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Design of Heat Exchanger for the Gas Turbine Air Inlet Cooling

by

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Dissertation submitted in partial fulfillment of
the requirements for the
Bachelor of Engineering (Hons)
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CERTIFICATION OF APPROVAL

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Approved by,

(Dr. Zainal Ambri Bin Abdul Karim)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
May 2011

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

MOHD NORSHAHRUL HAFIZI MOHD AZMI

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In the name of ALLAH S.W.T, the most merciful and compassionate, praise to ALLAH, he is the almighty, eternal blessing and peace upon the Glory of the Universe, our beloved Prophet Muhammad (S.A.W), and his family and companions.

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ABSTRACT

Efficiency of a gas turbine is closely related to the intake air temperature. An increase in the intake air temperature will results in a decreasing of the turbine efficiency. This report basically discusses the results from the research done and basic understanding of the chosen topic, which is design of a heat exchanger for gas turbine inlet air cooling system. The project aims at providing a constant air temperature at the inlet side of the gas turbine irrespective of the change in ambient temperature by employment of a heat exchanger to reduce the temperature at constant low temperature. In this paper, focus is given to the best fin geometry for plate and fin heat exchanger in order to improve the efficiency of the gas turbine. This paper also discussed about the selection of a few parameters based on given standard rating in designing high performance plate and fin heat exchanger with high transfer rate of heat and a few numerical solution to find the suitable sizing of heat exchanger for gas turbine at Gas District Cooling area. A clear methodology to complete this project was also proposed in this paper. As a result from this research, it was said to achieve the objectives made with satisfied parameters that can be applied to the real world problem to reduce the air temperature. The results were done with the helped of assumptions made and with set of analytical equations to identify the parameters desired. Fin surface geometry of 1/8 – 16.00 (D) can be use for air side while as for the water side, geometry of 1/8 – 19.82 (D) can be use. Both of the fin surface geometry was tested and proved to reduce the air and water temperature and yield a satisfied heat transfer coefficient, as well as efficiency.

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CHAPTER 1
INTRODUCTION

1.1 Gas Turbine

Gas turbine is used to provide utility in term of electricity and in certain industry type, it is used as part of a system to provide steam to the costumers. As to many of the power production and oil refinery company, gas turbine plays an important role in economic growth of the companies. If the systems by means of external factors degrade the performance of the gas turbine, it would mean that the output that is being produced by the gas turbine will steadily decreases with time. During sunny day or high ambient air temperature, costumers demand for more production of electricity due to increment usage of air conditioning unit, which if the gas turbine is working with rated performance, high demands will be an opportunity in growing the economics of the company. But with lower efficiency of gas turbine, higher cost will be needed to provide the fuel consumption just to produce approximately the same quantity of output in comparison to the higher efficiency of gas turbine.

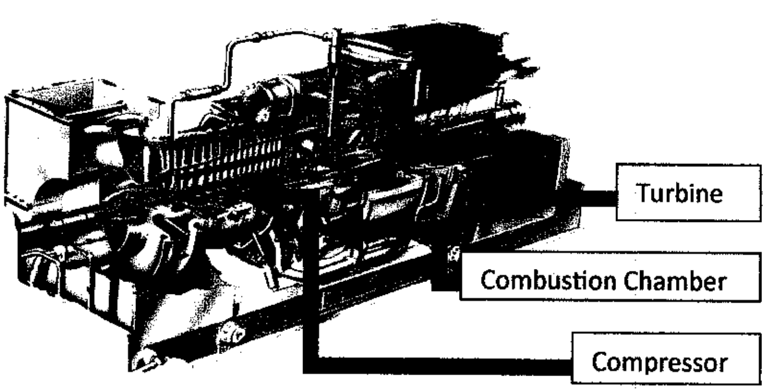

Figure 1.1: General Structure of Gas Turbine [18]

Malaysia is a tropical, warm country, with averages temperature of the surroundings in the range of 27 to 36°C during sunny day. As to relate to the power output that a gas turbine can produce, one of the common and unattractive characteristic of all the gas turbines is that their power outputs decreases as the inlet ambient air temperature increases.

A gas turbine is consider to be operated in low efficiency in terms of its working condition as a gas turbine can be assume to contain a considerable "hidden capacity". The capacity is the performance of a gas turbine by conducting analysis of the output produce by the gas turbine. As for the economics, a company can generate more profits out the plant if the initiative to improve the efficiency of the gas turbine is conducted.

1.2 Problem Statement

Gas turbine is design to handle certain fix volumetric flow rate of air according to the size of gas turbine. Even though the volumetric capacity of compressor is fixed, the mass flow rate of air it delivers to gas turbine changes with the change of climatic condition. With the increase in ambient air temperature, the density of air will decrease which will also degrades the mass flow rate of air. A lower mass flow rate of air will results in lower power output of the gas turbine due to higher specific compressor work and eventually causes higher specific heat consumption.

Based on ISO (International Standards Organization) the rated capacities of all gas turbines are based on the standard ambient condition of 15°C, pressure at sea level with 60% humidity of air. Higher ambient temperature will deteriorates in terms of power output and efficiency of gas turbine, especially during high demand of electricity which is unfortunate because the efficiency is highly dependable to the climatic condition.
There are a few steps that can be used to counter the problem occurred at the gas turbine, and one of the methods has been identified that can influence the performance of the gas turbine which is by installing cooler at the inlet compressor of the gas turbine.

1.3 Objectives

The main goal of this research is to propose the suitable method to be implementing to the current system at Gas District Cooling site in order to increase and maintain the high efficiency of gas turbine especially during high ambient air temperature. In achieving the main goal, the research also attempt to accomplish a list of following objectives:

1. To suggest a suitable air inlet chiller to enhance the performance of gas turbine.
2. To develop a heat transfer model of the new cooler system.
3. To develop a spread sheet for mathematical modeling.
4. To construct an analytical equation and solid modeling of the system using Catia V5

1.4 Scope of Study

The purpose of this study is because of the possibility that arises in terms of gas turbine enhancement of efficiency and performance, by cooling the air that enters the gas turbine prior to the production of electricity by generator. The methods studied are then compared to find the suitable solution with the current problem which will then implemented to the current system of gas turbine. This thesis is divided into two parts of studies, through Final Year 1 and Final Year 2. For the first part of the studies, it was conducted by 1) Reviewing the thesis in application area for design of air cooling system for gas turbine; 2) Proposing analytical equation. The later part of the studies then continued to second part of Final Year Project and the studies continued by 3) Completing the analytical equation by doing some comparison to different types of fin geometrics; 4) Constructing solid modeling for the optimized gas turbine system using the high performance fin calculated; 5) Completed the project with the development of heat transfer modeling of the selected fin at desired parameters.

CHAPTER 2
LITERATURE REVIEW

2.1 Design of Heat Exchanger for Gas Turbine Air Inlet Cooling System.

Deficiency in performance during the process can cause a significant loss of power production by the gas turbine. Deficiency by means of lower performance below the designated value will cause a counter productivity in terms of economics for the company if the problem is not being solved. Several external factors are believe to cause such problem which are high ambient air temperature, quality of the fuel for combustion, external deposits on the blades of the compressor and turbine and many others. Focusing on one of the main factor that is the ambient air temperature, a gas turbine loses approximately 7% of its nominal power when the intake temperature increases from 15°C, ISO conditions, when in summer, the ambient temperature increases above the 25°C, the losses are still bigger, reaching even 15% of the power rating with 36°C [1]. In the present paper, improvement of the gas turbine performance is investigated. It is hypothesized that by implementing a cooling system prior to the compressor will significantly improve the gas turbine performance in terms of production of useful power output from the power generation production process.

In a research article [2], three elements which are ambient temperature, pressure and humidity are important factors which determine the performance of an operating gas turbine. These facts are also being supported by a parametric study done on the effect of ambient air, pressure, humidity and gas turbine inlet air temperature and also thermal efficiency [3], where the researcher stated that, the ambient air temperature has the greatest effect on the efficiency of the gas turbine which will increase the power production rate and also compressor ratio of the turbine. A comprehensive study was conducted at one of the refinery gas turbine by using the refinery natural gas pressure drop station to be able to capture the cooling ability created [4]. Natural-gas properties, flow rate at the pressure-drop station, as well as exergy and energy analysis, amount of available works and cooling capacity could be calculate. The result of this study support the criteria mention it the articles where the ambient temperature, pressure and humidity are important factors which manipulate the efficiency of operated gas turbine. Each of the criteria which are quantity of mass flow rate and inlet temperature of pressure-drop station was compared with other gas turbine inside Khangiran refinery and each was then constructed with a modeling in relation to the time. From the modeling, the significant different can be seen where, during the winter season, where the temperature drop significantly for the gas turbine without pre-heater, while thermal efficiency can be considered as considerably high efficiency for gas turbine with cooling and pre-heating.

Next, the topic of attachment in relation to Design of Heat Exchanger for Air Inlet Cooling system is address according to type of combustion turbine inlet cooling system. In a research article [1], the introduction of combustion turbine air inlet cooling to the gas turbine system was implemented and studied. The studies focusing on method to decrease the inlet temperature in order as air inlet temperature are is main key that determine the efficiency of operated gas turbine. Extreme decreasing of air inlet temperature below 6°C - 8°C should be avoid in practice to avoid ice crystallization at the intake compartment in which will produce erosion and wear at the intake vane. The study concluded a few methods which implemented technology into account to decrease the air inlet temperature. Evaporative cooling, Absorption Chiller and also Thermal Energy Storage system (TES) are a few which is being proposed in the studies. All of the methods were proposed based on fundamental understanding of heat transfer process.

It was reported that, a media cooler can increase the relative humidity of ambient air to about 90% from the norm and so will increase the power output by 5 – 10% and the efficiency by 1.5 – 2.5% [5]. Evaporative cooling system used a direct interaction between the air and particles of water and evolves with the compressor intake flow stream. Evaporative cooling is capable of cooling the air up to 85 – 90% saturation. In the system, the evolution of fine water drops can produce and increase of extra power. In addition to increasing the incoming mass flow, evaporation of drops will absorb latent heat, reducing the temperature of the mass that passes through the compressor, reducing the compression work that depends on the intake flow temperature. The drawback of the method is that it is highly dependable to the ambient condition and will cause pressure drop during the operation [1], and according to study prior to the need of water for evaporative cooling for gas turbine in Saudi Arabia, the cooling system used up to 12 655 ton of water for 40MWatt of gas turbine annually [5]. Another method is by using an absorption chiller, using a refrigerant cooling to cool down the air inlet. The working fluids might be water, lithium bromide or ammonia. Although ammonia presents better properties of refrigerant than water, but due to toxicity, water is preferable to be use in application [6]. Evaluation of combustion turbine inlet air cooling system, lithium bromide is used as a suitable refrigerant for cooling. The benefit of the system is that, it was reported that, inlet air temperature can goes lower than 10°C which will give benefit to the system to achieve higher efficiency of the system [6].The result of the study indicates that, the temperature of the inlet air can be cooled down to achieve the desired value which should be higher than freezing point at 4°C. The improvement of the inlet air temperature according to the methods proposed, will help to increase up to 27% of the power output if the inlet temperature was cooled down to 4°C [7]. The improvement based on the method proposed will need a capital cost which in certain case, the modification to the gas turbine should be done prior to suits the inlet cooling systems. With the increment of efficiency, useful power output will increase and will directly increase the annual revenue of the power plant and oil refinery companies.

Finally, the topic attachment in relation to the Design of Heat Exchanger for Air Inlet Cooling system is addressed using a selection of a suitable heat exchanger based on advantage and disadvantage of each method. In a research article [8], an evaluation study has been extendedly conducted to obtain a generic comparisons result of different inlet cooling technologies using analytic method. The main focuses of the cooling technologies were on absorption chillers and also saturated evaporative coolers (inlet fogging). The study was conducted by taken into account the off-design performance of gas turbine combine cycle and annual average climatic data without using the historical data of the process operation. The result indicate that gas turbine inlet cooling with absorption chiller is better than inlet fogging in terms of efficiency ratio at climatic condition of relative humidity bigger than 0.4 and ambient temperature higher than 25°C. But, in terms of climatic condition of ambient temperature decreasing, method using inlet fogging begins to be higher than the inlet chilling, inlet fogging also was proved to be superior to the inlet chilling in the temperature in range of 15°C - 20°C [8].

2.2 Evaluation and Selection of Heat Exchanger.

Based on articles which discuss the usage of coolers to be implemented into the designated process, a heat exchanger that is used for transfer of thermal energy (enthalpy) between two or more fluids, between solid surface and fluid, or between solid particulates and a fluid, at different temperature and in thermal contact, usually with external heat and work interactions. In most of heat exchanger, the fluids are separated by a heat transfer surface and ideally they are not mix with each other. These types of cooling devices are referred as indirect transfer type. In contrast to the availability of heat transfer surface which separate the fluids they are commonly referred as direct transfer or known as regenerators. Heat exchangers can be classified according to transfer process, construction, flow arrangement, surface compactness, number of fluids and heat transfer mechanism or according to process functions. Based on article [1], installation of combustion turbine inlet air cooling systems in operating plants presents some associated inconveniences, as they require investment in equipment and maintenance costs. In some cases, it even requires remodelling of some gas turbine sections. Besides, the equipments installed at the turbine air intake sections create additional pressure drops that will depend on the Combustion Turbine Inlet Air Cooling (CTIAC) technology installed. Also, the profitability of introducing a CTIAC system depends on many external factors that are not controllable, like climatology, electricity tariffs and fuel price. So, it requires a particular viability analysis for each power plant. Based on current trend in energy conservation and optimization, CTIAC system can be group into a few main categories which are Evaporative Coolers, Mechanical Cooling Compression (vapor compression), Absorption Chillers and also Hybrid Systems.

