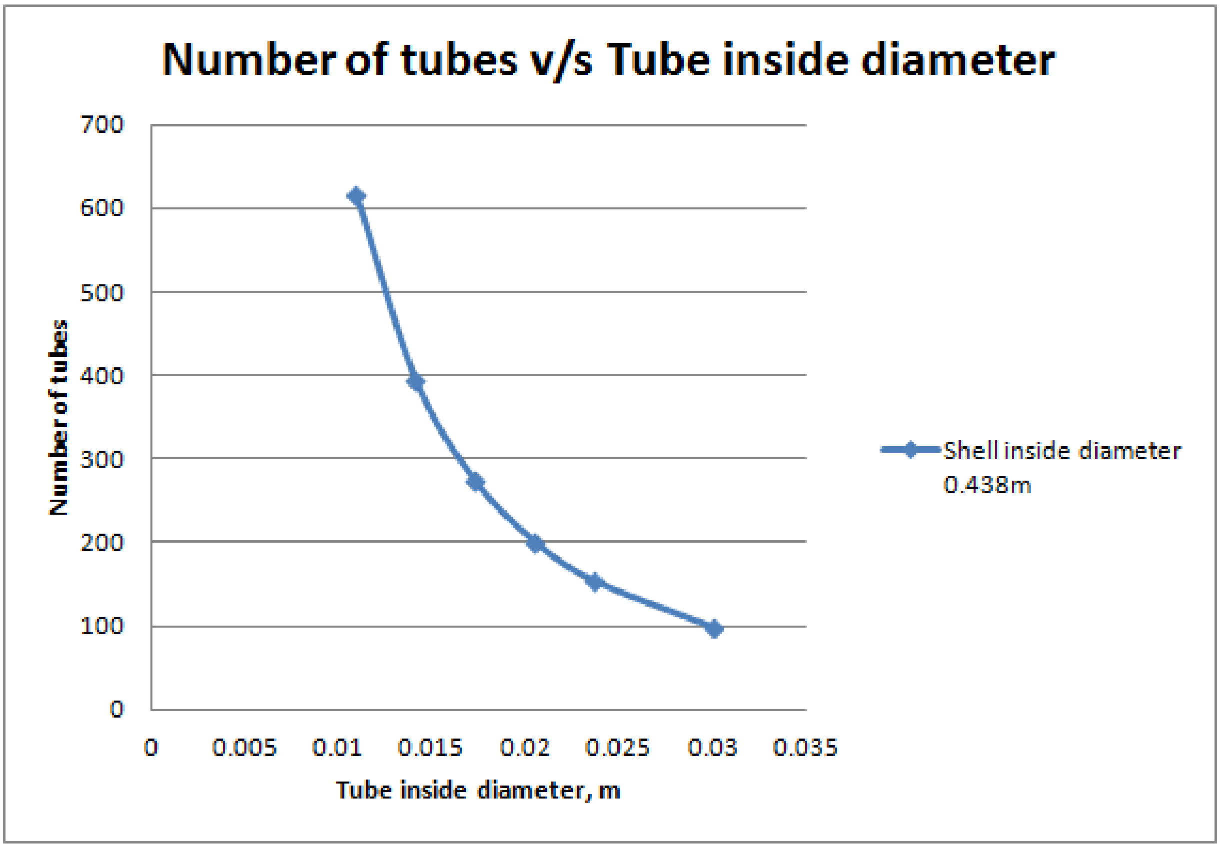

Figure 4.5: Number of tube v/s Tube inside diameter

As shown in the Fig.4.5 as the tube inside diameter increases, the number of tubes decreases for the constant shell inside diameter.

4.2 Variation of the different parameters of the heat exchanger with respect to change in inner diameter of the shell

Here the variation of various parameters of the heat exchanger is by varying the inside diameter of the shell for achieving the heat duty and pressure drop.

4.2.0.5 Variation of tube side pressure drop with the change in shell inside diameter

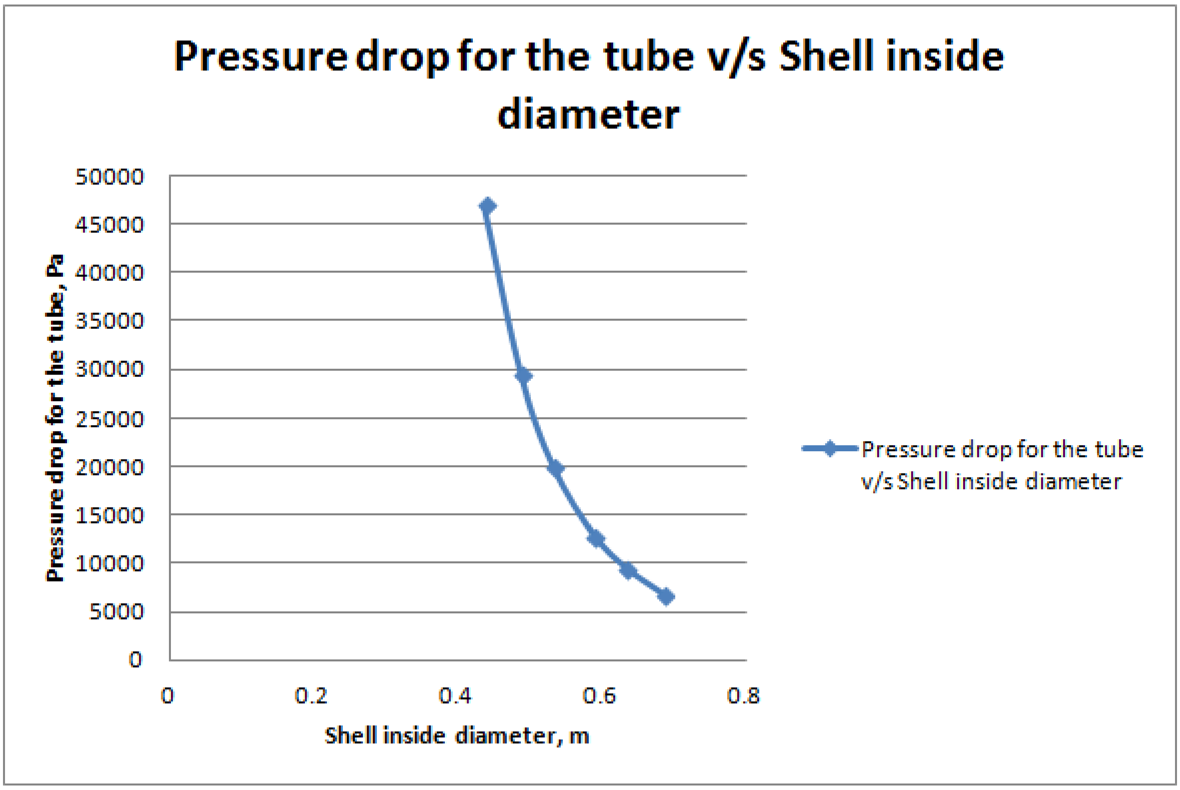

Figure 4.6: Pressure drop for the tube vs Shell inside diameter

As shown in Fig.4.6 pressure drop inside the tube decreases as the inner diameter of the shell increases. This happens because of the fact that with increasing the shell inside diameter, number of tube increases, so that the velocity in the tube decreases. So the pressure drop decreases as the inside diameter of the shell increases. From above graph it follows that after sometimes the decrease in the pressure drop is small with the variation in the inside diameter of the shell. This may helpful in the final optimization of the heat exchanger.

4.2.0.6 Variation of the pressure drop inside the shell

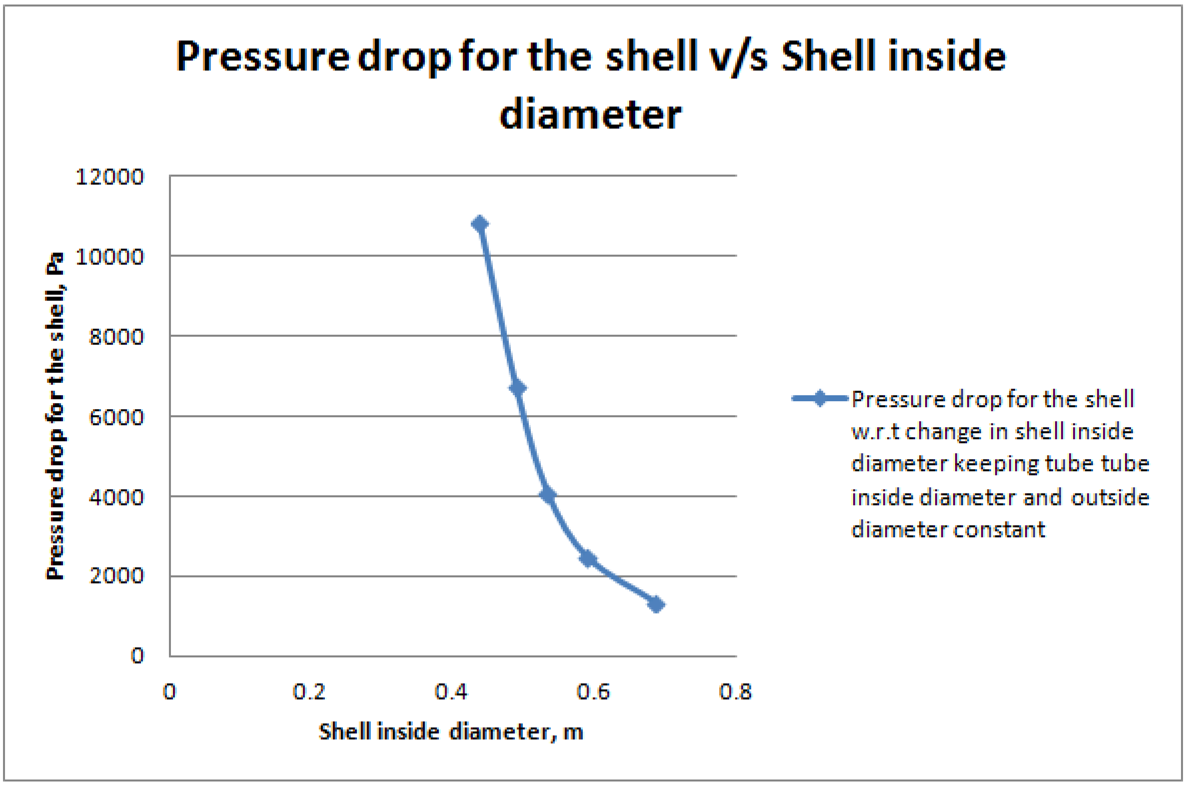

Figure 4.7: Pressure drop for the shell vs Shell inside diameter

Fig.4.7 the variation of the pressure drop in the shell side flow with the changing the inner diameter of the shell. As shown from the graph, as the inner diameter of the shell increases, pressure drop inside the shell side flow decreases. This happen due to the fact that as the inside diameter of the shell increases, the number of tubes increases and the shell side shell side velocity decreases, So Reynold number for the shell side flow also decreases. Because of this, the shell side pressure drop decreases as the inner diameter of the shell increases. In the graph pressure drop for the certain diameter does not decreases drastically.

4.2.0.7 Variation of length of the tube

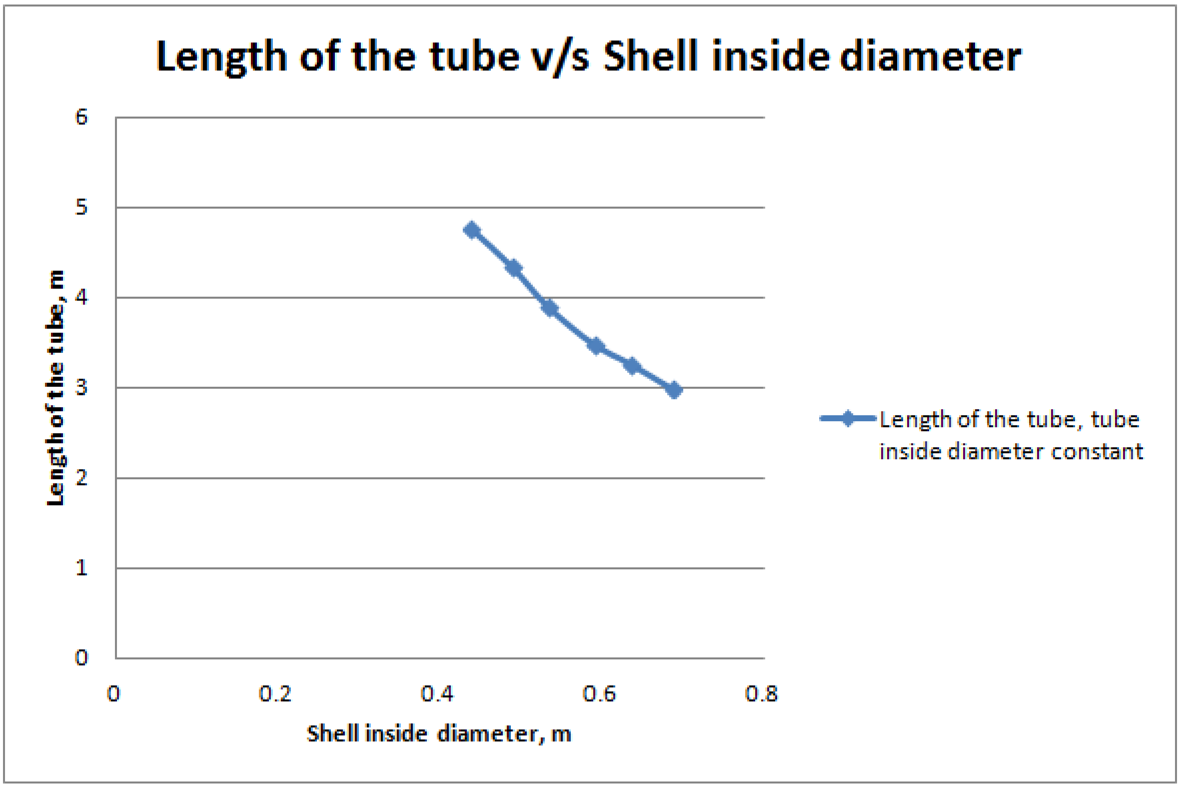

Figure 4.8: Length of the tube vs Shell inside diameter

From the Fig.4.8 it is clear that as the inside diameter of the shell increases, as length of the tube decrease, because the surface area available for the flow increases.

4.2.0.8 Variation of the volume of shell

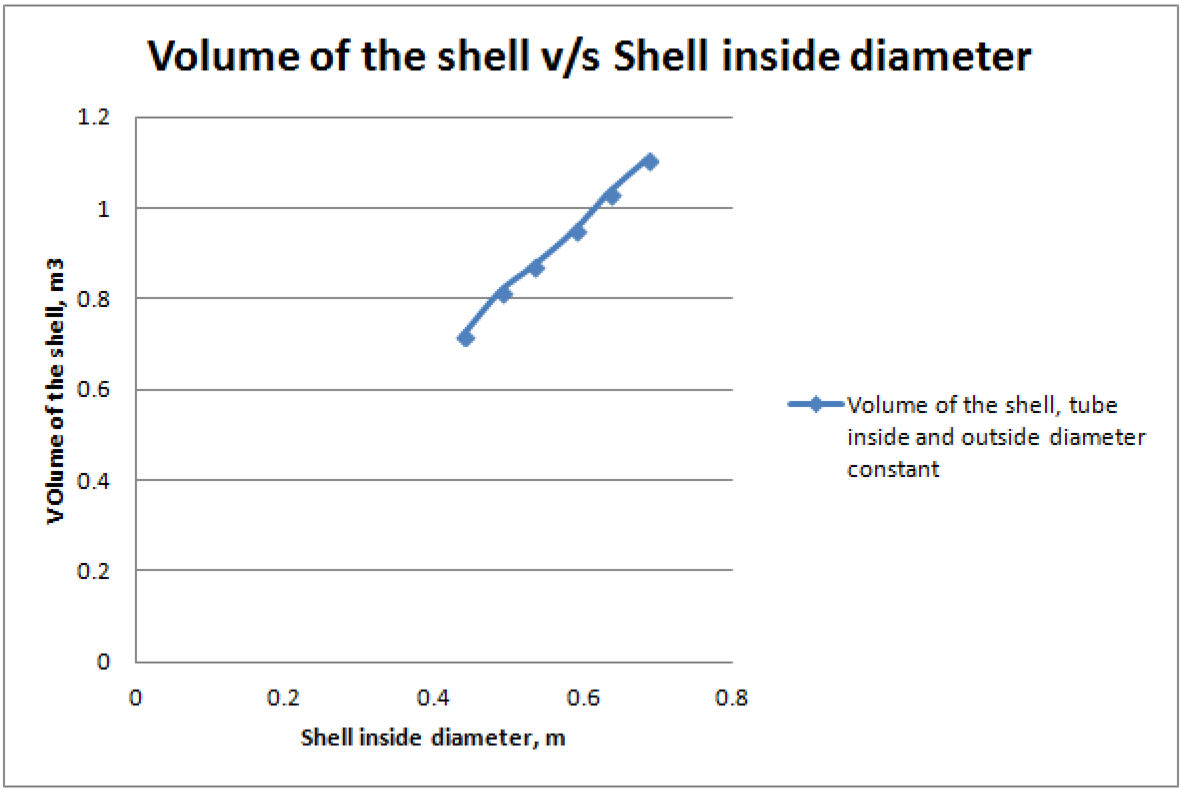

Figure 4.9: Volume of the shell vs Shell inside diameter

As shown in the Fig.4.9 as the shell inner diameter increases, the volume of the shell also increases. This is obvious because of the volume is directly depend upon the diameter of the shell and also the length of heat exchanger. But the increase in the volume of the shell due to the increases in the shell inner diameter is more than the increase in volume of the shell due to the decrease in the length of the tube.

