DESIGN AND DEVELOPMENT OF A SHELL AND TUBE HEAT EXCHANGER

BY

BABATUNDE, VICTOR SHOLA
(MEE/12/0857)

OF

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TO

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FEDERAL UNIVERSITY, OYE EKITI. EKITI STATE

STUDENT DECLARATION

I BABATUNDE VICTOR SHOLA with matriculation number MEE/12/0857 hereby declare that this work in this project is my own except for quotations and summaries which have been duly acknowledged. This project has not been accepted for any degree and is not concurrently submitted in candidate of any other degree.

Signature: ……………………………..

Name:

Matriculation number: MEE/12/0857

Date: November 2017

CERTIFICATION

This is to certify that this project was carried out by BABATUNDE VICTOR SHOLA with matriculation MEE/12/0857 a student of mechanical engineering, faculty of engineering, Federal University, oye-ekiti.

PROF. B.O BOLAJI	------------------------------------
Project supervisor's name	SIGNATURE AND DATE
DR. OYELARAN	Oyelaran Ayani 7/11/17
Co-supervisor's name	SIGNATURE AND DATE
DR ADELEKE	E Adeleke 20/12/2018
Head of department name	SIGNATURE AND DATE
PROF. ABOLARINWA	Abolarin 9/11/17
External Supervisor	SIGNATURE AND DATE

DEDICATION

This study is dedicated to my parents, Dr. & Mrs. Babatunde for their continued love over the years. Thanks for everything.

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I acknowledge the supernatural assistance of the Lord of host in this project work, when I was weak, He gave me strength

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ABSTRACT

This project is mainly focusing on designing one type of a heat exchanger which is shell and tube heat exchanger. Step by step on designing, analysis was done with the use of the effective NTU method. This method is technically simpler than LMTD where it has to refer to a complex graph to calculate certain outputs that needed. The software used is ANSYS 14. The language in the program consists of formulas of calculating parameters in heat exchanger such as heat transfer coefficient, heat transfer rate or heat capacity rate. This program design should match the heat exchanger software in the market. To design this heat exchanger, many considerations were taken. The shell size must be adaptable to the water flow rate. To determine how many tubes that are used also depends on the size of the shell. Water flow rate can be determined by using ball valve opening. To read the temperature, a thermometer digital is attached at inlet and outlet for both hot and cold fluid. The manual calculation is done to check the program in the software whether it is compatible. We used the exhaust gases from the diesel generator used in ikole campus to raise the temperature of water. This was achieved after rigorous activities of analysis.

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CHAPTER ONE

1.1 Background of the Study

Heat exchanger is a device built for efficient heat transfer from one medium to another, whether the media are separated by a solid wall so that they never mix, or the media are in direct contact, or a heat exchanger is a heat-transfer devise that is used for transfer of internal thermal energy between two or more fluids available at different temperatures. In most heat exchangers, the fluids are separated by a heat-transfer surface, and ideally they do not mix.

They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, and natural gas processing. One common example of a heat exchanger is the radiator in a car, in which a hot engine-cooling fluid, like antifreeze, transfers heat to air flowing through the radiator. To design a heat exchanger, many criteria have to be taken before making any decision. From there, the software runs and the main point is to create a new windows program that is similar to another existed software of calculating heat transfer in market. The major purpose is to obtain a desired output by using this program. Thus, the ideal shell and tube heat exchanger is produced at the end.

An extensive research work has been done till date on the Shell and Tube heat exchangers by changing different parameters to meet the industry requirements. Lunsford (1998) provided some methods for increasing shell-tube exchanger performance. The methods considered whether the exchanger is performing correctly to begin with, excess pressure drop capacity in existing exchangers, the re-evaluation of fouling factors and their effect on exchanger calculations, and the heat transfer in shell and tube heat exchangers for staggered has slight contribution to the local heat transfer at the surfaces of the external tubes of the tube bundle, but reduces greatly the per-compartment average heat transfer. Morcos and Shafey (1995) carried out an experimental analysis to study the performance analysis of a plastic shell and tube heat exchanger. Qiao He and Wennan Zhang (2001) presented a theoretical analysis and an experimental test on a shell and tube latent heat storage exchanger. The prediction by the mathematical model on the performance of the heat storage exchanger is reasonable and in agreement with experimental measurements. Rozzi et al (2007) worked on convective heat transfer and friction losses in helically enhanced tubes for both Newtonian and non-Newtonian fluids. Four fluid foods, namely, whole milk, cloudy orange juice, apricot and apple puree, are tested in a shell and tube heat exchanger. Both fluid heating and cooling conditions are considered. The experimental outcome confirms that helically corrugated tubes are particularly effective in enhancing convective heat transfer for generalized Reynolds number ranging from about 800 to the limit of the transitional flow regime.

Hosseini et al (2007), they 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). Corrugated and microfin tubes have shown degradation of performance at a Reynolds number below a certain value ($N_{Re} < 400$). At a higher Reynolds number the performance of the heat exchanger greatly improved for microfinned tubes. Babu and Munawar (2007) discussed about the differential evolution strategies for optimal design of shell and tube heat exchanger. Zubair et al (2000) presents a simple probabilistic approach to shell and tube heat exchanger. Probabilistic approach is used to characterize various fouling growth models in terms of the risk level p and scatter in growth rate of the process. These random fouling growth models are then considered in the performance evaluation of a shell and tube heat exchanger in a crude oil preheat train to demonstrate the influence of risk level and scatter parameter on important thermal parameters of the heat exchangers. Although the analysis presented in the paper is applicable to shell-and-tube heat exchangers, the procedure can easily be modified to include other types of heat exchanger such as double pipe, plate and frame and other compact heat exchangers.

Even while the traditional research is going on shell and tube heat exchangers by changing various parameters, in 1990's the research trend has changed towards modeling and simulation, without involving much experimental work. Taveb et al (1991) developed a new dynamic model for a shell and tube heat exchanger with water-water system and they linearized non-linear relations for the development of the dynamic model representing the system. Georgiadis et al (1998) developed a mathematical model for single.

Shell and Tube heat exchangers under milk fouling. Mandavgane et al (2006) developed mathematical models for the estimation of hot & cold fluid outlet temperatures as a function of flow rates and inlet temperatures for a shell and tube heat exchanger. Tan and Fok (2006) developed an educational computer aided design tool for shell and tube heat exchanger that integrates thermos hydraulics analysis with mechanical design. Dirkse et al (2006) has modelled a shell and tube heat exchanger based on natural convection using computational fluid dynamics.

spacing, baffle cut, tube size, shell pass number, shell size, etc., on the average heat transfer coefficient, thermal performance and thermal efficiency of the shell and tube heat exchangers. Joydeep Barman and Ghoshal (2007) considered an optimum design problem for the different constraints involved in the designing of a shell-and-tube heat exchanger consisting of longitudinally finned tubes. A Mat lab simulation has been employed using the Kern's method of design of extended surface heat exchanger to determine the behavior on varying the values of the constraints and studying the overall behavior of the heat exchanger with their variation for both cases of triangular and square pitch arrangements, along with the values of pressure drop. Simulations were performed to analyze the effect of tube pitch on the heat transfer. Results showed that the heat transfer coefficients can change when the longitudinal and transverse tube pitches are varied and the best values of these parameters are found. Good agreement is observed between the computed values and the literature values. Dirkse et al (2006) developed a computational fluid dynamics model for the design of a natural convection shell and tube heat exchanger with baffles.