Evaporative Coolers based on article [1] used water evaporation as it evaporated after it absorb latent heat from the incoming air, and thus reduce the dry bulb temperature. The minimum temperature of cooled air is limited to 10°C to avoid the risk of freezing at inlet air compartment. They are capable of cooling air to within 85-90% saturation. In addition to increasing the incoming mass flow, evaporation of drops will absorb latent heat, reducing the temperature of the mass that passes through the compressor, reducing the compression work that depends on the intake flow temperature.

Absorption Chillers is a type where the cooling effect is driven by hot water. The heat source may come from any number of industrial sources including waste heat from the industrial process. The primary heat sources come from solar thermal installation or from exhaust of a gas turbine. Based on article [1], double effect LiBr–H2O absorption refrigeration cycle improves the performance of the single effect cycle. Also, different heat sources have been considered, direct heat source in an auxiliary boiler or taking advantage of the residual heat of the plant, for the single effect cycle with hot water at 115°C and for the double effect cycle, medium pressure steam (8.3–9.86 a.t.m). However, the energy requirement for heating process is large for single effect cycle and double the capacity for double effect cycle. Even for absorption chillers consume large energy for heating purposed, from energy-efficiency point-of-view, absorption chiller is optimum when the heat source is cheap and easy to be get.

Mechanical Cooling Compression [1] used classical mechanical compression cycle, with an Intermediate refrigerant fluid, which cools the intake air passing through heat exchangers located in the gas turbine inlet. These systems can produce a stronger cooling effect than the evaporative systems, cooling the air down to 7.22°C. Compact heat exchanger also known as fin-plate heat exchanger is one of heat exchanger which applied this type of cooling process. The structure is consists of a stack of alternate flat plates called parting sheet by flowing along the passages made by fins brazed together as a block parting sheets [9]. The fin serves both as a secondary heat transfer surface and mechanical support for the internal pressure between layers. The process occur between two flow stream of cold (cooler) and hot fluids by transferring heat through alternating layer which separating the two flow stream.

In design and selection of heat exchanger based on article [1], there are three types of flow arrangements which are counter-flow, parallel-flow, and cross-flow. In the counter-flow heat exchanger, both fluids entered the exchanger from opposite sides. In the parallel-flow heat exchanger, the fluids come in from the same end and move parallel to each other as they flow to the other side. The cross-flow heat exchanger moves the fluids in a perpendicular fashion. Compare to other flow arrangements counter–flow is the most efficient design because it transfers the greatest amount of heat.

In design selection of a heat exchanger, there are a few elements which needed to be specified first before the process proceeds to a more details specification which are, operating temperature which usually set by process conditions and it is dictated by material considerations, safety, economics and ASME Code requirements. Another one is effective temperature different between two fluids which is calculated from counter current log mean temperature with a correction factor applied to account for the actual flow arrangement [1]. Fouling factor also is taken into account in design selection of a heat exchanger. Fouling in engineering terms means that, a resistant to both heat transfer and fluid flow that are usually cause by deposits on a heat transfer surface. Pressure drop also is set as an element in heat exchanger selection. This is because when pressure drop happen inside a heat exchanger, based on thermodynamic law, the specific volume of air will also affected and which decrease the mass flow rate of air inside heat exchanger.
The mentioned specifications are applied to all of heat exchanger. In application of the usage of fin plate heat exchanger, additional requirement and specification are still in need in design selection in finding a suitable plate and fin heat exchanger for the plant process. One of them is surface selection of parting sheet. The objective in specifying surface selection of the plate and fin heat exchanger is to produce the smallest weight or volume by using a simple technique which has been developed referred as volumetric performance index.

In article [9], the study has been done extensively in finding the high performance surface. The studies compared two types of surface which were plain type fin and also louvered type fin in objective to find the right sizing of plate and fin heat exchanger using a new algorithm with the best solution of surface selection. The studies which focusing on the numerical analysis using the equation and theoretical understand to create a new algorithm which later compared with the default value. Based on the numerical solution, a few parameters have to be stated first which are pressure drop, heat transfer coefficient and also Reynolds Number of a given flow stream through the heat exchanger before the methodology proposed can proceed in finding the right sizing for heat exchanger with minimize dimension within constraints of block dimensions [9].

Next, the topic of attachment in relation to evaluation and selection of heat exchanger is address according to various aspects characteristic which affect the heat transfer and fluid flow of the heat exchanger, which based on heat transfer plate [10]. Named of a few major heat transfer that widely used in industrial process which are wavy fin and also offset fin. In one of the research, the author has included studies by researchers on the matter of the non-uniformity analysis in the compact heat exchanger and also the methodology applied in finding the solution to reduce the non uniformity of fluid flow. As for offset and strip plates, the fin thickness, one of the triggering factor for the increasing value of pressure drop in the plate and fin heat exchanger [11]. One of the journals reviewed in the article, the effect of larger fin thickness in offset fin thickness has been reported in Kays [13], Patankar and Prakash [14] and also Cur and Sparrow [15] have observed an increment of up to 25% increases in pressure drop for the fin thickness of 0.01 inch as compared to 0.006 inch. The increment of fin thickness also has been observed to affect the Reynolds Number which increased the Reynolds Number in other words also changing the transition flow from laminar to turbulent flow. As for wavy plates, the author has discussed about the heat transfer characteristic with comparison to the parallel-plate. In range of 1500 – 25,000 of Reynolds Number flow, the enhancement of heat transfer as compared to parallel-plate was about a factor of 2.5. Even in lower Reynolds Number of 1000, wavy fin plate has been observed to have a high performance factor which about 2.2. But the value for performance factor decreased as lower Reynolds Number encountered. As for non-uniformity flow which usually developed at the inlet of compact heat exchanger, using software like Computational Fluids Dynamics (CFD) and also Particle Image Velocimetry (PIV) for quantification of flow maldistribution of the fluid, Seetharamu, Ranganayakulu et al. have proved that with implementation of baffle plate, the flow maldistribution has been drastically reduce. Wen et al. has improved the findings by stating that, punched baffle could effectively improve fluid flow distribution in the header. A few other findings which could improve fluid flow distribution are second header installation, modification of heat exchanger inner hot side inlets and outlet header in reducing the pressure drop and heat exchanger effectiveness.

Finally, the topic attachment in relation to design selection of heat exchanger for air inlet cooling system is addressed in a research article [12], by conducting experimental studies on the air-side heat transfer and pressure drop characteristic for 16 types of offset strips fins and flat tube heat exchanger. The research focusing on the general correlation of Colburn j-factor and Fanning friction f-factor which were developed by regression analysis F significance test which can predict 95% and 90% of the experimental data within ±10%. In comparison with the test data, the average deviations of the present correlations for j and f factors are 0.19% and 1.2%, and mean deviations are 4.2% and 5.3%. The correlations can be used over a wide variety of offset strip fins configurations and wide Reynolds region, Re = 500–7500, especially for the heat exchangers with shorter flow length. The performance data were analyzed using the effectiveness-NTU method. As a result from the extensive studies, a few parameters were found which affect the performance for the heat exchanger which are, fin thickness, fin space, fin height, fin length and flow length. From the finding of the affecting factors, the heat coefficient and pressure drop were found to increase with the adjustment of the parameters.

CHAPTER 3
PLANT CONFIGURATION

3.1 Introduction

The profile of the Gas District Cooling (GDC) cogeneration plant situated at Universiti Teknologi PETRONAS which is used for the research is explained in this chapter. The plant situated at the back end of Universiti Teknologi PETRONAS operated at maximum of 8.4 MW is one of three Gas District Cooling operated by Gas District Cooling (Putrajaya) Sdn Bhd which is wholly owned by PETRONAS. System of the plant is described and the study focuses mainly on the heat exchanger system.

3.2 Description of the plant

Cogeneration plant is a plant that uses heat engine or power station to simultaneously generate both electricity and useful heat, as in the case of Gas District Cooling plant, the heat is use to generate electricity and chilled water for space cooling in UTP. GDC comprises of two (2) gas turbines, heat recovery steam generator (HRSG), two (2) steam absorption chiller, (4) air cooled chiller, one (1) cooling tower and one (1) thermal energy storage (TES). The function for each of the component is briefly explain as follow; **1)** Gas Turbine drive the electric generator by converting the kinetic energy to electrical energy for generation of electricity; **2)** Heat recovery steam generator (HRSG) utilizes the exhaust heat from the gas turbine by recovering the heat to produce steam for other process, in this case for heating purpose in the steam absorption chiller. At GDC, two system is used, one for each gas turbine; **3)** Steam absorption chiller is use to provide air cooling (conditioning) for the buildings in UTP by producing chilled water later being direct to the buildings, the plant is consists of two units of steam absorption chillers; **4)** Four unit of air cooled electric chillers are used to produce chilled water for space cooling in UTP and to change the thermal energy storage **5)** Cooling tower provides a cooling space to cool down the recycling water from the steam absorption chiller; **6)** Thermal energy storage provide a function to store chilled water produced into tank for later use during peak hour/period. Schematic diagram of process flow of gas turbine system is shown in the Figure 3.1

3.3 Gas Turbine Engine.

Gas Turbine System is designed for the generation of electricity and also part of the system to produce steam in HRSG process system. It is a type of TAURUS 60-7301S power generation, single shaft equipped with axial compressor, combustion system and turbine. Operated using natural gas and diesel, it is consists of five main components which are as follow

Table 3.1: Components inside a Gas Turbine

Compartment	Function
Air Inlet System	Provide filtered air for compressor process, and reduces air pressure variations and turbulence at the compressor inlet.
Compressor	Increase air pressure to approximately 10 bars by compression
Combustion System	Direct compressed air to the combustion chambers, Disperse and mix fuel with the proper amount of combustion air and ignite the mixture.
Turbine	Convert the high pressure and temperature gases produced by the combustion of gas fuel, into mechanical energy and drives the compressor and the gas turbine generator.
Exhaust	Redirect combustion product (through silencers) either to atmosphere or to the HRSG inlet duct.

Table 3.2: Taurus 60-7301S Gas Turbine Engine Specification

Description	Data
COMPRESSOR	
Type	Axial
Number of Stages	12
Compression Ratio	12:1
Speed	14944 rpm
COMBUSTION CHAMBER	
Type	Annular
Ignition	Torch
Number of Fuel Injectors	12
TURBINE	
Type	Axial
Number of Stages	3

3.4 Gas Turbine Operation

Gas turbine consists of a few main components which act based on sequence of each other where the air will enter the duct of air inlet system to filter out dust or any external particles that it may bring with it. Later the filtered air will be compressed through twelve stages of compression generating high pressure air for the later process. After that, it will be directed into a number of combustion chamber and the mixing of compressed air with the fuel helped with the spark plug will combusted. The combustion product will then leave the combustion chamber at high pressure and temperature and flow into the turbine section. The expansion of the combustion product will convert the kinetic energy stored from the high velocity of the combustion into mechanical energy by rotating the turbine blades and thus, generating electrical energy stored done by the generator which connected to the gas turbine shaft. The detailed operation is discussed below. Figure 3.1 describes the operation of gas turbine as follows:

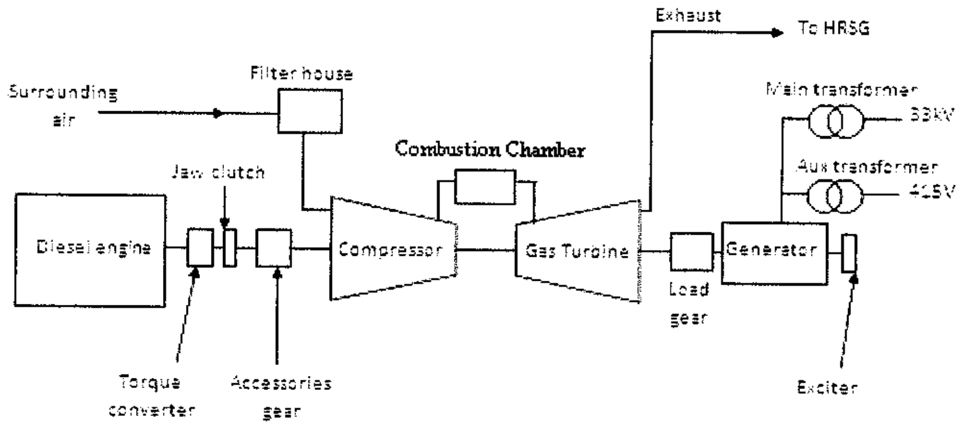

Figure 3.1: General Operation of Gas Turbine System [18]

3.4.1 Air Inlet System

Air inlet system is used to protect the blading of the gas turbine from dust, fumes and salt which could corrode or form deposits on the compressor and turbine blading and reduce their efficiency. It consists of an integrated self-cleaning filter house and support structure, a cleaning compressed air heat exchanger, inlet ducting and silencing sections. Filter unit inside the filter house consists of a single stage, self cleaning system utilizing cylindrical and conical filter cartridges sequentially cleaned by reverse flow pulses of pressurized air from the compressor fifth stage extraction.