4.3 Variation of Number of Tubes for different shell diameter with the change in tube inside diameter

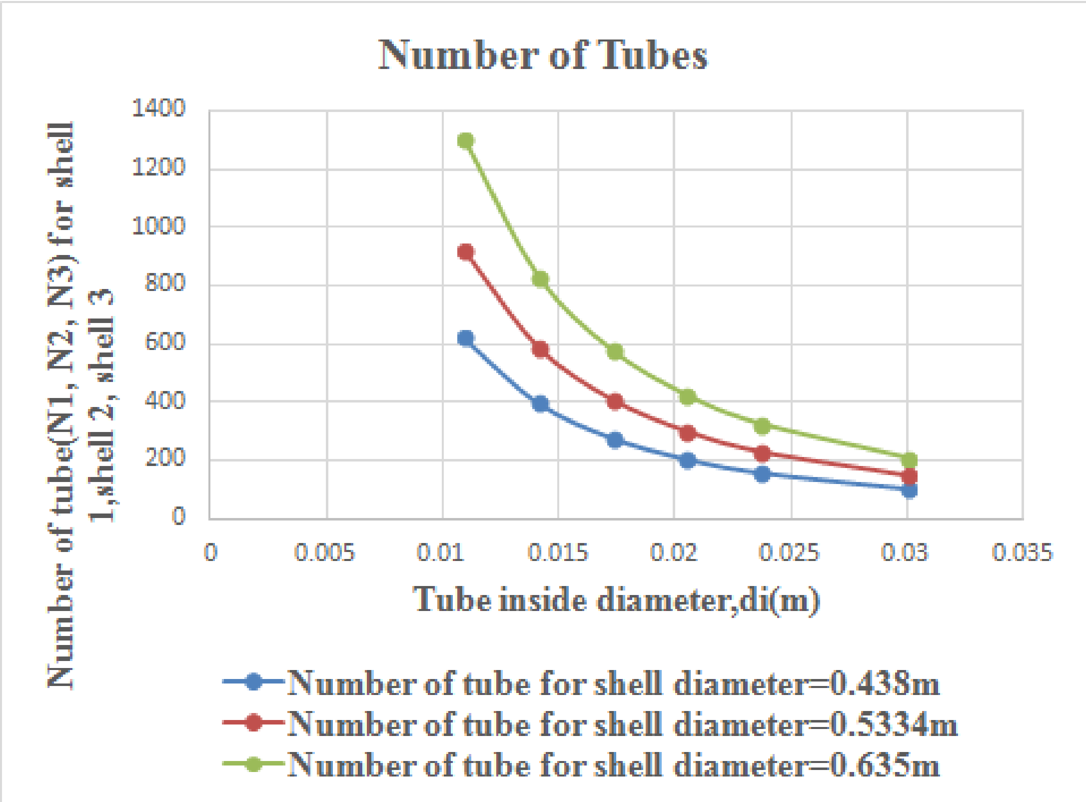

Figure 4.10: Variation of Number of Tubes for different shell inside diameter with changing tube inside diameter

As shown in the Fig.4.10 as the tube inside diameter increases, number of tubes for the different shell inside diameter also changes. Also from the Fig.4.10graph it is clear that as the tube inside diameter increases, the number of tube for the respective shell inside diameter decreases, it is because of the fact that for the constant shell inside diameter tube inside diameter increases, number of tube must have to be decreased.

From the graph, highest number of tubes is for the shell inside diameter of 0.635 m and lowest is for the shell inside diameter of 0.438 m.

4.4 Variation of pressure drop for the tube and for the shell

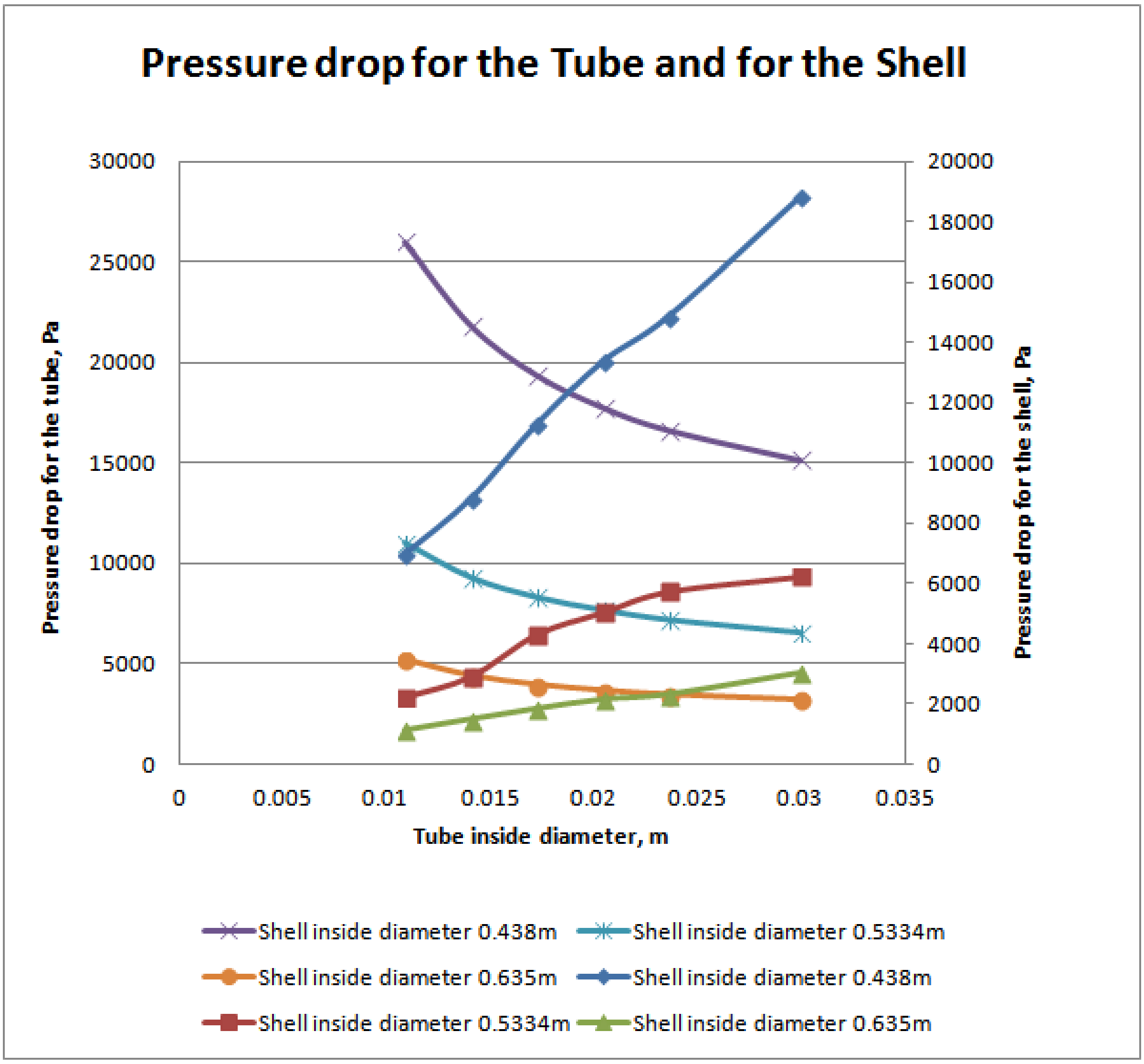

Figure 4.11: Pressure drop for the tube and pressure drop for the shell for w.r.t change in tube inside diameter for the different shell

As shown in the graph, as the tube inside diameter increases, pressure drop for the tube decreases, and pressure drop for the shell increases. As the pressure drop limit for the cold fluid which is water is 0.2 bar and the pressure drop for the hot fluid which is oil is 0.1 bar. And the water is inside the tube so that the pressure drop limit for the tube side fluid flow is 0.2 bar and oil is flowing inside the shell, so that the pressure drop limit for shell side fluid flow is 0.1 bar. It is clear from the graph that the optimum point is having shell inside diameter 0.5334 $m$and tube inside diameter is of 0.020447 m and outside diameter of the tube is of 0.022225 m.

4.5 Pressure drop for the tube and Length of the tube with respect to change in tube inside diameter

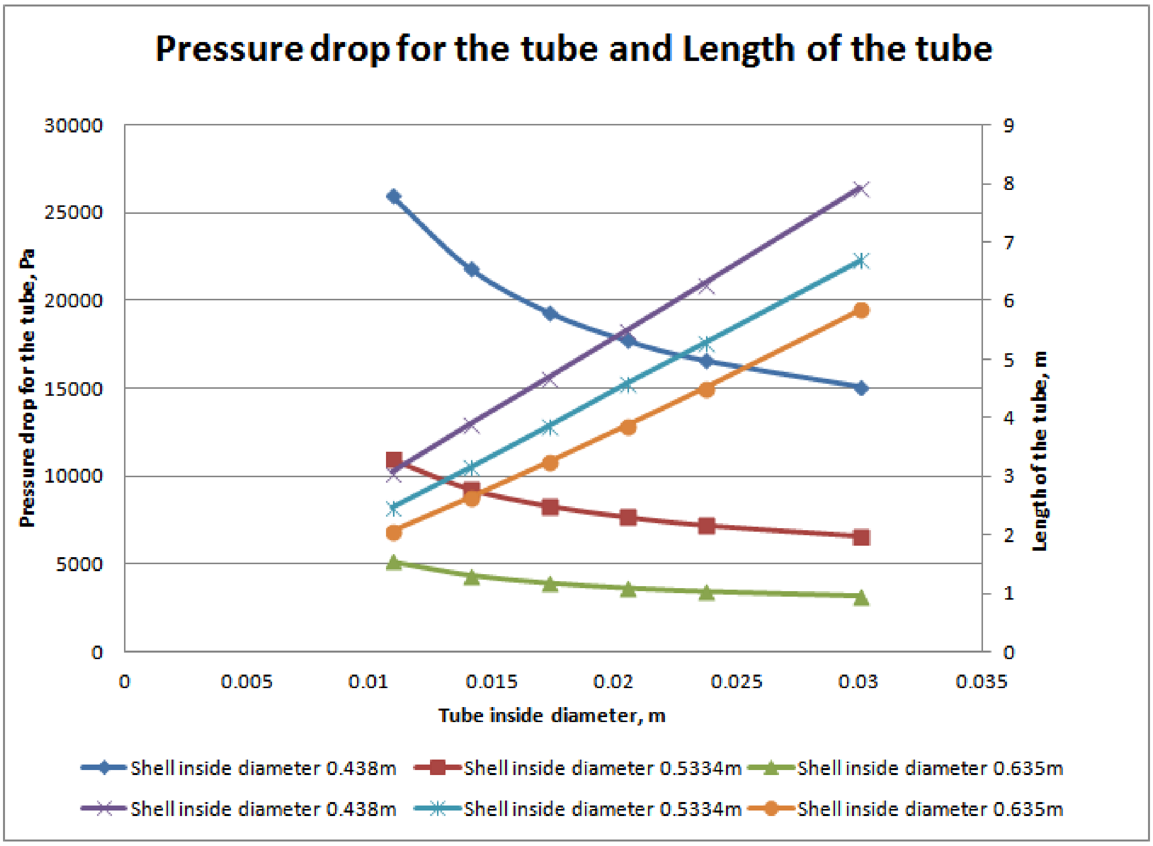

Figure 4.12: Pressure drop for the tube and Length of the tube with respect to change in Tube inside diameter

4.6 Longitudinal Fin

Parameters like fin heigth, fin thickness, fin density i.e., fins per tube are to be changed and optimum would be selected.

4.6.1 Optimization for the fin height

Length of the tube

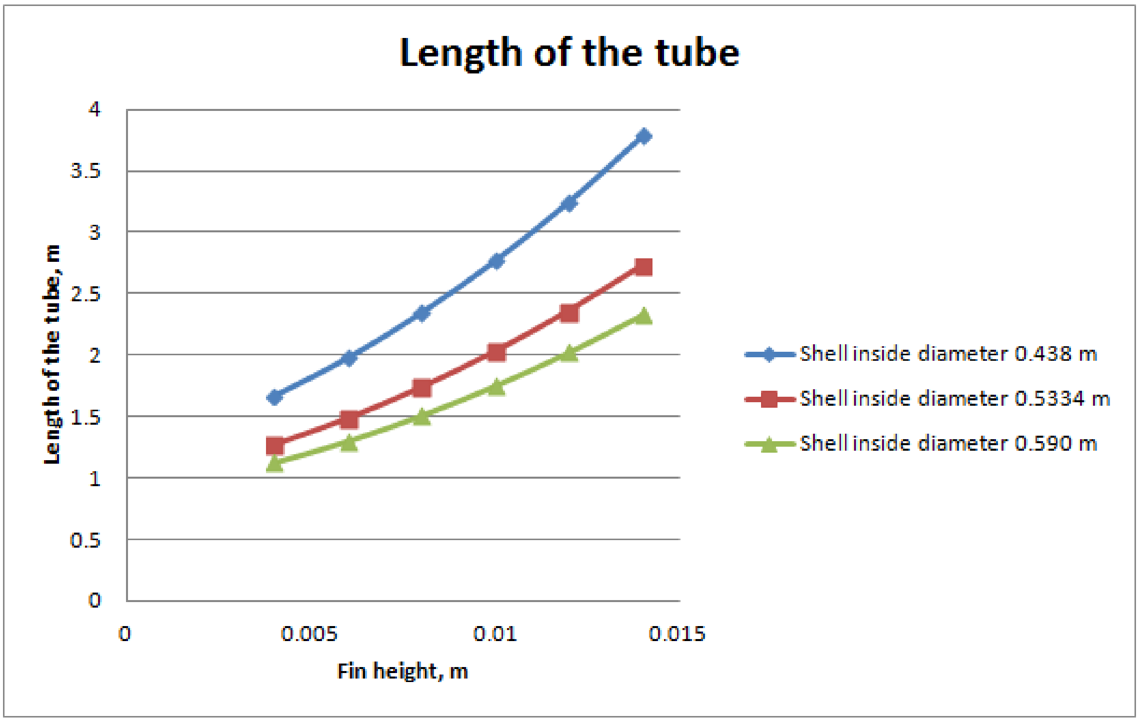

Figure 4.13: Length of the tube w.r.t change in Fin height keeping shell inside diameter constant

As shown in the Fig.4.13, as the fin height for the constant shell inside diameter increasing, the length of the tube increases, the number of tube decreases to prevent from the intersection of the fin to the next fin. Velocity of the fluid inside the tube increases. $U_i A_i$ per meter decreases so the length of the tube increases. This is happen because of the decrease in the efficiency of the fin with the increase in height of the fin.