1.2 Aims and Objective of the Project

The aim of the study is to:

- Develop an apparatus that can be of use in Fuoye for the generator by recycling the exhaust gases to raise the temperature of water

The objectives of this project are:

- To design and simulate a shell and tube type heat exchanger using ANSYS software.

1.3 Justification of the Study

The project helps in revealing the ingenuity in the engineering field and helps to realize the beauty of engineering field

The project solves real life problem in restoring the eco system to a balanced one.

The project also helps in renewing the heat that is supposed to be lost from the exhaust of the generator

The project will in the long run helps to increase the life span of the generator and this will reduce the failure rate.

1.4 Problem Statement

The shell and tube heat exchanger is one type of heat exchanger that is common used in industry. This is because it contains a large number of tubes packed in a shell with the axis parallel to that shell. There is a heat transferred when one fluid flows inside the tubes and other fluid flows outside the tubes through the shell. In reality, shell and tube heat exchanger is not suitable f0or use in aircraft and automotive purposes because the size is relatively large and also weighty. In this project, the main focus is in designing a small shell and tube heat exchanger which is available and easy to do an experiment in the lab. There is a choice of formula to utilize in defining the desired outputs. We can use the LMTD (Log Mean Temperature Difference) or ε-NTU(effectiveness-Number of Transfer Units). The method used in constructing the formula is using ε-NTU which is introduced in 1955 by Kays and London. This method is more greatly simplified heat exchanger analysis than LMTD. In this project, the problem stated are to check

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

Heat exchanger is one of devices that is convenient in industrial and household application, these include power production, chemical industries, food industries, electronics, environmental engineering, manufacturing industry, and many others. It comes in many types and function according to its uses.

Heat exchanger is a device that is used to transfer thermal energy between two or more fluids, between a solid surface and a fluid at different temperatures and in thermal contact. There are usually no external heat and work interactions. In most heat exchangers, heat transfer between fluids takes place through a separating wall or into and out of a wall in a transient manner. (Shah R.K., 2003)

This chapter will discuss about the uses and application of shell and tube heat exchanger, type of heat exchangers, and shell and tube heat exchanger.

2.2 Uses and Applications of Heat Exchanger

Heat exchangers are used to transfer heat from one media to another. It is most commonly used in space heating such as in the home, refrigeration, power plants and even in air conditioning. It is also used in the radiator in a car using an antifreeze engine cooling fluid. Heat exchangers are classified according to their flow arrangements where there are the parallel flow, and the counter flow. Aside from this, heat exchangers also have different types depending on their purpose and

Heat exchangers mostly can be found in industries which produce a heat stream. In this case, heat exchangers usually circulate the output heat to put it as input by heating a different stream in the process. The fact that it really saves a lot of money because when the output heat no longer needed then it can be recycled rather than to come from an external source as heat is basically recycled. When used in industries and in the home, it can serve to lower energy costs as it helps recover wasted heat and recycle it for heating in another process. Typically, most heat exchangers use fluid to store heat and heat transfer can take the form of either absorption or dissipation. For instance, heat exchangers are used as oil coolers, transmission and engine coolers, boiler coolers, waste water heat recovery, condensers and evaporators in refrigeration systems. In residential homes, heat exchangers are used for floor heating, pool heating, snow and ice melting, domestic water heater, central, solar and geothermal heating. Of course, heat exchangers have different designs which depend on the purpose it is intended for. Brazed heat exchangers, a collection of plates which are brazed together, are used for hydronic systems like swimming pools, floor heating, snow and ice melting. The shell and coil heat exchanger design is best for areas with limited spaces as it can be installed vertically.

2.3 Type of Heat Exchangers

Different heat transfer applications require different types of hardware and different configurations of heat transfer equipment. The attempt to match the heat transfer hardware to the heat transfer requirements within the specified constraints has resulted in numerous types of innovative heat exchanger designs.

2.3.1 Tubular Heat Exchangers

The tubular types are consists of circular tubes. One fluid flows inside the tubes and the other flows on the outside of the tubes. The parameters of the heat exchanger can be changed like the tube diameter, the number of pitch, tube arrangement, number of tubes and length of the tube can be manipulate. The common type of heat exchangers lie under this categories are double-pipe type, shell-and-tube type and spiral tube type. The tubular heat exchangers can be designed for high pressure relative to the environment and high pressure difference between the fluids. These exchangers are used for liquid-to-liquid and liquid-to-vapor phase. But when the operating temperature or pressure is very high or fouling on one fluid side, it will used gas-to-liquid and gas-togas heat transfer applications.

2.3.1.1 Double-Pipe Heat Exchanger

According to SadicKakac, a double-pipe heat exchanger consists of smaller and larger diameter pipe where the smaller pipe fitted concentrically into the larger one in purpose to give direction to the flow from one section to another. One set of these tubes includes the fluid that has to be cooled or heated. The second fluid runs over the tubes being cooled or heated in order to provide heat or absorb the heat. A set of tubes is the tube bundle and it can be made up of several types of tubes such as longitudinally plain, longitudinally finned, and more. If the application requires an almost constant wall temperature, the fluids may flow in a parallel direction. It's easy to clean and convenient to disassemble and assemble. The double-pipe heat exchanger is one of the simplest. Usually, it is used for small capacity applications because it is so expensive on a cost

