DEVELOPMENT OF A SYSTEM FOR COOLING INLET AIR FOR GAS TURBINE USING CHILLED WATER

By

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
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Approved by,

____

(Dr Zainal Ambri b. Abdul Karim)

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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.

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ABSTRACT

The strong influence of climate on gas turbine behavior is well known. The output of gas turbine falls to a value that is less than the rated output under high temperature conditions that often occur during the daylight. In Malaysia, during daylight ambient temperature varies from 23°C to 38°C. This temperature is relatively high. Cooling of the turbine air intake can increase output power substantially. This is because cooled air is denser, giving the turbine higher mass flow rate and resulting in decreasing compressor specific work and specific fuel consumption.

In order to maintain inlet air at constant low temperature, an air cooling system is required to be developed at the air intake of gas turbine. A lot of researches have been done proved that by cooling the air at inlet gas turbine will increase the efficiency of gas turbine. The gas turbine engine at Gas District Cooling Plant (GDC) plant in Universiti Teknologi PETRONAS (UTP) has an advantage due to availability of a chilled water system that is produced by cogeneration plant. Heat from gas turbine exhaust is recovered to produce chilled water by the steam absorption chiller (SAC). The chilled water is then distributed to the academic building at a temperature of 6°C by pipeline into the air handling unit (AHU) in every building at UTP for air conditioning purpose and return back to GDC at a temperature of 13°C.

This study analyzed the power requirement by the cooling system, the material used, sizing and feasibility of new cooling system. Mathematical equations have been developed. The air cooling system consist of 83 units of copper tubes 19.05 mm outer diameter and 77.26 m length for each tube. A 0.35 kW pumping power is required to circulate 11.72 kg/s mass flowrate of chilled water through the system to cool 19 kg/s mass flowrate of air from variable high temperature to 20°C.

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1.0 INTRODUCTION

1.1 Background of study

Gas turbine basically is a rotary engine that converts flowing fluid into useful works. The fluid can be described as liquid or gas. A simple gas turbine is comprised of three main sections which are a compressor, a combustor, and a power turbine. The gas-turbine operates on the principle of the Brayton cycle, where compressed air is mixed with fuel, and burned under constant pressure conditions. The resulting hot gas is allowed to expand through a turbine to perform work. In a 33% efficient gas-turbine approximately two / thirds of this work is spent compressing the air, the rest is available for other work (mechanical drive, electrical generation).

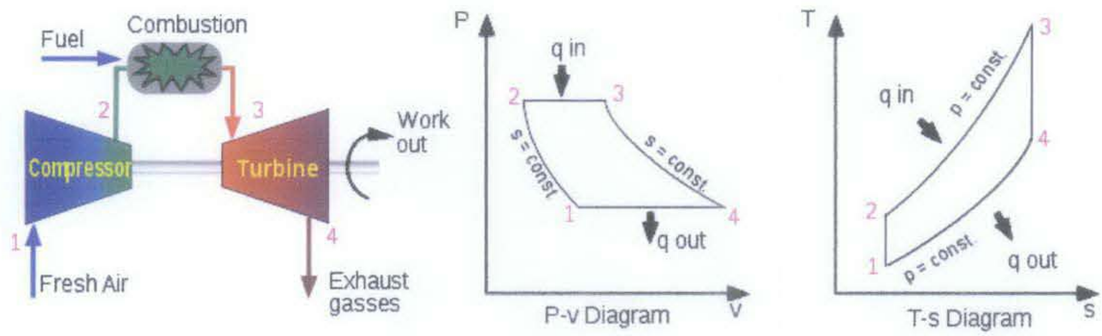


Figure 1: PV and TS diagram of a Brayton cycle mapped to actual of a gas turbine engine [15]

Efficiency of gas turbine is inversely proportional to the air ambient temperature. It is known that the efficiency of the gas turbine engine is relatively low when the ambient air temperature increases. The present trend in design towards improving efficiency and power output of gas turbine by increasing engine pressure ratio and lower the compressor inlet temperature. Improving the performance of gas turbine generator can be achieved by precooling of the inlet air. Low temperature of air increases the mass flow through the compressor, resulting in a significant decrease the compression work and increase in gas turbine power output, with a slight improvement in efficiency. In GDC plant, the air intake temperature to gas turbine varies from 23°C to 38°C. The development of air cooling system is required to decrease the temperature at air intake gas turbine to maintain air at a constant low temperature.

GDC is a power plant that supplies chilled water and electricity to Universiti Teknologi PETRONAS. It mainly consists of 2 units of 4.2 MW gas turbine. The gas turbine is used to generate electricity for the whole UTP. The hot exhaust gas is recovered to produce steam in heat recovery steam generator (HRSG) and further the steam is used in steam absorption chiller (SAC) to produce the chilled water.

In this project, the development of cooling system of air intake using chilled water is proposed. The system consists of air cooler at the air intake gas turbine, using chilled water to cool the hot air. This project will analyze the feasibility of the cooling system at air intake gas turbine using chilled water and what are the effects to the steam absorption chiller if the system is developed.

1.2 Problem statement

The current gas turbines in GDC draw air from surrounding at temperature which varies from 23°C to 38°C. To increase efficiency of the gas turbine, the air intake temperature to the gas turbine need to be reduced. The increasing of temperature of air intake is inversely proportional to the performance of the gas turbine. One effective method to reduce the temperature of the air at the intake of gas turbine is by using air cooler. The cooling liquid can be obtained from the chilled water that the GDC plant produced.

Currently GDC is operated by 2 unit of main gas turbine each produces 4.2 MW at maximum power. These turbines produce electricity from the generators and distribute to the whole of UTP. The exhaust heat from gas turbine is recovered to produce steam for SAC each having a capacity of 1250 tons refrigeration (RT). SAC produces chilled water at 6°C and distributed into the AHU as the main air-conditioning system for UTP building.

This project will analyze requirement and feasibility of the cooling system at air intake gas turbine generator using chilled water that produced by GDC and how it will effects performance of SAC with additional cooling system.

1.3 Objective

The objectives and scope of study for this project are as follows:

- To analyze the cooling requirement of the gas turbine
- To investigate the effect on SAC performance with additional cooling demand
- To develop a system for circulating chilled water for the air cooling system

1.4 Scope of studies

This study will be divided into two parts which are Final Year Project Part 1 and Final Year Project Part 2.

1.4.1 Final year project part 1

Final Year Project Part 1 scope of studies:

- Investigate gas turbine Air Intake Requirement.
- Study the Principle of Heat Transfer between circulating chilled water and air through coils and tubes.
- Study of Steam Absorption Chiller operation and capacity.

1.4.2 Final year project part 2

Final Year Project Part 2 scope of studies:

- Study of Chilled Water Distribution System.
- Design the Cooling System.
- Full Analysis and model development using software

2.0 LITERATURE REVIEW

Cooling system for air intake gas turbine is precooling of air using heat transfer concept. There are 2 type of fluid using in cooling system which are liquid and gas. In this project, the development of cooling system will use liquid (chilled water). Chilled water is produced by SAC at lower temperature of 6°C and return at 13°C. Development of cooling system will add a small slot of chilled water at 13°C flow into additional design of heat exchanger that will be place at air intake gas turbine. The hot air will flow through the heat exchanger and cooled to maintain the air at lower temperature before entering the compressor.

2.1 The effect of low ambient temperature to the gas turbine performance

Recently, El-Hadik [1] carried out a parametric study on the effects of ambient temperature, pressure, humidity and turbine inlet temperature on power and thermal efficiency. He concluded that the ambient temperature has the greatest effect on gas turbine performance, which increases with the turbine inlet temperature and pressure ratio. Reductions of power and efficiency due to a 1 K temperature change were found to be around 0.6 and 0.18% respectively [2, 3].

The most practical method up to date for inlet-air cooling is evaporative cooling, with its subsequent increase in mass flow through the turbine and the heat recovery boiler, thus boosting power and steam production [4]. However, evaporative systems are excellent in regions of high temperature and low humidity [5].

Output of gas–steam combined cycle (GTCC) is a strong function of the inlet air temperature. When the inlet air temperature drops, GTCC power output increases considerably and heat rate varies slightly [6]. A simple strategy to improve GTCC performance under high ambient temperature is to employ GTCC inlet air cooling (IAC) technologies.

A study regarding enhancement of performance of gas turbine engine by inlet air cooling and cogeneration system by Yousef [7] concluded that:

- Suction air precooling in a combined system improves power output by about 21%, overall thermal efficiency by about 38% and overall specific fuel consumption by 28%.
- Performance of the combined system is relatively less sensitive to variations in operating variables.
- Thermoeconomic evaluation shows that the combined system is viable.