Pressure drop for the tube with respect to change in fin height

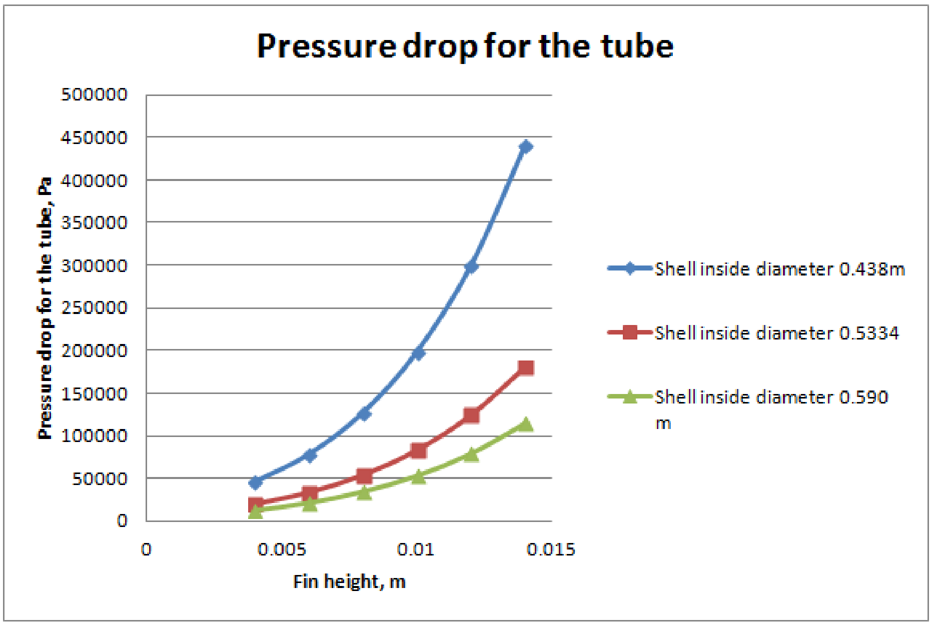

Figure 4.14: Pressure drop for the tube with respect to change in Fin height

This is due to the increase in the length of the tube as the fin height is increases, and the pressure drop for the tube is proportional to the length of the tube of the heat exchanger.

Pressure drop for the shell with respect to change in fin height

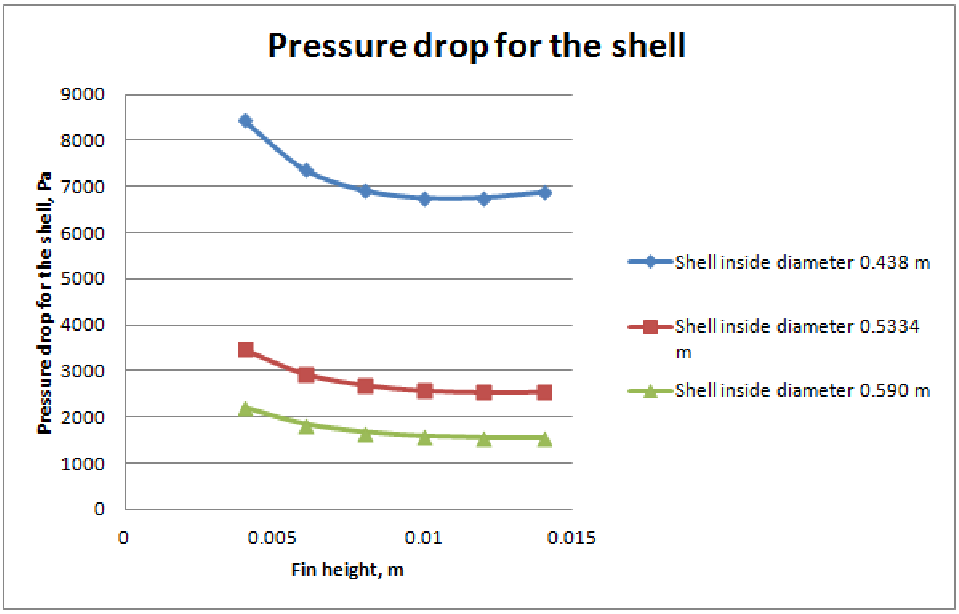

Figure 4.15: Pressure drop for the shell w.r.t change in Fin height, keeping shell inside and tube inside diameter constant

As shown in the Fig.4.15 , as the fin height increases, the free flow area available for the shell side flow decreases as the other parameters of the fin like fin density and fin thickness kept constant. So the pressure drop for shell side flow should have to be decrease.

From above Fig.4.15, it is concluded that the fin height of 0.004 m is optimum.

4.6.2 Optimization for the Fin Thickness

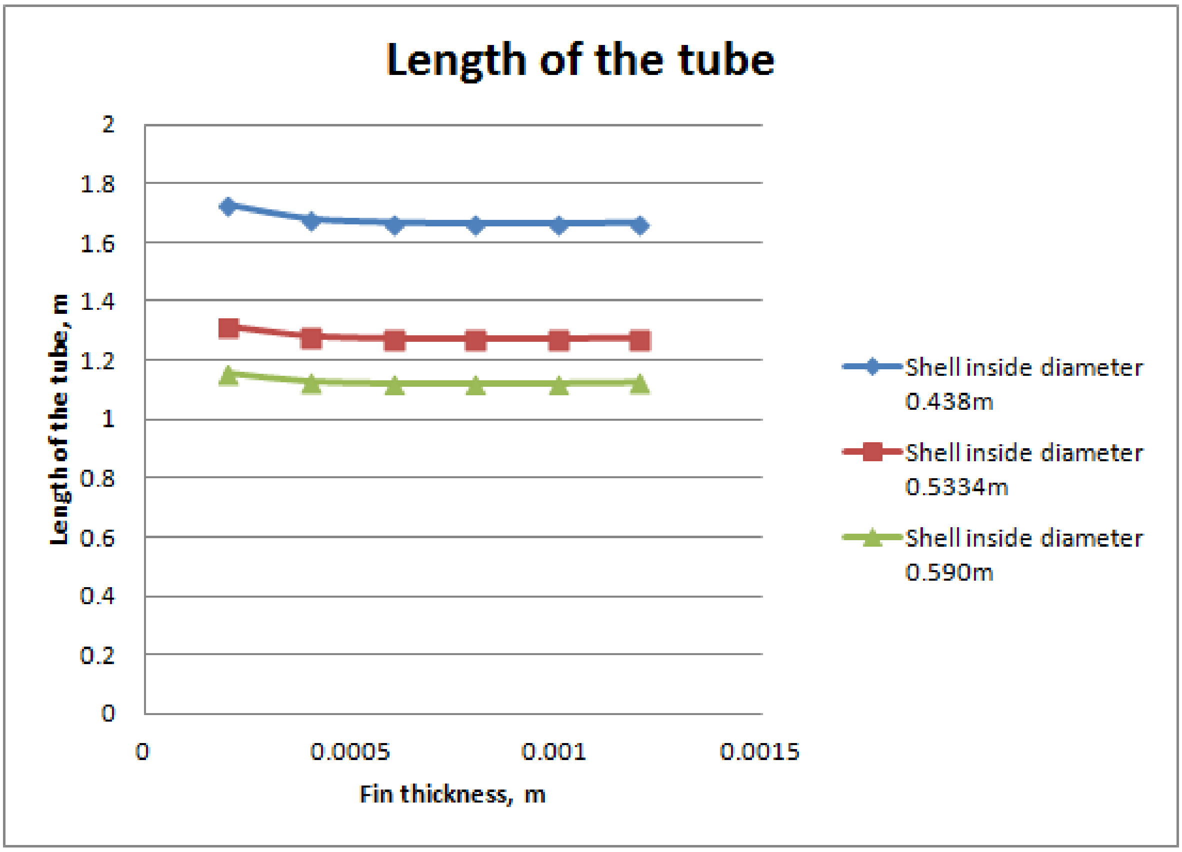

Figure 4.16: Length of the tube w.r.t. change in Fin thickness by keeping shell inside and tube inside diameter constant

From the Fig.4.16, as the fin thickness increases and fin density and fin height kept constant, than the total effective surface area for the fluid flow increases. So the length of tube and length of the shell decreases as the fin thickness inncreases for the constant tube inside and outside diameter.

Pressure drop for the tube with respect to change in fin thickness

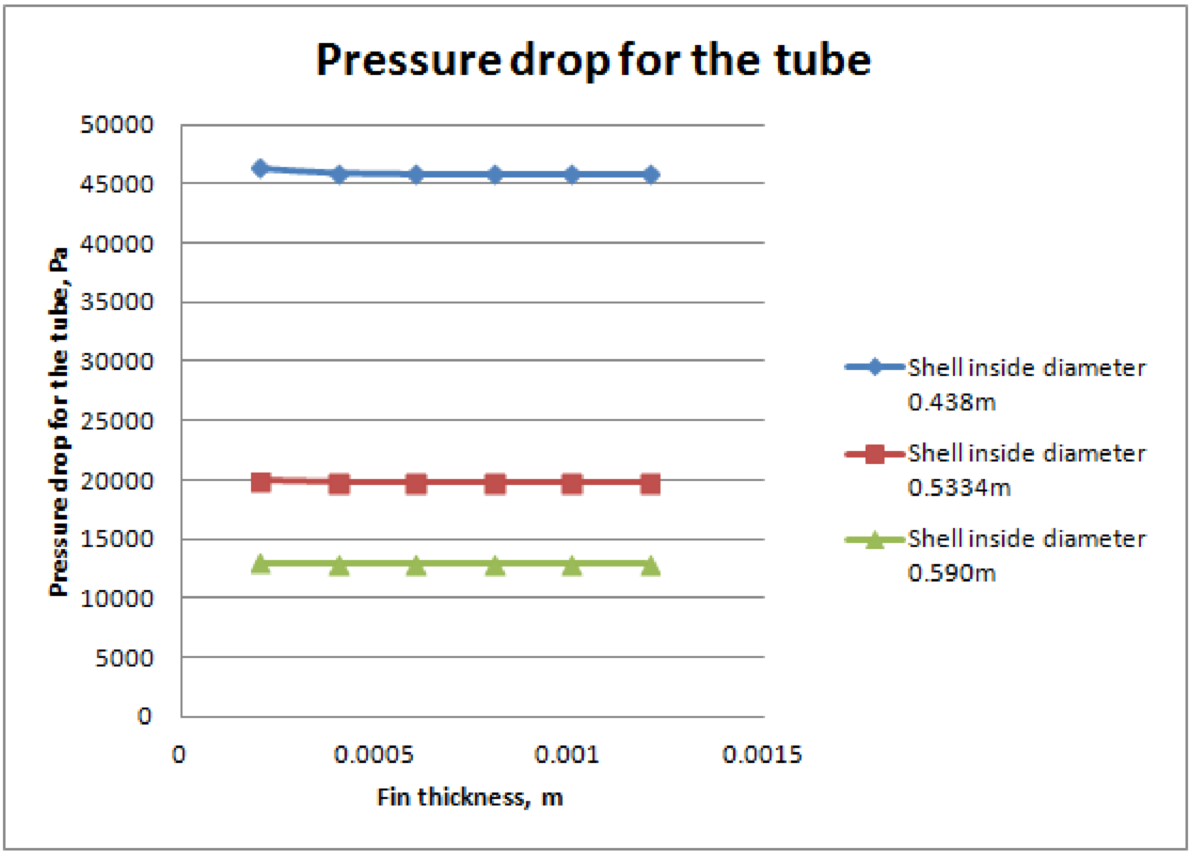

Figure 4.17: Pressure drop for the tube with respect to change in Fin Thickness by keeping shell inside diameter and tube inside diameter constant

As shown in the Fig.4.17, as the fin thickness increases for the constant fin density and constant fin height, the pressure drop for the tube should be decreases, this is happen due to the decrease in the length of the tube as the fin thickness increases. And pressure drop for the tube is directly proportional to the length of the tube.

Pressure drop for the shell with respect to change in fin thickness

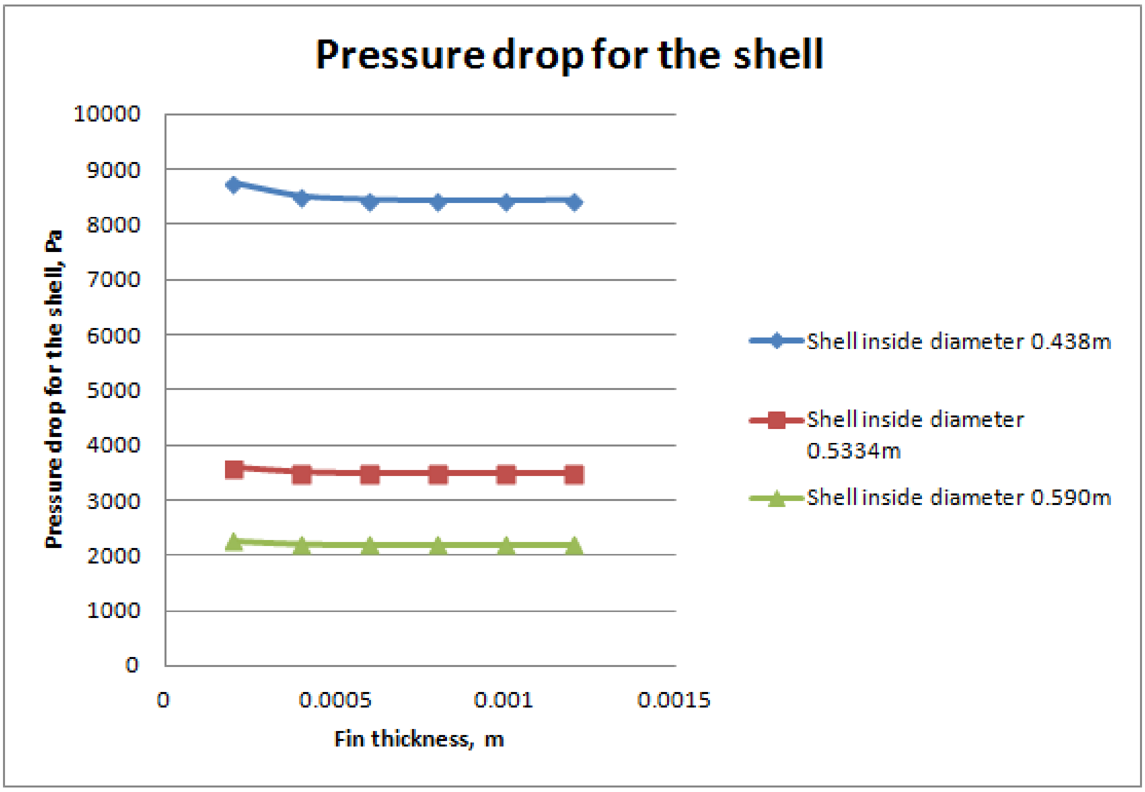

Figure 4.18: Pressure drop for the shell w.r.t change in fin thickness by keeping shell inside and tube inside diameter constant

As shown in the graph, as the fin thickness increases, the free flow area for shell side flow decreases. So pressure drop for the shell decreases.

From the above graph, optimum fin thickness is $0.0012\,m$.

