Cold Energy Recovery from LNG Regasification Process

by

Amirul Ariff Bin Sazali

15068

Dissertation submitted in partial fulfillment of

the requirements for the

Bachelor of Engineering (Hons)

(Chemical)

January 2015

Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

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Approved by,

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(Associate Professor Dr. Shuhaimi Mahadzir)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

January 2015

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 that as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by underspecified sources or persons.

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AMIRUL ARIFF BIN SAZALI

ABSTRACT

Energy sector has been getting a lot of attention these past years as the gap between supply and demand of energy is getting bigger day by day. Energy resources must be used efficiently to ensure that there will be a continuous and uninterrupted energy source in the future. This also applies to Liquefied Natural Gas (LNG) which nowadays is one of the widely used sources of energy. LNG must go through regasification process before being used for industrial and domestic purposes. Known to be at a very cold temperature (-162°C), this process normally uses the thermal energy of sea water as heating medium. Unfortunately, this process releases a large amount of energy (about 800 kJ per kg of LNG) as the cold sea water will be discarded back to the sea. This study proposed to recover the large amount of cold energy from LNG regasification process using the Rankine and Brayton power cycles for electricity generation. Aspen Hysys software is used to design and simulate an improved system using the Kelloggs process as the base case. The results show that after simulation and parameter manipulation, the proposed combine cycle has thermal efficiency of 38.8% and thermal efficiency of 65.52% using water and carbon dioxide as the working fluid.

ACKNOWLEDGEMENT

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CHAPTER 1: INTRODUCTION

1.1 Background of study

Natural gas (NG) is a fossil fuel that is formed after being pressurized and heated for millions of years underneath thousands meters of soil and rock. It is a form of hydrocarbon made up of compounds of hydrogen and carbon. Qiang, Yanzhong and Jiang (2004) suggested that NG is becoming the third biggest energy resource and known to be the cleanest burning fossil fuel as it produces less emissions and pollutants than both coal and petrol.

Normally, NG will be transported from the gas field far from the land to the power plant in the form of Liquefied Natural Gas (LNG). It is condensed into liquid at atmospheric pressure by being cooled to about -162 °C. NG is transported in liquid phase because in that form, the transportation of LNG is more reliable and appealing. NG will be compressed to about 1/600th of the original volume and thus having 600 times more energy density in liquid form rather than gas (Kim & Kim, 2014). After that, to be used for industrial and domestic purposes, LNG must be turned back to gas form at the regasification terminal.

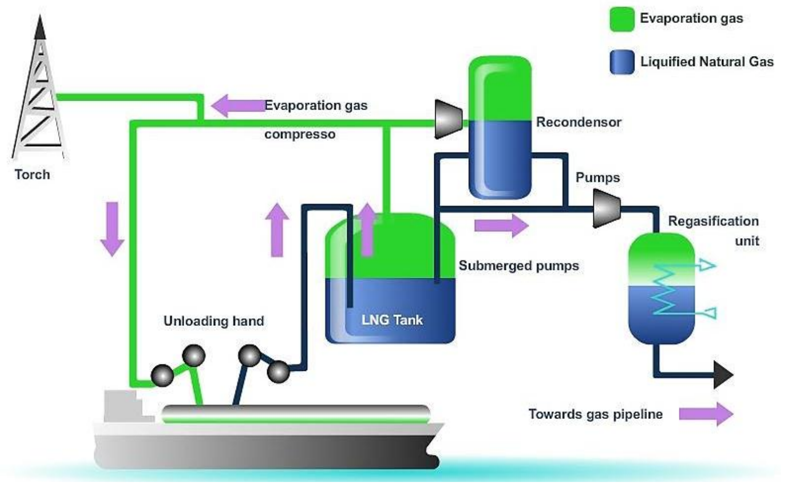

Figure 1: LNG regasification process flow

LNG regasification releases a large amount of cold energy, about 800 kJ/kg of LNG and usually, sea water will be used as the heating medium and will be discarded back into the sea (Gomez et al, 2014). According to figure 1, a conventional regasification terminal would start with the unloading ship transferring the LNG to the tank. Due to the heat from the pumping and solar radiation, a small part of the liquefied gas turned to vapour and will compensate the unloading process from the tank while some will be reinjected into the recondenser. If both of the processes exceeded their capacity, the balance ‘boil off’ gas will be burned in the torch. High pressure pump will pump the LNG to the vaporizer (regasification unit) to be heated into gas form and send to the national pipelines grid. Normally, sea water will be used as the heating medium and will be discarded at a very cold temperature.

With the rising energy prices and environmental effects, it is very crucial to recover the energy lost and thus improvements must be done to the current regasification process. There are several of ways to recover the energy such as CO_2 capture technology, air separation and also agro-food industry (Gomez et al, 2014). Thus, this paper deals in improving one of the regasification processes to be the base case which is the Kellogg process to combine the Rankine with Brayton cycle.

1.2 Problem Statement

Conventionally, the regasification process of LNG uses the thermal energy of sea water as heating medium. However, a large amount of energy is being wasted as the cold used sea water is returned to the sea (Gomez et al, 2014). Other than that, this action also has the potential to cause degradation to the underwater ecosystem. Dispenza et al. (2009) suggested that there is still much potential to improve in the recovery of cold energy from the regasification process such as utilizing the power cycles. Through this study, a new method of combining Brayton and Rankine cycle to recover the cold energy will be designed and simulated. Therefore, besides producing more power for industrial usage, energy wastage and environment issues can be resolved.

1.3 Objectives

The objectives of this study are;

1. To design and simulate a combined power cycle system that uses the cold energy from regasification process to generate more power by using Aspen Hysys simulation software.

2. To evaluate potential economic impact of the new proposed system based on the power generation.

1.4 Scope of Study

The scope for this project covers the following topics;

After understanding a typical LNG regasification process, the working principle of Ranking and Brayton cycle is studied. It is found that the waste cold energy from the process can be used as a cooling medium to cool down the working fluid in the power cycles. This concept is then being used as the principle to design a new process combining these two cycles. Further research is being carried out to find the best working parameters for the operation of the new designed system.

1.5 Relevancy and Feasibility of the Project

As the demand for energy increasing from days to days, it is vital to make full use of all the energy available. The cold energy from the regasification process is better to be converted into useful energy such as electricity rather than just releasing it to the environment. This project is relevant to the course of chemical engineering as it applies back some of the engineering knowledge inside the project. Mass and energy balance are used widely in the form of sofware simulation and not to forget the thermodynamic principle of Rankine and Brayton cycle.

1.6 Feasibility of the Project

This project is feasible to be carried out for as it is within the scope and also the time frame. The period of 8 months is enough for the simulation and the process optimization to be carried out. The help from the open literature and previous researches provide the suitable planning and sample of how the project going to take place. Furthermore, no sophisticated equipment and chemicals required for this project as it is totally simulation based. This means that no workstation is required and only the software needed is the Aspen Hysys.