Referring to Farzaneh et al. [8] the gas turbine efficiency will increase when the temperature is low. Figure 2 show the result of average daily efficiency of gas turbine using cooling system or without cooling system

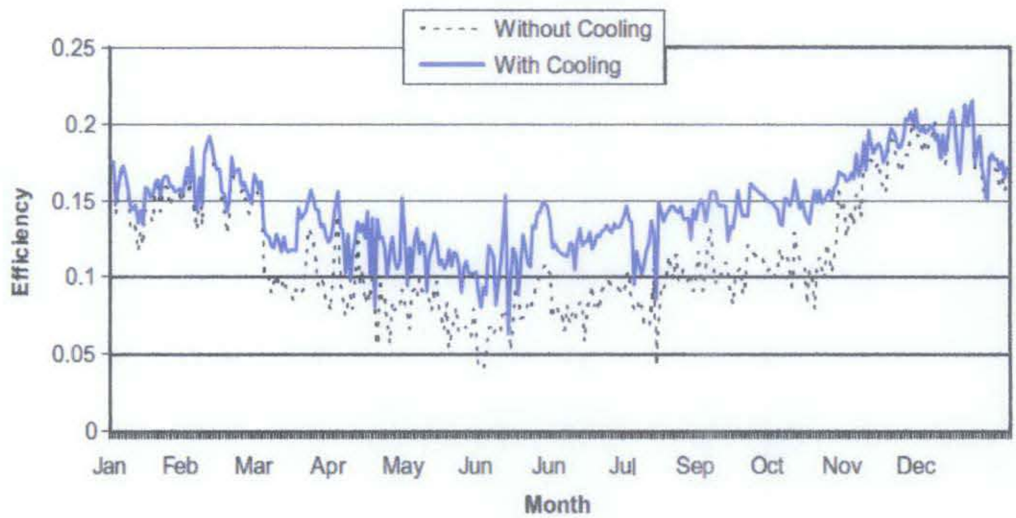


Figure 2: Average daily turbine thermal efficiency [8]

The enhanced efficiency of the gas turbine in case of inlet air cooling has been also presented. As it can be seen the actual efficiency of the gas turbine is quite low and ranging for 10% in summer to 23% during winter. Note from the figure, by cooling the inlet air temperature, the efficiency could be improved for almost 10 months and this improvement is ranging from 1.5% to 5%.

Jean Perre et. Al. [9] made a research on gas turbine performance increase using an air cooler with a phase change energy storage. According to them the possibility of improving the electric output is by cooling the gas turbine inlet air. The selected turbine is a land turbo-alternator used for Combined Heat and Power generation. Figure 3 illustrates the relationship between ambient air temperature and mechanical power for the studied gas turbine.

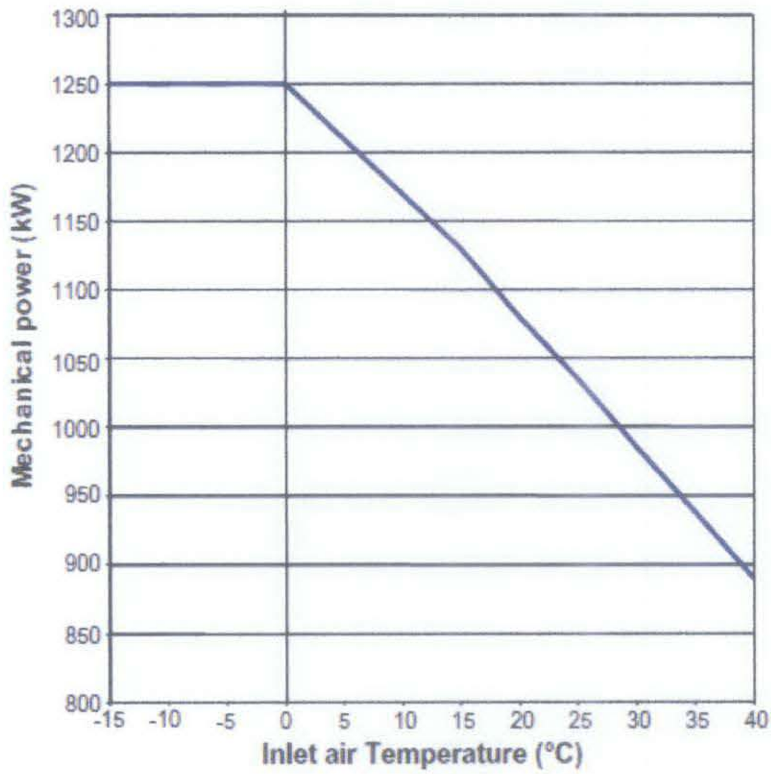


Figure 3: Effect of inlet air temperature to the mechanical power [9]

The mechanical power is of 1115 kW with an ambient air temperature of 15°C (ISO conditions, 60 % relative humidity) and power drop is 9 kW when the temperature of the air increases 1°C. So, with 40°C, the power is only 890 kW compared to the standard conditions (15°C), a fall of 225 kW that is to say 20% of the capacity. The electric output is equal (taking into account the efficiency of the alternator) to 96% of the mechanical power.

2.2 The inlet air cooling system

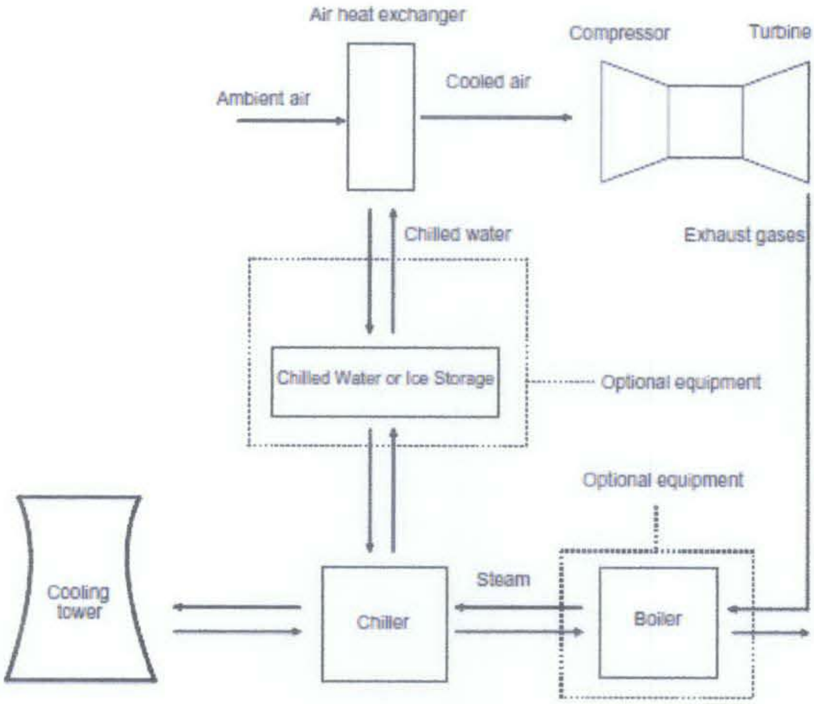


Figure 4: Generic inlet air cooling system [10]

Figure 4 is the generic inlet air cooling system according to Dawoud et al. [10]. The basic building blocks are the chiller, its cooling tower, the air heat exchanger, and the interconnecting piping. The exhaust gases of the gas-turbine may be used directly to drive the absorption chiller. Some of the available chillers are hot water or steam driven, while the others are direct fired. For this reason, a steam boiler is incorporated as optional equipment. Cold fluid from the chiller is pumped through the air heat exchanger, where the coolant is heated and returned to the chiller, while the inlet air is cooled prior to entering the compressor. The required cooling water for both the condenser and absorber of the absorption chiller is provided by using the cooling tower. Alternatively, evaporative or air-cooled condensers and absorbers might be used with some types of chillers. Including storage and its associated piping loop increases the number of system components, but allows the chiller and cooling tower components to be downsized.

A design compressor inlet air temperature of 14C has been assigned to LiBr–water chilling systems. These technologies have been compared with respect to their effectiveness in power boosting of small-size gas-turbine power plants used in two oil fields at Marmul and Fahud in the Sultanate of Oman. The LiBr-H_2O cooling reduces energy up to 11.5%.

2.3 Chilled water cooling system

Jian and Zaheeruddin [11] from Concordia University said that to model a Chilled Water Cooling System, it is necessary to consider not only the vapor compression system but also an evaporative cooler, a storage tank and building load. They also said that if the load is constant, increasing compressor motor speed would generate a higher quantity of gas refrigerant flow rate, therefore increasing the refrigerant capacity of the system.

The study on how to Improved Energy Performance of air cooled centrifugal chillers with variable chilled water flow have been made by Yu and Chan [12]. A Thermodynamic model for the chillers was developed and validated using a wide range of operating data and specifications. Based on the validated model, it was found that optimizing the control of condensing temperature and varying the evaporator's chilled water flow rate enable the COP to increase by 0.8–191.7%, depending on the load and ambient conditions. A cooling load profile of an office building in a subtropical climate was considered to assess the potential electricity savings resulting from the increased chiller COP and optimum staging of chillers and pumps. There is 16.3–21.0% reduction in the annual electricity consumption of the building's chiller plant.

2.4 Absorption refrigeration system

According to Yang et al. [13] a further possibility of cooling system for gas turbine air intake is the utilization of an absorption refrigeration machine that is driven by the heat recovered from the exhaust gases of the engine. The latter heat-recovery method can be modified to have a waste-heat boiler in the exhaust duct of the steam power-plant. The tail-end gases coming from the waste-heat boiler can be used to power the generator of the absorption machine [14].