4.6.3 Optimization for the fin density (Number of fins per tube)

Length of the tube with respect to change in fin density by keeping shell inside and tube inside diameter constant

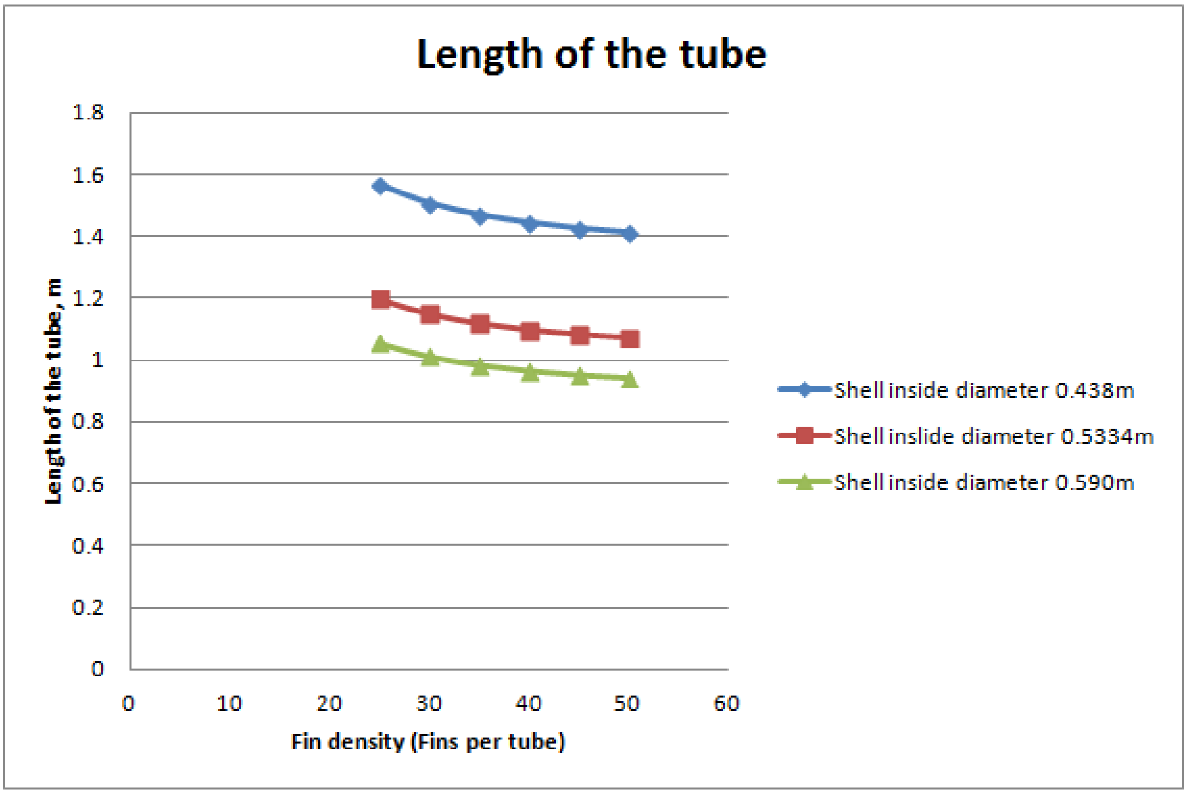

Figure 4.19: Length of the tube with respect to change in the Fin density by keeping shell inside and tube inside diameter constant

As shown in the Fig.4.19, as the fin density increases, kepping fin thickness and fin height constant, the length of the tube decreases. This is due to the fact that as the fin density increases, total effective surface area for the fluid to transfer heat is increases. So length of the tube decreases for the constant shell inside diameter.

Pressure drop for the tube with respect to change in the fin density, keeping shell inside and tube inside diameter constant

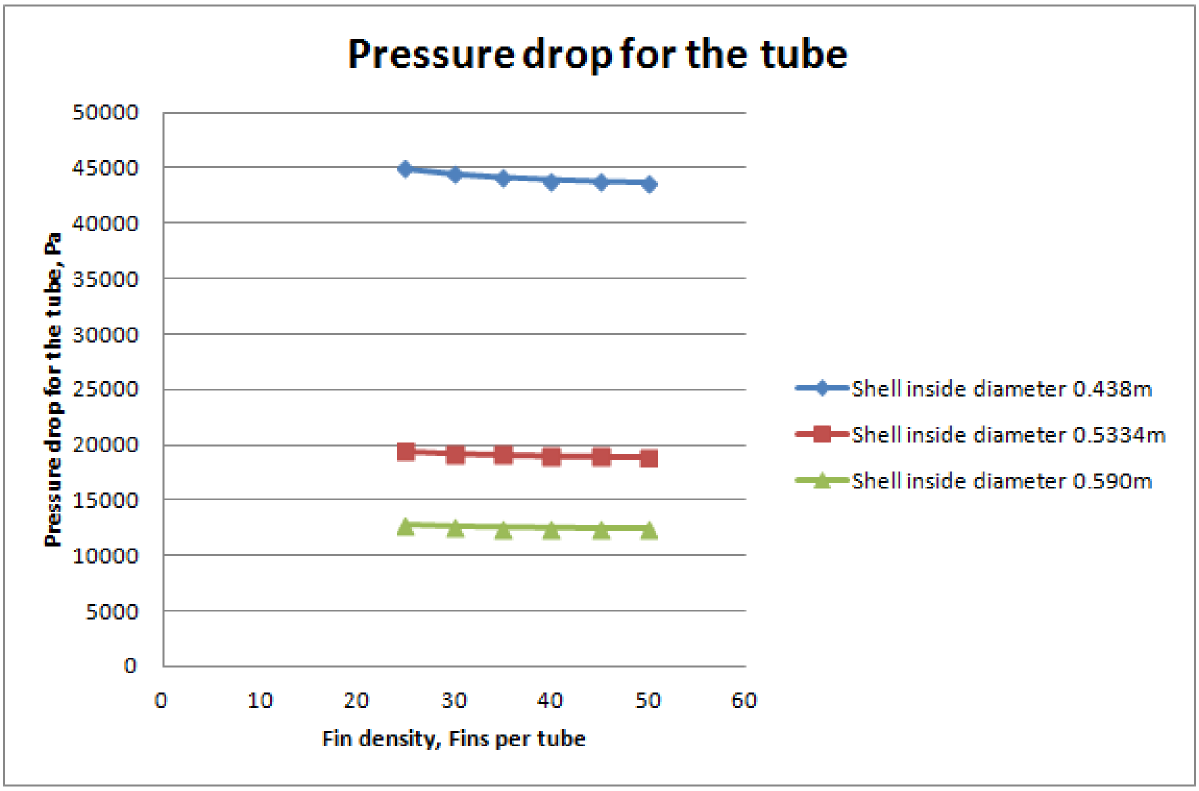

Figure 4.20: Pressure drop for the tube w.r.t change in the fin density, keeping shell inside diameter and tube inside diameter constant

As the fin density increases, length of tube decreases. And tube side pressure drop is proportional to the length of the tube.

Pressure drop for the shell with respect to change in fin density, keeping inside diameter of the shell and inside diameter of the tube constant

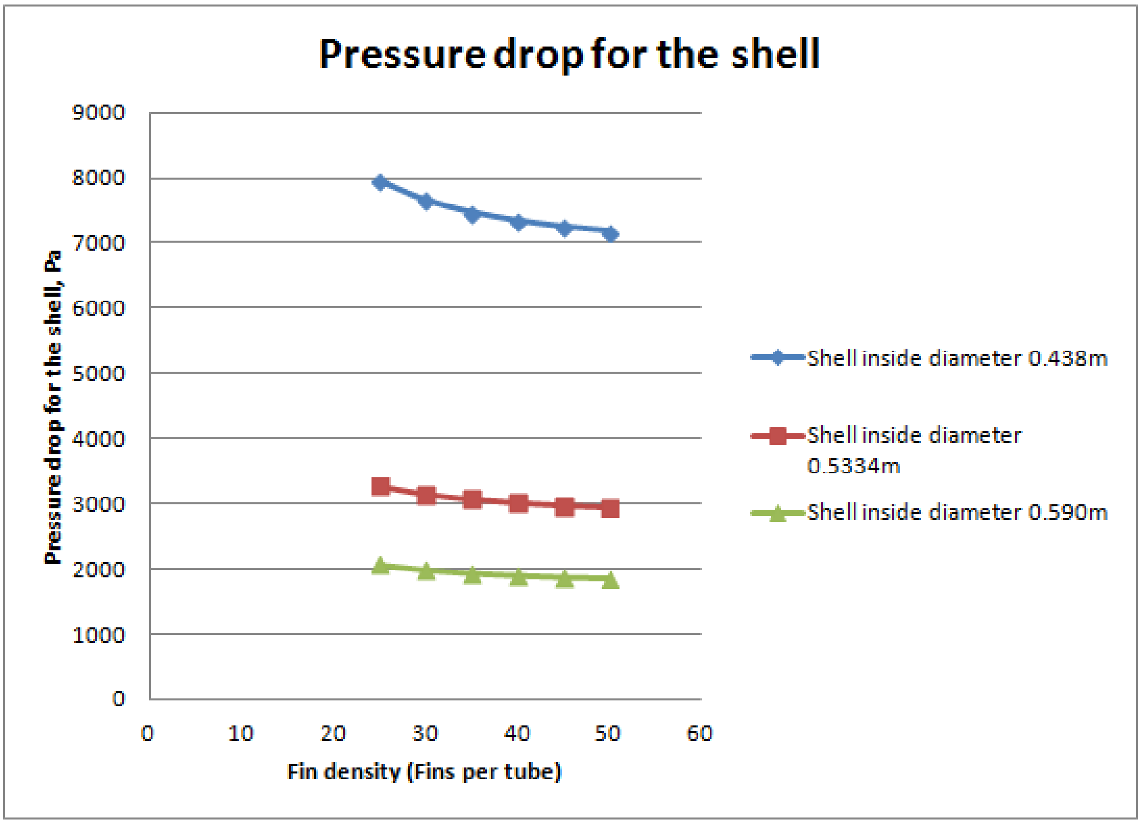

Figure 4.21: Pressure drop for the shell w.r.t change in fin density, keeping shell inside diamete and tube inside diameter constant

As the fin density increases for the constant fin density and constant fin thickness, free flow area for the shell decreases. So pressure drop for shell side flow is decreases.

From the above graph, the optimum fin density is 50.

4.7 Radial Fin

4.7.1 Optimization for the fin thickness for different fin density

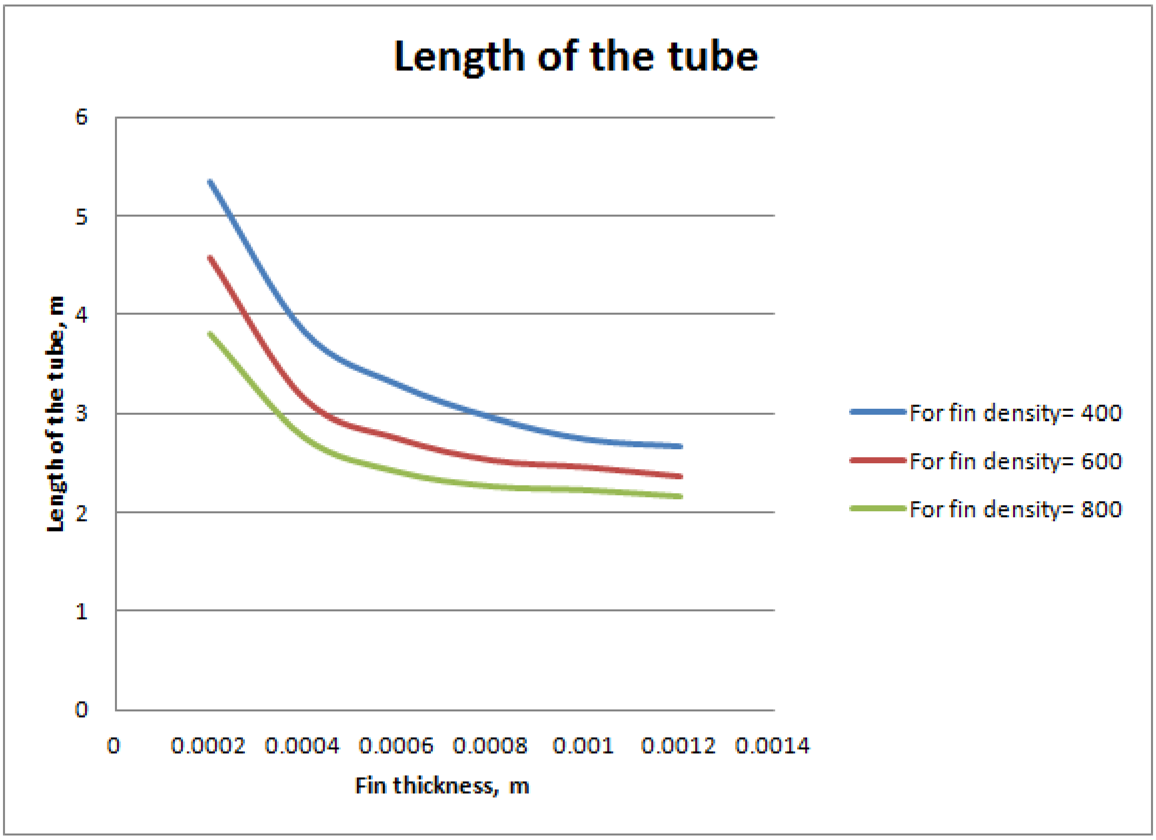

Figure 4.22: Length of the tube with respect to change in fin thickness, keeping shell inside diameter for different fin density

As shown in the Fig. 4.22, as the fin thickness increases for the constant fin density, total effective surface area for the fluid to transfer the heat from one fluid to another fluid increased, so the length of the tube should be decreased.

Pressure drop for the tube with respect to change in tube inside diameter, keeping shell inside diameter for different fin density

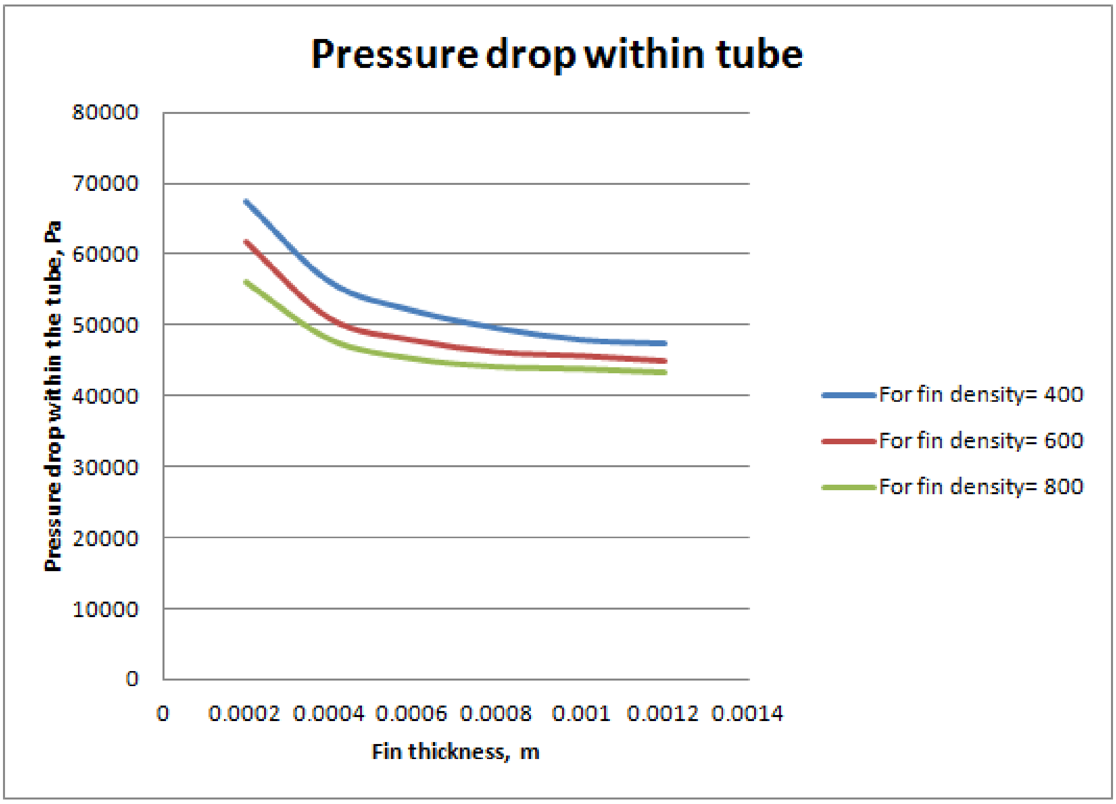

Figure 4.23: Pressure drop for the tube with respect to change in fin thickness, keeping shell inside diameter constant for different fin density

As the length of the tube decreases, as the fin thickness increases for the constant fin density, the tube side pressure drop should be decreased, because the tube side pressure drop directly proportional to the length of the tube.