CHAPTER 2: LITERATURE REVIEW

2.1 Regasification Process

Generally, after being transported in the form of LNG to the terminals, the liquefied gas is loaded into the LNG storage tanks. Before being used or delivered to the consumers, the liquid needs to be vaporized based on the demand. This process is called regasification. This is a process where the LNG is changed from liquid phase to gas phase using vaporizer.

There are three types of vaporizers commonly used in LNG regasification process. They are Open Rack Vaporizer (ORV), Intermediate Fluid Type Vaporizer (IFV) and Submerged Combustion Vaporizer (SCV). The ORV as shown in Figure 2 uses sea water which is usually above 5°C as the heating medium to regasify the LNG (Patel et al, 2013). The main part of an ORV is hundreds heat transfer tubes made of aluminum alloy forming panels that have excellent heat conductivity which is about 300 W/mK (Sigli et al, 2010). The LNG will go up inside the panels counter current with the sea water which is going down. The LNG is being heated along the way and transformed into gas phase. The ORV uses sea water as the heat source, thus the running cost of ORV will only come from the pumping work of sea water.

The typical length of the heat transfer tube is 10 m and a longer heat transfer tube will yield a better performance. However, the pump will require working harder and this will result in a higher operating cost. According to research done by Yamazaki et al (1998), they found that the optimum length of of the heat transfer tube is 8 m. They also developed an ORV that contains a vaporizing section of duplex tube construction and has a better performance comparing to the conventional type. The rate of LNG that can be vaporized is 3 to 5 times more and it required about 15% lesser sea water comparing to the conventional one. The new vaporizer has a capacity of 350 kg/h or LNG per heat transfer tube and uses the ratio of 30:1 sea water/LNG (Yamazaki et al, 1998). At 4 MPa, the LNG will enter the ORV at about -150°C and exiting at about -86°C which is the saturated temperature at that condition. Meanwhile, sea water enters at about 10°C and comes out at about 7°C. The decrease in temperature is because the heat has been transferred to the LNG (Jin et al, 2014).

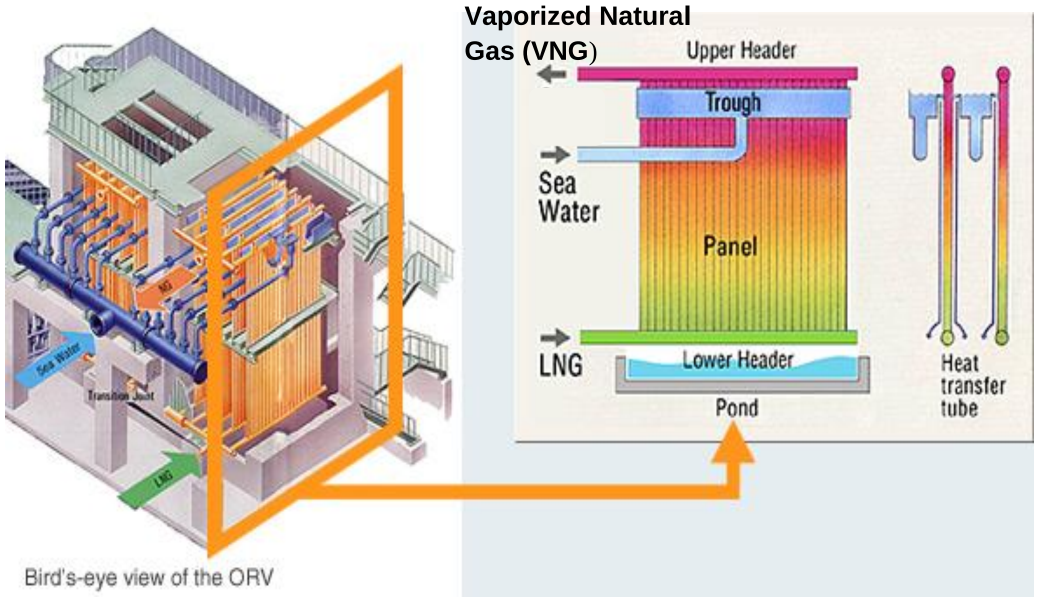

Figure 2: Open Rack Vaporizer

The IFV (Figure 3) uses the sea water as the heat source but does not vaporize the LNG directly. The sea water is used to heat a heating medium (intermediate fluid) which will normally be propane. Using hydrocarbon can prevent freezing problem faced by the seawater thus allowing the use of cold sea water as cold as 1°C (Patel et al, 2013). This is because such intermediate fluid meets the requirement of large heat flow even up to 50 MMBTU/h and fluids like propane has very low freezing temperature which is -180°C (Fengxia et al, 2013).

There are three shell and tube heat exchangers involved in this vaporizer (E1, E2 &E3).Normally, an IFV operates at 0.45 MPa where the saturated temperature of LNG is 1.65°C (Fengxia et al, 2013).. As for the process first, the LNG will be introduced into E2 which is also known as the condenser, at -161°C. This is where heat is transferred between LNG in liquid form and the intermediate fluid gas. The LNG will be almost completely vaporized and transferred to the E3 which is known as the thermolator. This is where the balance LNG exchanges heat with sea water and completely turned to gas at normal temperature exiting the IFV at about 2-3°C.

The intermediate fluid on the other hand, after the heat exchange process in E2, it will turn into liquid form and flows into E2, which is also known as evaporator. This is where it will be heated by seawater, vaporizing it again in to gas form to repeat the whole process.