3.4.2 Compressor

Compressor compartment is an axial flow compressor where it consists of seventeen stages. The compressor compartment consists of 3 major components which are rotor, compressor casing (includes inlet casing, compressor casing and discharge casing) and also compressor blading. As for the sub division inside a compressor compartment, rotor is an assembly of fifteen wheels, two stubshafts, through bolts, and the compressor rotor blades.

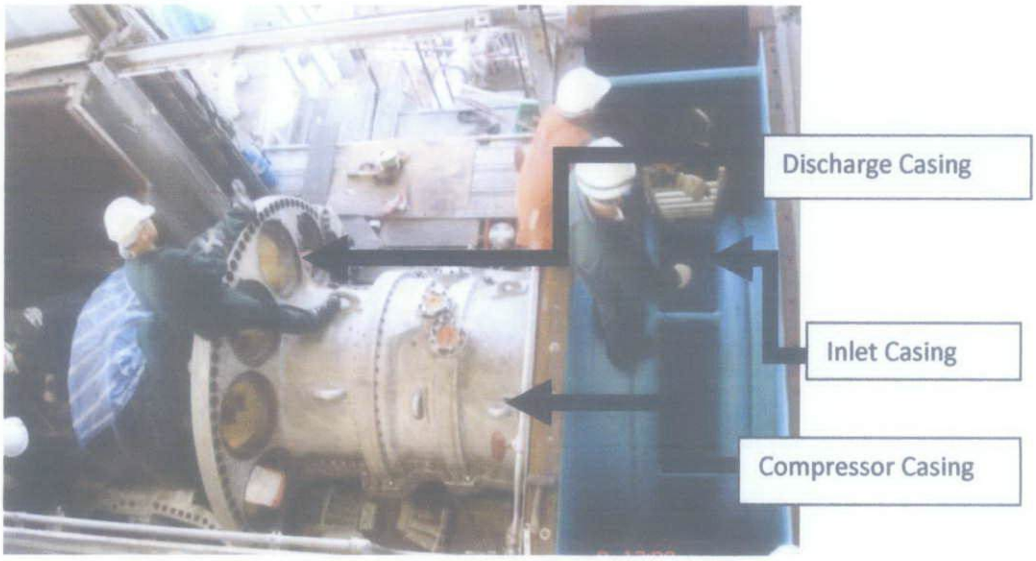

Fig.3.2: Compressor Compartment [20]

While for compressor blading, they are airfoil-shaped and were designed to compress air efficiently at high blade tip velocities. The blades are attached to their wheels by dovetail arrangements. The dovetail is very precise in size and position so as to maintain each blade in the desired position and location on the wheel.

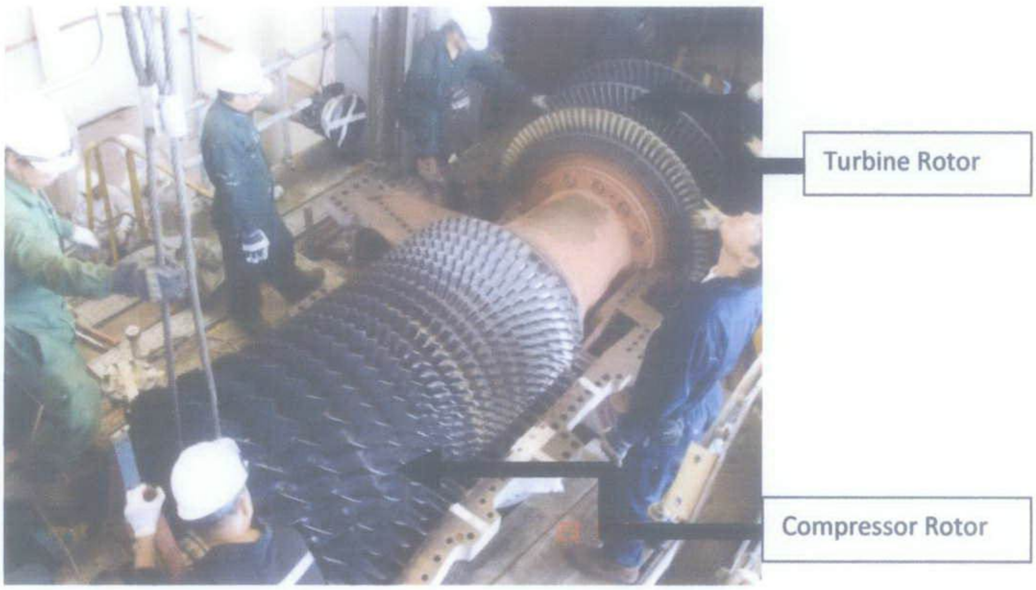

Fig.3.3: Location of Compressor rotor and turbine rotor. [20]

3.4.3 Combustion Chamber

As for combustion compartment, it is actually consists of a few components which act as a chamber for combustion to occur. It consists of twelve combustion cans that are arranged around the periphery of the compressor compartment. The twelve combustion chambers are identical except for those fitted with spark plugs (1 and 10), or flame detectors (2, 3, 7 and 8). After the combustion happened inside the cans, it will then transfer to the first stage turbine bucket one of main components inside a turbine compartment through transition piece in between combustion chamber and turbine bucket first stage.

3.4.4 Turbine

Turbine compartment is where the kinetic energy of the combustion product is converted into mechanical energy by passing the high pressure combustion gas through sets of blades inside the turbine casing. As much as 60% of gas turbine power generation is used by axial-flow compressor. The major components of the turbine section include the rotor assembly (rotating elements) and the stator assembly (stationary elements).

The information explained above is important as to describe the actual performance of gas turbine engine where as compare to ideal, the actual gas turbine needs more energy to generate power electricity due to many parameters which includes air surrounding temperature as well as performance of each component inside a gas turbine engine.

CHAPTER 4
HEAT EXCHANGER SYSTEM

4.1 Introduction

Based on the literature review and needs of heat exchanger for the application to reduce air inlet temperature of the gas turbine, the author proposed the using of plate and fin heat exchanger (extended surface). This chapter introduces the plate and fin heat exchanger and discussion regarding the fin geometries that are available and also flow friction and heat transfer characteristic. This chapter also discusses about the merits and drawbacks of the plate and fin heat exchangers with regards to other types of heat exchangers that are available in the market.

4.2 Plate and Fin Heat Exchanger

A plate and fin heat exchanger is a type under the category of compact heat exchanger, which consists of a number of stacking alternative layers of corrugated fins and flat separators known as parting sheets. Figure 4.1 below describes the components of the plate and fin heat exchanger.

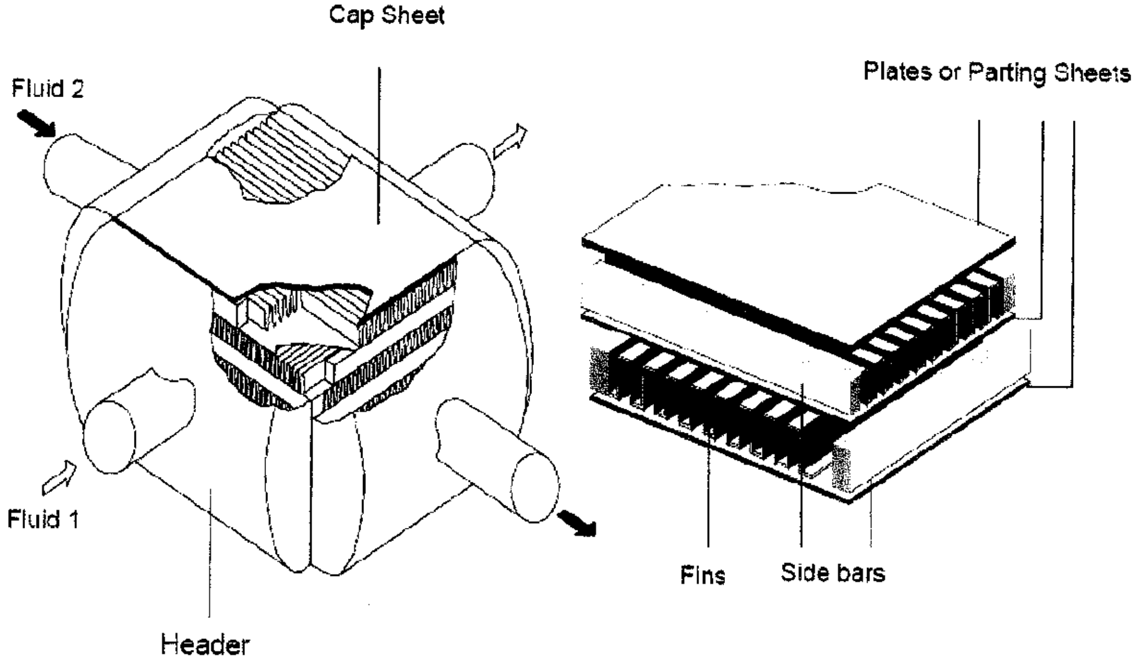

Figure 4.1 Plate and Fin Heat Exchanger assembly and components [16]

The working principle of a heat exchanger is based on heat equilibrium between the coolants of heat exchanger with working fluid that mend to be decrease in temperature. The heat exchange process occurred along the flow line of the passage corrugations between the parting sheets. Each stream of the fluid will enter from its own header through the ports in the side bars of appropriate layers and these streams will leave in similar fashion without mixing inside the heat exchanger body. The edges of corrugated layers are sealed and the corrugations and its side bars are brazed to the parting sheets on both sides to produce rigid, strong pressure-containing voids. Cap sheets as per above details, is an outer layer of the plate and fin heat exchanger, it usually is thicker than parting sheets due to reason for external protection from physical disturbance and it is also use to provide support from excessive pressure.

4.3 Advantages and Drawbacks.

4.3.1 Advantages

Application of plate and fin heat exchanger provides several advantages if compared to other designs. [21]

1) High thermal efficiency and close (Temperature approach between single phase fluid streams as low as 3K and between condensed and boiling fluids 1K is fairly common).
2) Large heat transfer surface area per unit volume (Typically 1000 m^2/m^3).
3) Easy for transportation and installation due to low weight if compared to other design heat exchanger.
4) Can be built up in multiple-stream operations (Up to ten stream can exchange heat in a single heat exchanger).
5) Pure counter-flow operation (Shell and tube heat exchanger use the principle of mixture of cross and counter flow).

4.3.2 Disadvantages

1) Limited range of temperature and pressure operation parameters.
2) Due to its compact design, the maintenance will be difficult especially to clean the inside of the plate and fin heat exchanger.
3) If broken or needs repair, it will take sometime to repair the leakages or defects.

4.4 Flow Arrangement

A plate and fin heat exchanger accept 2 or more stream flow for heat exchange process which may flow in the similar directions or in counter direction to one another or one of the stream can flow in perpendicular direction or in other terms as cross flow. So, in the plate and fin heat exchanger, there are three types of flow; **1)** Counter Flow; **2)** Cross Flow; **3)** Parallel flow. Based on thermodynamic principle, counter flow provides with the highest heat (or cold) recover, while parallel flow provide lowest heat (or cold) recovery compared to other flow arrangement. While cross flow provide an intermediate solution, it provides with the easy mechanical installment and superior heat transfer properties. Under certain condition, hybrid of cross-counter flow arrangement give a greater heat (or cold) recovery with higher heat transfer performance

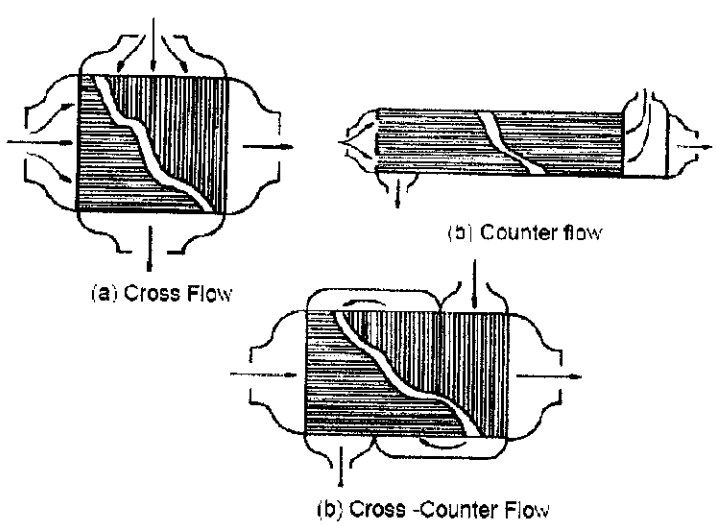

Figure 4.2: Heat Exchanger Flow Arrangements [19]

4.4.1 Cross Flow

Usually for this flow arrangement, only two streams are operated by means eliminating the needs for distributors. As the header tanks are located at all four sides of heat exchanger core, the result of the applications reduce in complexity and cheaper cost. In a condition of high effectiveness is not necessary or two streams are widely differing volume flow rates and also if either one or both streams are nearly in isothermal condition, the cross flow is preferred. Common applications include aircraft heat exchangers and automotive radiators.

4.4.2 Counter Flow

This flow arrangement provides higher thermal efficiency for the used of heat recovery or cold from the process stream. This flow arrangement needs proper design due to the complexity of geometry of the headers and the distributor channels. Typical applications include cryogenic refrigeration and liquefaction equipment.

4.4.3 Cross-Counter Flow

Cross-counter flow is hybrid geometry of cross and counters flow arrangements. The benefits of the hybrid arrangement provide the thermal effectiveness of counter flow properties with high heat transfer characteristic of cross flow configuration. In the flow arrangement, one of the streams flows in a straight path while the second stream follows a zigzag path normal to the first stream. While flowing through such path, the fluid stream covers the length of the heat exchanger in a direction opposite to direct stream. Cross-counter flow usually applied to the field similar as the cross flow arrangements but with more flexibility in design. In application, they are particularly suitable for the applications where two streams have characteristics of considerable different volume flow rates or permits significant different pressure drops. With the application of hybrid flow arrangement, the overall geometry of the heat exchanger can be optimized.