Pressure drop for the shell with respect to change in fin thickness, keeping shell inside diameter constant for different fin density

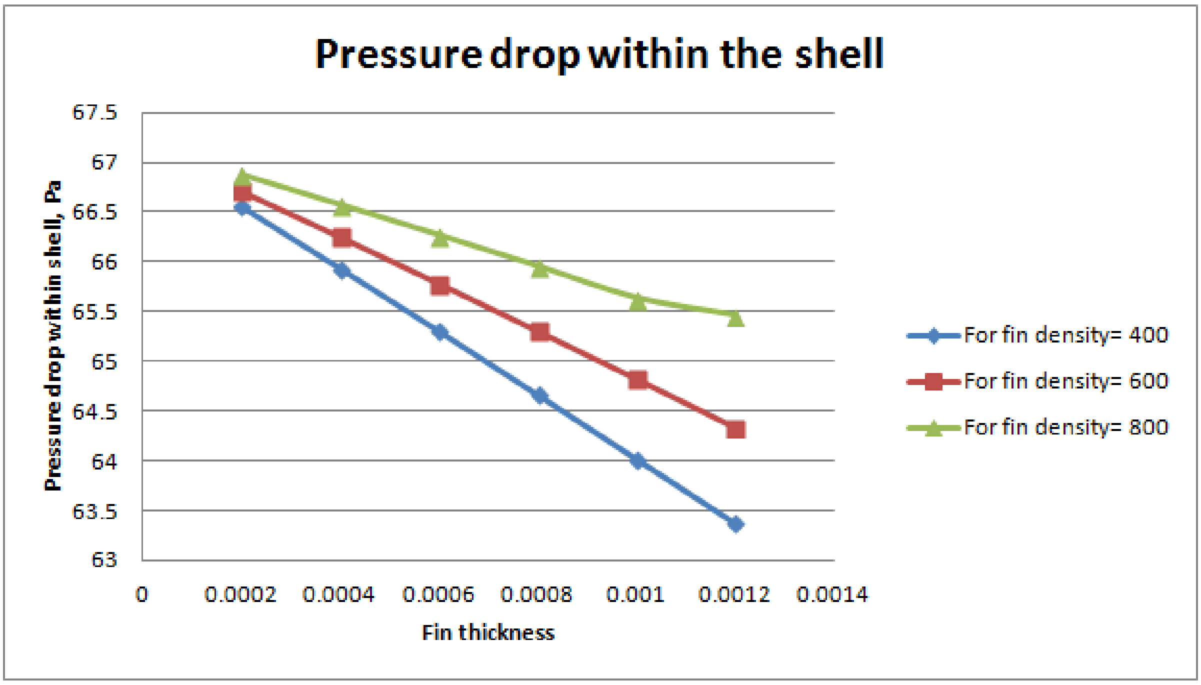

Figure 4.24: Pressure drop for the shell with respect to change in fin thickness, keeping shell inside diameter constant for different fin density

As shown in the Fig.4.24as the fin thickness increases, free flow area for the shell side flow decreases, and because of that the pressure drop for the shell decreases.

From the Fig.4.24, it is concluded that the optimum fin thickness is 0.001 m and optimum fin density is800.

4.7.2 Optimization for the fin height

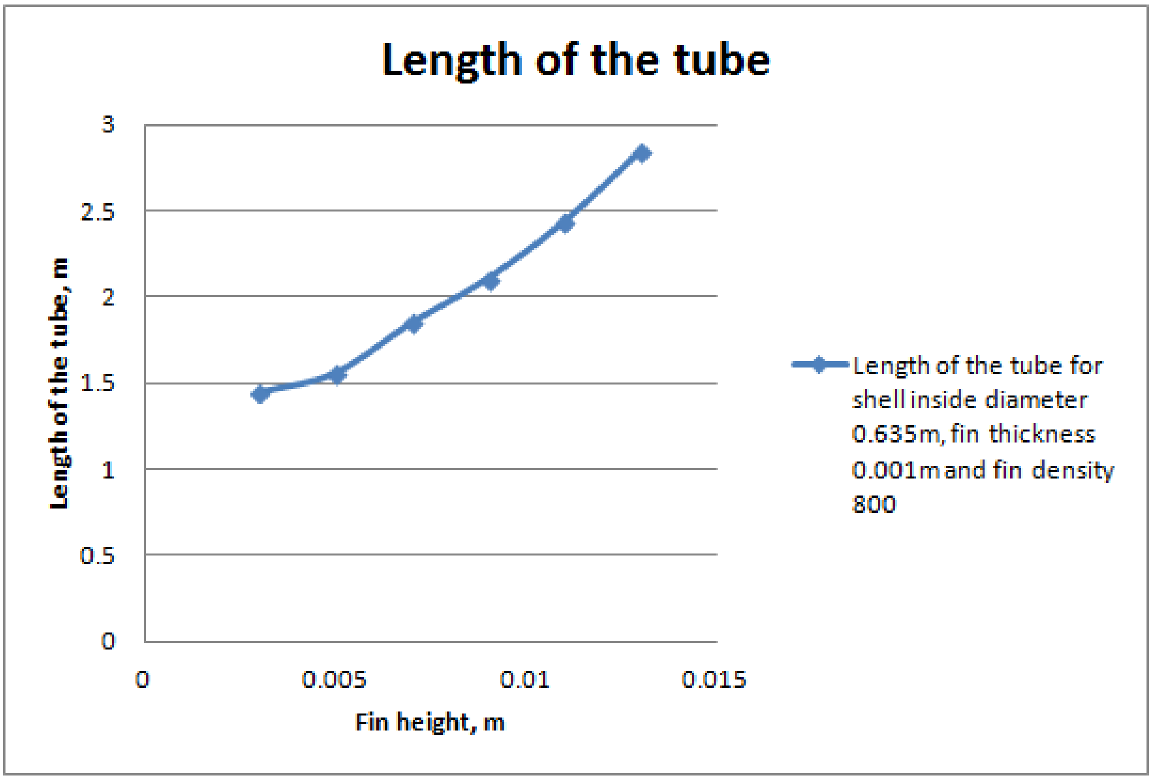

Figure 4.25: Length of the tube for constant fin thickness of 0.001m and constant fin density of 800 for constant shell inside diameter

As fin height increases for the constant fin thickness and constant fin density, length of the tube increases. This is because of the decrease in the number of tube.

Pressure drop for the tube w.r.t. change in fin height for constant fin thickness and fin density and shell inside diameter

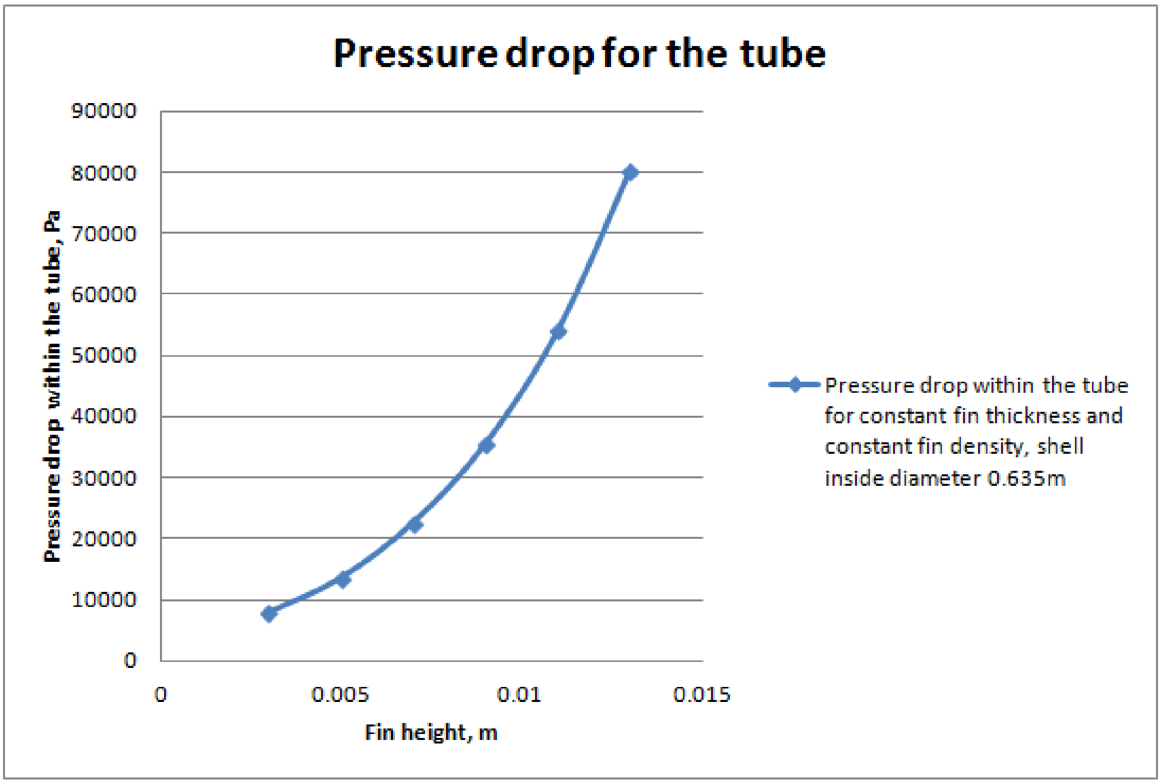

Figure 4.26: Pressure drop for the tube w.r.t. change in fin height for constant fin thickness and fin density and shell inside diameter

As the fin height increases for the constant fin thickness and constant fin density, the tube side pressure drop increased. And this is because of decrease in number of tubes and because of that the length of tube decreased. And tube side pressure drop is direclty proportional to the length of the tube.

Pressure drop for the tube at fin height of 0.003 m is minimum, So from the above graph, it is concluded that the optimum fin height is of 0.003 m.

4.8 Final Results

4.8.1 Optimum result without fin (Validation with HTRI Result)

Table 4.1: Optimum result shell and tube heat exchanger without fin and validation with HTRI result

Sr. No	Parameters	Analytical Value	HTRI value	% variation
1	Shell inside diameter (m)	0.5334	0.5334	-
2	Tube inside diameter (m)	0.020447	0.020447	-
3	Tube outside diameter (m)	0.022225	0.022225	-
4	Number of tubes	300	330	9.09
5	Lnegth of the tube (m)	4.58	4.58	-
6	Volume of the shell (m^3)	1.023	1.023	-
7	Pressure drop for the tube (bar)	0.07698	0.09888	22
8	Pressure drop for the shell (bar)	0.05038	0.05775	12.76
9	Over design (%)	17	18.33	7.25

It was also tried for the copper tube material in place of stainless steel, and it is observed that the length of the tube is changed and is decreased from 4.58 $m$to 4.55 m and volume of the shell is decreased from 1.023 m^3 to 1.01 m^3.

4.8.2 Optimum result with Radial Fin

Radial fins are to be used to increase total effective surface area for heat transfer and because of this, the length of the tube enhance the total volume of the shell would be decreased.

Table 4.2: Optimum result with radial fin outside the tube of shell and tube heat exchanger

Sr. No.	Parameters	Value
1	Shell inside diameter(m)	0.635
2	Tube inside diameter(m)	0.018923
3	Tube outside diameter(m)	0.022225
4	Number of tubes	263
5	Fin thickness(m)	0.001
6	Fin height(m)	0.003
7	Fin density	800
8	Length of the tube(m)	1.45
9	Volume of the shell(m^3)	0.4589
10	Pressure drop for the tube (bar)	0.08074
11	Pressure drop for the shell (bar)	0.01207

4.8.3 Optimum result with Longitudinal fin

Longitudinal fin is also used in design of the shell and tube heat exchanger. In this design tube is designed with external fin.

Table 4.3: Optimum result with longitudinal fin outside the tube of the shell and tube type of heat exchanger

Sr. No.	Parameter	Value
1	Shell inside diameter(m)	0.59
2	Tube inside diameter (m)	0.018923
3	Tube outside diameter (m)	0.022225
4	Fin thickness (m)	0.0012
5	Fin height (m)	0.004
6	Fin density	50
7	Length of the tube(m)	0.94
8	Volume of the shell(m^3)	0.2568
9	Number of tubes	198
10	Pressure drop for the tube(bar)	0.123
11	Pressure drop for the shell(bar)	0.0186

If copper is used in place of the stainless steel with thermal conductivity of $387 W/mK$, then the length of the tube changes from $0.94m$ to $0.85m$. And volume of the shell decreases from $0.2568m^3$to $0.2341m^3$.

From the above result, it is clear that longtudinal fin is more effective than the redial fin. And in this case the fin is useful to make a more heat transfer.

4.9 Cost analysis of heat exchanger

The cost of the stainless steel is lesser than the cost of the copper. And for the copper the decrease in the length of tube enhence the length of the shell and volume of the shell is less. So material of the tube of the heat exchanger should be of the stainless steel. And the shell of the heat exchanger is of carbon steel.

4.9.1 Plain tube heat exchanger

Tube material is stainless steel 321

Shell material is carbon steel

Shell inside diameter: $0.5334\,m$

Shell outside diameter: $0.5588\,m$

Shell thickness: $0.00553\,m$

Tube inside diameter: $0.020447\,m$

Tube outside diameter: $0.022225\,m$

Tube thickness: $0.000889\,m$

Number of tube: 300

Length of the tube:$4.58\,m$

Density of stainless steel 321: 8027 kg/m^3

Density of carbon steel: 7801 kg/m^3

Volume of tube material

$V_{tube} = \frac{\pi}{4}\left(d_o^2 - d_i^2\right) LN_t$

$= \frac{3.14}{4}\left(0.022225^2 - 0.020447^2\right) * 4.58 * 300$

$= 0.08083\,m^3$

Volume of the shell

$V_{shell} = \frac{\pi}{4}\left(D_o^2 - D_i^2\right) L$

$= \frac{3.14}{4}\left(0.5588^2 - 0.5334^2\right) * 4.58$

$= 0.09974\,m^3$

Material used in kg= $(0.09974 * 8027) + (0.0808 * 7801) = 1430.93\,kg$

Capital cost= $1430.93 * 400 = 5.72 * 10^5\ Rs$

Power= $\dfrac{\Delta P * m}{\eta_{pump*\rho}}$

$= \dfrac{7698 * 43.041}{0.85 * 1000}$

$= 389\,W$

$Power\ *\ time\ = .389 * 24 * 365 = 3414.64\,KWh$

$Operating\,cost\ = 3414.64 * 5 = 17073.2\,Rs/KWh$

Considering discount rate= 20%

Payback period N=20 year

$CRF = \left[\frac{d(1+d)^n}{(1+d)^n-1}\right]$

$= \left[\frac{0.2*(1+0.2)^{20}}{(1+0.2)^{20}-1}\right]$

$= 0.2053$

Anual life cycle cost

$ALCC = (Co * CRF) + AC$

$ALCC = (5.72 * 10^5 * 0.2053) + 17073.2$

$ALCC = 13.4558 * 10^4 Rs$

4.9.2 Circular fin

Volume of the tube material

$V_{tube} = \frac{\pi}{4}\left(d_o^2 - d_i^2\right) LN_t + \left[\frac{\pi}{4}\left(d_f^2 - d_o^2\right) * t_f\right] * \rho_f * L * N_t$

$= \frac{3.14}{4}\left(0.022225^2 - 0.018923^2\right)*1.45*263+\left[\frac{3.14}{4}\left(0.025225^2 - 0.022225^2\right) * 0.001\right]*$ $800 * 1.45 * 263$

$= 0.07476\,m^3$

Volume of the shell

$V_{shell} = \frac{\pi}{4}\left(D_o^2 - D_i^2\right) L$
$= \frac{3.14}{4}\left(0.660^2 - 0.635^2\right) * 1.45$
$= 0.03685\, m^3$

Material used in$kg = (0.0747 * 8027) + (0.03685 * 7801) = 887.08\, kg$

Capital cost$= 887.08 * 400 = 3.54 * 10^5\, Rs$

Power $= \dfrac{\Delta P * m}{\eta_{pump*\rho}}$
$= \frac{8074*43.021}{0.85*1000}$
$= 0.408\, KW$

Power * time$= 0.408 * 24 * 365 = 3579.76\, KWh$

Operating cost$= 3579.76 * 5 = 17898.82\, Rs/KWh$

$ALCC = (Co * CRF) + AC$

$ALCC = (3.54 * 10^5 * 0.2053) + 17898.82 = 9.057 * 10^4\, Rs$

4.9.3 Longitudinal fin

Volume of the tube material

$V_{tube} = \frac{\pi}{4}\left(d_o^2 - d_i^2\right) LN_t + h_f * L * t_f * \rho_f * N_t$
$= \frac{3.14}{4}\left(0.022225^2 - 0.018923^2\right) * 0.94 * 198 + (0.004 * 0.94 * 0.0012 * 50 * 198)$
$= 0.06452\, m^3$

Volume of the shell

$V_{shell} = \frac{\pi}{4}\left(D_o^2 - D_i^2\right) L$
$= \frac{3.14}{4}\left(0.6096^2 - 0.590^2\right) * 0.94$
$= 0.01734\, m^3$

Material used in$kg = (0.06452 * 8027) + (0.01734 * 7801) = 653.24\, kg$

Capital cost $= 653.24 * 400 = 2.61 * 10^5\, Rs$

Power$= \dfrac{\Delta P * m}{\eta_{pump*\rho}}$
$= \frac{12300*43.021}{0.85*1000}$
$= 0.6225\, KW$

Power*time$= 0.6225 * 24 * 365 = 5453.1\, KWh$

Operating cost$= 5453.1 * 5 = 27265.5\, Rs$

$ALCC = (Co * CRF) + AC$

$ALCC = (2.61 * 10^5 * 0.2053) + 27265.5$

$ALCC = 8.0848.8 * 10^4\, Rs$

So from the above analysis for the different heat exchanger it is clear that ALCC for the Longitudinal fin heat exchanger is lowest than the other type of heat exchanger. So one should have to go for this type of heat exchanger.