3.0 METHODOLOGY

3.1 Project flow

The overall project work will follow the chart as below

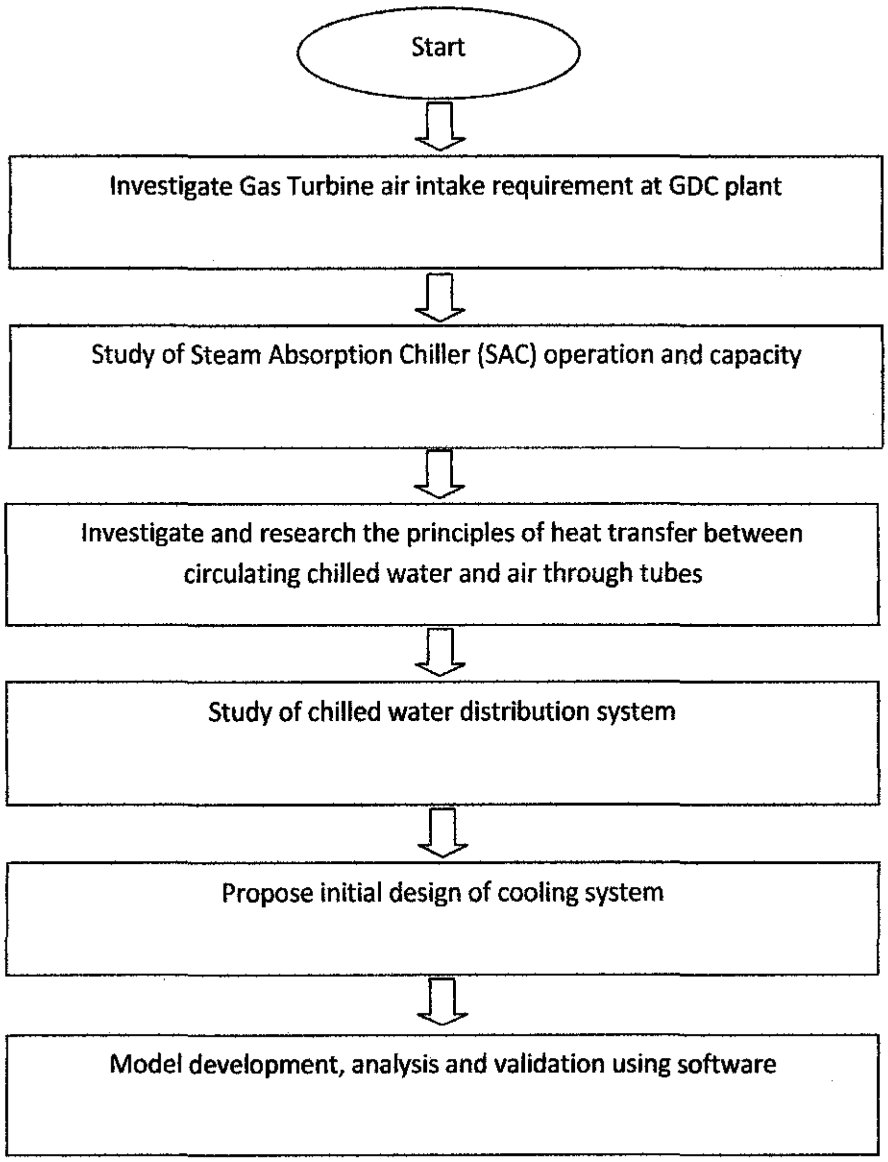


Figure 5: flow of work

3.2 Modeling flow

Overall modeling flow using Microsoft excel 2007 software:

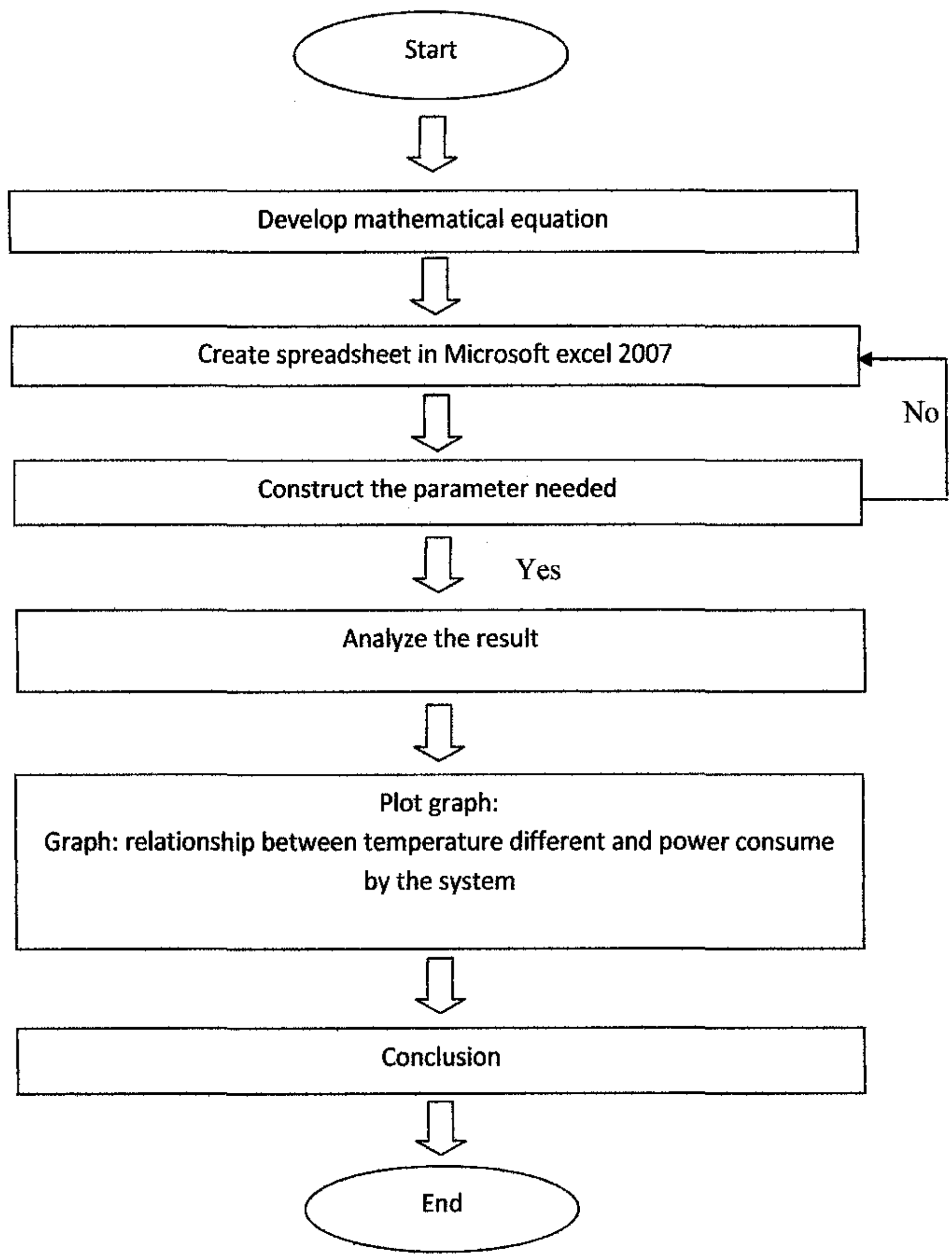


Figure 6: modeling flow work by Microsoft excel

3.3 Project Tools

The tools that have been used are as follows:

- **Software for modeling**

  For this project the author will do the modeling by using Microsoft Excel to determine the feasibility of the system.

- **Information Resource Centre**

  The library is the source of information that can be used to extract important information for this project. The main focus of research is at the journal section and the available book. Information gain from this location is the fundamentals of thermodynamic and heat transfer studies.

- **UTP Gas District Cooling (GDC) plant**

  GDC plant is the source for the author to get information regarding gas turbine performance and look at the current condition and situation. Information gain from this source is the parameters and the actual reading of gas turbine.

- **Online journal reviews**

  Online journals can be found on the internet. The author used sciencedirect.com website to find the journals. All available journals are helpful and related to the author project.