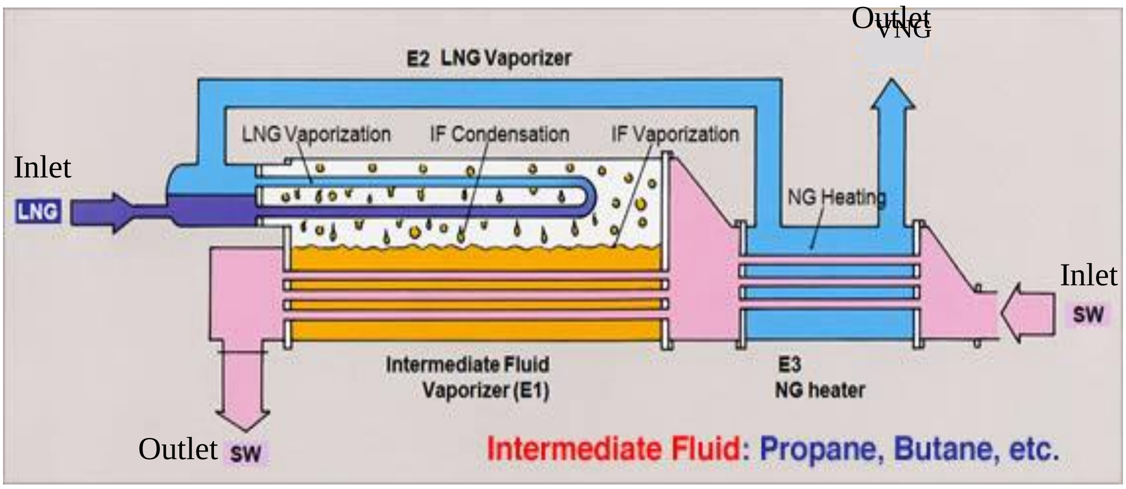

Figure 3: Intermediate Fluid Vaporizer

Another type of LNG vaporizer is the SCV (Figure 4) which is submerged underwater burner. It works by burning about 1.5% of the vaporized LNG to generate heat to turn the LNG to gas phase. According to CHIV international (2007), the SCV has very high thermal efficiencies reaching up to above 95%. The combustion gas will be exhausted to the water and thus creating a relatively low temperature from about 12°C to 37°C to be a stable heat source for vaporization of LNG (CHIV International, 2007). One of the features of SCV is that even though the combustion of burner stops, the high heat capacity of water (4.18 kJ/kg.C) can still continue providing the heat for a limited time.

Other than that, the water bath during the operation of SCV has the tendency to become acidic when it absorbs the product from the combustion. Therefore, chemicals with basic properties such as sodium carbonate and soda must be added to the water bath in order to monitor the pH level.

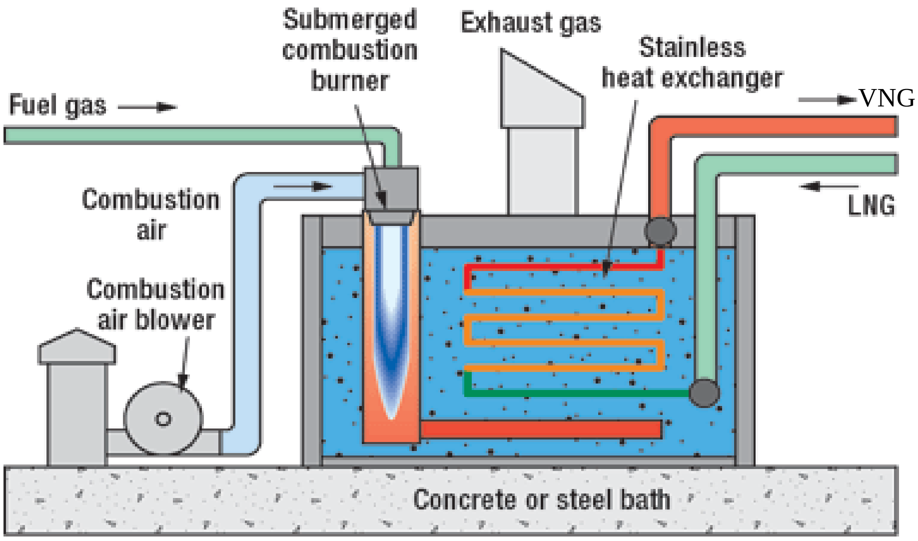

Figure 4: Submerged Combustion Vaporizer

In terms of the performance, even though Eisentrout et al (2006) suggested that the SCV is much favorable, there is also a study saying that the ranking of the vaporizers depends on the ambient of the locations (Patel et al, 2013).

In a warm condition, it is best to use the IFV with glycol water as intermediate fluid and air as the heat source. On the other hand, in a cold environment, the best vaporizer will be ORV with sea water as the heat source combined with SCV producing heat from fuel gas (Patel et al, 2013).

According to Lu & Wang (2009), it is important to have effective utilization of cryogenic energy associated with LNG vaporization. Thus, there are many studies and researches done on regasification process and how to recover the cold energy being discarded to the sea (Sun et al, 2014; Choi et al, 2013; Zhang & Lior, 2007; Shi & Che, 2009; Miyazaki et al, 2000). This is usually done by incorporating the LNG into the thermodynamics power cycles like Rankine and Brayton cycle. Among them is the base case of this study, Kellogg process which uses the Rankine cycle to generate more power. Nevertheless, issues and findings from their research works will be further discussed in this chapter.

2.2 Rankine Cycle

Rankine cycle is commonly found to be used in steam turbines. Most of the current power generation plant has been reported to be using rankine cycle as their working principle (Kim & Kim, 2014).

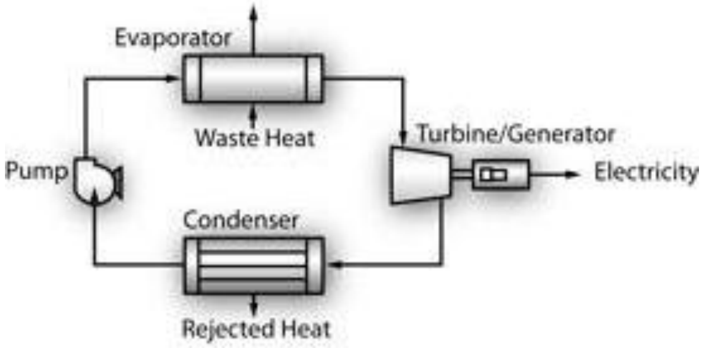

Figure 5: Example of close loop Rankine cycle

A Rankine cycle that consisted of multiple stages of organic Rankine cycles was simulated by Choi et al (2013). It was a study to analyse and optimize a cascade Rankine cycle for liquefied natural gas cold energy recovery. After the optimization process, it was found that as the stages increases, the power output, thermal and exergy efficiencies also increase. Propane showed the best performance in this study as the working fluid within the cycle.