4.10 Summary: Thermal Design Optimisation

Based on the thermal design conducted, the following points are summarized:

(i) The optimum result without fin is obtained with the tube inside diameter of0.020447 m, tube outside diameter of 0.022225 m, and length of the tube 4.58 m.

(ii) The optimum result with radial fin is obtained with tube inside diameter of 0.018923 m, tube outside diameter 0.022225 m, length of the tube 1.45 m, fin thickness of 0.001 m, fin height of 0.003 m, fin density800. The pressure drop is observed to be within the limits.

(iii) The optimum result with longitudinal fin is obtained with tube inside diameter of 0.018923 m, tube outside diameter of 0.022225 m, length of the tube is 0.94 m, fin thickness 0.0012 m, fin height 0.004 m, fin density 50. The pressure drop is observed to be within limits.

(iv) TheAnnual Life Cycle Cost of longitudinal fin is minimum as compared to other types of heat exchange surfaces.

(v) The shell and tube heat exchager with longitudinal fin as heat exchange surface is most preferred for the application.

Chapter 5

Conclusion and Proposed work

5.1 Conclusion

The following conclusions can be drawn from the present study:

1. A shell and tube heat exchanger was designed to meet the heat duty involved for oil cooler in Liquid Helium Plant.

2. A program was prepared using Excel to optimise the design of the heat exchanger based on tube inside diameter, shell inside diameter, etc.

3. The analytical results for thermal design calculations are compared with the HTRI results and found to have satisfactory acceptance.

4. The optimum result without fin is obtained with a tube inside diameter of $0.020447\,m$, tube outside diameter of $0.022225\,m$, and length of the tube $4.58\,m$.

5. The optimum result with radial fin is obtained with a tube inside diameter of $0.018923\,m$, tube outside diameter $0.022225\,m$, length of the tube$1.45\,m$, fin thickness of $0.001\,m$, fin height of $0.003\,m$, fin density 800. The pressure drop observed to be within the limits.

6. The optimum result with longitudinal fin is obtained with a tube inside diameter of $0.018923\,m$, tube outside diameter of $0.022225\,m$, length of the tube is $0.94\,m$, fin thickness $0.0012\,m$, fin height $0.004\,m$, fin density 50.The pressure drop observed to be within limits.

7. TheAnnual Life Cycle Cost of longitudinal fin is minimum as compared to other types of heat exchange surfaces.

8. The shell and tube heat exchager with longitudinal fin as heat exchange surface is most preferred for the application.

5.2 Proposed work

The mechanical design of the heat exchanger and stress analysis may be undertaken.

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Appendix A

Sample Calculation for the Design of the Heat Exchanger for the Compressor After-Cooler of the Helium plant in Excel

Input data

- Heat duty-900KW
- Inlet temperature of hot fluid- 90 $^\circ C$
- Outlet temperature of hot fluid- 35 $^\circ C$
- Inlet temperature of the cold fluid- 20 $^\circ C$
- Outlet temperature of the cold fluid- 25 $^\circ$C
- Hot fluid- Breox oil(oil) cold fluid- Water
- Heat exchanger type- Shell and tube heat exchanger

Parameters	DESIGN		
T_{hi}	90	90	90
T_{ho}	35	35	35
T_{ci}	20	20	20
T_{co}	25	25	25
Q	900000	900000	900000
Cp_h	2006	2006	2006

Cp_c	4182	4182	4182
ρ_h	925	925	925
ρ_c	1000	1000	1000
k_h	0.146	0.146	0.146
k_c	0.6	0.6	0.6
Pr_h	88.96	88.96	88.96
Pr_c	6.97	6.97	6.97
μ_h	0.006475	0.006475	0.006475
μ_c	0.001	0.001	0.001
m_h	8.1573	8.1573	8.1573
m_c	43.0416	43.0416	43.0416
Di	0.635	0.635	0.635
di	0.014224	0.0049276	0.007058
do	0.01905	0.00635	0.009525
t	0.002413	0.0007112	0.0012446
CL =Tube layout constant	0.87	0.87	0.87
CTP =Tube count calculation constant	0.9	0.9	0.9
PR =Pitch ratio	1.25	1.25	1.25
$N_t = 0.785\left(\frac{CTP}{CL}\right)\frac{D_i^2}{(PR)^2 d_o^2}$	577.47	5197.24	2309.88
$F=$ Temperature correction factor(Taken from graph)	0.92	0.92	0.92
$\triangle T_{lmtd} = \dfrac{(T_{hi}-T_{ho})-(T_{ho}-T_{ci})}{ln\frac{(T_{hi}-T_{co})}{(T_{ho}-T_{ci})}}$	34.098	34.098	34.098
$\Delta Tcf = \Delta Tlmtd * F$	31.370	31.370	31.370
$Npass=$ Number of pass	2	2	2
$A_{tp} = \frac{\pi}{4}d_i^2\left(\frac{N_t}{N_{pass}}\right)$	0.0458	0.0495	0.0451
$u_{mt} = \frac{m_t}{\rho_t A_{tp}}$	0.938	0.868	0.953
$R_e = \frac{\rho_t u_{mt} d_i}{\mu_t}$	13350.4	4281.93	6726.30
$f = (1.58lnRe - 3.28)^{-2}$	0.00726	0.01013	0.00882
$\frac{f}{2}$	0.00363	0.00501	0.0044
$Nu_t = \frac{\left(\frac{f}{2}\right)(Re-1000)Pr_t}{1+12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)}$	103.3243089	34.15120375	54.44080885
$h_i = \frac{Nu_t k_c}{d_i}$	4358.44	4158.35	4628.00
B =Central baffle spacing = 0.5of the shell inside diameter	0.3175	0.3175	0.3175
$Pt=$ Pitch= $PR * do$	0.02381	0.00793	0.0119

C =Clearance= $Pt - do$	0.0047	0.0015	0.002381
$A_s = \frac{D_i CB}{P_t}$	0.0403	0.0403	0.0403
$Re_{shell} = \frac{d_o m_s}{\mu_s A_s}$	595.19	198.39	297.59
$a1$	0.593	0.593	0.593
$a2$	-0.477	-0.477	-0.477
$a3$	1.45	1.45	1.45
$a4$	0.519	0.519	0.519
$a = \frac{a_3}{1+0.14(Re_s)^{a_4}}$	0.2985	0.4559	0.3928
$J_i = a_1 \left(\frac{1.33}{\frac{Pt}{d_o}}\right)^a (Re_s)^{a_2}$,	0.02868	0.0489	0.0401
$\mu_{s,wall}$ =Viscosity at average wall temperature	0.00925	0.00925	0.00925
$h_{ideal} =$ $J_i C_{ps} \left(\frac{m_s}{A_s}\right) \left(\frac{k_s}{C_{ps}\mu_s}\right)^{\frac{2}{3}} \left(\frac{\mu_s}{\mu_{s,wall}}\right)^{0.14}$	555.59	947.50	777.83
Diameter of outer tube limit, D_{otl}	0.5937	0.5937	0.5937
$D_{ctl} = D_{otl} - d_o$	0.57465	0.58735	0.584175
Baffle cut= $0.25 * D_i$	0.15875	0.15875	0.15875
$\left(\frac{D_s - 2l_c}{D_{ctl}}\right)$	0.552	0.540	0.543
$\theta_{ctl} = 2cos^{-} \left(\frac{D_s - 2l_c}{D_{ctl}}\right)$	1.970	1.999	1.992
$F_w = \frac{\theta_{ctl}}{2\pi} - \frac{sin\theta_{ctl}}{2\pi}$	0.167	0.173	0.171
$F_c = 1 - 2F_w$	0.665	0.652	0.656
$J_c = 0.55 + 0.72F_c$	1.0292	1.020	1.022
$1 - \frac{2l_c}{D_s}$	0.5	0.5	0.5
$\theta_b = 2cos^{-} \left(1 - \frac{2l_c}{D_s}\right)$	2.094	2.094	2.094
Diametral shell baffle clearance,δ_{sb}	0.0044	0.0044	0.0044
$A_{o,sb} = \pi D_s \left(\frac{\delta_{sb}}{2}\right) \left(1 - \frac{\theta_b}{2\pi}\right)$	0.0029	0.0029	0.0029
Diametral clearance between tube and the baffle,δ_{tb}	0.00079	0.00079	0.00079
$A_{o,tb} = \frac{\pi D_s \delta_{tb} N_t (1-F_w)}{2}$	0.011	0.033	0.022
Transverse pitch, X_t for 30 $^\circ$ layout	0.02381	0.00793	0.0119
Longitudinal pitch, $X_l = \sqrt{\frac{3}{2}} P_t$	0.0206	0.0068	0.0103
$A_{o,cr} = \left[D_s - D_{otl} + \frac{D_{ctl}}{X_t}(X_t - d_o)\right] B_c$	0.0496	0.0504	0.0502
$r_{lm} = \frac{A_{o,sb} + A_{o,tb}}{A_{o,cr}}$	0.289	0.732	0.511
$r_s = \frac{A_{o,sb}}{A_{o,sb} + A_{o,tb}}$	0.205	0.0799	0.115
$J_l = 0.44(1 - r_s) +$ $[1 - 0.44(1 - r_s)] e^{-2.2 r_{lm}}$	0.693	0.523	0.587

Number of sealing strips, N_{ss}	2	2	2
$N_{r,cc} = \frac{D_s - 2l_c}{X_t}$	15.396	46.188	30.792
$N_{ss}^{+} = \frac{N_{ss}}{N_{r,cc}}$	0.129	0.043	0.064
Number of pass partition lane, N_p	2	2	2
Width of lane,W_p	0.01905	0.00635	0.009525
$A_{o,bp} = B(D_s - D_{otl} + 0.5N_pW_p)$	0.01916	0.01512	0.01611
$r_b = \frac{A_{o,bp}}{A_{o,cr}}$	0.3862	0.3001	0.3214
Constant, C	1.25	1.25	1.25
$C = -Cr_b\left[1 - (2N_{ss^+})^{\frac{1}{3}}\right]$,	-0.1747	-0.2091	-0.1982
$J_b = e^{-Cr_b\left[1-\left(2N_{ss}+\right)\frac{1}{3}\right]}$	0.8396	0.8112	0.8201
Inlet baffle spacing,$L_{b,i}$	0.361	0.361	0.361
Outlet baffle spacing, $L_{b,o}$	0.361	0.361	0.361
$L_i^+ = \frac{L_{b,i}}{L_{b,c}}$	1.1370	1.1370	1.1370
$L_o^+ = \frac{L_{b,o}}{L_{b,c}}$	1.1370	1.1370	1.1370
Length of tube, L	4	4	4
$N_b = \frac{L}{B} - 1$	11.59	11.59	11.59
$J_s = \frac{N_b - 1 + \left(L_i^+\right)^{(1-n)} + \left(L_o^+\right)^{(1-n)}}{N_b - 1 + L_i^+ + L_o^+}$	0.992	0.992	0.992
Constant n	0.333	0.333	0.333
Correction factor for temperature gradient, J_r	1	1	1
$h_{shell} = h_{ideal}J_cJ_lJ_bJ_sJ_r$	330.50	407.46	380.49
Number of pass in the tube, N_{pass}	2	2	2
Friction factor for ideal tube bank, f_i(from graph)	0.3	0.5034	0.163
$A_m = B\left[D_i - D_{otl} + \left(\frac{D_{otl} - d_o}{P_t}\right)(p_t - d_o)\right]$	0.0496	0.0504	0.0502
$F_{sbp} = \frac{\left[D_i - D_{otl} + \frac{1}{2}N_pW_p\right]B}{A_m}$	0.386	0.300	0.321
Tube pitch parallel to flow, p_p	0.0206	0.0206	0.0206
$N_c = \frac{D_i\left[1 - 2\left(\frac{l_c}{D_i}\right)\right]}{p_p}$	15.356	15.356	15.356
$\Delta P_{b,i} = \frac{4f_i m_s^2 N_c}{2\rho_s g A_m^2}\left(\frac{\mu_{s,w}}{\mu_s}\right)^{0.14}$	28.895	46.948	15.324
$N_{cw} = \frac{0.8l_c}{p_p}$	6.1425	6.1425	6.1425
$A_{wg} = \frac{D_i^2}{4}\left\{\frac{\theta}{2} - \left[1 - 2\left(\frac{l_c}{D_i}\right)\right] sin\left(\frac{\theta}{2}\right)\right\}$	0.0484	0.0484	0.0484
$A_{wt} = \frac{N_t}{8}(1 - F_c)\pi d_o^2$	0.0275	0.0285	0.0282
Area for flow through window, $A_w = A_{wg} - A_{wt}$	0.0209	0.0198	0.0201

$\triangle P_{w,i} = \frac{m_s^2(2+0.6N_{cw})}{2gA_mA_w\rho_s}$	20.121	20.844	20.658
$\frac{A_{o,sb}+A_{o,tb}}{A_m}$	0.289	0.732	0.511
$\frac{A_{o,sb}}{A_{o,sb}+A_{o,tb}}$	0.205	0.0799	0.115
Correction factor for baffle leakage,R_l	0.5	0.32	0.38
$F_{sbp} = \frac{\left[D_i-D_{otl}+\frac{1}{2}(N_pW_p)\right]B}{A_m}$	0.386	0.300	0.321
Number of sealing strips per crossflow section, N_{ss}	2	2	2
$\frac{N_{ss}}{N_c}$	0.13024	0.13024	0.13024
R_b	0.58	0.62	0.63
$l_{s,i}^* = \frac{l_{b,i}}{l_b}$	1.137	1.137	1.137
$l_{s,o}^* = \frac{l_{b,o}}{l_b}$,	1.137	1.137	1.137
$n'$for laminar flow	1	1	1
n' for turbulent flow flow	1.6	1.6	1.6
$R_s = \frac{1}{2}\left[\left(l_{s,i}^*\right)^{-n} + \left(l_{s,o}^*\right)^{-n}\right]$	0.879	0.879	0.879
$\triangle P_{shell} = \left[(N_b-1)\left(\triangle P_{b,i}\right)R_b + N_b\triangle P_{w,i}\right]R_l + 2\triangle P_{b,i}R_b\left(1+\frac{N_{cw}}{N_c}\right)R_s$	246.77	247.764	153.708
Internal fouling resistance, R_{fi}	0.000176	0.000176	0.000176
External fouling resistance, R_{fo}	0.000176	0.000176	0.000176
Internal heat transfer area, $A_i = \pi d_i L N_t$	103.167	321.660	204.767
External heat transfer area, $A_o = \pi d_o L N_t$	138.17	414.51	276.34
Thermal conductivity of tube material, k_w	40	40	40
$\frac{r_o}{r_i}$	1.339	1.288	1.349
$A = \frac{1}{2}\left(A_i + A_o\right)$	120.66	368.08	240.55
$U_iA_i = \frac{1}{\frac{1}{h_iA_i}+\frac{ln\left(\frac{r_o}{r_i}\right)}{2\pi kLN_t}+\frac{R_{fi}}{A_i}+\frac{R_{fo}}{A_o}+\frac{1}{A_oh_o}}$	36229.61	130066.23	82024.41
$U_iA_i permeter$	9057.40	32516.55	20506.10
$UA_{total} = \frac{Q}{\triangle T_{lmtd}}$	28689.20	28689.20	28689.20
$L = \frac{UA_{total}}{UA_{permeter}}$	3.167	0.882	1.399
$V = \frac{\pi}{4}D_s^2L$	1.002	0.279	0.442
$\triangle P_{tube} = \left[\frac{4fLN_{pass}}{d_i} + 4N_{pass}\right]\frac{\rho_t u_m^2}{2}$	9227.90	8502.65	9986.90