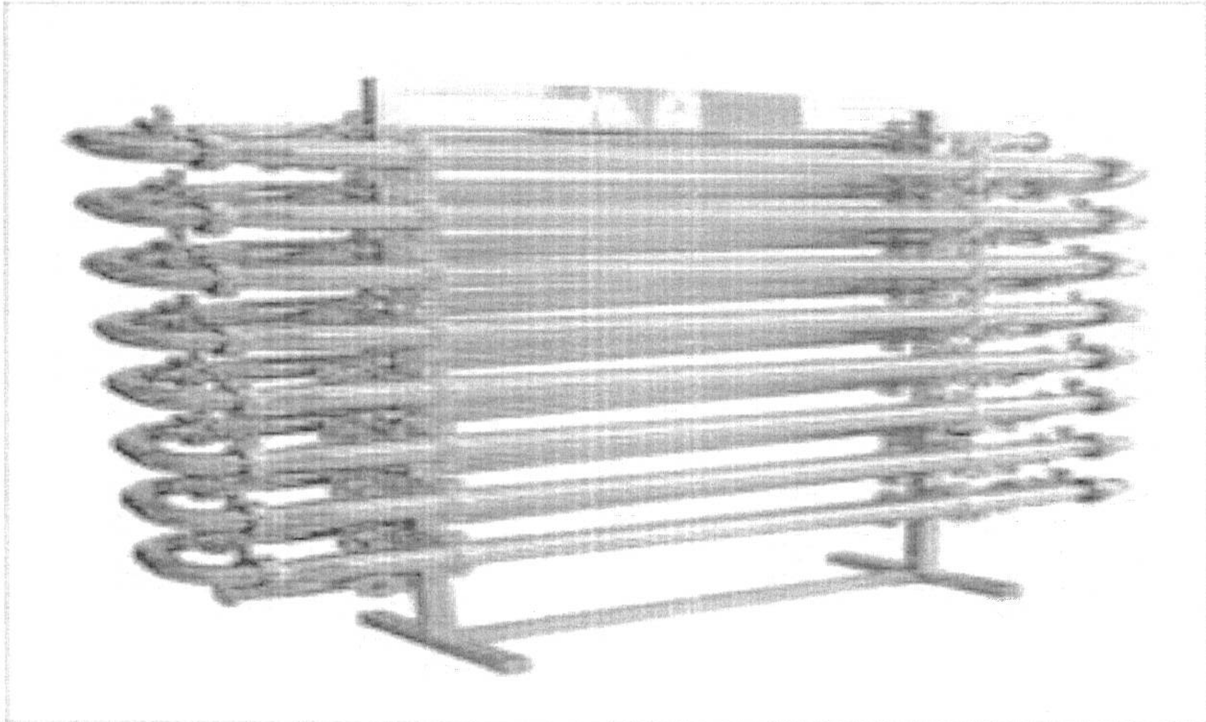


Plate 2. 1: double-pipe heat exchanger
Source: Ritai China

2.3.1.2 Shell-And-Tube Heat Exchanger

The most common type of heat exchanger in industrial applications is the shell-and-tube heat exchanger, shown in Plate 2.2. Shell-and-tube heat exchangers contain a large number of tubes (sometimes several hundred) packed in a shell with their axes parallel to that of the shell. Heat transfer takes place as one fluid flows inside the tubes while the other fluid flows outside the tubes through the shell. Baffles are commonly placed in the shell to force the shell-side fluid to flow across the shell to enhance heat transfer and to maintain uniform spacing between the tubes.

and-tube heat exchangers are further classified according to the number of shell and tube passes involved. Heat exchangers in which all the tubes make one U-turn in the shell, for example, are called one-shell-pass and two tube-passes heat exchangers. Likewise, a heat exchanger that involves two passes in the shell and four passes in the tubes is called a two-shell-passes andfour-tube-passes heat exchanger

The plate 2.2 shows the type of shell-and-tube heat exchanger.

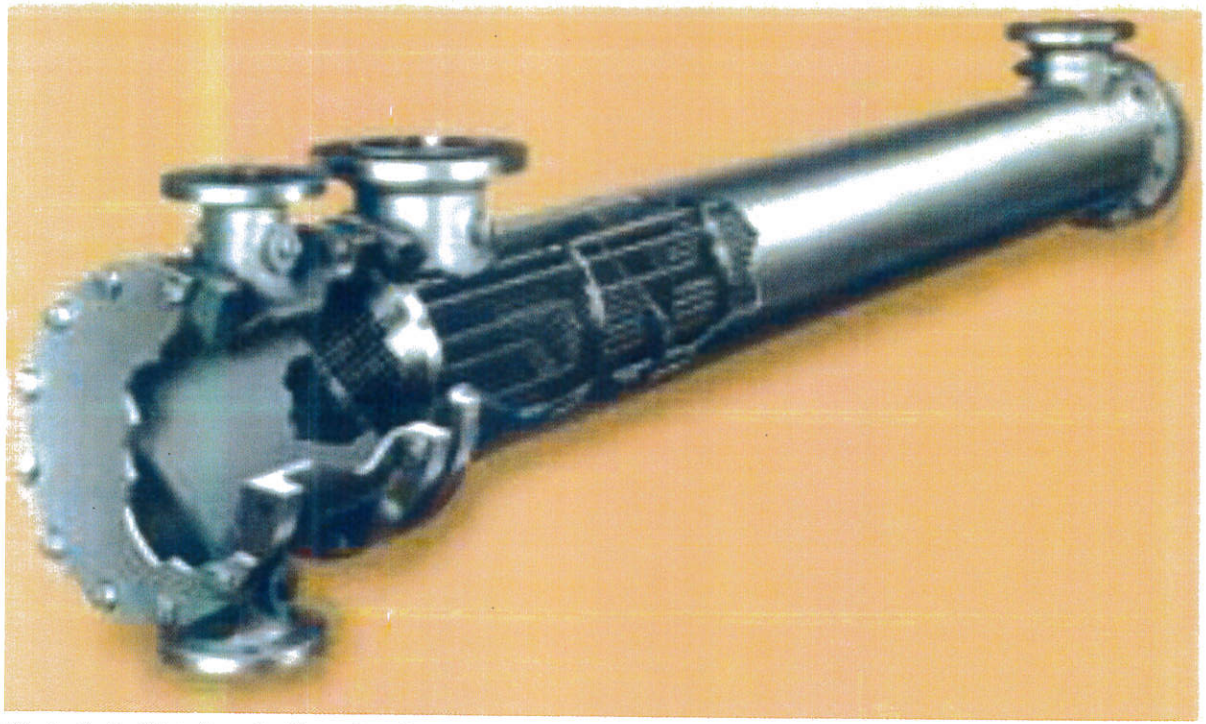


Plate 2. 2: U-tube shell-and-tube heat exchanger
Source: API Heat Transfer

2.3.1.3 Spiral-Tube Heat Exchanger

A spiral heat exchanger is a helical or coiled tube configuration. It consists of spirally and clean fluids. The biggest advantage of the spiral heat exchanger is its efficient use of space. A compact spiral heat exchanger can lower costs, while an oversized one can have less pressure drop, higher thermal efficiency, less pumping energy, and lower energy costs. Spiral heat exchangers are frequently used when heating fluids that have solids and therefore often foul the inside of the heat exchanger. Spiral heat exchangers have three types of flow arrangements. Firstly, the spiral flow and cross flow has one fluid in each. The spiral flow passages are welded at each side and this type of flow is good for handling low density gases which pass through the cross flow. This can be used for liquid-to-liquid applications if one fluid has a much greater flow rate than the other. A second type is the distributed vapor and spiral flow. The coolant moves in a spiral and exits through the top. The hot gases that enter will leave as condensate out of the bottom outlet. The third type is the countercurrent flow where both of the fluids will flow in opposite directions and are used for liquid-to-liquid applications. The spiral heat exchanger is good for pasteurization, heat recovery, digester heating, effluent cooling, and pre-heating. Plate 2.3 presents the spiral-tube heat exchanger.

2.3.2 Plate Heat Exchangers

A second type of heat exchanger is a plate heat exchanger. It has many thin plates that are slightly apart and have very large surface areas and fluid flow passages that are good for heat transfer. This can be a more effective heat exchanger than the tube or shell heat exchanger due to advances in brazing and gasket technology that have made this plate exchanger more practical. Large heat exchangers are called plate and frame heat exchangers and there allow for periodic disassembly, cleaning, and inspection. There are several types of permanently bonded plate heat exchangers like dip brazed and vacuum brazed plate varieties, and they are often used in refrigeration. These heat exchangers can further be classified as gasketed plate, spiral plate and lamella.