4.5 Fin Geometries

Another parameter that needs to be considered in application of plate and fin heat exchanger is the geometries of the fins. Common industries usages of the fin configurations include **1)** Plain (straight and uninterrupted) fin with rectangular, triangular and trapezoidal passages; **2)** Uninterrupted wavy fins; **3)** Interrupted wavy fins (Offset strip, Perforated, Louvered, and Pin fins);

4.5.1 Plain Fins

Plain fins are straight fins that continuous in the fluid flow direction. Although passages of triangular and rectangular cross section are more commonly use in industries but any desired shape can be given to the fins with only consideration of manufacturing constraints. Although they are cheaper than rectangular fins due to the faster and easier to make in manufacturing, but the structure are weaker if compared to rectangular geometry for same characteristic of passage size and thickness. Furthermore, it is lower in heat transfer performance especially in laminar flow. Usually, plain fin is use in application where the pressure drop is critical and requires a smaller flow frontal area compared to the interrupted fins for particular pressure drop, mass flow rate and heat transfer.

4.5.2 Wavy Fins

Wavy fins are characterized by uninterrupted fins surfaces with similar cross-sectional area similar to plain fins, but with cyclic lateral shifts perpendicular to the flow direction. The flow fin results in more effective interruptions and induces a complex flow field. Heat transfer is enhanced due to the creation of Goertler vortices. The counter-rotating vortices form while the fluid passes over the concave wave surfaces and produce a corkscrew-like flow pattern. The characteristic of heat transfer and pressure drop of a wavy fin surface can be identified to be between plain and offset strip fins. The friction factor is inversely related to the Reynolds number where, in the fin, the friction factor continues to fall with the increment of Reynolds number.

Typical application of this fin type usually in the hydrocarbon industry where the heat exchangers are designed to operated at high mass velocities and moderate thermal duties.

4.5.3 Offset Strip Fins

Offset strip fins is widely used in application of high performance plate and fin heat exchanger. This geometry is consists of type of interrupted surface. This geometry can be visualized as a set of plain fin cut perpendicular to the flow path at regular intervals which each of the cuttings is arrange at half setting of the fin spacing. The application of interrupted surface helps to enhance the performance of heat transfer by two important mechanisms. First, the interrupted surface helps to prevent the continuous increment of thermal boundary layer by periodically interrupting the growth. As the thermal boundary layer is interrupted regularly, the thermal resistance will be lower as compared to uninterrupted surface layer. Interrupted surface also will help to enhance the heat transfer by continuously bringing in fresh fluid by oscillations in the flow field in the form of vortices shed from the trailing edges. The application will then accompanied by increment in pressure drop. If compare to the plain fin surface, offset strip fin offer as much as 5 times in performance but as for its drawback, it will cause higher pressure drop. At specific heat transfer and pressure drop requirements, the offset strip fin demands a higher frontal area if compared to plain fin, but results in a shorter flow length and lower overall volume.

Offset Strip fins also has a few drawbacks during high Reynolds numbers, where the friction factor remains nearly constant due to form drag while the heat transfer performance goes down. Offset Strip Fins are less frequently used for application of high Reynolds numbers but in return, they are extensively being use in air separation and cryogenic applications where mass velocities are low and high thermal effectiveness is essential.

4.5.4 Louvered Fins

The geometry of a louvered fins are similar to the offset strip fins but instead of shifting the slit strip laterally, small segments of fin are slit and rotated 20 to 45 degrees relative to fluid flow. The surface base can be either triangular or rectangular shape, and it can be cut in many different forms.

Another type of louvered fins is multilouvered fins where the fins has highest heat transfer enhancement relative to pressure drop if compared to other fin types. Flows over louvered fins are similar in nature to that through the offset strip fin geometry, with boundary layer interruption and vortex shedding are few crucial components. The performances of louvered fins are much depends on the degree to which fluid follows the louvered where at low Reynolds number, the flow is nearly parallel to the axial direction while at high Reynolds number, the flow is in the direction of the louvered (boundary layer flow). Typical application of this fin type is in automotive sector.

4.6 Heat Transfer and Flow Friction

Heat transfer data are correlated based on individual surface basis by using Colburn type of presentation.

$$jh = \mathrm{St}\mathrm{Pr}^{2/3} = \frac{h}{G_{c_p}} \left(\frac{\mu c_p}{k}\right)^{2/3} \tag{1}$$

as a function of Reynolds number, which is obtained by employing the equivalent diameter

$$d_e = 4_{rh} : \mathrm{Re} = \frac{G4_{rh}}{\mu} = \frac{Gd_e}{\mu} \tag{2}$$

The Stanton number, St, is the ratio of the Nusselt number, Nu, to the product of the Reynold and Prandtl numbers, RePr[1]:

$$\mathrm{St} = \frac{Nu}{Re\mathrm{Pr}} = \frac{hd_e\,/\,k}{(Gd_e\,/\,\mu)(\mu c_p\,/\,k)} = \frac{h}{G} \tag{3}$$

The overall heat transfer coefficient may be based on the surface of the hot or cold side. In the absence of fouling, these relationships, based on the hot side are

$$\mathrm{U_h} = \frac{1}{\frac{1}{h_h \eta_{oh}} + \frac{1}{(s_h/s_c) h_h \eta_{oh}}} \tag{4}$$

The exchanger effectiveness or simply the effectiveness ϵ, is designated in the nomenclature of the ϵ-N_{tu} method by

$$\epsilon = \frac{C_h(T_1 - T_2)}{C_{min}(T_1 - t_2)} = \frac{C_c(t_1 - t_2)}{C_{min}(T_1 - t_1)} \tag{5}$$

where $C_h = \dot{m}_h . c_h$ is the product of the mass flow rate $\dot{m}_h$ and the specific heat c_h of the hot fluid working between inlet temperature T_1 and outlet temperature T_2.

In similar fashion, $C_c = \dot{m}_c . c_c$ is the product of the mass flow rate $\dot{m}_c$ and the specific heat c_c of the cold fluid working between inlet temperature t_1 and outlet temperature t_2. Moreover C_{min} is the smaller of the C_h or C_c values and it is noted that Є is a measure of the amount of the heat transferred if the exchanger had infinite surface.

The *number of transfer units* N_{tu} is defined by

$$N_{tu} = \frac{US}{C_{min}} = \frac{t_2 - t_1}{LMTD} \tag{6}$$

Kays and London (1984) suggest that the pressure drop ΔP in a compact heat exchanger to be computed from equation

$$\frac{\Delta P}{P_1} = \frac{G^2 v_1}{2 g_c P_1} [\Phi_1 + \Phi_2 + \Phi_3 - \Phi_4] \tag{7}$$

Where,

$\Phi_1 = 1 + Kc - \sigma^2$

$\Phi_2 = 2\ ((v_2/v_1) - 1)$

$\Phi_3 = f\ (S/A)\ (v_m/v_1)$

$\Phi_4 = (1 - \sigma^2 - Ke)\ (v_2/v_1)$

Where,

Kc = Entrance loss coefficient

Ke = Exit loss coefficient

ΔP = Pressure drop

σ = Contraction ratio of cross-sectional area

f = Fanning friction factor

Friction factors are correlated on an individual surface basis and are usually plotted as a function of Reynolds number. The entrance and exit loss coefficients differ for the various types of passages and are plotted as functions of the parameter σ and the Reynolds number.

CHAPTER 5
METHODOLOGY

The methodology of this study is presented in a flow chart in Figure 5.1. The research starts with the study of the performance of the gas turbine and the external factor which affecting gas turbine performance (pressure drop, humidity, ambient temperature). The study then proceed to method and ways to cool down the ambient air temperature before it will then proceed to the final stage of study where construction of numerical and solid modeling of gas turbine with implementation of inlet cooling system. Also, the design will be tested and analyze with the development of simulation modeling for the gas turbine in terms of performance.

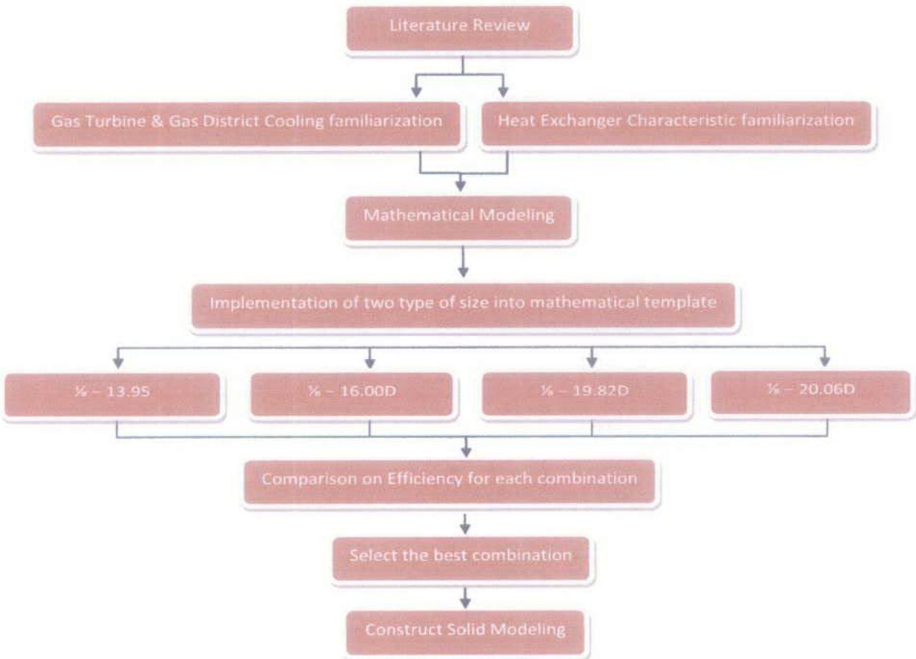

Fig. 5.1: Flow chart of the Methodology Employed in the Research

5.1 Gantt Chart

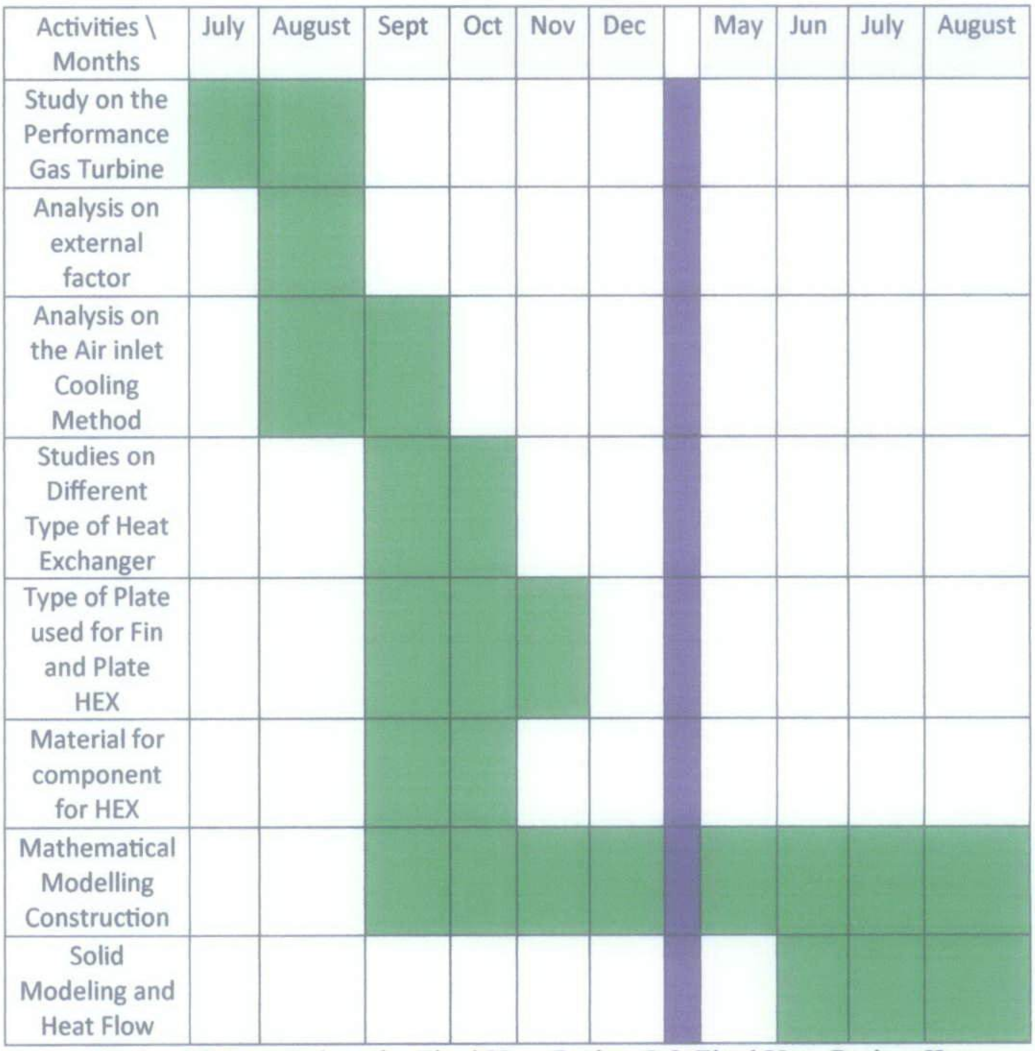

Figure 5.2: Gantt chart for Final Year Project I & Final Year Project II

Procedure

Steps in the design of or performance calculation for a compact heat exchanger with single hot and cold fluids:

1. Establish the heat balance
2. Assume an overall exchanger size (length, width, depth) and select the types of surfaces to be used. Using the geometric data for the assumed surfaces, compute the surface areas, free flow areas, and other physical parameters.
3. Obtain the thermal properties of the fluids at their bulk temperatures.
4. Compute the heat transfer coefficients.
5. Obtain the fin and overall passage efficiencies.
6. Determine the overall heat transfer coefficients.
7. Obtain R and N_{tu} and compute the actual exchanger surface effectiveness.
8. Calculate the pressure drops.