Appendix B

Sample calculation for the longitudinal fin shell and tube heat exchanger

Parameter	1^{st} Iteration	2^{nd}Iteration
T_{hi}	90	90
T_{ho}	35	35
T_{ci}	20	20
T_{co}	25	25
Q	900000	900000
C_{ph}	2006	2006
C_{pc}	4182	4182
ρ_h	925	925
ρ_c	1000	1000
k_h	0.146	0.146
k_c	0.6	0.6
P_{rh}	88.96	88.96
P_{rc}	6.97	6.97
μ_h	0.006475	0.006475
μ_c	0.001	0.001
m_h	8.157346143	8.157346143
m_c	43.04160689	43.04160689
D_i	0.635	0.438
d_i	0.018923	0.018923
d_o	0.022225	0.022225
t	0.001651	0.001651
Tube layout constant, CL	0.87	0.87
Tube count calulation constant, CTP	0.9	0.9

PR= Pitch Ratio	1.25	1.25
Fin height, h_f	0.003	0.004
d_f	0.028225	0.030225
$N_t = 0.785\left(\frac{CTP}{CL}\right)\frac{D_i^2}{(PR)^2 d_o^2}$	263.058	109.1411
Length of the tube, L	4	4
Width of the fin, W	4	4
Thickness of the fin, t	0.003	0.001
Number of fin per tube	20	20
Total number of fin	5261.17	2182.82
Circumference of the fin. $C = 2\,(W + t)$	8.006	8.006
Fin area, $A_{finned} = C * H * N$	0.039146	0.199146
Unfinned area, $A_{unfinned} = (\pi d_o - N * t) * W$	0.039146	0.199146
Total area available for the heat transfer for single tube	0.519506	0.839306
Total area available for the heat exchanger	136.6604998	91.60281444
$L_c = L + \frac{t}{2}$	0.0015	0.0005
$P = 2W + +2t$	8.006	8.002
$A_c = Wt$	0.012	0.004
$\triangle T_{lmtd} = \frac{(T_{hi}-T_{ho})-(T_{ho}-T_{ci})}{ln\frac{(T_{hi}-T_{co})}{(T_{ho}-T_{ci})}}$	34.09857192	34.09857192
Temperature correction factor, F	0.92	0.92
$\Delta Tcf = \Delta Tlmtd * F$	31.37068617	31.37068617
$UA_{total} = \frac{Q}{\triangle T_{lmtd}}$	28689.20352	28689.20352
Number of pass, N_{pass}	2	2
$A_{tp} = \frac{\pi}{4} d_i^2 \left(\frac{N_t}{N_{pass}}\right)$	0.036971926	0.015339391
$u_{mt} = \frac{m_t}{\rho_t A_{tp}}$	1.16416999	2.805952709
$R_e = \frac{\rho_t u_{mt} d_i}{\mu_t}$	22029.58872	53097.04312
$f = (1.58lnRe - 3.28)^{-2}$	0.006379341	0.005168124
$\frac{f}{2}$	0.00318967	0.002584062
$Nu_t = \frac{\left(\frac{f}{2}\right)(Re-1000)Pr_t}{1+12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)}$	161.2222431	346.2341076
$h_i = \frac{Nu_t k_c}{d_i}$	5111.94556	10978.19926
diameter of outer tube limit, D_{otl}	0.618	0.42
Root diameter of tube, d_r	0.022225	0.022225
Pt= tube pitch= $PR * d_o$	0.03528125	0.03778125
Central baffle spacing, $B = 0.5 * D_i$	0.3175	0.219

Free flow area for the shell side flow, $A_m = B\left[D_i - D_{otl} + \left(\frac{D_{otl}-d_o}{P_t}\right)(p_t - d_r) - 2h_f\right]$	0.075397896	0.03981025
Reynold number for the shell side flow, $Re_{shell} = \frac{d_o m_s}{\mu_s A_s}$	371.3570384	703.3248803
a_1	0.593	0.593
a_2	-0.477	-0.477
a_3	1.45	1.45
a_4	0.519	0.519
$a = \frac{a_3}{1+0.14(Re_s)^{a_4}}$	0.360794506	0.27855964
Colburn factor, $J_i = a_1\left(\frac{1.33}{\frac{P_t}{d_o}}\right)^a (Re_s)^{a_2}$,	0.036056574	0.026452465
$\mu_{s,wall}$ =viscosity at average wall temperature	0.00925	0.00925
$h_{ideal} = J_i C_{ps}\left(\frac{m_s}{A_s}\right)\left(\frac{k_s}{C_{ps}\mu_s}\right)^{\frac{2}{3}}\left(\frac{\mu_s}{\mu_{s,wall}}\right)^{0.14}$	373.5426679	519.0225359
Baffle cut$= 0.25 * D_i$	0.15875	0.1095
$D_{ctl} = D_{otl} - d_o$	0.595775	0.397775
$\left(\frac{D_s-2l_c}{D_{ctl}}\right)$	0.532919307	0.550562504
$\theta_{ctl} = 2cos^{-}\left(\frac{D_s-2l_c}{D_{ctl}}\right)$	2.017498909	1.97551683
$F_w = \frac{\theta_{ctl}}{2\pi} - \frac{sin\theta_{ctl}}{2\pi}$	0.177646886	0.168201328
$F_c = 1 - 2F_w$	0.644706229	0.663597344
$J_c = 0.55 + 0.72F_c$	1.014188485	1.027790087
$1 - \frac{2l_c}{D_s}$	0.5	0.5
$\theta_b = 2cos^{-}\left(1 - \frac{2l_c}{D_s}\right)$	2.094395102	2.094395102
Diametral shell baffle clearance, δ_{sb}	0.00155127	0.001550876
$A_{o,sb} = \pi D_s\left(\frac{\delta_{sb}}{2}\right)\left(1 - \frac{\theta_b}{2\pi}\right)$	0.001030764	0.000710803
Diametral clearance beteen tube and baffle, δ_{tb}	0.004	0.004
$A_{o,tb} = \frac{\pi D_s \delta_{tb} N_t(1-F_w)}{2}$	0.862668912	0.249712606
Transverse pitch, $X_t = p_t$	0.03528125	0.03778125
Longitudinal pitch, $X_l = \left(\frac{\sqrt{3}}{2}\right)p_t$	0.030554459	0.032719522
$A_{o,cr} = \left[D_s - D_{otl} + \frac{D_{ctl}}{X_t}\left[(X_t - d_o) - (d_f - d_o)\right]\right]B_c$	0.004803719	0.00368575
$r_{lm} = \frac{A_{o,sb}+A_{o,tb}}{A_{o,cr}}$	11.45522247	6.290425394
$r_s = \frac{A_{o,sb}}{A_{o,sb}+A_{o,tb}}$	0.001193429	0.002838406
$J_l = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]e^{-2.2r_{lm}}$	0.439474891	0.43875165
Number of sealing strips, N_{ss}	2	2
$N_{r,cc} = \frac{D_s-2l_c}{X_t}$	8.99911426	5.796526055
Number of pass partition lane, N_p	2	2

Width of lane, W_p	0.022225	0.022225
Cross flow area for flow bypass, $A_{o,bp} = B(D_s - D_{otl} + 0.5N_pW_p)$	0.012453938	0.008809275
$r_b = \frac{A_{o,bp}}{A_{o,cr}}$	0.165176193	0.221281578
Constant, C	1.25	1.25
$C = -Cr_b\left[1-(2N_{ss^+})^{\frac{1}{3}}\right]$,	-0.048898788	-0.032173005
$J_b = e^{-Cr_b\left[1-\left(2N_{ss^+}\right)\frac{1}{3}\right]}$	0.952277507	0.96833904
Inlet baffle spacing, $L_{b,i}$	0.3175	0.219
Outlet baffle spacing, $L_{b,o}$	0.3175	0.219
$L_i^+ = \frac{L_{b,i}}{L_{b,c}}$	1	1
$L_o^+ = \frac{L_{b,o}}{L_{b,c}}$	1	1
Length of the tube, L	4	4
Number of baffle, $N_b = \frac{L}{B} - 1$	11.5984252	17.26484018
n	0.333	0.333
$J_s = \frac{N_b - 1 + \left(L_i^+\right)^{(1-n)} + \left(L_o^+\right)^{(1-n)}}{N_b - 1 + L_{ii}^+ + L_o^+}$	1	1
J_r	1	1
$h_{shell} = h_{ideal}J_cJ_lJ_bJ_sJ_r$	158.5464364	226.6401472
$AC = W * t$	0.012	0.004
Thermal conductivity of fin material, k_f	350	350
$m = \sqrt{\frac{2h}{kt}}$	17.3779519	35.98731104
$L_c = L + \frac{t}{2}$	0.003	0.004
$\eta_f = \frac{tanh(mL_c)}{mL_c}$	0.999095004	0.993149646
Total fin area, A_f =Fin area for one tube* Total number of tube	2527.256189	1397.355856
Total unfinned area, $A_{unfinned}$ =Unfinned area*Total number tubes	205.9538071	434.7004331
Effective surface area, $A_o = A_{eff} = A_f * \eta_f + A_{unfinned}$	2730.92284	1822.483906
$A = A_{finned} + A_{unfinned}$	2733.209996	1832.056289
Overall surface efficiency,$\eta_o = \left[1-(1-\eta_f)\frac{A_f}{A}\right]$	0.999163198	0.994775061
Internal surface area, $A_i = \pi d_i L N_t$	62.52188535	25.93988929
η_i	1	1
R_{fi}	0.000176	0.000176
R_{fo}	0.000176	0.000176
$\frac{1}{U_iA_i} = \frac{1}{\eta_ih_iA_i} + \frac{R_{fi}}{\eta_iA_i} + \frac{ln\left(\frac{r_o}{r_i}\right)}{2\pi N_tLk} + \frac{R_{fo}}{\eta_oA_o} + \frac{1}{\eta_oh_oA_o}$	8.38941E-06	1.29949E-05

U_iA_i	119197.9439	76953.14707
$U_iA_i$per meter	29799.48598	19238.28677
$L = \frac{UA_{total}}{UA_{permeter}}$	0.962741557	1.491255633
Number of pass for the tube, N_{pass}	2	2
$\triangle P_{tube} = \left[\frac{4fLN_{pass}}{d_i} + 4N_{pass}\right]\frac{\rho_t u_m^2}{2}$	7180.66322	44320.20612
b_1	4.57	4.57
b_2	-0.476	-0.476
b_3	7	7
b_4	0.5	0.5
$b = \frac{b_3}{1+0.14(Re)^{b_4}}$	1.892972583	1.485304539
$f_i = b_1\left(\frac{1.33}{\frac{p_t}{d_o}}\right)^b (Re)^{b_2}$	0.307393821	0.221149762
Friction factor for ideal tube bank for the finned tube	0.15369691	0.110574881
Tube pitch parallel to flow,$p_p = p_t cos 30^0$	0.005442184	0.005827813
$N_c = \frac{D_i\left[1-2\left(\frac{l_c}{D_i}\right)\right]}{p_p}$	58.34054893	37.57842185
$\Delta P_{b,i} = \frac{4f_i m_s^2 N_c}{2\rho_s g A_m^2}\left(\frac{\mu_{s,w}}{\mu_s}\right)^{0.14}$	477.1110851	793.0626892
$N_{cw} = \frac{0.8l_c}{p_p}$	23.33621957	15.03136874
$A_{wg} = \frac{D_i^2}{4}\left\{\frac{\theta}{2} - \left[1-2\left(\frac{l_c}{D_i}\right)\right] sin\left(\frac{\theta}{2}\right)\right\}$	0.061913671	0.02945692
$A_{wt} = \frac{N_t}{8}\left(1-F_c\right)\pi d_o^2$	0.01812021	0.007118215
Area for flow through window, $A_w = A_{wg} - A_{wt}$	0.043793461	0.022338704
$\Delta P_{w,i} = \frac{m_s^2\left(2+0.6N_{cw}\right)}{2gA_{free\,flow}A_w\rho_s}$	174.3108977	445.6647675
$P = 0.8 - 0.15\left(1+r_s\right)$	0.649820986	0.649574239
$R_L = exp\left[-1.33\left(1+r_s\right)\left(r_l\right)^P\right]$	0.001512033	0.01222371
$\theta = 2cos^{-}\left(1-\frac{2l_c}{D_s}\right)$	2.094395102	2.094395102
$S_{sb} = \frac{\pi D_i \delta_{sb}}{2}\left(1-\frac{\theta}{2\pi}\right)$	0.001030764	0.000710803
$S_{tb} = \pi d_o \delta_{tb}\left(\frac{1}{2}\right)\left(1+F_c\right)N_t$	0.060386824	0.025341839
$\theta_{ds} = 2cos^{-}\left(1-2l_c\right)$	1.639237246	1.349061815
$S_{sb} = D_i\delta_{sb}\left(\pi - 0.5\theta_{ds}\right)$	0.002285707	0.001674753
$\theta_{ctl} = 2cos^{-}\left(\frac{D_s-2l_c}{D_{ctl}}\right)$	1.512440278	1.071137057
$F_c = 1 + \frac{1}{\pi}\left(sin\theta_{ctl} - \theta_{ctl}\right)$	0.836260347	0.938410449
$S_{tb} = 0.5\pi D_i \delta_{tb} N_t\left(1+F_c\right)$	1.926282868	0.581926301
r_s	0.001185183	0.002869688
r_l	25.57854621	14.65956758
p	0.649822223	0.649569547

R_l	1	1
$F_{sbp} = \frac{\left[D_i - D_{otl} + \frac{1}{2}(N_p W_p)\right]B}{A_{freeflow}}$	0.165176193	0.221281578
Number of sealing strips per crossflow section, N_{ss}	2	2
$\frac{N_{ss}}{N_c}$	0.034281474	0.053222033
$S_b = B\left(D_s - D_{otl}\right)$	0.0053975	0.003942
C_R	3.7	3.7
$R_b = exp\left[-C_R\left(\frac{S_b}{S_m}\right)\left(1 - \sqrt[3]{2r_{ss}}\right)\right]$	0.855163552	0.82469583
$n'$for laminar flow	1	1
$n'$for turbulent flow	1.6	1.6
$R_s = \frac{1}{2}\left[\left(l^*_{s,i}\right)^{-n} + \left(l^*_{s,o}\right)^{-n}\right]$	1	1
N_b	2.032256872	5.809386454
$\triangle P_{shell} = \left[\left(N_b - 1\right)\left(\triangle P_{b,i}\right)R_b + N_b \triangle P_{w,i}\right]R_l + 2\triangle P_{b,i} R_b \left(1 + \frac{N_{cw}}{N_c}\right)R_s$	1917.83602	7565.847681