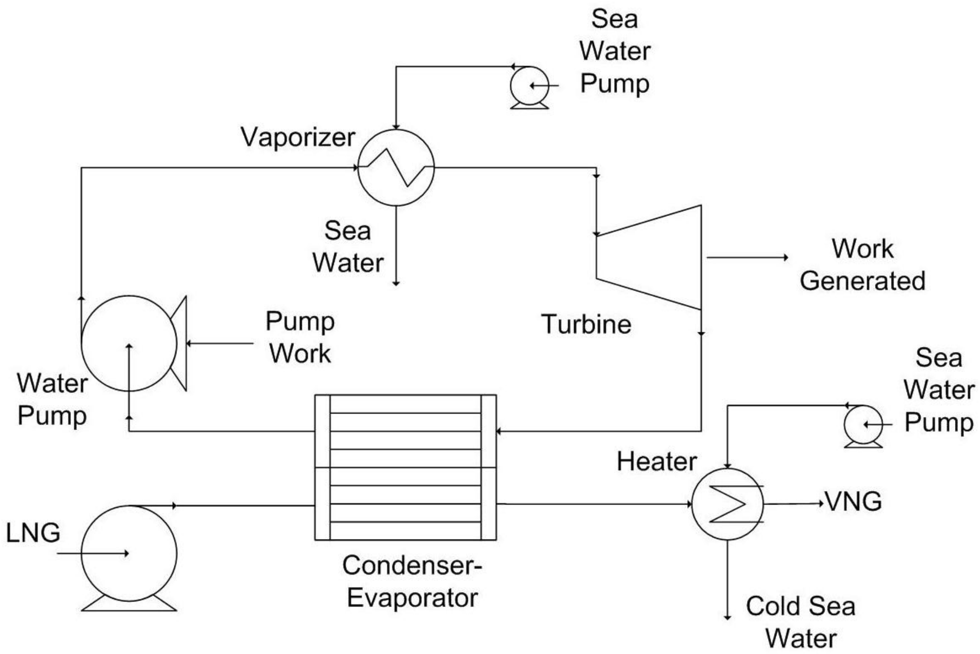

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Figure 6: Schematic of the organic Rankine cycle (ORC) (Choi et al, 2013)

According to Sun et al (2014) Rankine cycle is relatively simple and high efficiency can be achieved. Ethane is recommended to be used as a better working fluid comparing to methane and propane. They found that the optimum pressure for expander is ranging from 1400 to 2200 kPa. In terms of result, the output work increased from 1.023 kWh to 1.346 kWh comparing before and after LNG expansion. The exergy efficiency also varies from 29.58% to 49.68% based on different parameters and reheating temperature.

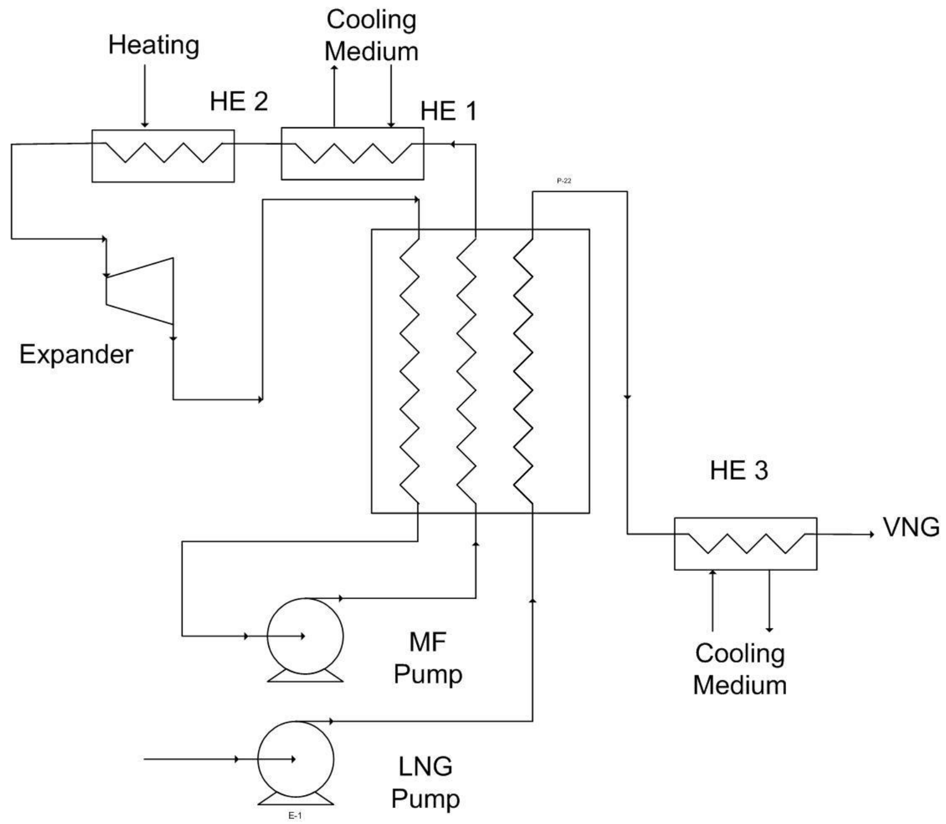


Figure 7: Process flow diagram for the proposed power cycle (Sun et al, 2014)

2.3 Brayton Cycle

Other than Rankine cycle, Brayton cycle is also one of the commonly used cycles for power generation. This thermodynamics cycle usually runs as an open system and normally found in gas turbine and jet engines. It usually consists of a compressor, a burner or heat exchanger and an expander or turbine.

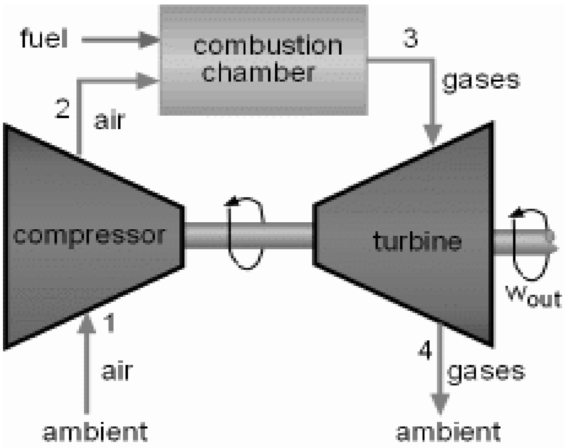


Figure 8: Example of Brayton cycle

In a similar study, Zhang & Lior (2007) used Brayton cycle with utilization of liquid hydrogen cryogenic exergy. Using nitrogen as working fluid, the liquid hydrogren keeps the inlet temperature of the compressor very low and the compression work to reduce significantly. As a result, the cycle has attractive thermal performance with 73% more power production and exergy efficiency of 45%.