A plate and fin heat exchanger for the air pre-cooler of a gas turbine for a Gas District Cooling system plant. Specifications for the unit entail the air with mass flow rate of 19.00 kg/s at temperature of 36°C to 23°C for cooling purpose cool down by cooling water flow of temperature of 6°C in which water temperature may rise up to 13 °C. Mass flow rate of water is acquire through the equation of energy equilibrium, with a constant value of 19 kg/s mass flow rate of air into the heat exchanger.

5.2 Establish the Heat Balance

For efficient heat absorption from air to water, the heat should be transferred from high energy value to lower energy, in this case, heat is transferred from air to water to achieve a thermal equilibrium and thus, reducing the temperature of air to desired temperature.

$Q_w = \dot{m}_w \, c_{pw}.(T_{w1} - T_{w2})$

$Q_a = \dot{m}_a \, c_{pa}.(T_{a1} - T_{a2})$

$Q_a = Q_w$

$\dot{m}_a \, c_{pa}.(T_{a1} - T_{a2}) = \dot{m}_w \, c_{pw}.(T_{w1} - T_{w2})$

$$\dot{m}_w = \frac{\dot{m}_a.Cpa.(Ta1 - Ta2)}{Cpw.(Tw1 - Tw2)}$$

$$\dot{m}_w = \frac{(19)(1.005)(36-20)}{(4.18)(13-6)} = 10.442 \text{ kg/s}$$

Where,

Q	= Heat Energy
C_p	= Specific heat for substance at designated temperature
$\dot{m}$	= Mass flow rate for substance
T	= Temperature of the substance

Using the formula above, the value for mass flow rate for water can be calculated by taken into account the constant value for:

Specific heat for water - 4.180 kJ/kg.K
Mass flow rate for air - 19.00 kg/s
Specific heat for air - 1.005 kJ/kg.K

Mass flow rate for water can be found to be 10.442 kg/s. A full list for mass flow rate of water at various water and air temperature can be found in Chapter 6: Result & Discussion.

5.3 Exchanger physical data

In the actual research of fin surface geometry, there is various specification of surface sizing specifically for offset strip fin, but as for this research, the data ranges are include up to four different sizing which are relatively common in industries. They are:

Table 5.1: Type of Surface Geometry for Offset

Strip Fins	⅛ - 13.95	⅛ - 16.00D	⅛ - 19.82D	⅛ - 20.06D
b_h (10e-3 m)	9.54	6.48	5.21	5.11
Fins per inch	13.95	16	19.82	20.06
d (10e-3 m)	2.68	1.86	1.54	1.49
δ (10e-3 m)	0.254	0.152	0.102	0.102
β (m^2/m^3)	1250	1804	2231	2290
S_f/S	0.84	0.845	0.841	0.843

The number in each sizing for example **⅛ – 16.00 (D)** the fraction indicate the length of the fin in the flow direction (inches) followed by numeral that indicates the number of fins per inch. The suffixes which can be either (D), (T) indicate **double** or **triple stacks**, while sizing without the suffixes indicates a **single stack**.

For the hot side, assume 200 single stack of offset strip fin surface $\frac{1}{8}$ **– 20.06 (D)**, 400 cm long by 400 cm wide. The surface is composed of a pair of surfaces, each 0.25 in. or 0.635 cm high with a single splitter plate 0.005 in. or 0.0127 cm thick. The total height of this surface is 0.255 in. or 0.6477 cm.

For the cold side, assume 201 single stack of offset strip fin surface $\frac{1}{8}$ **– 16.00 (D)**, 400 cm long by 400 cm wide. This surface is composed of a pair of surfaces, each 0.100 in. or 0.2540 cm high with a single splitter plate 0.005 in. or 0.0127 cm thick. The total height of this surface is 0.6477 cm.

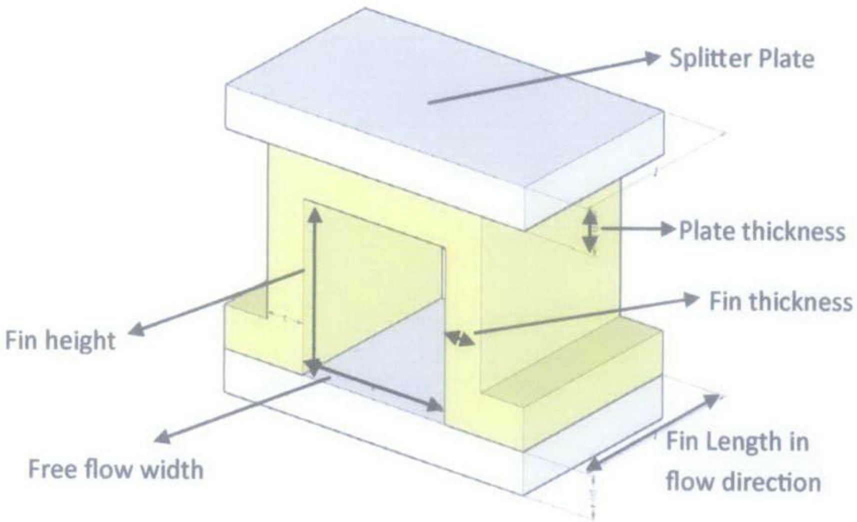

Figure 5.3: Geometric parameters of offset strip fin (individual) [16]

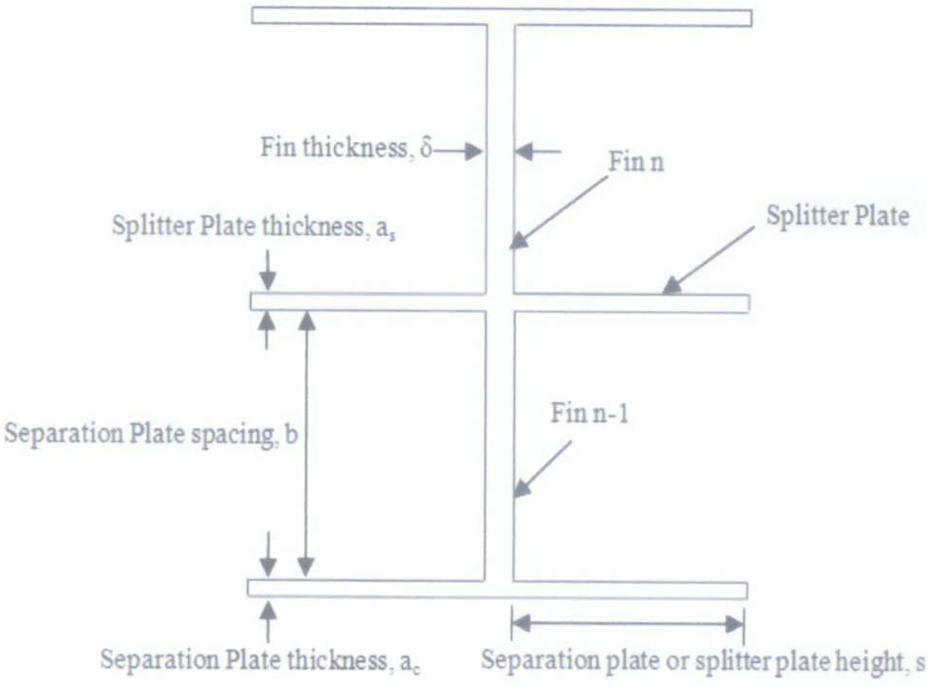

Figure 5.4: Double stack with single-side loading

The separation plates are to be aluminum and are selected to be 0.0152 cm thick so that the exchanger height is in accordance with:

$$H = n_1 (b_1 + a) + n_2 (b_2 + a)$$
$$= 200(0.6477 + 0.0152) + 201(0.6477 + 0.0152)$$
$$= 265.5422 \text{ cm}$$

Where,

H = Exchanger height

n = Number of stack

b = Separation plate spacing

a = Separation plate thickness, designated by a_c or, if a splitter plate by a_s

β = Ratio of the total surface area to the total volume on one side of the exchanger

d_e = Equivalent diameter used to correlate flow friction and heat transfer

Table 5.2: Physical Data

	Hot Side (air) $\frac{1}{8}$ – 20.06(D)	Cold Side (Chilled water) $\frac{1}{8}$ – 16.00(D)
β	2290	1804
S_f / S	0.843	0.845
d_e	1.49 x 10^{-3}	1.86 x 10^{-3}
δ	0.102 x 10^{-3}	0.152 x 10^{-3}

For hot side, using the above data:

Overall volume, V = WDH

$$= (400)(400)(265.5422)$$
$$= 42{,}486{,}752 \text{ cm}^3$$
$$= 42.487 \text{ m}^3$$

Frontal Area, $A_{f,h} = HW$

$= (265.5422)\ (400)$

$= 106{,}216.88\ cm^2$

$= 10.622\ m^2$

Where,

W = Width of the exchanger

D = Long of the exchanger

H = Height of the exchanger

Because there are two surfaces, it is necessary to employ the factor α, which is the ratio of the total surface on *one* side to the total surface on *both* sides of the exchanger.

$$\alpha_h = [\frac{b_h}{b_h+b_c+a}]\ (\beta_h)$$

$$= [\frac{0.511}{0.511+0.648+2(0.0152)}]\ (2290)$$

$$= 1144.82\ m^2/m^3$$

Total surface, $S_h = \alpha_h V$

$= (1144.82)\ (42.4868)$

$= 48639.66\ m^2$

Flow area to the frontal area, $\sigma_h = \frac{(\alpha_h\ d_{eh})}{4}$

$$= \frac{(1144.82)(0.00149)}{4}$$

$$= 0.42645$$

Flow areas, $A_h = \sigma_h A_{f,h}$

$= 0.42645 \times 0.142373$

$= 4.5296\ m^2$

While for cold side,

Frontal Area, $A_{f,c} = HD$

$= (265.5422)(400)$

$= 10{,}622 \text{ cm}^2$

$= 10.622 \text{ m}^2$

$\alpha_c = [\frac{b_c}{b_h+b_c+a}](\beta_c)$

$= [\frac{0.648}{0.511+0.648+2(0.0152)}](1804)$

$= 881.296 \text{ m}^2/\text{m}^3$

Total surface, $S_c = \alpha_c V$

$= (881.296)(42.4868)$

$= 37443.391 \text{ m}^2$

Flow area to the frontal area, $\sigma_c = \frac{(\alpha_c \, d_{ec})}{4}$

$= \frac{(881.296)(0.00186)}{4}$

$= 0.4098$

Flow areas, $A_c = \sigma_c A_{f,c}$

$= 0.4098 \times 10.622$

$= 4.3528 \text{ m}^2$

Where,

S = Total heat transfer surface on one side of the exchanger

σ = Ratio of the free flow area to the frontal area on one side of the exchanger

b = Separation plate spacing. This dimension is approximation of the fin height

a = Separation plate thickness

A_f = Frontal area on one side of the exchanger

α = Ratio of the total surface area on one side of the exchanger to the total volume on both side of the exchanger

5.4 Fluid properties at selected temperature

Table 5.3: Fluid Properties for fluids at already determined bulk temperatures

Property	Hot side	Cold side
Temperature (K)	309	286
Specific heat c (J/kg.K)	1005	4180
Thermal conductivity k (W/m.k)	1.398	0.282
Dynamic viscosity μ (kg/m.s)	0.00001889	0.001214
Pr =cμ / k	0.013579721	18.08712766
$Pr^{2/3}$	0.056919433	6.89043123

Where,

Pr = Prandtl numbers

c = Specific heat

5.5 Heat transfer coefficient.

For the hot side, with C_h = 19095 kJ/K

$$G_h = \frac{\dot{m}_h}{A_h}$$

$$= \frac{19}{4.5296}$$

$$= 4.19465994 \text{ kg/m}^2\text{.s}$$

Reynolds number for hot side, $Re_h = \frac{d_{eh}\, G_h}{\mu} = \frac{(0.00149)(4.1947)}{1.889 \times 10^{-5}}$

$$= 330.865183$$

From graph of Reynolds number versus Colburn Factor for strip fin compact heat exchanger,

$j_h = \frac{h_h}{c_h G_h} Pr^{1/3} = 0.01$ and friction factor, $f_h = 0.8$

Then,

Heat transfer coefficient for hot side, $h_h = \frac{j_h c_h G_h}{Pr^{2/3}}$

$$= \frac{(0.01)(4.1947)(1005)}{0.0569}$$

$$= 740.6316 \text{ W/m}^2\text{.K}$$

The cold-side computations

$G_c = \frac{\dot{m}_c}{A_c} = \frac{10.44}{4.3527}$

$= 2.39881739 \text{ kg/m}^2\text{.s}$

Reynolds number for cold side, $Re_c = \frac{d_{ec}\, G_c}{\mu} = \frac{(0.00186)(2.3988)}{0.001215}$

$$= 3.67226366$$

From graph of Reynolds number vs Colburn Factor for strip fin compact heat exchanger,