3.4 Project planning for final year project 1

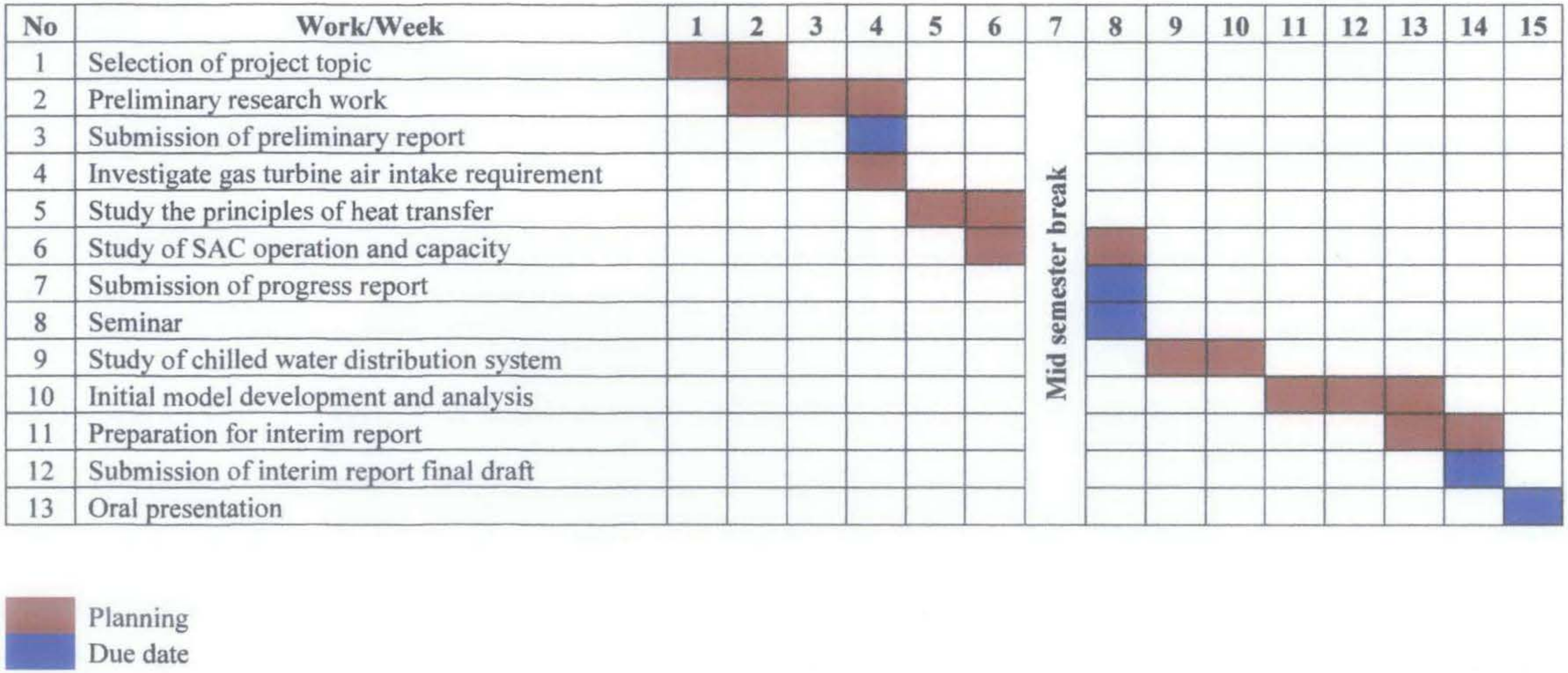


Figure 7: Gantt chart for fyp 1

3.5 Project Planning for final year project 2

No	Work/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	GDC request letter															
2	Survey material for HE tube,diameter @ GDC															
3	Determine U value															
4	Determine Heat Exchanger sizing															
5	Determine power consume by HE and develop graph															
6	Determine cost of power consume and develop graph															
7	Determine power usage from current system															
8	Determine power saving															
9	Determine power loss in the system															
10	Summarize overall system design															
11	Develop model															
12	Progress Report															
13	Project work continue															
14	Pre-edx															
15	Submission of draft report															
16	Submission of dissertation (soft-bound)															
17	Submission of technical paper															
18	Oral presentation															
19	Submission of dissertation (hard-bound)															

Planning
Due date

Figure 8: Gantt chart for fyp 2

4.0 MATHEMATICAL MODEL

4.1 Parameters

All known and unknown parameter have been listed below. In some cases, assumptions need to be done for the sake of simplicity of the analysis.

Table 1: Parameters

Parameters	Value
Maximum capacity of SAC, $Q_{s,max}$	2500 RT
Mass flowrate of water in chilled water main pipe, $m_{w,main}$	278 kg/s
Chilled water main pipe inlet diameter, D	750 mm
Mass flowrate of air at inlet of GTE, m_a	19 kg/s
Chilled water inlet temperature, T_{cws}	6 ^{0}C
Chilled water outlet temperature, T_{cwr}	13 ^{0}C
Velocity of water, V_w	0.629 m/s
Surface area, $A_{surface}$	unknown
Number of tube, N	unknown
Mass flowrate of water, m_{water}	unknown
Tube length, L	unknown
Heat transfer coefficient, U	unknown
Constant specific heat of water at constant pressure, $C_{p,water}$	4.188 KJ/kg-^{0}K
Density of water, $\dot{\rho}_{water}$	1000 kg/m3
Pipe inlet diameter, D_i	16.91 mm
Pipe outlet diameter, D_o	19.05 mm
Pipe thickness, t	1.07 mm

4.2 Assumptions

The purpose of assumption is to solve the problem. All assumptions for this project are as below:

- The cool air at outlet air cooler is 20^{o}C
- The temperature of chilled water supply and return are fix
- The mass flowrate of air is constant
- Ambient temperature varies from 23^{o}C to 38^{o}C

4.3 Current Steam Absorption System capacity

Current capacity of chilled water system produced by SAC is calculated. GDC plant is able to produce 2500 RT chilled water capacity by 2 units of SAC. Current energy used can be calculated by

$$
\begin{aligned}
Q_{actual} &= m_{w,main} \text{ x } C_{pw} \text{ x } (T_{cwr}-T_{cws}) \\
&= (278 \text{ kg/s}) \text{ x } (4.188\text{KJ/kg-K}) \text{ x } (7\text{K}) \\
&= 8149.8 \text{ kJ/s x } (1 \text{ RT}/3.517 \text{ kJ/s}) \\
&= 2317.26 \text{ RT}
\end{aligned}
$$

By knowing the value, the amount of capacity after new cooling system is applied shall not be more than the maximum capacity allowed which is 2500RT. Thus GDC has remaining availability of 183 RT capacity for the new cooling system. Further analysis will determine feasible amount of power required in generating new cooling system.

4.4 Principle of heat transfer between circulating chilled water and air

The heat equilibrium between circulating chilled water and flowing air is calculating according to this equation:

$$Q_c = m_a \text{ x } C_{pa} \text{ x } (T_{ha}-T_{ca}) = m_w \text{ x } C_{pw} \text{ x } (T_{cwr}-T_{cws})$$

Where,

Q_c = total of energy, KW

m_a = mass flowrate of air, 19 kg/s

m_w = mass flowrate of chilled water, kg/s

C_{pa}= average specific heat at constant pressure of air, 1.005 KJ/kg-K

C_{pw}=water respectively, 4.188KJ/kg-K

T_{cws} = chilled water temperature inlet, 279K

T_{cwr} = chilled water temperature outlet, 286K

T_{ha} = maximum ambient ambient air temperature, 311K

T_{ca} = cool air temperature, 293K

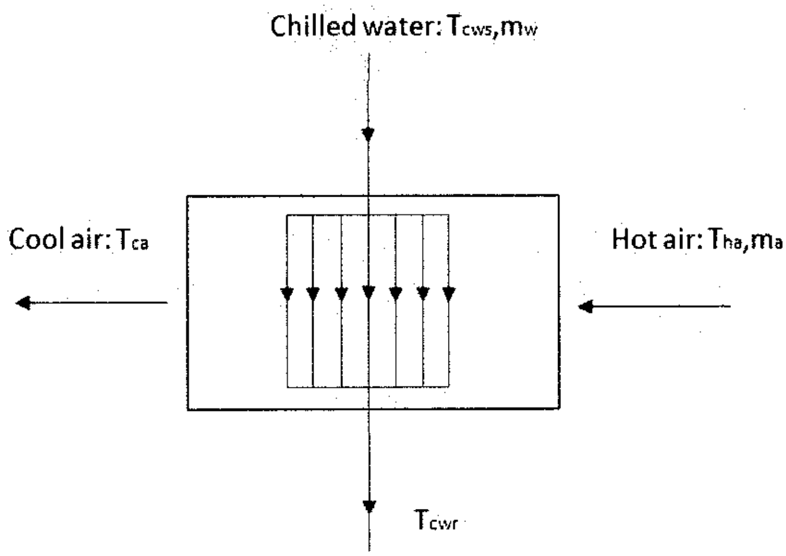


Figure 9: System diagram

4.4.1 Mass flowrate of water in tubes

The m_w can be calculated using this equation

$m_w = m_a \times C_{pa} \times (T_{ha}-T_{ca}) / C_{pw} \times (T_{cwr}-T_{cws})$

$= [(19\ kg/s)(1.005 KJ/Kg\text{-}K)(311-293)\ K] / [(4.188 KJ/kg\text{-}K)(286-279)]\ K$