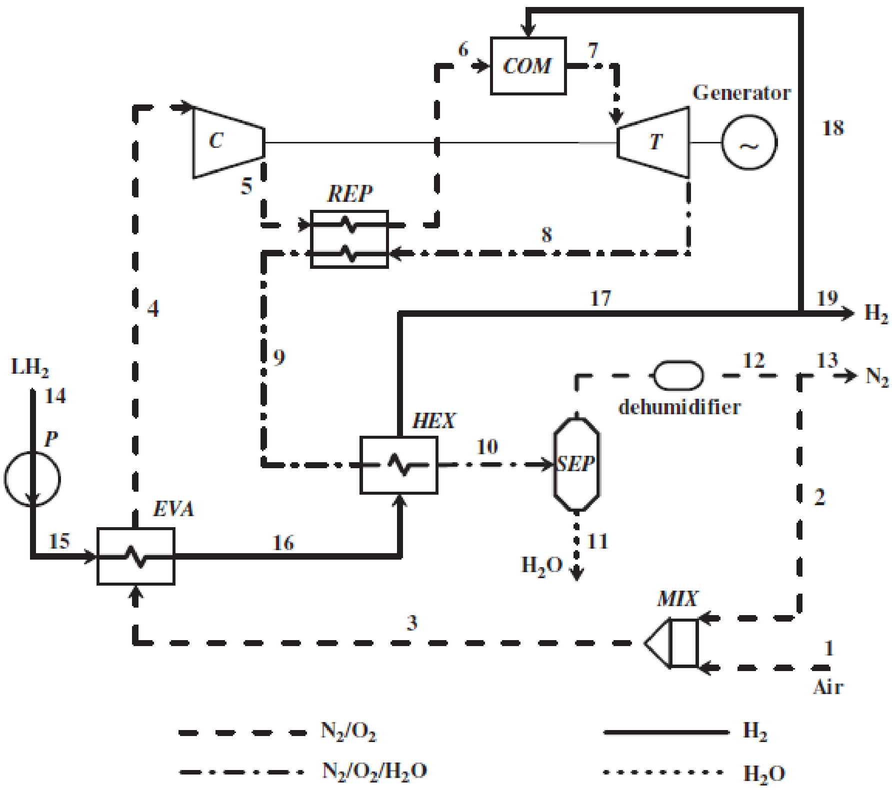


Figure 9: Flow sheet of the studied cycle (Zhang & Lior, 2007)

Using nitrogen as the working fluid, Angelino & Intermezzi (2011) exploited Brayton cycle and achieves efficiency as high as 63% comparing to the perfect gas efficiency of just 56%. The best results for both real and ideal cycle were found to be at 800°C. However, such temperature is bounded by the materials that can be used therefore even at 500-600°C the performance of the cycle is commendable.

2.4 Combined cycle

Apart from the existing power plant system, there are also combined cycles which consist of two or more thermodynamic cycles. This is due to the increase in demand of power and awareness of people regarding environmental pollution. Multiple processes can be combined to recover and utilize the residual heat in the hot exhaust gas. Shi et al (2010) suggested that an advanced conventional combined cycle power plant has the potential to achieve thermal efficiency as high as 58%.

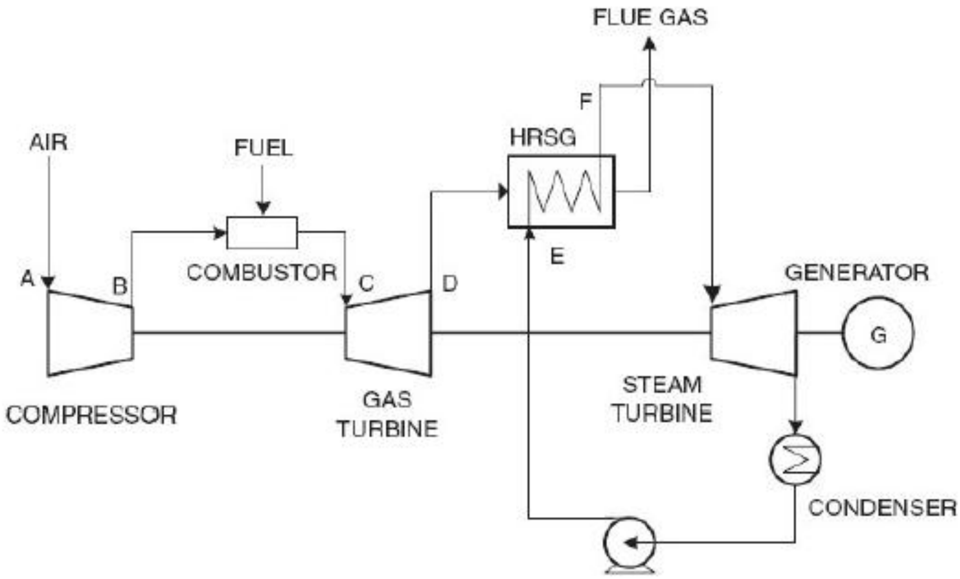


Figure 10: Example of combined cycle power plant

Miyazaki et al (2000) proposed a Rankine cycle with refuse incineration combining with LNG cold energy cycle. After comparing with the conventional steam Rankine cycle, the result found that the combined cycle is 1.53 and 1.43 times better in terms of thermal and exergy efficiencies respectively. As shown in figure 11**,** the system incinerates garbage about 600 tons/day as a heat source to heat up the working fluid through the heat exchanger (HX 1).

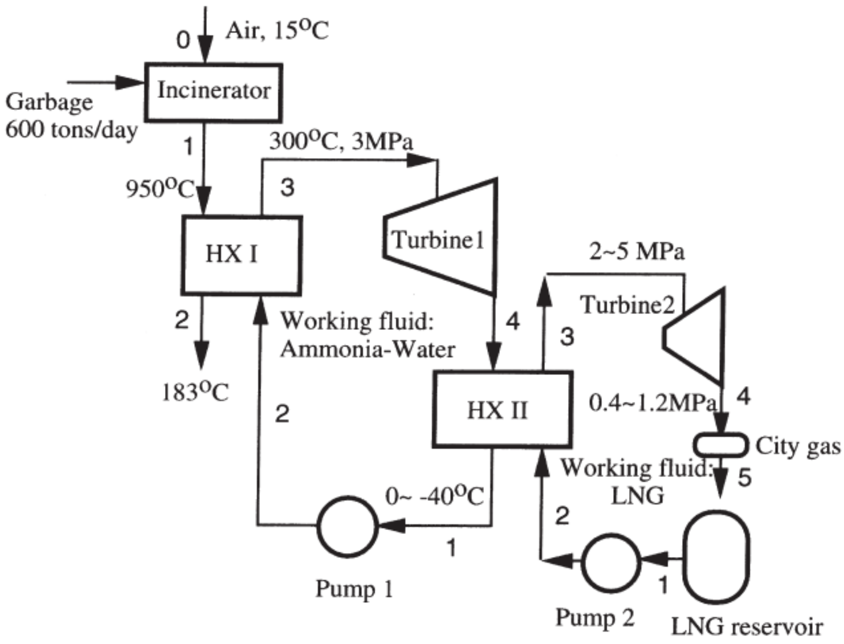


Figure 11: Combined power cycle using refuse incineration and LNG cold energy (Miyazaki et al, 2000).

A proposed system that can effectively recover low temperature waste heat and efficiently use the cold energy from liquefied natural gas has also been studied. This research has found to achieve very high waste heat recovery efficiency reaching up to 86.57%. Figure 12 shows the schematic diagram of the proposed system that uses ammonia-water as the working fluid. As a result, the system has successfully generated about 1.25 MWh per kg of the mixture (Shi & Che, 2009).