2.3.2.1 Gasketed Plate Heat Exchangers

A gasketed plate heat exchanger consists of a series of thin plates that have wavy surface which function as separating the fluids. The plates come with corner parts arranged so that the two media between which heat is to be exchanged flow through interchange exclaim spaces. Appropriate deisgn and gasketing permit a stack of plates to be held together by compression bolts joining the end plates. Gaskets prevent leakage to the outside and direct the fluids in the plates as desired. The flow patern is generally chosen so that the media flow countercurrent to each other. Since the flow passages are quite small, strong eddying gives high heat transfer coefficients, high pressure drops, and high local shear which minimizes fouling. These exchangers provide a relatively compact and lightweight heat transfer surface. Gasketed plate are typically used for

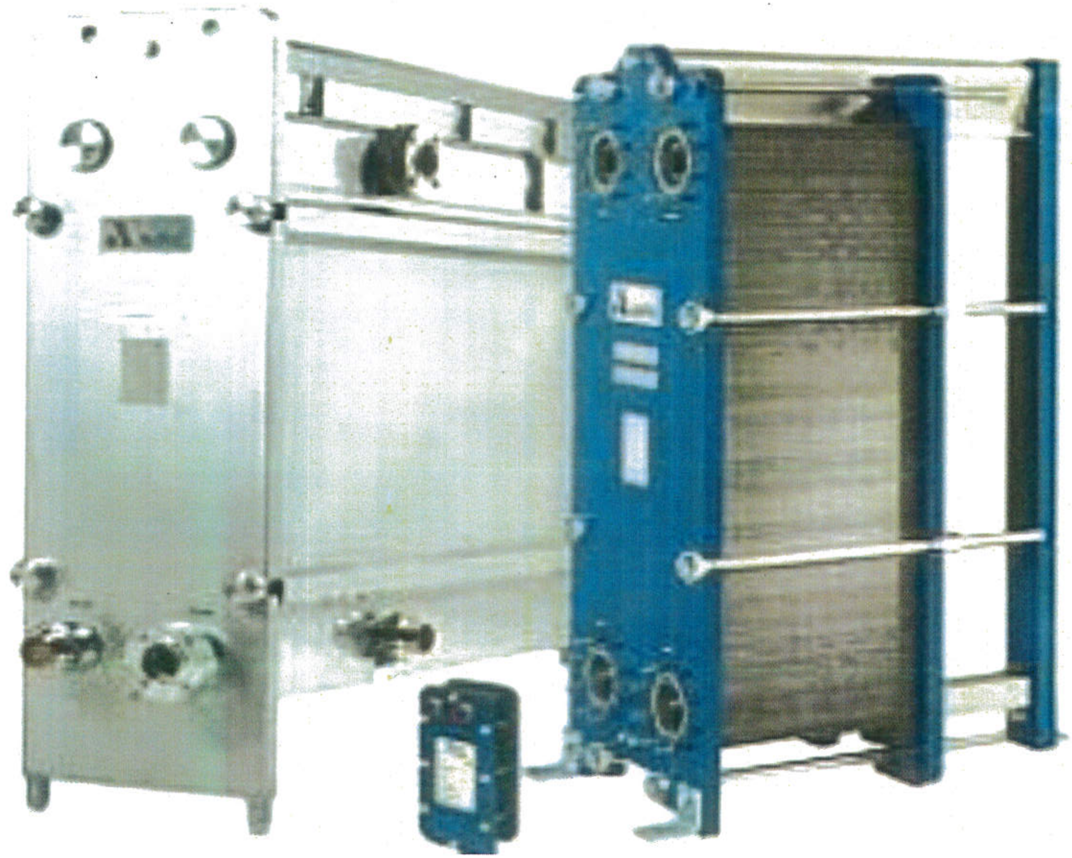


Plate 2. 4: Gasketed plate heat exchanger

2.3.2.2 Spiral Plate Heat Exchanger

Spiral heat exchangers are formed by rolling two long, parallel plates into a spiral using a mandrel and welding the edges of adjacent plates to form channels. The distance between the metal surfaces in both channels is maintained by means of distance pins welded to the metal sheet. The two spiral paths introduce a secondary flow, increasing the heat transfer and reducing fouling deposits. These heat exchangers are quite compact but are relatively expensive

Type Ihas flat covers over the spiral channels. The media flow countercurrent through the channels via the connections in the center and at the periphery. This type is used to exchange heat between media without phase changes such as liquid-liquid, gas-liquid, or gas-gas. One stream enters at the center of the unit and flows from inside outward. The other stream enters at the periphery and flows towards the center. Thus the counterflow is achieved.

Type II is designed for cross flow operation. One channel is completely seal welded, while the other is open along both sheet metal edges. The passage with the medium in spiral flow is welded shut on each side, and the medium in cross flow passes through the open spiral annulus. This type is mainly used as a surface condenser in evaporating plants. It is also highly effective as a vaporizer. Two spiral bodies are often built into the same jacket and are mounted below each other.

Type III, the third standard type is in principle similar to type I with alternately welded up channels, but type III is provided with a specially designed top cover. This type of heat exchanger is mainly intended for condensing vapors with sub-cooling of condensate and non-condensable gases. The top cover, therefore, has a special distribution cone where the vapor is distributed to the uncovered spiral turns in order to maintain a constant vapor velocity along the channel opening. The plate 2.5 and 2.6 presents the types of spiral plate.

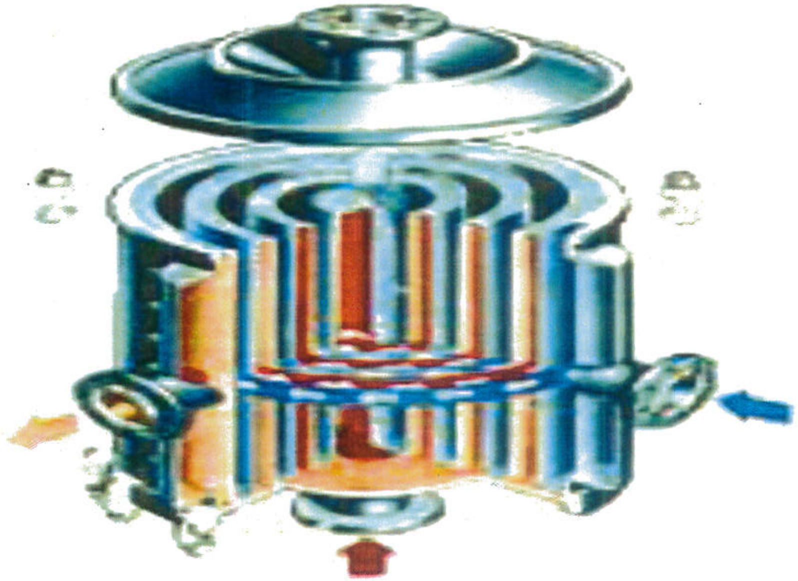


Plate 2. 5: Spiral plate heat exchanger Type I

2.3.2.3 Lamella Heat Exchangers

The lamella type of heat exchanger consists of a set of parallel, welded, thin plates channels are lamellae placed longitudinally in a shell. It is a modification of the floating-type shell-and-tube heat exchanger. These flattened tubes, called lamellae are made up of two strips of plates, profiled and spot- or seam-welded together in a continuous operation. The forming of the strips creates space inside the lamellae and bosses acting as spacers for the flow sections outside the lamellae on the shell side. The lamellae are welded together at both ends by joining the ends with steel bars in between, depending on the space required between lamellae. Both ends of the lamella bundle are joined by peripheral welds to the channel cover which at the outer ends is welded to the inlet and outlet nozzle. The lamella side thus completely sealed in by welds. At the fixed end, the channel cover is equipped with an outside flange ring which is bolted to the shell flange. Plate 2.7 presents the lamella heat exchanger.