$= 11.72\ kg/s$

Where,

m_a = mass flowrate of air after cooled and enter air intake, kg/s

m_w = mass flowrate of water entering the heat exchanger tube, kg/s

C_{pa} = constant specific heat of air, kJ/kg-K

C_{pw} = constant specific heat of water, kJ/kg-K

T_{cws} = chilled water inlet, K

T_{cwr} = chilled water outlet, K

T_{ha} = maximum ambient temperature, K

T_{ca} = cool air temperature, K

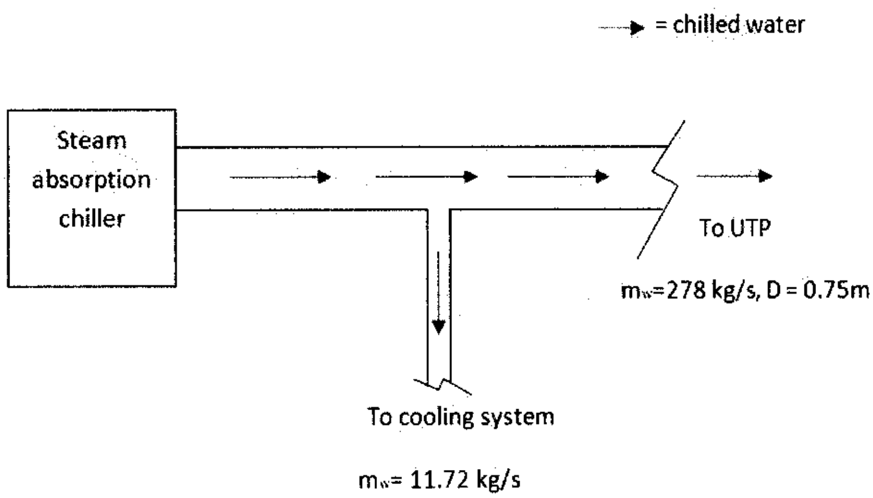


Figure 10: distribution of chilled water mass flowrate

4.4.2 Energy extracted by air cooler

The mass flowrate of water is determined. The Energy used by the system can be calculated using equation:

$$Q_c = m_w \times C_{pw} \times (T_{cwr} - T_{cws}) = (11.72\text{kg/s}) \times (4.188\text{kJ/kg-K}) \times (7\text{K})$$

$$= 343.58 \text{ KW}$$

$$= 97.7 \text{ RT}$$

To cool the air at maximum ambient temperature of 38°C at 19 kg/s needs amount of 11.72 kg/s mass flowrate of chilled water and 97.7 RT of energy.

4.4.3 Material and overall heat transfer coefficient

For the heat exchanger tubes the material used is copper. For composite material in series, the overall heat transfer coefficient U due to combined conduction and convection heat transfer is given by

$$
\begin{aligned}
(1/U) &= (1/h_A) + \sum(x/\lambda) + (1/h_B) \\
&= (1/55) + (0.00107/401) + (1/5250) \\
&= 0.0184\ m^2K/W
\end{aligned}
$$

Where,

h_A= heat transfer coefficient for air, W/m^2K

h_B= heat transfer coefficient for water, W/m^2K

λ= thermal conductivity of materials, W/mK

x= tube wall thickness in, m

In this study assume $h_A = 55\ W/m^2K$, $h_B = 5250\ W/m^2K$, λ=401W/mK for copper [17] and assume wall thickness is 1.07mm [20] giving the overall heat transfer equation equal to 54.42 W/m^2K.

4.4.4 Log mean temperature difference

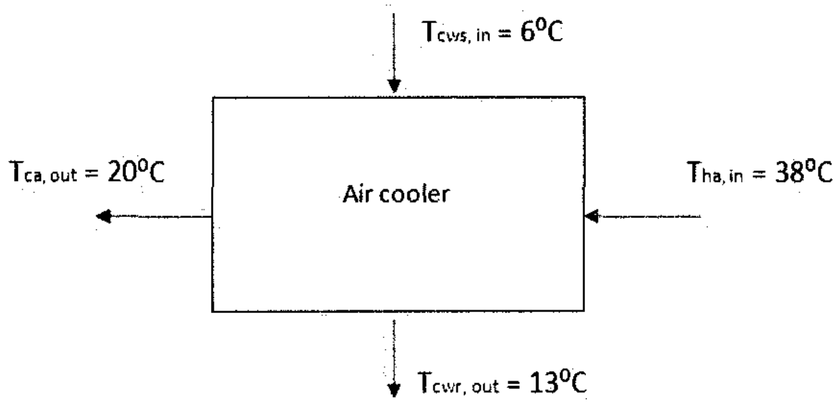


Figure 11: inlet and outlet temperature

The log mean temperature difference (LMTD) is used to determine the temperature driving force for heat transfer in flow systems,

$$\Delta T_{lm} = \Delta T_{in} - \Delta T_{out} / \ln (\Delta T_{in} / \Delta T_{out})$$
$$= (32-7)\ K / \ln(32/7)$$
$$= 16.45\ K$$

Where,

$$\Delta T_{in} = T_{ha,in} - T_{cws,in}$$
$$\Delta T_{out} = T_{ca,out} - T_{cwr,out}$$

4.4.5 Heat exchanger surface area

After finding U, log mean temperature and power consume by the system, the surface area of heat exchanger can be determined by using this equation,

$$A_{surface} = Q_c / U\ \Delta T_{lm}$$
$$= (343580W) / (54.42\ W/m^2\text{-}K \times 16.45K)$$
$$= 383.80\ m^2$$

Where,

Q_c = energy extracted by chilled water to cool the hot air, kW

U = overall heat transfer coefficient, W/m^2-K

$A_{surface}$ = total contact area with hot air, m^2

ΔT_{lm} = log mean temperature, ΔT_{in}-ΔT_{out}/In ($\Delta T_{in}/\Delta T_{out}$), K

4.4.6 Number of tubes in heat exchanger

Assume Inner diameter of copper pipe is 16.91mm [21], and the pipe velocity is 0.629 m/s. Thus the number of tubes in heat exchanger can be determined by

$$N = m_w / (\dot{\rho}_{water} \text{ x } A_{tube} \text{ x } V_w)$$
$$= (11.72 \text{ kg/s}) / (1000\text{kg/m}^3 \text{ x } 2.25\text{x}10^{-4}\text{m}^2 \text{ x } 0.629\text{m/s})$$
$$= 83 \text{ tubes}$$

Where,

$\dot{\rho}_{water}$ = density of water, kg/m^3

A_{tube} = cross-sectional area of tube, m^2

V_w = velocity of water in tube, m/s

N = number of coils in heat exchanger

m_w = chilled water mass flowrate, kg/s

4.4.7 Length of tubes

Assume outer diameter of tube is 19.05mm [21], the length of single tube is determined by

$$L = A_{surface} / \pi \text{ x } D_o \text{ x } N$$
$$= 383.80 \text{ m}^2 / (\pi \text{ x } 0.01905\text{m x } 83\text{tubes})$$
$$= 77.26 \text{ m}$$

Where,

$A_{surface}$ = surface area of air cooler, m^2

D_o = outer diameter of tubes, m

L = length of single coil, m

N = number of tube in air cooler

4.4.8 Type of flow in tubes

Type of flow can be determined using Reynolds number given by,

$$Re = \rho \times V_w \times D_i / \mu$$
$$= [(1000 kg/m^3) \times (0.629 m/s) \times (0.01691 m)]/(1.307 \times 10^{-3} kg/m.s)$$
$$= 8138 > 4000 \text{ (Turbulent)}$$

Where,

ρ = density of fluid, kg/m^3

V_w = velocity of water, m/s

D_i = internal diameter of tube, m

μ = dynamic viscosity, kg/m.s

The Reynolds number is more than 4000 thus the type of flow is turbulent. To find pressure drop along the tube for turbulent flow,

$$P_{drop} = (f \times L/D_i) \times (\rho \times V_w^2 / 2) \times 83 \text{ tubes}$$
$$= [(0.033 \times 77.26)/0.01691] \times [(1000 \times 0.629^2)/2] \times 83$$
$$= 2475565.51 \text{ N/m}^2 \text{ or (Pa)}$$

Where,

f = friction coefficient determined using moody chart, relative pipe roughness (ε/D_i) of 8.87x10-5 for copper tube [appendix 1]

L = length of tube, m

D_i = pipe inner diameter, m

$\dot{\rho}$ = density of water, kg/m^3

V_w = velocity of water, m/s

4.4.9 Pumping power

The pumping work required to circulate chilled water to overcome pressure losses through the loop is given by,

$$P_{pump} = P_{drop} \text{ x } A_{tube} \text{ x } V_w$$
$$= 2475.565 \text{ x } [\pi/4 \text{ x } (0.01691)^2] \text{ x } 0.629$$
$$= 0.35 \text{ kW}$$

Where,

A_{tube} = cross-sectional area of tube, m^2

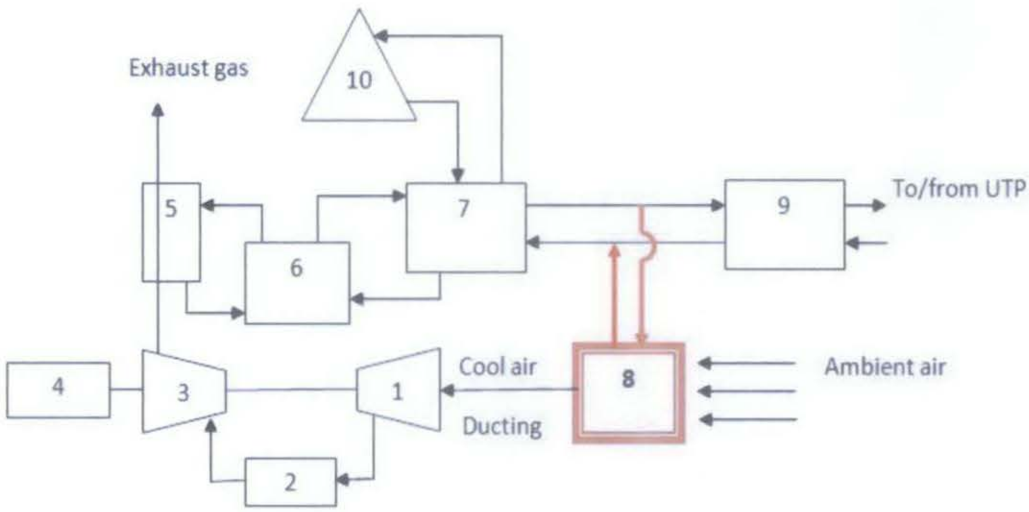
P_{drop} = pressure drop in pipe, kPa

V_w = chilled water velosity, m/s

P_{pump} = pumping power, kW

5.0 RESULTS AND DISCUSSION

5.1 Basic design



1-compressor; 2-combustor; 3-turbine; 4-generator; 5-HRGS; 6-boiler;