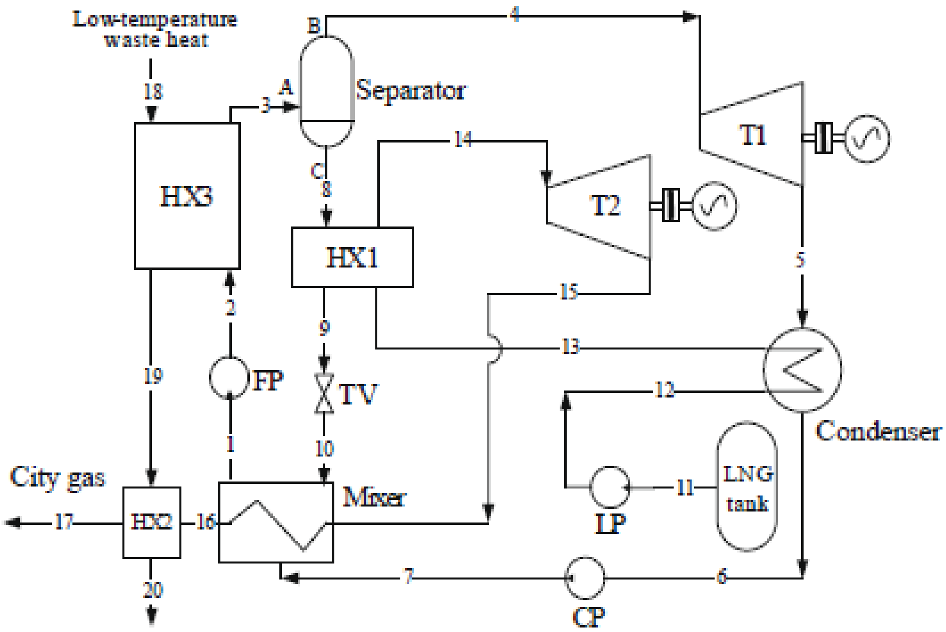


Figure 12: A schematic diagram of combined power cycle (Shi & Che, 2009).

Kim and Kim (2014) also performed an analysis of a combine cycle, which is a combination of Rankine cycle and LNG cycle using a low grade heat source. Ammonia-water mixture is chosen as the working fluid for the Rankine cycle. This study also investigates the effects of influential parameters like mass fraction of ammonia, turbine inlet pressure and condensation temperature. Mass fraction of ammonia is found to be the most influential parameter where the higher the mass fraction of ammonia, the higher the work generated. This is because of the change in the bubble point of the mixture that will occur at a different temperature as the ammonia mass fraction changes.

2.5 Kellogg's Process

For this project, Kellogg's process is used as the base case. This process uses the Rankine cycle as the working principle with water as the working fluid. The cycle starts with water from the tank to be pumped at 55 bar into the fired heater. Then, water will turn into steam and go through the steam turbine. The turbine will produce shaft work of 11MW reducing the temperature and pressure to 100°C and 1 bar respectively. Lastly, the cycle continues to the condenser (EX2) to turn the mixture into liquid form and back to the water tank. Balance heat from the fired heater is used as the heating medium to heat the LNG from -162°C at gas phase to 4 °C at liquid phase.

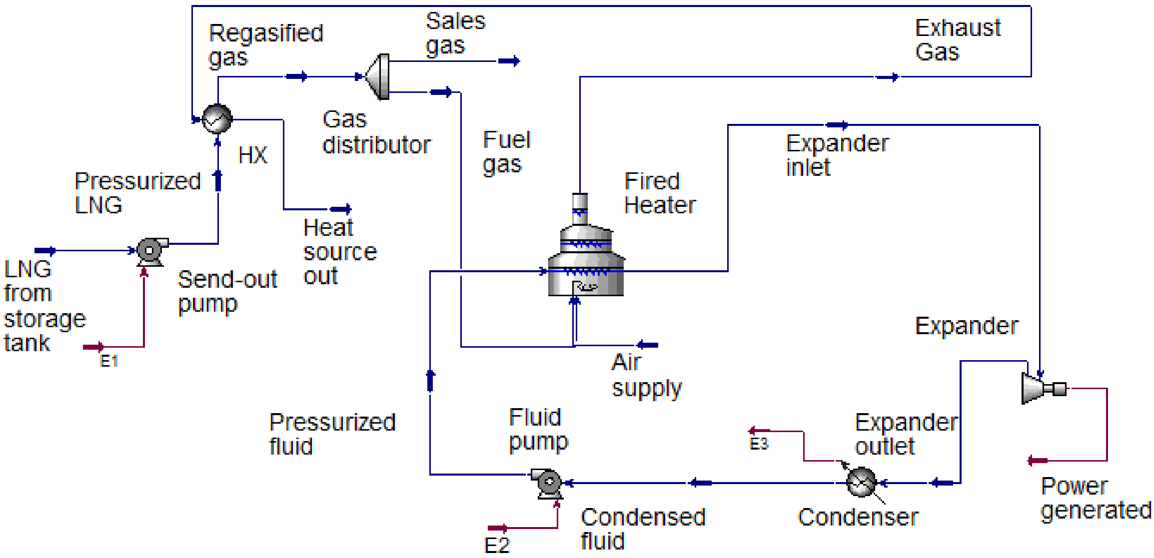


Figure 13: Process Flow Sheet of Kellogg's process

CHAPTER 3: METHODOLOGY/PROJECT WORK

For this study, the methodology is divided into three parts. The first part is the simulation of the proposed power plant for the cold energy recovery. Second part is the performance analysis to evaluate the efficiency of the suggested system and lastly is the parameter manipulation to find the best working parameters of the proposed system.

3.1 Process Simulation

3.1.1 Selection of working fluid

The simulation is started by determining the best working fluid to be used in the proposed system. The working fluid chosen for this study is further explained in the following chapter. There are few properties that must be considered before deciding the best fluid which are;

Table 1: Properties for selecting working fluid

Criteria of selection fluid.	
Critical temperature	Flammability
Heat capacity	Toxicity
Thermal conductivity	Global warming potential
Latent heat of vaporization	Ozone depletion potential

Some of the working fluids that have been determined are helium, carbon dioxide, ammonia, propane, air, refrigerant R-218, and nitrogen. These working fluids are changed in the simulation and the working fluid with the best performance is chosen.

3.1.2 Process scheme and description

The next step is to design the process flow. This is the improvement options that can be made from the existing Kellogg process which only using Rankine cycle to generate power. The proposed system consists of 2 cycles, Rankine cycle (Pump – Furnace – Steam turbine – Heat exchanger) and Brayton cycle (Compressor – Heat exchanger – Gas turbine – Heat exchanger). The hot flue gas from the furnace is also being used to heat up the cold LNG to increase the temperature. These proposed systems are portrayed in figure 14.