CHAPTER THREE

MATERIALS AND METHODS

3.1 Materials

Table 3. 1: Materials used for the design

Materials	Dimension/Specification	Application
Stainless steel pipe	89mm	Used as the shell
Copper pipe	18mm	Used as the pipe
Stainless steel plate	20mm	Used as the baffle
Mild steel pipe	92mm	Used as a housing for the insulator and protects the shell
313CDW Hose	8mm	Used to link the pipe to the pump
U curves	18mm	Used with the pipe
Water pump	0.5hp	Used for pumping water
¼" mild steel pipe	-	Used as the stand for the exchanger
A & B Chemical	2litres	Used as the insulator
Paint	1 litre	Used to paint the whole project

3.1.1 Shell

The shell barrel must be straight and have no out-of-roundness, as a tightly fitting tube always staggered. The inside diameter of a rolled shell should not exceed the design inside diameter by more than 3.2 mm (1/8 in) as determined by circumferential measurement. All internal welds must be made flush.

When welding large nozzles to the shell "shrinkage" may occur at the nozzle/shell junction and effective measures, such as the use of temporary stiffening, must be taken to avoid it. Shrinkage reduces the shell diameter at the nozzle/shell junction so that the baffle diameter must be reduced accordingly. The increased clearance between baffle and shell may result in reduced thermal performance.

Standard pipe less than 450 mm in diameter is usually available, and this will be used for the shell and head barrels instead of rolled plate. Depending on the fabricators roll capacity, at thicknesses of the order of 80 mm and greater or large thickness/diameter ratios, it may be necessary to use forged instead of rolled barrels.

When an expensive barrel metal is required for corrosion resistance purposes only, the barrel is formed from the selected metal if its thickness is less than about 15 mm. Above this the use of clad metal should be investigated, as it may provide a cost saving. The clad metal will usually comprise a steel plate, having a thickness suitable for the pressure and temperature conditions with a layer of the required corrosion resistant metal, about 3 mm thick, bonded to it. The cladding may be applied by explosive, roll bonding, or weld deposition methods. TEMA specifies minimum shell and head barrel thicknesses, which depend on barrel diameter, metal and TEMA class.

Plate 3. 1: **shell of the heat exchanger**

3.1.2 Water Pump

A water pump is a device that is used for raising, compressing or transferring fluids (liquids or gases) from one medium to another by means of mechanical action. Pumps can be classified into three major groups according to the method they use in moving the fluid. The function of the water pump is to pump water that will enter the shell and tube heat exchanger which will be heated by the exhaust gas coming out from the generator. The water pump used for the heat exchanger is a Marcuise (model Q860, 0.5hp, 220v) water pump

Plate 3. 2: marcuise water pump

3.1.3 Tube

The tubes are the basic component of the shell and tube heat exchanger, providing the heat transfer surface between one fluid flowing inside the tube and other fluid flowing across the outside of the tubes. The tube may be seamless or welded and most commonly made of copper or steel alloys. Other alloys of nickel, titanium, or aluminum may also be used for specific applications. The tube may be either bare or extended surface on the outside. Extended or enhanced surface tubes are used when one fluid has a substantially lower heat transfer coefficient then the other fluid. Doubly enhanced tubes that is, with enhancement both inside and outside are available that can reduce the size and cost of the exchanger. Extended surfaces (finned tubes) provide two to

Tubes should be able to withstand the following

1. Operating temperature and pressure on both sides

2. Thermal stresses due to the differential thermal expansion between the shell and the tube bundle

3. Corrosive nature of both the shell-side and the tube-side fluids

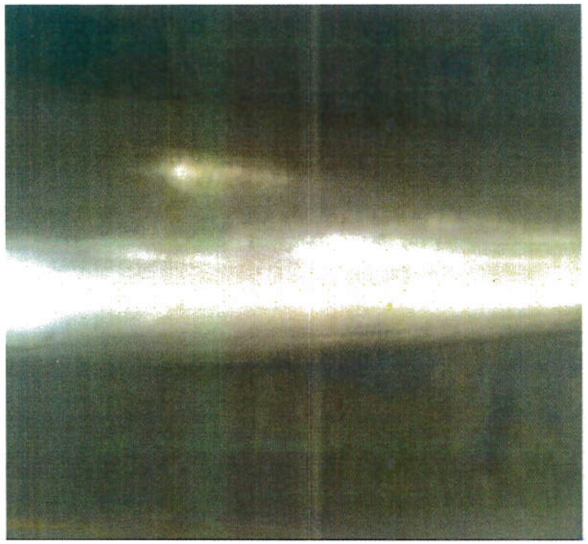
There are two types of tubes: straight tubes and U-tubes. The tubes are further classified as

1. Plain tubes

2. Finned tubes

3. Duplex or bimetallic tubes

4. Enhanced surface tubes



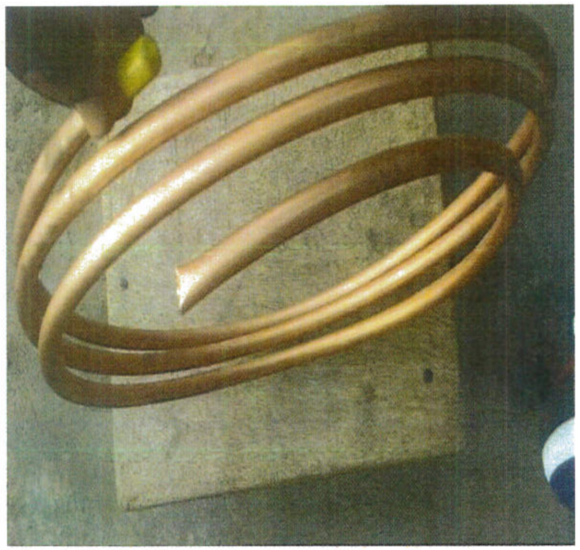
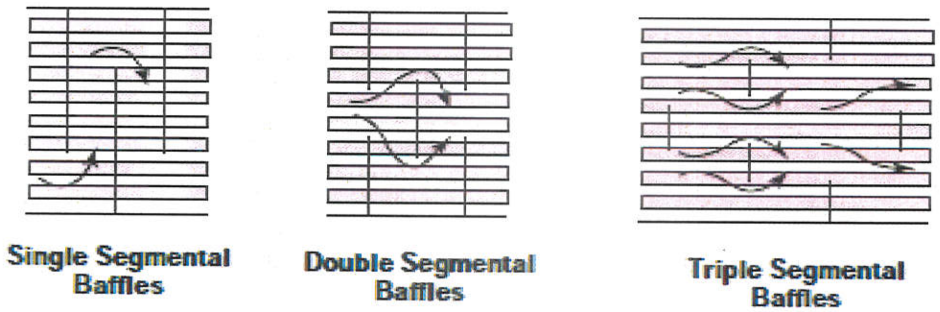


Plate 3. 3: Tube

3.1.4 Baffle

Baffles are used to support tubes, enable a desirable velocity to be maintained for the shell side

Plate baffles may be single-segmental, double-segmental, or triple-segmental, as shown in Plate.



The Baffle spacing is the centerline-to-centerline distance between adjacent baffles. It is the most vital parameter in Shell and rube heat exchanger design. The TEMA standards specify the minimum baffle spacing as one-fifth of the shell inside diameter or 2 in., whichever is greater. Closer spacing will result in poor bundle penetration by the shell side fluid and difficulty in mechanically cleaning the outsides of the tubes.

3.1.5 313CDW Hose

A hose is a flexible hollow tube designed to carry fluids from one location to another. Hose design is based on a combination of application and performance. The hose is been used to carry water that is been pumped by the water pump into the heat exchanger for heating.