$j_c = \frac{h_c}{c_c G_c} Pr^{1/3} = 0.8$ and friction factor, $f_c = 0.9$

Then,

Heat transfer coefficient for cold side, $h_c = \frac{j_c c_c G_c}{Pr^{2/3}}$

$$= \frac{(0.8)(4180)(2.3988)}{6.89}$$

$$= 1164.172 \text{ W/m}^2\text{.K}$$

Where,

j	= Heat transfer factor
f	= Fanning friction factor
St	= Stanton number
Nu	= Nusselt number
Re	= Reynolds number
G	= Thermal conductance

5.6 Fin and overall passage efficiencies.

Both of the surfaces employed in this design may be treated as double stacks with equal heat dissipation on both sides. The configuration is shown below and equation below will yield the fin efficiency. The thermal conductivity of aluminum (99% pure) is listed by Lienhard (1987) as k = 211 W/m.K

For the hot side, pertinent data are

$b_h = 5.11 \times 10^{-3}$ m

$s = \frac{0.0012662 - 0.000102}{2} = 5.821 \times 10^{-4}$ m

$\delta_{fh} = 1.02 \times 10^{-4}$ m

$a_s = 1.27 \times 10^{-4}$ m

$h_h = 740.6316$ W/m^2.K

Then for the fin,

$$m_f = \left(\frac{2h_h}{k\delta_f}\right)^{1/2} = \left[\frac{(2)(740.6316)}{(211)(1.02 \times 10^{-4})}\right]^{1/2} = 262.34622 \text{ m}^{-1}$$

$m_f b = (262.34622)(1.341) = 1.34058919$

$\tanh m_f b = \tanh 1.34058919 = 0.87181369$

and for unit fin length,

$$Y_{of} = (2h_h\, k\delta_{fh})^{1/2} = [(2)(740.6316)(211)(1.52 \times 10^{-4})]^{1/2} = 5.64621535 \text{W/K}$$

For the splitter plate with $a_s = 1.27 \times 10^{-4}$

$$m_s = \left(\frac{2h_h}{ka_s}\right)^{1/2} = \left[\frac{(2)(740.6316)}{(211)(1.27 \times 10^{-4})}\right]^{1/2} = 235.111048 \text{ m}^{-1}$$

$m_s s = (235.111048)(0.0005821) = 0.1368586$

$\tanh m_s s = \tanh 0.1368586 = 0.13601049$

and for unit fin length,

$Y_{o,s} = (2h_h\, ka_s)^{1/2}$

$= [(2)(740.6316)(211)(1.27\ x\ 10^{-4}\ )]^{1/2} = 4.45496417$ W/K

Fin efficiency, $\eta_{fh} = \dfrac{Y_{o,f}}{2(b+s)Lh} \dfrac{\tanh m_f b+(2Y_{o,s}/\,Y_{o,f})\tanh\ m_s s}{1+\ (2Y_{o,s}/\,Y_{o,f})\tanh m_s s\ \tanh\ m_f b}$

$$\eta_{fh} = \frac{5.646}{2(0.00511+5.821x10^{-4})(1.5\ X\ 740.632)}\left[\frac{0.872+(\frac{2(4.455)}{5.646})\ 0.136}{1+(2(\frac{4.455}{5.646})(0.136\ X\ 0.872)}\right]$$

$= 0.47$

And, for overall passage efficiency,

$\eta_{oh} = 1 - \frac{s_f}{s}\,(1-\eta_{fh})$

$= 1 - 0.843\ (1\text{-}0.47)$

$= 0.55314792$

For the cold side, the same procedure is followed. Pertinent data are

$b_c = 6.48\ x\ 10^{-3}$ m

$s = \dfrac{0.0015875-0.000152}{2} = 0.00071775$ m

$\delta_{fc} = 1.52\ x\ 10^{-4}$ m

$a_s = 1.27\ x\ 10^{-4}$ m

$h_c = 1164.17175\ W/m^2.K$

Then for the fin

$m_f = (\frac{2h_c}{k\delta_f})^{1/2} = [\frac{(2)(1164.17175\ )}{(211)(1.52\ x\ 10^{-4})}]^{1/2} = 269.439028\ m^{-1}$

$m_f b = (269.439028)(6.48\ x\ 10^{-3}) = 1.7459649$

$\tanh m_f b = \tanh 1.7459649 = 0.94091455$

and for unit fin length,

$Y_{o,f} = (2h_c\ k\delta_{fc})^{1/2}$

$= [(2)(1164.17175)(211)(1.52\ x\ 10^{-4})]^{1/2} = 8.641$ W/K

For the splitter plate,

$m_s = \left(\frac{2h_c}{ka_s}\right)^{1/2} = \left[\frac{(2)(1164.17175)}{(211)(1.27\ x\ 10^{-4})}\right]^{1/2} = 1743.411\ m^{-1}$

$m_s s = (1743.411)(5.8977 \times 10^{-4}) = 0.21157025$

$\tanh m_s s = \tanh 0.2116 = 0.208469$

and for unit fin length,

$Y_{o,s} = (2h_c\ ka_s)^{1/2}$

$= [(2)(1164.17175)(211)(1.27\ x\ 10^{-4})]^{1/2} = 7.899$ W/K

Fin efficiency, $\eta_{fc} = \frac{Y_{o,f}}{2(b+s)Lh} \frac{\tanh m_f b + (2Y_{o,s}/Y_{o,f}) \tanh\ m_s s}{1 + (2Y_{o,s}/Y_{o,f}) \tanh m_s s\ \tanh\ m_f b}$

$= \frac{8.641}{2(0.00648+71775x10^{-4})(1.5\ X\ 1164.17)} \left[\frac{0.94091455 + 2\left(\frac{7.899}{8.641}\right) 0.208469}{1+(2\left(\frac{7.899}{8.641}\right)(0.208469\ X\ 0.94091455)}\right]$

$= 0.41987615$

And, for overall passage efficiency,

$\eta_{oc} = 1 - \frac{s_f}{s}(1 - \eta_{fh})$

$= 1 - 0.845\ (1 - 0.41987615)$

$= 0.5098$

Where,

b	= Fin height or passage height, m
δ	= Fin thickness, m
m	= Fin performance parameter, m^{-1}
a_s	= Splitter plate thickness, m
k	= Thermal conductivity constant of the material, W/m.K

η_f = Fin Efficiency, dimensionless

η_o = Overall passage efficiency, dimensionless

s = Separation plate or splitter plate height, m

Y_o = Unit fin length, $Y_{o,f}$ for fin, $Y_{o,s}$ for splitter plate

5.7 Overall heat transfer coefficient.

In the absence of fouling, the overall heat transfer coefficient can be obtained because $C_h = C_{min}$

$$U_h = \frac{1}{\frac{1}{h_h \eta_{oh}} + \frac{1}{(s_h/s_c) h_h \eta_{oh}}}$$

$$= \frac{1}{\frac{1}{(740.632)(0.5531)} + \frac{1}{\left(\frac{48639.66}{37443.39}\right)(740.632)(0.5531)}}$$

$$= 267.521 \text{ W/m}^2\text{.K}$$

Where,

U = Overall heat transfer coefficient

s = Separation plate or splitter plate height, m

5.8 Effectiveness.

The required effectiveness with $C_h = C_{min}$ is

$$\epsilon = \left| \frac{T_1 - T_2}{T_1 - t_1} \right| = \left| \frac{36-20}{36-13} \right| = 0.696$$

For this design, the number of transfer unit, N_{tu} is defined by

$$N_{tu} = \frac{S_h U_h}{C_{min}} = \frac{(48639.66)(267.521)}{19095} = 524.58$$

and then,

Capacity rate ratio, $R = \frac{C_{min}}{C_{max}} = \frac{19095}{43645.71} = 0.438$

The actual exchanger effectiveness may be computed by first calculate the parameters of,

$\Lambda = e^{-RN_{tu}{}^{0.78}} - 1$

$= -1$

Then, exchanger effectiveness can be calculated,

$\epsilon = 1 - e^{(\Lambda/\mathrm{R})(N_{tu})^{0.22}}$

$= 1 - e^{(-1/0.438)(524.581)^{0.22}}$

$= 0.9998$

This exceed the required $\epsilon = 0.696$ and $\epsilon = 0.9998$, by a factor of 0.9998/0.696 = 1.44 and is slightly conservative.

Where,

ϵ	= Exchanger effectiveness
t	= Water temperature
T	= Air temperature
N_{tu}	= Number of transfer unit
e	= Exponential component
R	= Capacity rate ratio
U	= Overall heat transfer coefficient
C	= Heat transfer coefficient

CHAPTER 6

RESULT AND DISCUSSION

6.1 Introduction

Detailed analysis results of the application of offset strip fin heat exchanger type are presented in this chapter. Respective parameters are taken into account to investigate the effect of heat transfer as well as the efficiency of the heat exchanger. The research taken into account a few type of common fin surface geometry that are available in the market for more accurate data for implementation of the system later. By selecting fin surface geometry as an independent variable, other parameters are investigated to achieve the most optimum configuration for heat transfer process. To name a few affecting parameters that will help to identify the suitable configuration including Reynolds Number, Number of unit transfer, heat transfer coefficient, fin efficiency, overall passage efficiency. In this study, the model was developed and implemented using various related formula and by the usage of Microsoft Excel to manage the data and for faster, wide range of data. It must be noted that the spread sheet created using Microsoft Excel is solely for one type of Fin surface due to specific formula usage for offset strip fin.

6.2 Mass flow rate required for water

To achieve an optimum heat transfer to ensure the temperature of air is cool down to the desired value, the mass of water is a critical parameter where the mass flow rate should be sufficiently enough to transfer all the heat from the air and cool down the temperature. Taken into calculation a constant value of mass flow rate enters into the compressor of gas turbine with a value of 19 kg/s, and with assumption of:

1. The specific heat c_p is constant both for air and water
2. Mass flow fate for air is constant
3. The heat exchanger operates under steady-state conditions (constant flow rates and fluid temperature)
4. Heat losses to or from the surrounding are negligible (adiabatic process)
5. No energy sources or sinks in the exchanger walls or fluids
6. The temperature of each fluid is uniform over every cross section
7. Wall thermal resistance is distributed uniformly in the entire heat exchanger
8. The individual and overall heat transfer coefficient is constant throughout the system
9. Temperature are in range based on data
    1. Air: ` 23°C to 35°C
    2. Water: 06°C to 13°C

Table 6.1: Mass Flow Rate of Water at various Temperature Different (δT)

Hot Air Temp		23	24	25	26	27	28	29	30	31	32	33	34	35	36
Diff Temp (Hot Air)		3	4	5	6	7	8	9	10	11	12	13	14	15	16
Cold Water Temp	Diff Temp Cold Water	mass flow rate of water at different (dT of water & dT of air)													
13	7	1.958	2.610	3.263	3.916	4.568	5.221	5.873	6.526	7.179	7.831	8.484	9.136	9.789	10.442

Using principle of Energy Conservation demonstrated in Chapter 5 or simplified equation of $\dot{m}_w = (\dot{m}_a.c_p.\delta T) / (c_{pw}.\delta T)$, a table was constructed in Table 6.1.

Based on the table for respective range of temperature for water and air, the highest mass flow rate required would be at 10.442 kg/s which in turns provide us the data for calculation at needed heat transfer to be equilibrium with air according to the assumption stated. Based on the template created using equation of rate heat transfer which is $Q=\dot{m}_a.c_p.\delta T$, the highest heat transfer at equilibrium condition for both water and air would be 248.235 kWatt.

A full list of heat transfer at equilibrium for water and air at various temperatures can be seen in Appendix B.

6.3 Type of Fin Geometry Available

Table 6.2: Physical data for respective side

Physical Data	Air			Water		
Fin surface geometry	1/8 - 16.00 (D)			1/8 - 19.82 (D)		
Heat Exchanger Height	H_h	265.5422	cm	H_c	265.5422	cm
Length of Fin Passage	L_f	1.5	m	L_f	1.5	m
No. of Stack	N	200	N/A	N	201	N/A
Long	L	400	cm	L	400	cm
Wide	W	400	cm	W	400	cm
High of Surface	b	0.647	cm	b	0.647	cm

Based on these four different sizing discussed in previous chapter and with the mathematical modeling that has been constructed based on assumptions pre-defined above, and proved correct, the sizing parameters are combined into two different sizing for each combination.

Parameters of the physical data are set to constant as to simplify research to identify the most optimum where as the only changing variable is fin surface geometry along with data for each geometry fin.

Table 6.3: Combination of Sizing for Strip Fins

Combination		
No #	Air	Water
#1	1/8 - 20.06 (D)	1/8 - 13.95
#2	1/8 - 20.06 (D)	1/8 - 19.82 (D)
#3	1/8 - 20.06 (D)	1/8 - 16.00 (D)
#4	1/8 - 19.82 (D)	1/8 - 16.00 (D)
#5	1/8 - 19.82 (D)	1/8 - 13.95
#6	1/8 - 16.00 (D)	1/8 - 13.95
#7	1/8 - 19.82 (D)	1/8 - 20.06 (D)
#8	1/8 - 13.95	1/8 - 16.00 (D)
#9	1/8 - 13.95	1/8 - 19.82 (D)
#10	1/8 - 16.00 (D)	1/8 - 20.06 (D)
#11	1/8 - 13.95	1/8 - 20.06 (D)
#12	1/8 - 16.00 (D)	1/8 - 19.82 (D)

With each combination has been inserted into the Mathematical Modeling template, each of the data are then compiled and two graph are constructed based on constructed based on their Fin Efficiency, Overall Passage Efficiency and also Heat Transfer Coefficient characteristics for further analysis.