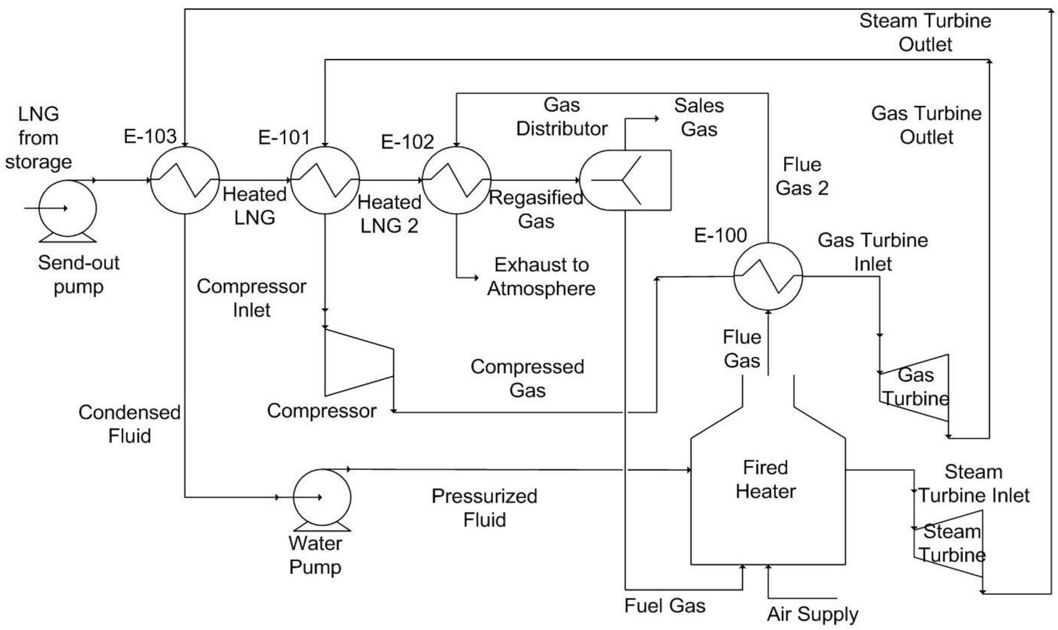


Figure 14: Process flow of the proposed system

3.1.3 Simulation modeling by Aspen Hysys

After finishing the previous steps, Aspen Hysys is chosen to be the software for simulating the overall process. After determining the components that exist in the system, the right thermodynamic package which is the Peng Robinson package is chosen. This is because of the compability of the package with the components in the simulation. Other than that, most of oil and gas based simulation will usually use this package in the simulation as well. Then, the equipment involved in the proposed system is arranged in their order and the parameters of the streams are specified.

By specifying the involving parameters, choosing the right thermodynamics packages, and following the process scheme, a real plant behavior can be simulated.

3.2 Performance Analysis

Each component must be analyzed to determine the performance and the efficiency of the proposed system. This must be done in terms of the energy and exergy balances (Gomez et al, 2014). There are some assumptions being made to the system which are;

- Flue gas have ideal gas behavior,
- Each equipment are well insulated,
- The flow is in steady state
- Kinetic and potential energy lost is neglected.

3.2.1 Energy equations and thermal efficiency

Normally, the performance of a power plant is evaluated through calculation of thermal efficiency. Basically, thermal efficiency is the ratio between the net thermal output power over the heat input. It can be calculated as shown by Equation (5).

The energy balance and heat transfer equations of equipment can be calculated as follows;

Compressor and pump;

$$W_i = m_{i,inlet}\left(h_{i,outlet} - h_{i,inlet}\right) \quad (1)$$

Where h = enthalpy of the stream

m= mass flowrate of the stream

i= Compressor, Pump

Turbine or Expanders;

$$W_j = m_{j,inlet}\left(h_{j,inlet} - h_{j,outlet}\right) \quad (2)$$

Where h = enthalpy of the stream

m= mass flowrate of the stream

j= Turbine 1, Turbine 2

Fired Heater;

$$Q_{in} = m_{fuel} LHV_{fuel} \tag{3}$$

$$LHV_{fuel} = HHV_{fuel} - \left(\frac{m_{H_2O}}{m_{fuel}}\right) \Delta H_{vap,H_2O} \tag{4}$$

Where LHV = Lower Heating Value (kJ/kg)

HHV = Higher Heating Value (kJ/kg) = -ΔH_C

Q_{in} = Heat input

m_{fuel} = mass flowrate of fuel gas

Thermal efficiency of the combined cycle;

$$\eta_{thermal} = \frac{(\Sigma W_j - \Sigma W_i)}{Q_{in}} \times 100\% \tag{5}$$

Where $\eta_{thermal}$ = Thermal efficiency

ΣW_j = Summation of work by turbine

ΣW_i = Summation of work by compressor and pump

Q_{in} = Amount of heat put in

3.2.2 Exergy equations and exergy efficiency

Exergy is defined as the maximum amount of work that can be obtained from a system in a steady state environment. On the contrary with energy, exergy does not follow the laws of conservation as it will always be destroyed when there is a temperature change. This destruction of exergy is increasing along with the increase in entropy of the system. Thus, for a better performance evaluation, it is better to analyse the system using in terms of exergy.

When analyzing exergy, the parameters that we need to look at are the enthalpy and the entropy of the stream. The exergy source, E_{source} and exergy sink, E_{sink} must be calculated for all equipment.

For the fluid of unit mass, the exergy is defined as;

$$e = (h - h_0) - T_0 (S - S_0) \tag{6}$$

Where h_0 = Enthalpy at reference temperature

h = Enthalpy at respective temperature

S_0 = Entropy at reference temperature

S = Entropy at respective temperature

T_0 = Reference temperature

Turbine or expander;

The exergy source, $E_{turbine}$ for turbine or expander is

$$E_{turbine} = m_{turbine} \left(e_{turbine,outlet} - e_{turbine,inlet}\right) \quad (7)$$

Where $E_{turbine}$ = Exergy source of turbine

$m_{turbine}$ = mass flowrate of turbine

$e_{turbine,outlet}$ = exergy of outlet stream of turbine

$e_{turbine,inlet}$ = exergy of inlet stream of turbine

The exergy sink of turbine can be obtained from the HYSYS simulation.

Pump and compressor;

The exergy sink, E_j for pump or compressor is

$$E_j = m_j \left(e_{j,outlet} - e_{j,inlet}\right) \quad (8)$$

Where E_j = Exergy sink of pump or compressor

m_j = mass flowrate of pump or compressor

$e_{j,outlet}$ = exergy of outlet stream of pump or compressor

$e_{j,inlet}$ = exergy of inlet stream of pump or compressor

The exergy source of pump or compressor can be obtained from the HYSYS simulation.

Heat exchanger;

For heat exchanger, the exergy source is the exergy coming from the hot stream while the exergy sink is the exergy coming from the cold stream.