3.1.6 A and B Chemical

A and B Chemical is used for insulation, it is used to preventing heat from escaping, or it is used to reduce the rate of heat transfer into the surrounding

3.2 Methods

The conceptual design of the project was started with the use of Ansys 14.00 for the modelling of the project and Ansys 18.0 for the simulation

For the simulation to be possible, initial conditions are very important. The designed exchanger was modelled with the following initial boundary condition were applied Also, the flow table shown below is used in the identification of other fow parameters

Table 3. 2: Fouling factors for some fluid (j.pholman)

Intake & Exhaust System

Combustion Air Consumption	m^3/min	27
Max. Intake Restriction	KPa	5
Exhaust Temperature (Before Turbo)	°C	660
Exhaust Temperature (After Turbo)	°C	510
Max. Exhaust Back Pressure	Kpa	5
Exhaust Gas Flow	m^3/min	33
Turbo Bellows Diameter	mm	DN100-150
Exhaust Flange Diameter	mm	DN150

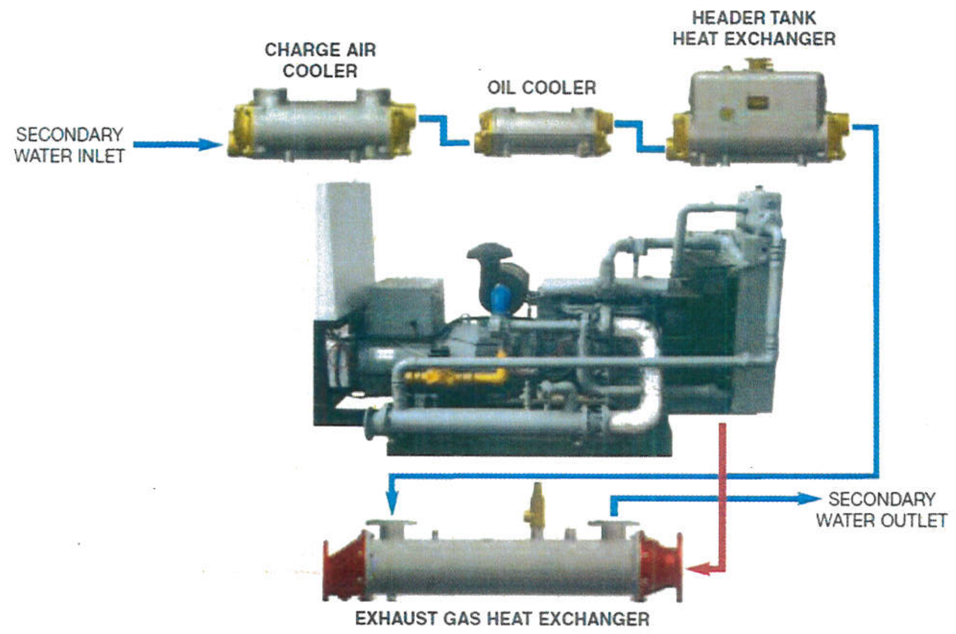


Table 3. 3: Flow tude table

parameters	Flow rate(m^3/s)	Velocity(m/s)	Area (m^2)	Diameter (m)
Pump	5.833e-4	0.726	8.04e-4	0.032
Tube	2.9165e-4	3.713	7.855e-5	0.01
Shell	0.55	96.83	5.68	0.085

3.3 Modelling Of the Exhaust

The modelling of the exhaust was done with Ansys 14.0 and the result is as shown below

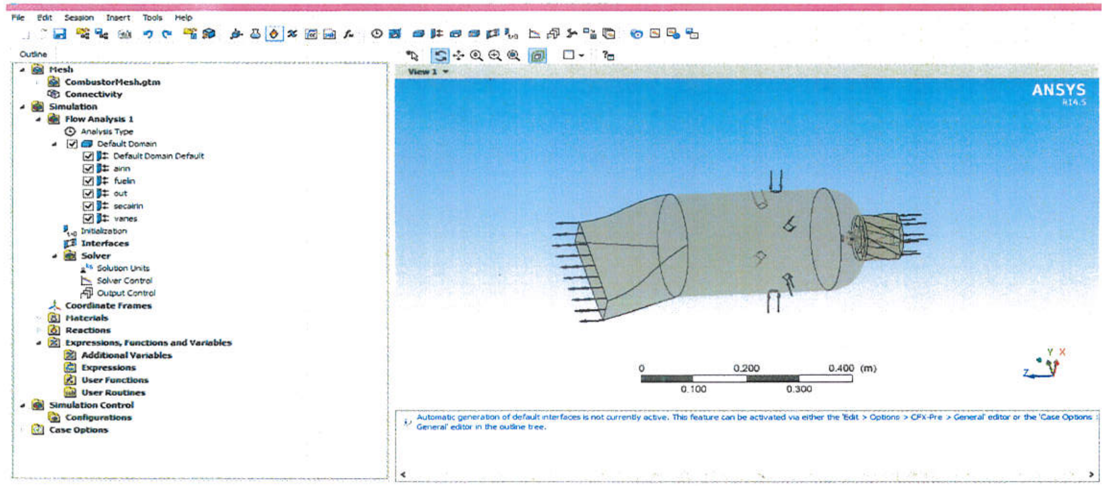


Plate 3. 5: Result of simulated exhaust pipe

3.4 Design Calculations

Marcuise Water Pump Model QB60

3.4.1 Specifications for Pump

H_{max} = 35m

Q_{max}= 35l/min

V=220v

F=50

A=2.5A

P=0.5

Pipe = 2.5m

D= 8.9cm

Taking the efficiency of pump as = 0.85

Average flow rate = 0.85 X 35 = 29.75L/min $\frac{29.75}{60} = \mathbf{0.496L/sec}$

Taking the transfer to be at the minimum of 350^0c, and the tube size is that of 3cm=0.03m, than the water temperature is at room temperature = 25^0c, and the properties of water is extracted from steam table as follow:

Knowing that the outlet temperature of water is not to exceed 100^0C, we take a counter flow exchanger design with the following parameters, i.e

$T_{c1} = 25^0$c

$T_{c2} = 100^0$c

$T_{h1} = 350^0$c

$T_{h2} = ?$

$C_{pe} = ?$

$C_{pw} =$ 4.2kj/kgk

M_w = 35 L/min = 0.035m^3/min

M_e = 33m^3

$\boldsymbol{we\ know\ power\ confirmed\ by\ the\ pump\ is\ P = IV = 220\ X\ 2.5 = 550w}$

The pressure with which the water flows at a mean height of 3.5m = P

$\boldsymbol{P = pgh = 1000\ x\ 9.81\ x\ 3.5 = 34.335kPa}$

$$\mathbf{0.035X4.2(100-25) = 3.3(1.0049)(350 - th2)}$$

$$\mathbf{th2 = 350 - 0.333 = 349.67^0}\text{c}$$

$\Phi m=(\Phi 1- \Phi 2)/(\ln\Phi 1/ \Phi 2)$

$\Phi m=349.84^0c$

$Q=UA\ \Phi m$

$A=\pi DL$

D=0.089m

$A=\pi * 0.089 * L$

Q=11.025kj

U=54w/m. ^0c

$11.025=54*\pi*0.089*L*349.84$

L=0.021m=21.00cm

Using n for the length of the tube

Using n for the length of the shell

3.4.2 Design for Shell

$$area\ of\ shell = \frac{\pi d^2}{4}$$

$$where\ d = 89cm = 0.089m$$

$$area = \frac{\pi\ X\ 0.089^2}{4}$$

$$area = 0.02853m^2$$

T_{h1} = temperature of hot exhaust

T_{h2} = temperature of cold exhaust

M_w = mass flow rate of water

M_e = mass flow rate of exhaust

C_{pe}= heat capacity of exhaust

C_{pw}= heat capacity of water

Θ_m = logarithmic mean temperature difference

$\Theta_1 = t_{h1} - t_{c2}$

$\Theta_2 = t_{h2} - t_{c1}$

Q = quantity of heat transferred

For maximum heat, the counterflow arrangement is used.