Appendix C

Sample calculation of the Radial fin

Parameter	1^{st} Iteration	2^{nd} Iteration	3^{rd} Iteration
T_{hi}	90	90	90
T_{ho}	35	35	35
T_{ci}	20	20	20
T_{co}	25	25	25
Q	900000	900000	900000
C_{ph}	2006	2006	2006
C_{pc}	4182	4182	4182
ρ_h	925	925	925
ρ_c	1000	1000	1000
k_h	0.146	0.146	0.146
k_c	0.6	0.6	0.6
P_{rh}	88.96	88.96	88.96
P_{rc}	6.97	6.97	6.97
μ_h	0.006475	0.006475	0.006475
μ_c	0.001	0.001	0.001
m_h	8.157346143	8.157346143	8.157346143
m_c	43.04160689	43.04160689	43.04160689
D_i	0.635	0.635	0.635
d_i	0.014224	0.018923	0.018923
d_o	0.01905	0.022225	0.022225
t	0.004826	0.001651	0.001651
Tube layout constant constant, CL	0.87	0.87	0.87
Tube count calculation constant, CTP	0.9	0.9	0.9
PR =Pitch Ratio	1.25	1.25	1.25
Fin height, h_f	0.01	0.01	0.003

d_f	0.03905	0.042225	0.028225
$N_t = 0.785\left(\frac{CTP}{CL}\right)\frac{D_i^2}{(PR)^2 d_o^2}$	137.4289606	117.5387375	263.0585591
r_1	0.01905	0.022225	0.022225
t_f	0.0002	0.0002	0.001
r_2	0.02905	0.032225	0.025225
r_{2c}	0.02915	0.032325	0.025725
Fin area, $A_{finned} = 2\pi\left(r_{2c}^2 - r_1^2\right)$	0.00305723	0.003459997	0.001053941
Tube length, L	4	4	4
Fin density	1428.571429	400	800
Number of fin per tube	5714.285714	1600	3200
Gap between two fins		0.0023	0.00025
Total fin area, $= A_f N_f$	17.46988343	5.53599584	3.3726112
Unfinned area, $A_{unfinned} = 2\pi r_1\left(H - N_f t_f\right)$	0.341811429	0.51362864	0.1116584
Thermal conductivity of fin material, k_f	40	40	40
$m = \sqrt{\frac{2h}{kt}}$	234.3789096	249.4530526	103.2937391
$c_2 = \frac{\frac{2r_1}{m}}{r_{2c}^2 - r_1^2}$	0.333916619	0.323419956	2.564136324
mr_1	4.464918228	5.544094094	2.295703351
mr_{2c}	6.832145215	8.063569925	2.657231437
$I_o\left(mr_1\right)$	17	44.3139	2.8177
$I_1\left(mr_{2c}\right)$	133.6	399.7464	2.8874
$I_1\left(mr_1\right)$	14.91044	40.1117	2.0945
$K_0\left(mr_1\right)$	0.006732	0.002012	0.0792
$K_1\left(mr_{2c}\right)$	0.000545	0.0001553	0.0616
$K_1\left(mr_1\right)$	0.00792	0.0022341	0.09524
$\eta_f = c_2\frac{K_1(mr_1)I_1(mr_{2c}) - I_1(mr_1)K_1(mr_{2c})}{I_0(mr_1)K_1(mr_{2c}) + K_0(mr_1)I_1(mr_{2c})}$	0.385851291	0.353591075	0.930508371
Total fin area= $N_f * Fin\,area\,per\,tube$	2400.867921	650.6939619	887.1942427
Total unfinned area	46.97478933	60.3712619	29.37269782
$A_{effective} = A_{finned}\eta_f + A_{unfinned}$	973.3527769	290.4508392	854.9143675
Total area, A	2447.84271	711.0652238	916.5669405
$\eta_o = \left[1 - (1 - \eta_f)\frac{A_f}{A}\right]$	0.397636978	0.408472851	0.93273533
$A_i = \pi d_i L N_t$	24.55215656	27.93577026	62.52188535
η_i	1	1	1
R_{fi}	0.000176	0.000176	0.000176

R_{fo}	0.000176	0.000176	0.000176
$\frac{1}{U_iA_i} = \frac{1}{\eta_i h_i A_i} + \frac{R_{fi}}{\eta_i A_i} + \frac{ln\left(\frac{r_o}{r_i}\right)}{2\pi N_t L k} + \frac{R_{fo}}{\eta_o A_o} + \frac{1}{\eta_o h_o A_o}$	2.41565E-05	4.64849E-05	1.26499E-05
U_iA_i	41396.73428	21512.3661	79052.26579
U_iA_i per meter	10349.18357	5378.091525	19763.06645
$\triangle T_{lmtd} = \frac{(T_{hi}-T_{ho})-(T_{ho}-T_{ci})}{ln\frac{(T_{hi}-T_{co})}{(T_{ho}-T_{ci})}}$	34.09857192	34.09857192	34.09857192
Temperature correction factor, F	0.92	0.92	0.92
$\Delta Tcf = \Delta Tlmtd * F$	31.37068617	31.37068617	31.37068617
$UA_{total} = \frac{Q}{\triangle T_{lmtd}}$	28689.20352	28689.20352	28689.20352
N_{pass}	2	2	2
$A_{tp} = \frac{\pi}{4} d_i^2 \left(\frac{N_t}{N_{pass}}\right)$	0.010913434	0.016519643	0.036971926
$u_{mt} = \frac{m_t}{\rho_t A_{tp}}$	3.943910643	2.605480428	1.16416999
$R_e = \frac{\rho_t u_{mt} d_i}{\mu_t}$	56098.18498	49303.50614	22029.58872
$f = (1.58 lnRe - 3.28)^{-2}$	0.005104172	0.005256263	0.006379341
$\frac{f}{2}$	0.002552086	0.002628132	0.00318967
$Nu_t = \frac{\left(\frac{f}{2}\right)(Re-1000)Pr_t}{1+12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)}$	363.0701882	324.7572782	161.2222431
$h_i = \frac{Nu_t k_c}{d_i}$	15315.10917	10297.22385	5111.94556
D_{otl}	0.593	0.593	0.593
d_r	0.01905	0.022225	0.022225
$Pt = PR * d_o$	0.0488125	0.05278125	0.03528125
B	0.3175	0.3175	0.3175
$A_m = B\left[D_i - D_{otl} + \left(\frac{D_{otl}-d_o}{P_t}\right)(p_t - d_r)\right] -2N_f t_f h_f]$	0.122631482	0.11773994	0.078874029
Reynold number for the shell side flow,$Re_{shell} = \frac{d_o m_s}{\mu_s A_s}$	195.7050908	237.808339	354.9906067
$\mu_{s,wall}$ =viscosity at average wall temperature	0.00925	0.00925	0.00925
Colburn factor, J_i	0.07	0.07	0.055
$h_{ideal} = J_i C_{ps}\left(\frac{m_s}{A_s}\right)\left(\frac{k_s}{C_{ps}\mu_s}\right)^{\frac{2}{3}}\left(\frac{\mu_s}{\mu_{s,wall}}\right)^{0.14}$	445.8728685	464.3967944	544.6827791
$D_{ctl} = D_{otl} - d_o$	0.57395	0.570775	0.570775
Baffle cut $= 0.25 * D_i$	0.15875	0.15875	0.15875
$\left(\frac{D_s-2l_c}{D_{ctl}}\right)$	0.553184075	0.556261224	0.556261224
$\theta_{ctl} = 2cos^{-}\left(\frac{D_s-2l_c}{D_{ctl}}\right)$	1.969229539	1.961832852	1.961832852

$F_w = \frac{\theta_{ctl}}{2\pi} - \frac{sin\theta_{ctl}}{2\pi}$	0.166808844	0.165178083	0.165178083
$F_c = 1 - 2F_w$	0.666382312	0.669643833	0.669643833
$J_c = 0.55 + 0.72F_c$	1.029795265	1.03214356	1.03214356
$1 - \frac{2l_c}{D_s}$	0.5	0.5	0.5
$\theta_b = 2cos^{-}\left(1 - \frac{2l_c}{D_s}\right)$	2.094395102	2.094395102	2.094395102
Diametral shell baffle clearance, δ_{sb}	0.004445	0.004445	0.004445
$A_{o,sb} = \pi D_s \left(\frac{\delta_{sb}}{2}\right)\left(1 - \frac{\theta_b}{2\pi}\right)$	0.002953546	0.002953546	0.002953546
Diametral clearance beteen tube and baffle, δ_{tb}	0.00079375	0.00079375	0.00079375
$A_{o,tb} = \frac{\pi D_s \delta_{tb} N_t (1-F_w)}{2}$	0.090610813	0.077648305	0.173781441
Transverse pitch, $X_t = p_t$	0.0488125	0.05278125	0.03528125
Longitudinal pitch, $X_l = \left(\frac{\sqrt{3}}{2}\right) p_t$	0.042272865	0.045709903	0.030554459
$A_{o,cr} = \left[D_s - D_{otl} + \frac{D_{ctl}}{X_t}\right] \left[(X_t - d_o) - (d_f - d_o)\, t_f N_f\right] B_c$	0.103112927	0.112754441	0.055742976
$r_{lm} = \frac{A_{o,sb}+A_{o,tb}}{A_{o,cr}}$	0.907396991	0.714844133	3.17053377
$r_s = \frac{A_{o,sb}}{A_{o,sb}+A_{o,tb}}$	0.031566998	0.036643649	0.016711722
$J_l = 0.44\,(1 - r_s) + [1 - 0.44\,(1 - r_s)]\, e^{-2.2r_{lm}}$	0.504067993	0.543418396	0.433177206
Number of sealing strips, N_{ss}	2	2	2
$N_{r,cc} = \frac{D_s - 2l_c}{X_t}$	7.510728214	6.945978372	10.39128208
$N_{ss^+} = \frac{N_{ss}}{N_{r,cc}}$	0.266285764	0.287936399	0.192469032
Number of pass partition lane, N_p	2	2	2
Width of lane, W_p	0.01905	0.02225	0.02225
Cross flow area for flow bypass, $A_{o,bp} = B(D_s - D_{otl} + 0.5N_pW_p)$	0.019383375	0.020399375	0.020399375
$r_b = \frac{A_{o,bp}}{A_{o,cr}}$	0.187982008	0.180918594	0.36595418
Constant, C	1.25	1.25	1.25
$C = -Cr_b \left[1 - (2N_{ss^+})^{\frac{1}{3}}\right]$,	-0.044510859	-0.037999174	-0.124680762
$J_b = e^{-Cr_b\left[1-\left(2N_{ss^+}\right)\frac{1}{3}\right]}$	0.956465214	0.962713736	0.882778674
Inlet baffle spacing, $L_{b,i}$	0.361	0.361	0.361
Outlet baffle spacing, $L_{b,o}$	0.361	0.361	0.361
$L_i^+ = \frac{L_{b,i}}{L_{b,c}}$	1.137007874	1.137007874	1.137007874
$L_o^+ = \frac{L_{b,o}}{L_{b,c}}$	1.137007874	1.137007874	1.137007874
Length of the tube, L	4	4	4
Number of baffle, $N_b = \frac{L}{B} - 1$	11.5984252	11.5984252	11.5984252

n	0.333	0.333	0.333
$J_s = \frac{N_b - 1 + \left(L_i^+\right)^{(1-n)} + \left(L_o^+\right)^{(1-n)}}{N_b - 1 + L_i^+ + L_o^+}$	0.992605807	0.992605807	0.992605807
J_r	1	1	1
$h_{shell} = h_{ideal} J_c J_l J_b J_s J_r$	219.733893	248.9073018	213.3919306
Friction factor for ideal tube bank	0.5	0.3886	0.3164
Friction factor for ideal tube bank for the finned tube	0.25	0.1943	0.1582
Tube pitch parallel to flow	0.206756	0.206756	0.206756
$N_c = \frac{D_i\left[1 - 2\left(\frac{l_c}{D_i}\right)\right]}{p_p}$	1.535626536	1.535626536	1.535626536
$\Delta P_{b,i} = \frac{4 f_i m_s^2 N_c}{2 \rho_s g A_m^2} \left(\frac{\mu_{s,w}}{\mu_s}\right)^{0.14}$	3.860948294	3.255240735	5.90603959
$N_{cw} = \frac{0.8 l_c}{p_p}$	0.614250614	0.614250614	0.614250614
$A_{wg} = \frac{D_i^2}{4}\left\{\frac{\theta}{2} - \left[1 - 2\left(\frac{l_c}{D_i}\right)\right] sin\left(\frac{\theta}{2}\right)\right\}$	0.061913671	0.061913671	0.061913671
$A_{wt} = \frac{N_t}{8}\left(1 - F_c\right) \pi d_o^2$	0.006530658	0.007528122	0.016848376
Area for flow through window, $A_w = A_{wg} - A_{wt}$	0.055383013	0.05438555	0.045065295
$\Delta P_{w,i} = \dfrac{m_s^2\left(2 + 0.6 N_{cw}\right)}{2 g A_{free\,flow} A_w \rho_s}$	12.54382891	13.30458549	23.96803515
$\dfrac{A_{o,sb} + A_{o,tb}}{A_{free\,flow}}$	0.762971776	0.68457527	2.240724726
$\dfrac{A_{o,sb}}{A_{o,sb} + A_{o,tb}}$	0.031566998	0.036643649	0.016711722
R_l	0.3	0.31	1
$F_{sbp} = \frac{\left[D_i - D_{otl} + \frac{1}{2}(N_p W_p)\right] B}{A_{freeflow}}$	0.391811094	0.416718194	0.622059955
Number of sealing strips per crossflow section, N_{ss}	2	2	2
$\frac{N_{ss}}{N_c}$	1.3024	1.3024	1.3024
R_b	1	1	1
$l_{s,i}^* = \frac{L_{b,i}}{L_{b,c}}$	1.137007874	1.137007874	1.137007874
$l_{s,o}^* = \frac{L_{b,o}}{L_{b,c}}$	1.137007874	1.137007874	1.137007874
n' for laminar flow	1	1	1
n' for turbulent flow	1.6	1.6	1.6
$R_s = \frac{1}{2}\left[\left(l_{s,i}^*\right)^{-n} + \left(l_{s,o}^*\right)^{-n}\right]$	0.879501385	0.879501385	0.879501385
$\triangle P_{shell} = [(N_b - 1)\left(\triangle P_{b,i}\right) R_b + N_b \triangle P_{w,i}]$ $R_l + 2\triangle P_{b,i} R_b \left(1 + \frac{N_{cw}}{N_c}\right) R_s$	65.43057612	66.54829461	1207.941786
$L = \frac{UA_{total}}{UA_{permeter}}$	2.772122393	5.33445803	1.451657494

Number of pass for the tube, N_{pass}	2	2	2
$\triangle P_{tube} = \left[\frac{4fLN_{pass}}{d_i} + 4N_{pass}\right]\frac{\rho_t u_m^2}{2}$	124109.0938	67389.9521	8074.20075

Appendix D

HTRI Results

Rating - Horizontal Multipass Flow TEMA AES Shell With Single-Segmental Baffles

See Data Check Messages Report for Informative Messages.

See Runtime Message Report for Warning Messages.

Process Conditions		Hot Shellside		Cold Tubeside	
Fluid name		polyalkline glycol		water	
Flow rate	(kg/s)		8.4500		43.0102
Inlet/Outlet Y	(Wt. frac vap.)	0.000	0.000	0.000	0.000
Inlet/Outlet T	(Deg C)	90.00	35.00	20.00	25.00
Inlet P/Avg	(kPa)	0.000	0.000	0.000	0.000
dP/Allow.	(kPa)	5.775	0.000	9.888	0.000
Fouling	(m2-K/W)		0.000176		0.000176

Exchanger Performance					
Shell h	(W/m2-K)	421.61	Actual U	(W/m2-K)	326.67
Tube h	(W/m2-K)	3637.44	Required U	(W/m2-K)	276.06
Hot regime	(--)	Sens. Liquid	Duty	(MegaWatts)	0.9139
Cold regime	(--)	Sens. Liquid	Area	(m2)	103.186
EMTD	(Deg C)	31.6	Overdesign	(%)	18.33

Shell Geometry			Baffle Geometry		
TEMA type	(--)	AES	Baffle type	(--)	Single-Seg.
Shell ID	(mm)	533.000	Baffle cut	(Pct Dia.)	25.00
Series	(--)	1	Baffle orientation	(--)	Perpend.
Parallel	(--)	1	Central spacing	(mm)	266.000
Orientation	(deg)	0.00	Crosspasses	(--)	15

Tube Geometry			Nozzles		
Tube type	(--)	Plain	Shell inlet	(mm)	102.261
Tube OD	(mm)	22.225	Shell outlet	(mm)	102.261
Length	(m)	4.580	Inlet height	(mm)	18.339
Pitch ratio	(--)	1.2500	Outlet height	(mm)	18.339
Layout	(deg)	30	Tube inlet	(mm)	154.051
Tubecount	(--)	330	Tube outlet	(mm)	154.051
Tube Pass	(--)	2			

Thermal Resistance, %		Velocities, m/s		Flow Fractions	
Shell	77.48	Shellside	0.15	A	0.065
Tube	9.76	Tubeside	0.80	B	0.305
Fouling	12.00	Crossflow	0.23	C	0.153
Metal	0.757	Window	0.39	E	0.177
				F	0.300

Figure D.1: HTRI Result