7-steam absorption chiller; 8-air cooler; 9-heat exchanger; 10-cooling tower

Figure 12: Basic design of cooling system

Figure 12 illustrates the basic system diagram consist of the current system in GDC and the new cooling system (red line marking). Hot exhaust gas from turbine is recovered heat recovery steam generator (HRSG) to produce steam. Steam is used in SAC to produce chilled water at 6°C and 278 kg/s at main pipe. A small tapping of pipe with mass flowrate of 11.72 kg/s enter into the air cooler to cool the hot ambient air from variable (23°C to 38°C) to 20°C.

5.2 Feasibility of system

To examine the feasibility of the new system there must be no interruption with current system capacity. Thus,

$$(Q_{sac,max} - Q_{actual}) - Q_c = (2500\text{-}2317.26)\ \text{RT} - 97.7\ \text{RT}$$
$$= 85.04\ \text{RT}$$

Total amount energy usage by UTP is about 2317.26RT. SAC is able to produce maximum capacity of 2500RT. After the new system is added the total amount is still below the maximum capacity that SAC can provide. Development of new system is feasible since capacity of SAC still can sustain about 85.04 RT remaining. Thus developing this air cooling system will not affect current system in GDC power plant.

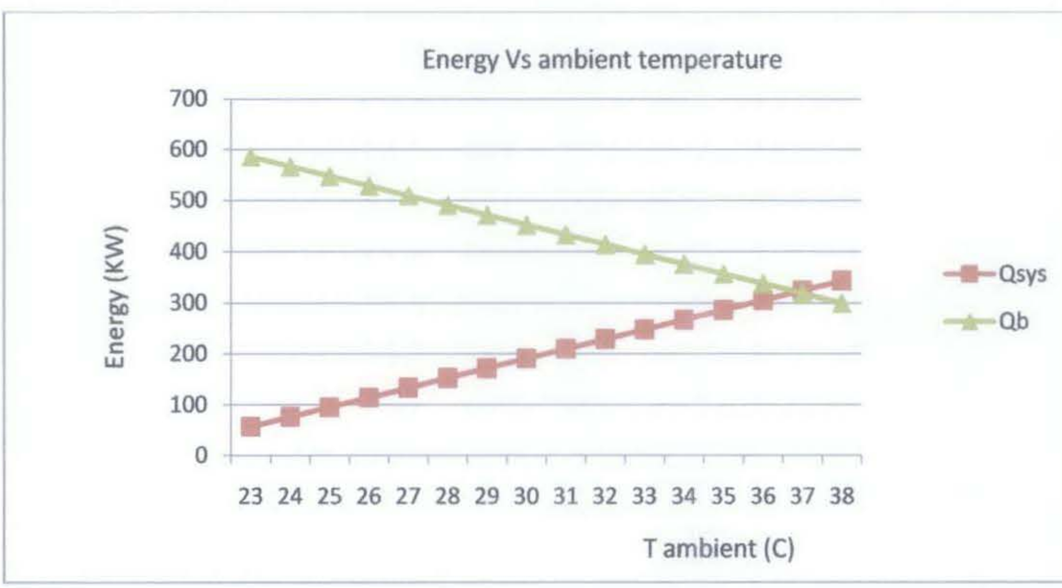


Figure 13: Energy versus ambient temperature

Figure 13 shows the capacity of cooling system,Q_{sys} and remaining capacity,Q_b. It shows that when maximum power required by cooling system is achieved at 38°C the SAC has remaining energy capacity about 299.79 kW. The added of new cooling system will not affect SAC current capacity and other system at GDC. Thus, any modification to the current system is not required.

5.3 Capacity of the air cooling system

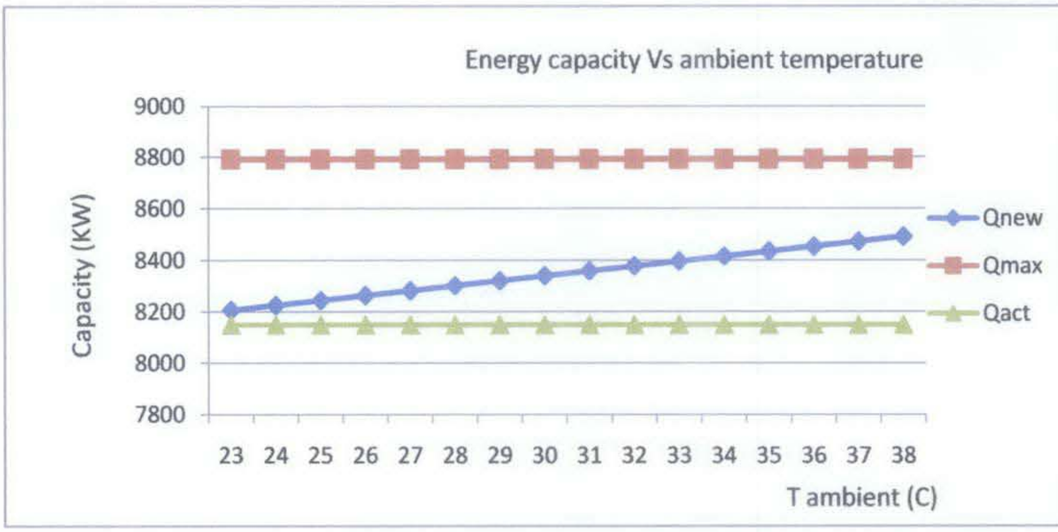


Figure 14: Energy capacity versus ambient temperature

Figure 14 shows the new energy capacity, actual energy capacity and maximum energy capacity (Q_{new},Q_{act} and Q_{max} respectively). It shows that the new system capacity, when added with current capacity is still on ranged where the maximum it can reach is 8492kW. This is not exceed the maximum capacity by steam absorption chiller. It is important to know that the power plant not need to add additional SAC or use any other supportive input machine (e.g electrical chiller) to produce chilled water for the new system. Thus, the cost can be minimized.

5.4 Development of air cooling system

Calculation is done in determining the mass flowrate, surface area, number of tubes and length of tubes of heat exchanger. Copper is chosen as tube material. Copper has good conductivity and gives the highest overall heat transfer coefficient compared to other material. In addition density of cooper is lower than stainless steel, resulting less weight of design. Figure 15 shows the minimum and maximum mass flowrate of water with respect to energy used by the air cooler.

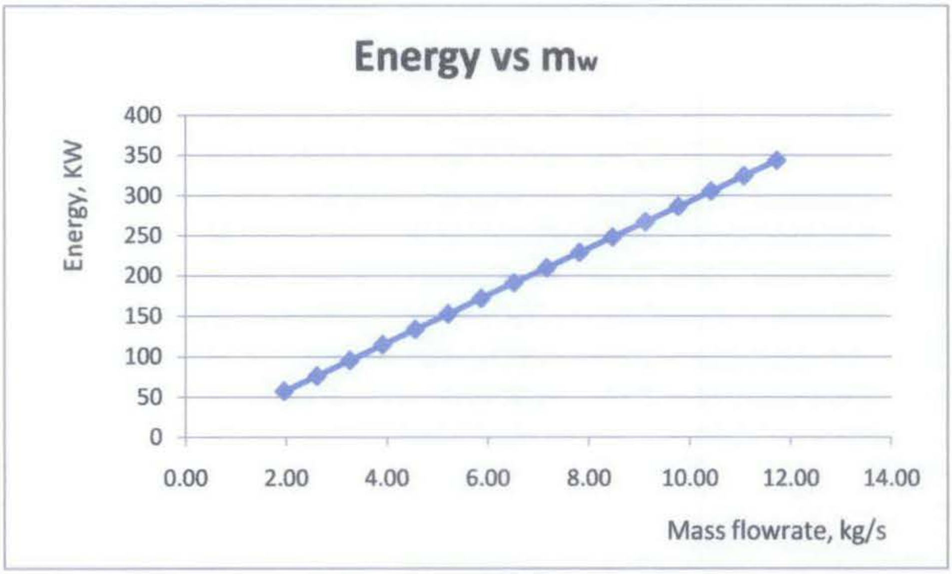


Figure 15: Energy versus water mass flowrate

To cool the air at temperature of 38°C, the more energy is required resulting mass flowrate of water to increase. In this design, the maximum energy used is about 343.58 kW and water mass flowrate of 11.72 kg/s. Air is estimated to cool from maximum ambient temperature of 38°C to 20°C using 83 tubes and length of each tube is 77.26 m. The total amount of surface area of air cooler is calculated to be 383.80 m^2.