Figure 6.1: Overall Passage Efficiency for all Sizing

Figure 6.2: Fin Efficiency for all Sizing

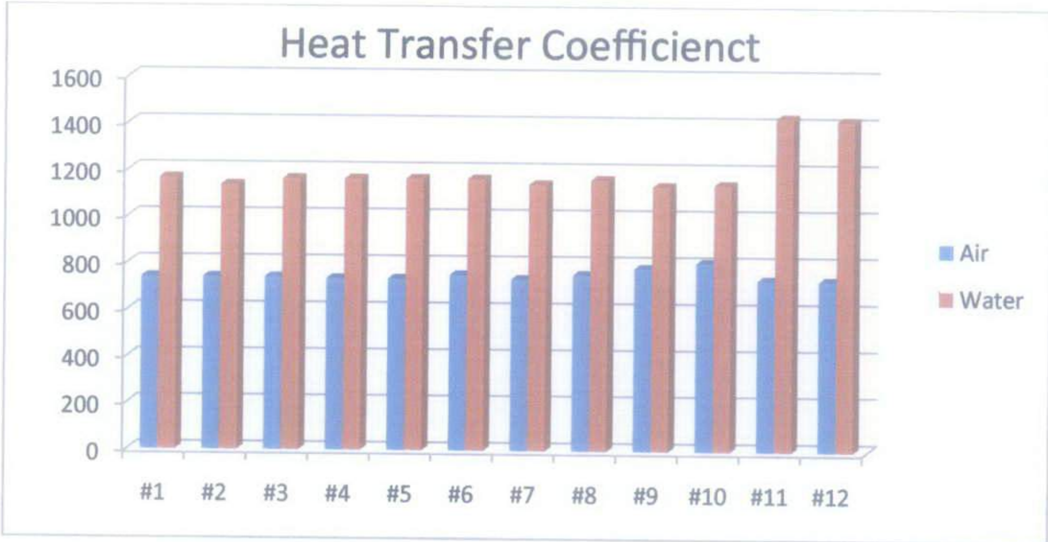

Figure 6.3: Heat Transfer Coefficient for all sizing

With the combination presented along with the constructed graph analysis consists of graph for Fin Efficiency, Overall Passage Efficiency and also Heat Transfer Coefficient a thorough research is proceed to acquire a suitable combination with high efficiency for fin selection for Plate and fin Heat Exchanger to be implemented to the Gas Turbine system at Gas District Cooling plant.

6.4 Result Analysis

Based on the two graphs presented above which are Fin Efficiency and Overall Passage Efficiency, a few potential combination which are Combination #2, #3, #4 and #7 are identified to be suitable fin configuration based on data available with satisfied efficiency for gas turbine to operate with higher performance compared to current condition at Gas District Cooling plant. But, with the inclusion of the third graph, which is Heat Transfer Coefficient graph, Combination #3 presenting a more suitable candidate, with higher heat transfer coefficient for both air and water.

With higher heat transfer coefficient and moderate efficiency for both fin and overall passage, a suitable fin can be constructed to absorb heat from surrounding air and lowering the temperature based on the desired value. Based on that, Combination #3 is selected and detailed output parameters are shown in table below. Full lists of detailed output parameters are presented in Appendix B

Table 6.4: Parameters acquired using spread sheet (Mathematical Modeling Template)

Physical Data	Air			Water		
Fin surface geometry	1/8 - 20.06 (D)			1/8 - 16.00 (D)		
Heat Exchanger Height	H_h	265.5422	cm	H_c	265.5422	Cm
Length of Fin Passage	L_f	1.5	m	L_f	1.5	M
No. of Stack	N	200	N/A	N	201	N/A
Long	L	400	cm	L	400	Cm
Wide	W	400	cm	W	400	Cm
High of Surface	b	0.647	cm	b	0.647	Cm

Properties	Air			Water		
Fin surface geometry	1/8 - 20.06 (D)			1/8 - 16.00 (D)		
Temperature	T_h	309.00	K	T_c	286.00	K
mass flow rate	$\dot{m}$	10.44	kg/s	$\dot{m}$	19.00	kg/s
Flow Area	A_h	4.53	m^2	A_c	4.35	m^2
Reynolds Number	Re_h	330.87	N/A	Re_c	3.67	N/A
Heat transfer Coefficient	h_h	740.63	$W/m^2.K$	h_c	1164.17	$W/m^2.K$
Fin Efficiency	η_{fh}	0.47	N/A	η_{fc}	0.42	N/A
Overall Passage Efficiency	η_{oh}	0.55	N/A	η_{oc}	0.51	N/A
Overall Heat Transfer Coefficient	U_h	267.52	$W/m^2.K$	U_c	205.94	$W/m^2.K$
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.166394	kPa	ΔP	0.046031	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	0.70	N/A			
Actual Effectiveness	Λ	-1.00	N/A			
Nusselt Number	N_{tu}	524.58	N/A			

6.5 Solid Modeling

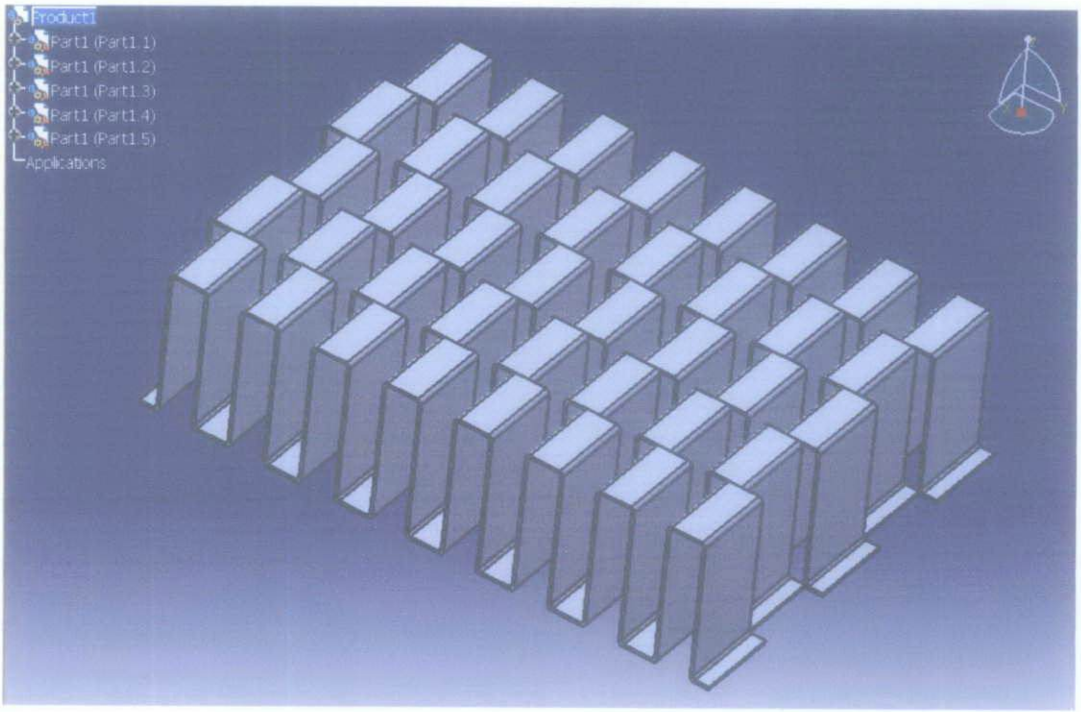

Figure 6.4: Isometric View of Offset Strip Fin

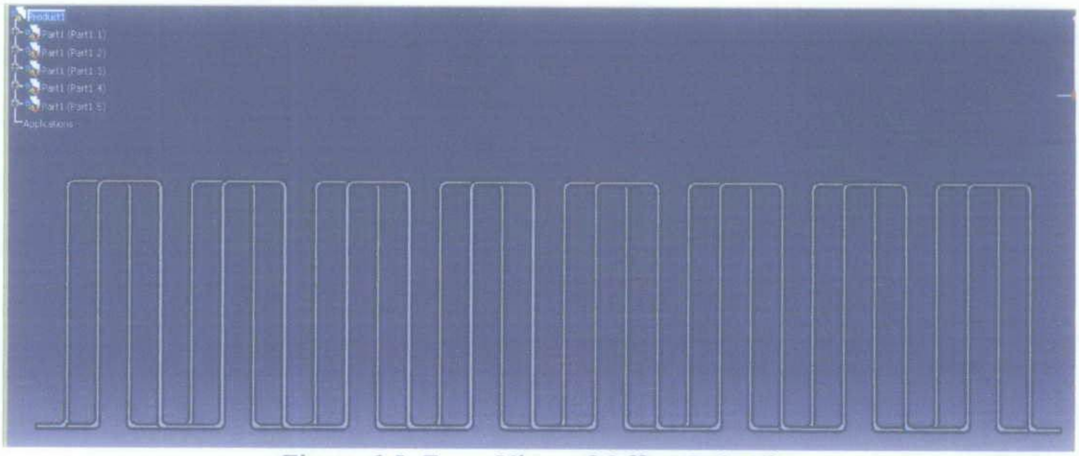

Figure 6.5: Front View of Offset Strip Fin

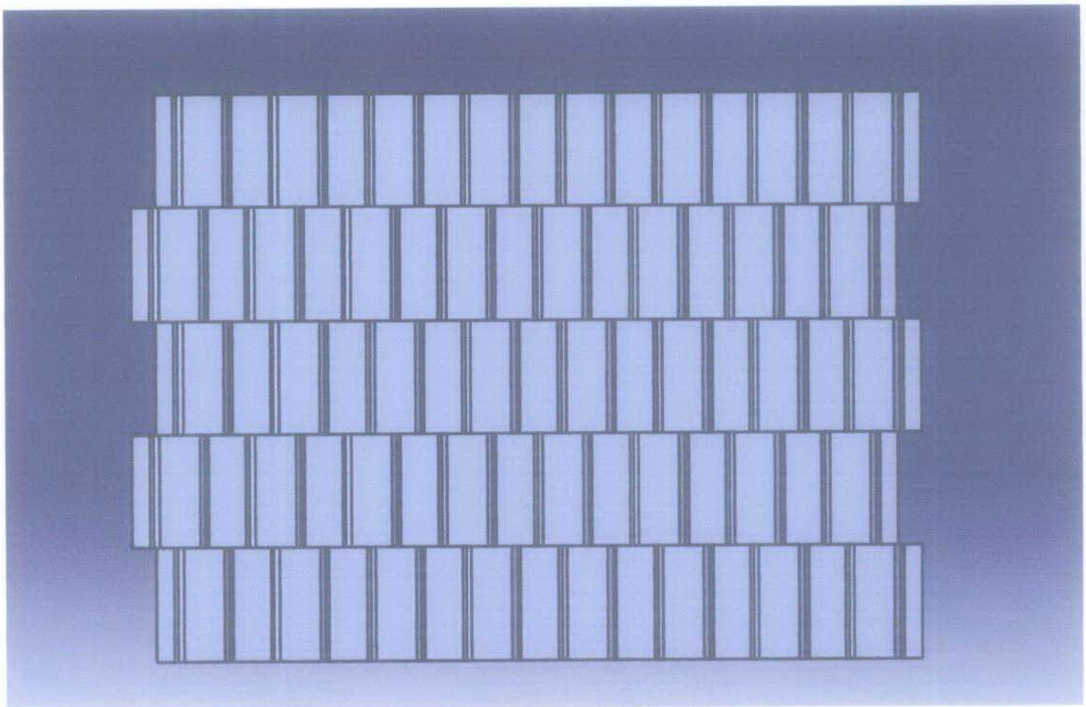
Figure 6.6: Upper view of Offset Strip Fin

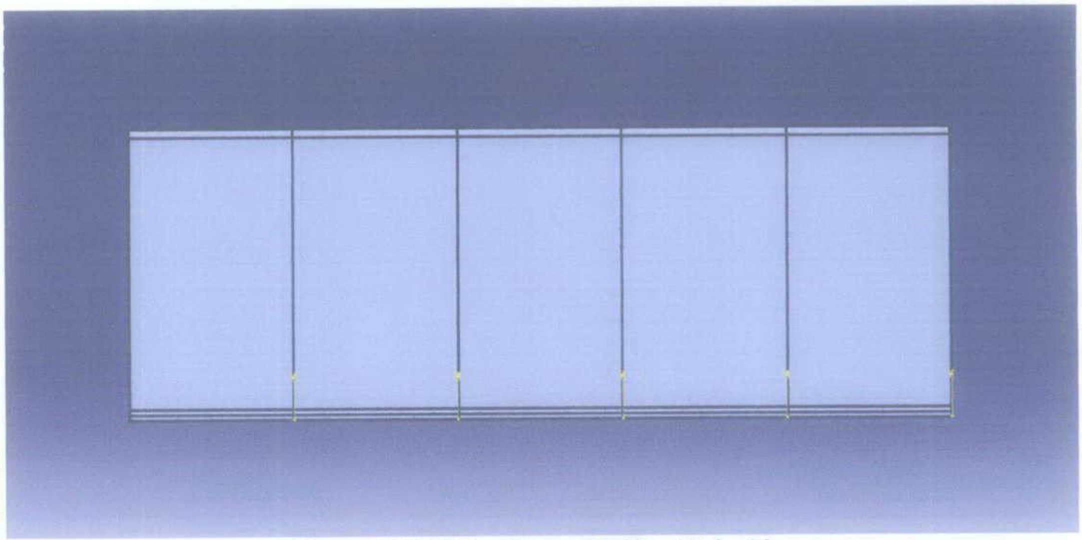
Figure 6.7: Side view of Offset Strip Fin

CHAPTER 7
CONCLUSION & RECOMMENDATIONS

7.1 Conclusion

An improvement of gas turbine efficiency is one of the current trends in the field of energy generation. Many researches have been done in order to achieve the desired performance. From the research, many methods have been developed prior to the gas turbine efficiency. One of the methods is plate and fin heat exchanger. Compact and high heat transfer rate has made the plate and fin heat exchanger as one of favorite device for many developers to enhance their heat exchanger. Implementation of heat exchanger with correct configuration will ensure the gas turbine to be working correctly without no side effect to its process (pressure drop, temperature rise) specifically gas turbine at Gas District Cooling plant which will ensure enhancement in terms of productivity with current load applied.