From the manual of the water pump engine, the mass flow of water

$M_e = 33m^3/min = 0.55m^3/sec$

By energy balance;

$M_e c_{pe} (t_{h1}-t_{h2}) = M_w c_{pe} (t_{c1}-t_{c2})$

$0.55\ X 1.033(350 - th2) = 0.0002916\ X\ 4.2\ X\ 73 =$

$0.5682(350 - th2) = 0.00012472(73)$

$0.582(350 - th2) = 0.0894$

350-th2=0.1536

$th_2 = 349.852$

$\Theta_m = \frac{250-322.85}{ln\frac{250}{322.85}} = \frac{-72.85}{ln\, 0.7744}$

$\Theta_m = 284.9^oc$

$Q = m_e c_{pe}(t_{h1}-t_{h2}) = 0.55 \ X\ 1.033\ X\ (350-349.85) = 0.0852 kw/m^2c$

$Q = UnA\Theta_m \ X\ F$

where F is the connection factor

$U_o =$ overall heat transfer coefficient $\frac{1}{ho} + \frac{1}{k} = 6.352\ m^2\ c/w$

$$\frac{1}{U} = \frac{1}{h} X \frac{di}{do} + \frac{\Delta x}{KA}$$

For negligible thickness of shell

Overall heat transfer $= \frac{1}{lm} X \frac{di}{do} + \frac{ox}{KA}$

for negligible thickness of shells $= \frac{1}{ho} + \frac{1}{k} = 6.352\ m^{2\,0}c/w$ R

$k = 54\ m^{2\,0}c/w$

$U = 5.684 kw/m^{2\,o}c$

$85.2 = 5.684\ X\ 0.032L\ X\ \pi\ X284.9\ X\ 0.70$

$$L = \frac{0.525}{0.70} = 75cm$$

$$Re = \frac{UD}{V}$$

For water at 100°C = 373 k, from appendix 1

$$V = 279\ X\ 10^{-6} \mu \frac{s}{m^3}$$

$$Re_{shell} = \frac{UD}{V}$$

From appendix two, by interpolation V at 623K (350°c) is calculated as 55.48m^3/s X 10^{-6}

$$Re_{shell} = \frac{96.83\ X\ 0.085\ X\ 10^6}{55.48} = 148351(\ turbulent)$$

3.4.3 Design for Fouling Factor (TUBE)

From table 4.2, F = 0.00009 m^{2o}c/w

$$F = (a\ \ln\frac{\left(\frac{1-p}{1-RP}\right)}{\frac{(1-R)\ ln((2-p)(b-a))}{(2-p)(b+a)}}$$

$$a = \sqrt{R^2 + H}$$

$$b = RH$$

$$R = \frac{th2 - th1}{th2 - tc2}$$

$$P = \frac{1}{\mathrm{R}}$$

$$a = \sqrt{R^2 + H} = \sqrt{R^2 + 1} = \sqrt{486.52^2 + 1}$$

$$a = 487.02$$

$$P = \frac{349.85 - 350}{27 - 350} = \frac{-0.15}{-323} = 0.000464$$

B = R+1 = 486.5+1= 487.5

$$F = \frac{4.87.02\ ln(\frac{1 - 0.000464}{1 - 0.000464\ X\ 486.5})}{(1 - 486.5)\ ln\frac{2 - 0.000464(489.5)}{2 - 0.000464(487.5 + 489.5)}}$$

$$L = \frac{A}{n\pi d} = \frac{0.00568}{2 X 3.142 X 0.01}$$

lenght of each tube = 9.04 cm

3.4.3 Design for Insulation

Thickness for insulation

$$r = \frac{h}{k} = \frac{0.352}{54}$$

$8.9 + 3 = 11.9 = 0.119m$

$h = 6.426$

$k = 54$

$r_o = 0.119m = 11.9cm$

$\Delta R = r_0 - r_1 = 11.9 - 8.9 = 3.0$ cm

CHAPTER FOUR

RESULT AND DICUSSION

4.1 Result

The plates below shows the fabricated heat exchanger

4.2 Test Result

The exchanger was tested with the generator that it was designed for and the results obtained is presented in table 4.1

Table 4. 1: the result obtained when the heat exchanger was tested

Parameters	Designed results	Tested results
Final tempertature of exhaust(°C)	349.66	245.62
Final temperature of water(°C)	100	87.53
Exchanger's effectiveness	0.226	0.1874

The relative efficiency of the exchanger is calculated as;

$$\text{Relative efficiency} = \frac{\text{effectiveness of the tested exchanger}}{\text{effectiveness of the designed exchanger}}$$

$$\text{Relativeefficiency} = \frac{0.1874}{0.226}$$

= 0.8292 or 82.92%

4.3 Discussion

The result of the tested exchanger has some disparities compared to the designed results of the exchanger, these may be due to the following reasons as highlighted below;

1. The generator was not working at full capacity because of students being on holiday and this might have reduced the exhaust flow rate designed.
2. The baffle material might have caused some constrictions in the the flow of the exhaust in the tube.
3. Approximation errors during design calculations and errors in the insulation thickness may be responsible for such a disparity.

However, relative efficiency of 0.8292 shows that the design was satisfactory.

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The results obtained from the testing of the heat exchanger show that the exchanger is satisfactory. Most exchangers' designs are not as efficient as this. However, the fact that the exchanger was designed into specifications is more than an enough reason to appreciate design and engineering. The project has also widened my knowledge and my ingenuity as an engineering student. The project has made me appreciate the use of software in designing and made me learn and understand the use of Ansys software

5.2 Recommendation

I recommend that further work be carried on the heat exchanger such as implementing and integrating the exchanger with flow meters so that the inlet and outlet flow parameters are easily known as this will make the exchanger a better one.

A similar exchanger may be designed to boil the water to a super-saturated level, such an exchanger finds an application in power plants.