It is important to know the type of flow inside the tube whether laminar or turbulent. For this study, the flow is turbulent. To find the pressure drop inside the pipe for turbulent flow, the tube friction coefficient of is found using moody chart. The relative pipe roughness ($\mathcal{E}/D_i$) line of 8.87×10^{-5} for copper tube intersect with Reynold's number in moody chart, give the value of 0.033. The total pressure drop in 83 tubes is 2475.57 kPa. To overcome the pressure loss inside the tubes, pumping power of 0.35 kW is needed to circulate chilled water through the system.

P_{drop} (Pa)	Tube length (m)
160.21	5
320.42	10
480.63	15
640.84	20
801.05	25
961.26	30
1121.47	35
1281.68	40
1441.89	45
1602.10	50
1762.31	55
1922.52	60
2082.73	65
2242.94	70
2403.15	75
2563.36	80

Table 2: Pressure drop versus tube length

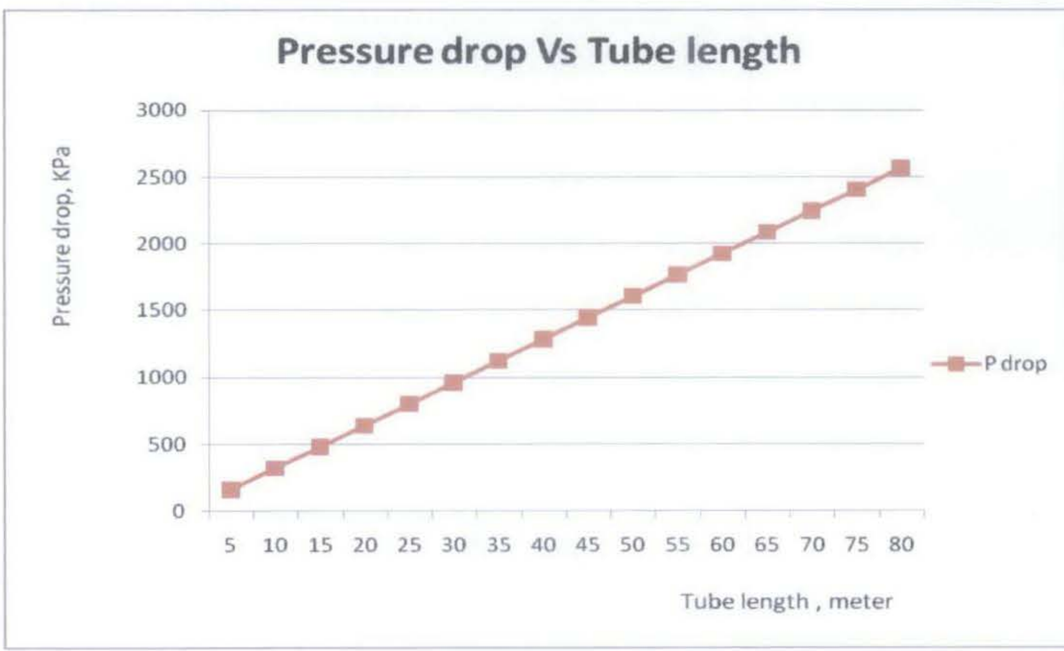


Figure 16: Graph pressure drop versus tube length

6.0 CONCLUSION AND RECOMMENDATION

By reducing the temperature of air at intake will increase the performance of gas turbine. To improve the efficiency of gas turbine by adding a cooling system at it air intake any interruption of current system must be avoided to make sure the additional system is reliable. Furthermore, the power need to run the system should be less than that the power improvement of the gas turbine so that this system can be successfully developed.

Based on analysis, the design of the cooling system consists of:

Table 3: design of air cooling system

Parameter	Value (unit)
Maximum energy required by air cooler	343.58 (kW)
Maximum mass flowrate of water	11.72 (kg/s)
Velocity of water	0.629 (m/s)
Surface area of air cooler	383.80 (m^2)
Tube inner diameter	16.91 (mm)
Tube outlet diameter	19.05 (mm)
Tube thickness	1.07 (mm)
Number of tubes	83
Length of tubes	77.26 (m)
Pumping power	0.35 (kW)

The development of new air cooling system for air inlet gas turbine in GDC does not affect the performance of current system and allowable capacity. Thus, the development of cooling system is feasible to develop.

In recommendation, the author suggested that the air cooler is turn on for 24 hours. The ambient air temperature at night is still high which is around 23°C to 29°C. The automatic control valve is suggested to be used in the system since the valve will control the amount of chilled water mass flowrate with respect to the ambient air temperature. This will save a lot of energy especially when the ambient air temperature is low (e.g, at 23°C), since the mass flowrate of chilled water is minimum. Furthermore, the author suggested that to calculate the surface area, it is required to add correction factor (f) in calculation. The correction factor is important for real condition compared to the theory. Thus it will give more accurate result and reduce errors. Last but not least, the author suggested that a small prototype of heat exchanger is created by a student to study the detail and any factors that make the system fails.

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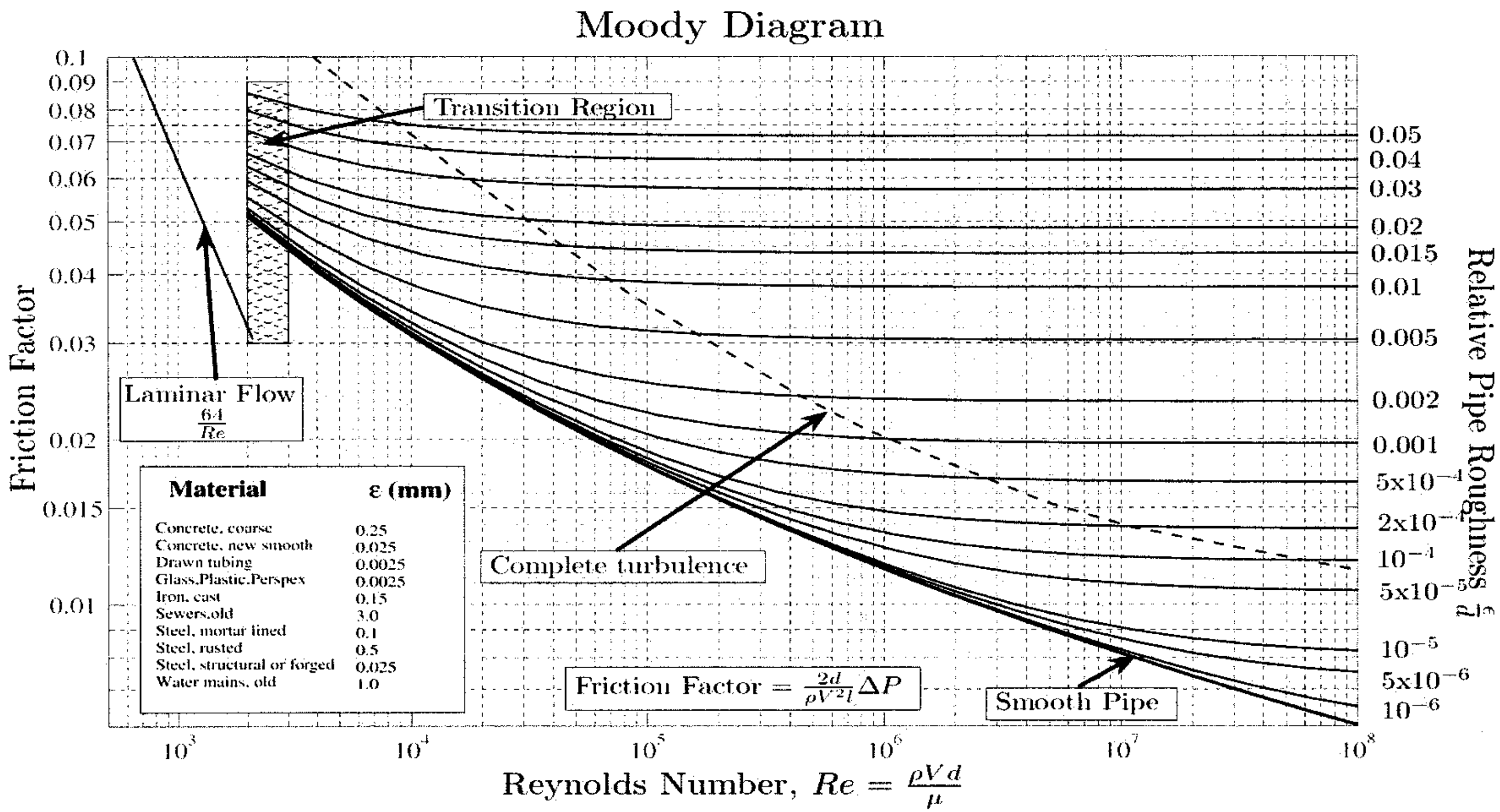
Appendix 1: Moody chart



Pipe/Duct material	Roughness (mm)
Glass, plastic	0.0 (smooth)
Drawn tubing (copper, brass, etc.)	0.0015
Commercial steel and wrought iron	0.045
Cast iron, asphalt coated	0.12
Galvanized iron	0.15
Cast iron, uncoated	0.26
Concrete	0.3-3.0
Riveted steel	0.9-9.0
Fiberglass duct, rigid	0.9
Fiberglass duct liner, spray coated	3.0
Flexible duct, fabric and wire, fully extended	1.0-4.6
Flexible duct, metallic, fully extended	1.2-2.1

Roughness for new pipe and ducts is a function of material and manufacturer. Values from Moody L.F, *Friction Factors for Pipe Flow* (1944).