7.2 Recommendations

Further studies should proceed on the various type of sizing available in market which is not included in the research. With more sizing are included, possibilities to achieve more accurate and higher exchanger efficiency can be achieved.

Before the research is ready to be implemented to the system, a thorough cost analysis should be done first to ensure the implementation project is feasible and economical to the company.

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Appendix A: Experimental Data

Strip Fins	$\frac{1}{8}$ – 13.95	$\frac{1}{8}$ – 16.00*D*	$\frac{1}{8}$ – 19.82*D*	$\frac{1}{8}$ – 20.06
b (10^{-3} m)	9.54	6.48	5.21	5.11
Fins per inch	13.95	16.00	19.82	20.06
d_e (10^{-3} m)	2.68	1.86	1.54	1.49
δ (10^{-3} m)	0.254	0.152	0.102	0.102
β (m^2/m^3)	1250	1804	2231	2290
S_f/ S	0.840	0.845	0.841	0.843

Table 1: Surface Geometry of Some Plate and Fin Surfaces (Strip Fins)

Table A.3: saturated liquids...*continued*

Temperature K	°C	ρ (kg/m³)	c_p (J/kg·K)	*k* (W/m·K)	α (m²/s)	ν (m²/s)	Pr	β (K^{-1})
				Water				
273.16	0.01	999.8	4220	0.5610	1.330×10^{-7}	17.91×10^{-7}	13.47	-6.80×10^{-5}
275	2	999.9	4214	0.5645	1.340	16.82	12.55	-3.55×10^{-5}
280	7	999.9	4201	0.5740	1.366	14.34	10.63	4.36×10^{-5}
285	12	999.5	4193	0.5835	1.392	12.40	8.91	0.000112
290	17	998.8	4187	0.5927	1.417	10.85	7.66	0.000172
295	22	997.8	4183	0.6017	1.442	9.600	6.66	0.000226
300	27	996.5	4181	0.6103	1.465	8.568	5.85	0.000275
305	32	995.0	4180	0.6184	1.487	7.708	5.18	0.000319
310	37	993.3	4179	0.6260	1.508	6.982	4.63	0.000361

Table 2: Some thermophysical properties of selected materials (water)

Table A.6 Thermophysical properties of gases at atmospheric pressure (101325 Pa)

T (K)	ρ (kg/m³)	c_p (J/kg-K)	μ (kg/m-s)	ν (m²/s)	*k* (W/m-K)	α (m²/s)	Pr
				Air			
100	3.605	1039	0.711×10^{-5}	0.197×10^{-4}	0.00941	0.251×10^{-5}	0.784
150	2.368	1012	1.035	0.437	0.01406	0.587	0.745
200	1.769	1007	1.333	0.754	0.01836	1.031	0.731
250	1.412	1006	1.606	1.137	0.02241	1.578	0.721
260	1.358	1006	1.649	1.214	0.02329	1.705	0.712
270	1.308	1006	1.699	1.299	0.02400	1.824	0.712
280	1.261	1006	1.747	1.385	0.02473	1.879	0.711
290	1.217	1006	1.795	1.475	0.02544	2.078	0.710
300	1.177	1007	1.857	1.578	0.02623	2.213	0.713
310	1.139	1007	1.889	1.659	0.02684	2.340	0.709

Table 3: Some thermophysical properties of selected materials (Air)

Appendix B: Output Data Analysis

Properties	Air			Water		
Fin surface geometry	1/8 - 20.06 (D)			1/8 - 13.95		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.529569	m2	A_c	4.345737	m2
Reynolds Number	Re_h	330.8652	N/A	Re_c	5.299811	N/A
Heat transfer Coefficient	h_h	740.6316	$W/m^2.K$	h_c	1166.062	$W/m^2.K$
Fin Efficiency	η_{fh}	0.469926	N/A	η_{fc}	0.383912	N/A
Overall Passage Efficiency	η_{oh}	0.553148	N/A	η_{oc}	0.482486	N/A
Overall Heat Transfer Coefficient	U_h	295.0696	$W/m^2.K$	U_c	157.392	$W/m^2.K$
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.019562	kPa	ΔP	0.017098	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	1.25863	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	400.9161	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 20.06 (D)			1/8 - 19.82 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.529569	m2	A_c	4.456962	m2
Reynolds Number	Re_h	330.8652	N/A	Re_c	2.969414	N/A
Heat transfer Coefficient	h_h	740.6316	$W/m^2.K$	h_c	1136.963	$W/m^2.K$
Fin Efficiency	η_{fh}	0.469926	N/A	η_{fc}	0.429602	N/A
Overall Passage Efficiency	η_{oh}	0.553148	N/A	η_{oc}	0.520295	N/A
Overall Heat Transfer Coefficient	U_h	246.8959	$W/m^2.K$	U_c	235.0507	$W/m^2.K$
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.019562	kPa	ΔP	0.028284	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	1.249685	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	598.7319	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 20.06 (D)			1/8 - 16.00 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.529569	m2	A_c	4.352794	m2
Reynolds Number	Re_h	330.8652	N/A	Re_c	3.672264	N/A
Heat transfer Coefficient	h_h	740.6316	W/m².K	h_c	1164.172	W/m².K
Fin Efficiency	η_{fh}	0.469926	N/A	η_{fc}	0.419876	N/A
Overall Passage Efficiency	η_{oh}	0.553148	N/A	η_{oc}	0.509795	N/A
Overall Heat Transfer Coefficient	U_h	267.5205	W/m².K	U_c	205.9405	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.166394	kPa	ΔP	0.046031	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	0.695652	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	524.5811	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 19.82 (D)			1/8 - 16.00 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.560951	m2	A_c	4.352794	m2
Reynolds Number	Re_h	339.6151	N/A	Re_c	3.672264	N/A
Heat transfer Coefficient	h_h	735.5356	W/m².K	h_c	1164.172	W/m².K
Fin Efficiency	η_{fh}	0.465591	N/A	η_{fc}	0.419876	N/A
Overall Passage Efficiency	η_{oh}	0.550562	N/A	η_{oc}	0.509795	N/A
Overall Heat Transfer Coefficient	U_h	263.1033	W/m².K	U_c	207.8963	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.018667	kPa	ΔP	0.024553	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	1.178535	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	515.9193	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 19.82 (D)			1/8 - 13.95		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.560951	m2	A_c	4.345737	m2
Reynolds Number	Re_h	339.6151	N/A	Re_c	5.299811	N/A
Heat transfer Coefficient	h_h	735.5356	W/m².K	h_c	1166.062	W/m².K
Fin Efficiency	η_{fh}	0.465591	N/A	η_{fc}	0.374149	N/A
Overall Passage Efficiency	η_{oh}	0.550562	N/A	η_{oc}	0.474285	N/A
Overall Heat Transfer Coefficient	U_h	289.0687	W/m².K	U_c	158.2687	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.018667	kPa	ΔP	0.017098	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	1.184273	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	392.7625	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 16.00 (D)			1/8 - 13.95		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.454008	m2	A_c	4.345737	m2
Reynolds Number	Re_h	420.0332	N/A	Re_c	5.299811	N/A
Heat transfer Coefficient	h_h	753.1961	W/m².K	h_c	1166.062	W/m².K
Fin Efficiency	η_{fh}	0.456823	N/A	η_{fc}	0.374149	N/A
Overall Passage Efficiency	η_{oh}	0.541016	N/A	η_{oc}	0.474285	N/A
Overall Heat Transfer Coefficient	U_h	271.8529	W/m².K	U_c	184.0876	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.016209	kPa	ΔP	0.017098	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	0.803319	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	369.3712	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 19.82 (D)			1/8 - 20.06 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.560951	m2	A_c	4.426296	m2
Reynolds Number	Re_h	339.6151	N/A	Re_c	2.89291	N/A
Heat transfer Coefficient	h_h	735.5356	W/m².K	h_c	1144.84	W/m².K
Fin Efficiency	η_{fh}	0.465591	N/A	η_{fc}	0.434369	N/A
Overall Passage Efficiency	η_{oh}	0.550562	N/A	η_{oc}	0.523173	N/A
Overall Heat Transfer Coefficient	U_h	241.3092	W/m².K	U_c	242.0434	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.018667	kPa	ΔP	0.029639	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	1.175808	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	600.6594	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 13.95			1/8 - 16.00 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.446086	m2	A_c	4.352794	m2
Reynolds Number	Re_h	606.2874	N/A	Re_c	3.672264	N/A
Heat transfer Coefficient	h_h	754.5381	W/m².K	h_c	1164.172	W/m².K
Fin Efficiency	η_{fh}	0.415235	N/A	η_{fc}	0.419876	N/A
Overall Passage Efficiency	η_{oh}	0.508798	N/A	η_{oc}	0.509795	N/A
Overall Heat Transfer Coefficient	U_h	200.7374	W/m².K	U_c	283.1656	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.011293	kPa	ΔP	0.024553	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	0.246012	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	393.6261	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 13.95			1/8 - 19.82 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.446086	m2	A_c	4.456962	m2
Reynolds Number	Re_h	606.2874	N/A	Re_c	2.969414	N/A
Heat transfer Coefficient	h_h	784.53	W/m².K	h_c	1136.963	W/m².K
Fin Efficiency	η_{fh}	0.316022	N/A	η_{fc}	0.429602	N/A
Overall Passage Efficiency	η_{oh}	0.425458	N/A	η_{oc}	0.520295	N/A
Overall Heat Transfer Coefficient	U_h	225.6126	W/m².K	U_c	393.585	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.011293	kPa	ΔP	0.028284	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	1.621438	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	547.1192	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 16.00 (D)			1/8 - 20.06 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.454008	m2	A_c	4.426296	m2
Reynolds Number	Re_h	420.0332	N/A	Re_c	2.89291	N/A
Heat transfer Coefficient	h_h	807.6707	W/m².K	h_c	1144.84	W/m².K
Fin Efficiency	η_{fh}	0.338221	N/A	η_{fc}	0.434369	N/A
Overall Passage Efficiency	η_{oh}	0.440796	N/A	η_{oc}	0.523173	N/A
Overall Heat Transfer Coefficient	U_h	300.642	W/m².K	U_c	372.963	W/m².K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.016209	kPa	ΔP	0.029639	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	3.326549	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	748.3489	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 13.95			1/8 - 20.06 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.446086	m2	A_c	4.426296	m2
Reynolds Number	Re_h	606.2874	N/A	Re_c	2.89291	N/A
Heat transfer Coefficient	h_h	735.4377	W/m^2.K	h_c	1431.05	W/m^2.K
Fin Efficiency	η_{fh}	0.304709	N/A	η_{fc}	0.402326	N/A
Overall Passage Efficiency	η_{oh}	0.415956	N/A	η_{oc}	0.496161	N/A
Overall Heat Transfer Coefficient	U_h	255.9355	W/m^2.K	U_c	458.2913	W/m^2.K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.014414	kPa	ΔP	0.055567	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	1.87133	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	637.0669	N/A			

Properties	Air			Water		
Fin surface geometry	1/8 - 16.00 (D)			1/8 - 19.82 (D)		
Temperature	T_h	309	K	T_c	286	K
mass flow rate	$\dot{m}$	10.44156	kg/s	$\dot{m}$	19	kg/s
Flow Area	A_h	4.454008	m2	A_c	4.456962	m2
Reynolds Number	Re_h	420.0332	N/A	Re_c	2.969414	N/A
Heat transfer Coefficient	h_h	732.3511	W/m^2.K	h_c	1421.203	W/m^2.K
Fin Efficiency	η_{fh}	0.343775	N/A	η_{fc}	0.397628	N/A
Overall Passage Efficiency	η_{oh}	0.44549	N/A	η_{oc}	0.493405	N/A
Overall Heat Transfer Coefficient	U_h	331.2027	W/m^2.K	U_c	400.2894	W/m^2.K
Pressure Inlet	P_1	4000	kPa	P_1	4000	kPa
Pressure Outlet	P_2	4000	kPa	P_2	4000	kPa
Pressure Drop	ΔP	0.137857	kPa	ΔP	0.053026	kPa
Overall						
Heat transfer rate at Equilibrium	q	305.52	kW			
Required Effectiveness	ϵ	3.146564	N/A			
Actual Effectiveness	Λ	-1	N/A			
Nusselt Number	N_{tu}	803.1792	N/A			