Works may also be carried out on the selection of header tube and on the separation of the baffles of the exchangers

Design may be done with other Computational Fluid Dynamics (CFD) software so that comparison can be made to know the best software for the design of heat exchangers especially a shell and tube heat exchanger

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APPENDICES

Appendix One

The values of μ, k, c_p, and Pr are not strongly pressure-dependent and may be used over a fairly wide range of pressures

T, K	ρ kg/m^3	c_p kJ/kg · °C	$\mu \times 10^5$ kg/m · s	$\nu \times 10^6$ m^2/s	k W/m · °C	$\alpha \times 10^4$ m^2/s	Pr
100	3.6010	1.0266	0.6924	1.923	0.009246	0.02501	0.770
150	2.3675	1.0099	1.0283	4.343	0.013735	0.05745	0.753
200	1.7684	1.0061	1.3289	7.490	0.01809	0.10165	0.739
250	1.4128	1.0053	1.5990	11.31	0.02227	0.15675	0.722
300	1.1774	1.0057	1.8462	15.69	0.02624	0.22160	0.708
350	0.9980	1.0090	2.075	20.76	0.03003	0.2983	0.697
400	0.8826	1.0140	2.286	25.90	0.03365	0.3760	0.689
450	0.7833	1.0207	2.484	31.71	0.03707	0.4222	0.683
500	0.7048	1.0295	2.671	37.90	0.04038	0.5564	0.680
550	0.6423	1.0392	2.848	44.34	0.04360	0.6532	0.680
600	0.5879	1.0551	3.018	51.34	0.04659	0.7512	0.680
650	0.5430	1.0635	3.177	58.51	0.04953	0.8578	0.682
700	0.5030	1.0752	3.332	66.25	0.05230	0.9672	0.684
750	0.4709	1.0856	3.481	73.91	0.05509	1.0774	0.686
800	0.4405	1.0978	3.625	82.29	0.05779	1.1951	0.689
850	0.4149	1.1095	3.765	90.75	0.06028	1.3097	0.692
900	0.3925	1.1212	3.899	99.3	0.06279	1.4271	0.696
950	0.3716	1.1321	4.023	108.2	0.06525	1.5510	0.699
1000	0.3524	1.1417	4.152	117.8	0.06752	1.6779	0.702
1100	0.3204	1.160	4.44	138.6	0.0732	1.969	0.704
1200	0.2947	1.179	4.69	159.1	0.0782	2.251	0.707
1300	0.2707	1.197	4.93	182.1	0.0837	2.583	0.705
1400	0.2515	1.214	5.17	205.5	0.0891	2.920	0.705
1500	0.2355	1.230	5.40	229.1	0.0946	3.262	0.705
1600	0.2211	1.248	5.63	254.5	0.100	3.609	0.705
1700	0.2082	1.267	5.85	280.5	0.105	3.977	0.705
1800	0.1970	1.287	6.07	308.1	0.111	4.379	0.704
1900	0.1858	1.309	6.29	338.5	0.117	4.811	0.704
2000	0.1762	1.338	6.50	369.0	0.124	5.260	0.702
2100	0.1682	1.372	6.72	399.6	0.131	5.715	0.700
2200	0.1602	1.419	6.93	432.6	0.139	6.120	0.707
2300	0.1538	1.482	7.14	464.0	0.149	6.540	0.710

°F	°C	c_p kJ/kg · °C	ρ kg/m³	μ kg/m · s	k W/m · °C	Pr	$\frac{g\beta\rho^2 c_p}{\mu k}$ 1/m³ · °C
32	0	4.225	999.8	1.79×10^{-3}	0.566	13.25	
40	4.44	4.208	999.8	1.55	0.575	11.35	1.91×10^{9}
50	10	4.195	999.2	1.31	0.585	9.40	6.34×10^{9}
60	15.56	4.186	998.6	1.12	0.595	7.88	1.08×10^{10}
70	21.11	4.179	997.4	9.8×10^{-4}	0.604	6.78	1.46×10^{10}
80	26.67	4.179	995.8	8.6	0.614	5.85	1.91×10^{10}
90	32.22	4.174	994.9	7.65	0.623	5.12	2.48×10^{10}
100	37.78	4.174	993.0	6.82	0.630	4.53	3.3×10^{10}
110	43.33	4.174	990.6	6.16	0.637	4.04	4.19×10^{10}
120	48.89	4.174	988.8	5.62	0.644	3.64	4.89×10^{10}
130	54.44	4.179	985.7	5.13	0.649	3.30	5.66×10^{10}
140	60	4.179	983.3	4.71	0.654	3.01	6.48×10^{10}
150	65.55	4.183	980.3	4.3	0.659	2.73	7.62×10^{10}
160	71.11	4.186	977.3	4.01	0.665	2.53	8.84×10^{10}
170	76.67	4.191	973.7	3.72	0.668	2.33	9.85×10^{10}
180	82.22	4.195	970.2	3.47	0.673	2.16	1.09×10^{11}
190	87.78	4.199	966.7	3.27	0.675	2.03	
200	93.33	4.204	963.2	3.06	0.678	1.90	
220	104.4	4.216	955.1	2.67	0.684	1.66	
240	115.6	4.229	946.7	2.44	0.685	1.51	
260	126.7	4.250	937.2	2.19	0.685	1.36	
280	137.8	4.271	928.1	1.98	0.685	1.24	
300	148.9	4.296	918.0	1.86	0.684	1.17	
350	176.7	4.371	890.4	1.57	0.677	1.02	
400	204.4	4.467	859.4	1.36	0.665	1.00	
450	232.2	4.585	825.7	1.20	0.646	0.85	
500	260	4.731	785.2	1.07	0.616	0.83	

DESIGN FOR THE FOULING FACTOR

Type of fluid	Fouling factor, $h \cdot ft^2 \cdot °F/Btu$	$m^2 \cdot °C/W$
Seawater, below 125°F	0.0005	0.00009
Above 125°F	0.001	0.002
Treated boiler feedwater above 125°F	0.001	0.0002
Fuel oil	0.005	0.0009
Quenching oil	0.004	0.0007
Alcohol vapors	0.0005	0.00009
Steam, non-oil-bearing	0.0005	0.00009
Industrial air	0.002	0.0004
Refrigerating liquid	0.001	0.0002

PARAMETERS EXTRACTED FROM THE GENERATOR MANUAL

Intake & Exhaust System

Combustion Air Consumption	m^3/min	27
Max. Intake Restriction	KPa	5
Exhaust Temperature (Before Turbo)	°C	660
Exhaust Temperature (After Turbo)	°C	510
Max. Exhaust Back Pressure	Kpa	5
Exhaust Gas Flow	m^3/min	33
Turbo Bellows Diameter	mm	DN100-150
Exhaust Flange Diameter	mm	DN150

a – local variable for equation 7

a_s – shell flow area, ft^2

a_t – tube flow area, ft^2

A – area for heat transfer, ft^2

b – local variable for equation 7

B – baffle spacing, ft

c' – shell clearance, ft

Cp_s – shell side heat capacity, $\frac{Btu}{lbm \cdot {}^\circ F}$

Cp_t – tube side heat capacity, $\frac{Btu}{lbm \cdot {}^\circ F}$

d_i – inner tube diameter, ft

d_o – outer tube diameter, ft

De – equivalent shell diameter, ft

D_s – shell diameter, ft

F – correction factor for 1 shell pass and 2 tube passes

h_i – tube side convective heat transfer coefficient, $\frac{Btu}{hr \cdot ft^2 \cdot {}^\circ F}$

h_o – shell side convective heat transfer coefficient, $\frac{Btu}{hr \cdot ft^2 \cdot {}^\circ F}$

k – thermal conductivity of tube material, $\frac{Btu}{hr \cdot ft \cdot {}^\circ F}$

L – tube length, ft

$\dot{m}_s$ – shell mass flow rate, $\frac{lbm}{s}$

$\dot{m}_t$ – tube mass flow rate, $\frac{lbm}{s}$

n_t – number of tubes

N_p – number of tube passes

Nu_{shell} – shell side Nusselt number

Nu_{tube} – tube side Nusselt number

P – local variable for equation 7

Pr_{shell} – shell side Prandlt number

Pr_{tube} – tube side Prandlt number

P_T – tube pitch, ft