Appendix 2: Thermal conductivity of materials

Thermal Conductivity - *k* - *W/(m.K)*			
Material/Substance	Temperature - °C		
	25	125	225
Acetone	0.16		
Acetylene (gas)	0.018		
Acrylic	0.2		
Air, athmosphere (gas)	0.024		
Alcohol	0.17		
Aluminum	250	255	250
Aluminum Oxide	30		
Ammonia (gas)	0.022		
Antimony	18.5		
Argon (gas)	0.016		
Asbestos-cement board	0.744		
Asbestos-cement sheets	0.166		
Asbestos-cement	2.07		
Asbestos, loosely packed	0.15		
Asbestos mill board	0.14		
Asphalt	0.75		
Balsa wood	0.048		
Bitumen	0.17		
Benzene	0.16		
Beryllium	218		
Bitumen	0.17		
Blast furnace gas (gas)	0.02		
Brass	109		
Breeze block	0.10 - 0.20		
Brick dense	1.31		
Brickwork, common	0.6 -1.0		
Brickwork, dense	1.6		
Cadmium	92		
Calcium silicate	0.05		
Carbon	1.7		
Carbon dioxide (gas)	0.0146		
Cement, portland	0.29		
Cement, mortar	1.73		
Chalk	0. 09		
Chlorine (gas)	0.0081		
Chrome Nickel Steel (18% Cr, 8 % Ni)	16.3		
Clay, dry to moist	0.15 - 1.8		
Clay, saturated	0.6 - 2.5		
Cobalt	69		
Concrete, lightweight	0.1 - 0.3		
Concrete, medium	0.4 - 0.7		
Concrete, dense	1.0 - 1.8		
Concrete, stone	1.7		
Constantan	22		
Copper	401	400	398
Corian (ceramic filled)	1.06		

Appendix 3: ASTM B 280 Copper Tube – Air Conditioning and Refrigeration

Seamless copper tube according ASTM B 280 – Standard Specification for Seamless Copper Tube for Air Conditioning and Refrigeration Field Service - is intended for use in the connection, repairs, or alternations of air conditioning or refrigeration units in the field.

AC/R Copper Tubes is used in the Air Conditioning and Refrigeration Field Service Industry and is designated by the tube's actual OD. B 280 tube is produced in straight lengths or coils.

Nominal Size *(inches)*	Annealed (A) or Drawn Temper (D)	Outside Diameter *(inches)*	Inside Diameter *(inches)*	Wall Thickness *(inches)*	Cross Sectional Area of Bore *(square inches)*
1/8	A	0.125	0.065	0.030	0.0033
3/16	A	0.187	0.128	0.030	0.0129
1/4	A	0.250	0.190	0.030	0.0284
5/16	A	0.312	0.248	0.032	0.0483
3/8	A	0.375	0.311	0.032	0.076
	D	0.375	0.315	0.030	0.078
1/2	A	0.500	0.436	0.032	0.149
	D	0.500	0.430	0.035	0.145
5/8	A	0.625	0.555	0.035	0.242
	D	0.625	0.545	0.040	0.233
3/4	A	0.750	0.680	0.035	0.363
	A	0.750	0.666	0.042	0.348
	D	0.750	0.666	0.042	0.348
7/8	A	0.875	0.785	0.045	0.484
	D	0.875	0.785	0.045	0.484
1 1/8	A	1.125	1.025	0.050	0.825
	D	1.125	1.025	0.050	0.825
1 3/8	A	1.375	1.265	0.055	1.26
	D	1.375	1.265	0.055	1.26

Appendix 4: Dynamic and kinematic viscosity in SI unit

Temperature - t - (^oC)	Dynamic Viscosity - μ - ($N\ s/m^2$) x 10^{-3}	Kinematic Viscosity - v - (m^2/s) x 10^{-6}
0	1.787	1.787
5	1.519	1.519
10	1.307	1.307
20	1.002	1.004
30	0.798	0.801
40	0.653	0.658
50	0.547	0.553
60	0.467	0.475
70	0.404	0.413
80	0.355	0.365
90	0.315	0.326
100	0.282	0.294

Appendix 5: Microsoft excel data spreadsheet 1

T_a(°C)	T_{chws}(°C)	T_{chwr}(°C)	Tcool	T_{amb}-T_{lm}	C_{pa}	C_{pw}	$\dot{\rho}_w$ (kg/m³)	m_a(kg/s)	m_w(kg/s)	Q_{sys}(KW)	U_{cop}(w/m²-k)	ΔT_{lm}	As(m²)	Di (mm)	Do(mm)	At (m²)	Vw(m/s)	P_{drop}(pa)	P_{pump}(KW)
23	6	13	20	11.73	1.005	4.188	1000	19	1.95	57.29	54.42	11.27	93.40	16.91	19.05	0.0002246	0.629	2475565.51	0.35
24	6	13	20	12.35	1.005	4.188	1000	19	2.61	76.38	54.42	11.65	120.51	16.91	19.05	0.0002246	0.629	2475565.51	0.35
25	6	13	20	12.98	1.005	4.188	1000	19	3.26	95.48	54.42	12.02	145.99	16.91	19.05	0.0002246	0.629	2475565.51	0.35
26	6	13	20	13.62	1.005	4.188	1000	19	3.91	114.57	54.42	12.38	170.01	16.91	19.05	0.0002246	0.629	2475565.51	0.35
27	6	13	20	14.26	1.005	4.188	1000	19	4.56	133.67	54.42	12.74	192.74	16.91	19.05	0.0002246	0.629	2475565.51	0.35
28	6	13	20	14.90	1.005	4.188	1000	19	5.21	152.76	54.42	13.10	214.30	16.91	19.05	0.0002246	0.629	2475565.51	0.35
29	6	13	20	15.55	1.005	4.188	1000	19	5.86	171.86	54.42	13.45	234.79	16.91	19.05	0.0002246	0.629	2475565.51	0.35
30	6	13	20	16.20	1.005	4.188	1000	19	6.51	190.95	54.42	13.80	254.32	16.91	19.05	0.0002246	0.629	2475565.51	0.35
31	6	13	20	16.86	1.005	4.188	1000	19	7.16	210.05	54.42	14.14	272.96	16.91	19.05	0.0002246	0.629	2475565.51	0.35
32	6	13	20	17.52	1.005	4.188	1000	19	7.82	229.14	54.42	14.48	290.79	16.91	19.05	0.0002246	0.629	2475565.51	0.35
33	6	13	20	18.18	1.005	4.188	1000	19	8.47	248.24	54.42	14.82	307.88	16.91	19.05	0.0002246	0.629	2475565.51	0.35
34	6	13	20	18.85	1.005	4.188	1000	19	9.12	267.33	54.42	15.15	324.28	16.91	19.05	0.0002246	0.629	2475565.51	0.35
35	6	13	20	19.52	1.005	4.188	1000	19	9.77	286.43	54.42	15.48	340.05	16.91	19.05	0.0002246	0.629	2475565.51	0.35
36	6	13	20	20.20	1.005	4.188	1000	19	10.42	305.52	54.42	15.80	355.22	16.91	19.05	0.0002246	0.629	2475565.51	0.35
37	6	13	20	20.87	1.005	4.188	1000	19	11.07	324.62	54.42	16.13	369.85	16.91	19.05	0.0002246	0.629	2475565.51	0.35
38	6	13	20	21.55	1.005	4.188	1000	19	11.72	343.71	54.42	16.45	383.96	16.91	19.05	0.0002246	0.629	2475565.51	0.35

Appendix 6: Microsoft excel data spreadsheet 2

T_{amb}	Q_{sys}(KW)	Qs,new(KW)	Qb (KW)	Qs,max	Qact
23	57.29	8206.29	586.22	8792.50	8149.80
24	76.38	8225.38	567.12	8792.50	8149.80
25	95.48	8244.48	548.03	8792.50	8149.80
26	114.57	8263.57	528.93	8792.50	8149.80
27	133.67	8282.67	509.83	8792.50	8149.80
28	152.76	8301.76	490.74	8792.50	8149.80
29	171.86	8320.86	471.65	8792.50	8149.80
30	190.95	8339.95	452.55	8792.50	8149.80
31	210.05	8359.05	433.46	8792.50	8149.80
32	229.14	8378.14	414.36	8792.50	8149.80
33	248.24	8397.24	395.26	8792.50	8149.80
34	267.33	8416.33	376.17	8792.50	8149.80
35	286.43	8435.43	357.08	8792.50	8149.80
36	305.52	8454.52	337.98	8792.50	8149.80
37	324.62	8473.62	318.89	8792.50	8149.80
38	343.71	8492.71	299.79	8792.50	8149.80