The exergy source and exergy sink, E_i for heat exchanger is

$$E_i = m_i (e_{i,outlet} - e_{i,inlet}) \quad (9)$$

Where E_i = exergy source and exergy sink of heat exchanger

m_i = mass flowrate of heat exchanger

$e_{i,outlet}$ = exergy of outlet stream of heat exchanger

$e_{i,inlet}$ = exergy of inlet stream of heat exchanger

Fired Heater;

For fired heater, the exergy source comes from the fuel gas, air mixture and also the flue gas.

The exergy source, E_k fired heater is

$$E_k = m_k (e_{k,fuel\ gas} + e_{k,air} - e_{k,flue\ gas}) \quad (10)$$

Where E_k = Exergy sink of fired heater

m_k = mass flowrate of fired heater

$e_{k,fuel\ gas}$ = exergy of fuel gas stream of fired heater

$e_{k,air}$ = exergy of air stream of fired heater

$e_{k,flue\ gas}$ = exergy of air stream of fired heater

The exergy sink comes from the water stream coming through the fired heater and the steam coming out.

The exergy sink, E_k fired heater is

$$E_k = m_k (e_{k,water} - e_{k,steam}) \quad (11)$$

Where E_k = Exergy sink of fired heater

m_k = mass flowrate of fired heater

$e_{k,water}$ = exergy of water stream of fired heater

$e_{k,steam}$ = exergy of steam stream of fired heater

Next, the efficiency of a power cycle can be obtained through the equation;

$$\eta_{exergy} = \frac{\Sigma E_{sink}}{\Sigma E_{source}} \times 100\% \qquad (12)$$

Where η_{exergy} = Exergy efficiency

ΣE_{source} = Summation of exergy source

ΣE_{sink} = Summation of exergy sink

The efficiency of the proposed system will be evaluated and compared with the current system and further optimized to get the best operating conditions.

3.3 Parameter Manipulation

The influential parameters must be operating at their optimum condition in order for the proposed system to produce the maximum amount of power. To study the influence of those parameters, their values will be varied within a range while other parameters will remain the same. The trend of how the changes in parameters affect the system performance (thermal and exergy efficiency) will be investigated. The involving parameters are;

- Pressure ratio of the compressor
- Inlet pressure of the turbine
- Outlet pressure of the turbine
- Turbine inlet temperature (TIT)
- Working fluid of the power cycle.

3.4 Project milestone

Beside the project activities, key milestone is also one of the important aspects that must be monitored. It is the indicator of project completion and usually it is the guideline of what to be completed by a certain date. It is a helpful tool to ensure the project runs effectively

The milestone for this projects divided into two; FYP1 and FYP 2. For FYP1, key milestones are submission of extended proposal submission, selection the working fluid, proposed system simulation and interim report. On the other hand, for FYP 2, the performance and thermodynamic analysis and process optimization shall be done. Progress report and oral presentation will also be conducted in FYP 2. Lastly is the submission of the technical report and dissertation. Key milestones for this project are summarized in Table 2.

Table 2: Key Milestone

Key Milestone	**Week**
FYP1	
• Extended proposal submission	7
• Selection on working fluid	8
• Interim report submission	13
• Proposed system simulation	14
FYP2	
• Performance and thermodynamic analysis	17
• Progress report	21
• Process optimization	22
• Submission of technical report	27
• Oral presentation	26
• Submission of dissertation	28

Gantt Chart and Key Milestone

Gantt Chart for FYP 1

No.	Detail	Week													
		1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Title Selection and Supervisor Allocation														
2	Understanding the Project														
3	Identifying the Objectives and Scope of Study														
4	Conducting Literature Studies on the Project														
5	Finalizing Proposed Combined System														
6	Preparation of Extended Proposal														
7	Submission of Extended Proposal														
8	Proposal Defence														
9	Continuation of Project Work (Simulation of proposed system)														
10	Preparation of Interim Report														
11	Submission of Interim Report														

Gant Chart for FYP 2

No.	Detail	Week													
		15	16	17	18	19	20	21	22	23	24	25	26	27	28
12	Project Work Continues (Performance & thermodynamic analysis)														
13	Submission of Progress Report														
14	Project work continues (Process optimization)														
15	Pre-SEDEX														
16	Submission of Draft Final Report														
17	Submission of Dissertation (soft bound)														
18	Submission of Technical Paper														
19	Viva														
20	Submission of Project Dissertation (hard bound)														

Process

Suggested Milestone

CHAPTER 4: RESULTS AND DISCUSSION

4.1 Selection of working fluid

Selecting the right working fluid is important as it can affect the performance and the economics of a plant. Water is chosen as the working fluid for the Rankine cycle part as this study is an improvement of the Kellogg process. However, water has its own advantage and disadvantages. Besides being readily available and easy to be handled, water has high specific heat capacity (4.18 kJ/kg.C). Other than that, one cubic metre of water will occupy 1600 cubic metre of steam after being vaporized. Therefore, a large amount of energy can be put into each kilogram of steam. Nevertheless, using water also has its challenges such as high compressor outlet temperature and also water requires high compressor work (Kilicarsian & Muller, 2005). Water is more suitable for high temperature application and large centralized system (Tchanche et al, 2011). For a small or medium power plant, selecting a better working fluid can partially lessen the problems when using water such as the need of superheating to prevent condensation during expansion and also risk of erosion of turbine blades.

As for the Brayton cycle part, helium is chosen first as the working fluid as it satisfies most of the criteria. According to Gomez et al (2014), helium has high specific heat and has the ability to generate power at high and low temperature. Other than that, although using helium can contribute to the improvement of power density, it also requires complex storage vessels and the cost of helium is expensive.

There are also some power plants that use other fluids like air, nitrogen and ammonia as working fluids. However, air is not suitable as it contains oxygen and can cause oxidation in the equipment. Nevertheless, some other working fluids such as organic compounds for Rankine cycle and nitrogen and carbon dioxide (Chen, 2011) for Brayton cycle will be tested in the simulation to get the optimum performance of the system. There are also studies that recommended using refrigerants (Rovira et al, 2013) and binary mixture of the fluids (El-Genk & Tournier, 2009) to find the best working fluid. The working fluid that can provide highest efficiency and generate most power will be chosen. Therefore, for the initial simulation, water is chosen as the working fluid for the Rankine cycle, and helium is chosen for the Brayton cycle.

4.2 Simulation result of proposed model

Simulation on the proposed simulation has been done using the Hysys software as shown in figure 15. This is to replicate the real plant environment and how the process is going to take place. As per discussed in Chapter 3, the proposed cycle consisted of two power cycles; Rankine cycle and also Brayton cycle. The Rankine cycle starts where water is pumped at 55 bar to the fired heater to change the water into steam. The steam produced will go through the turbine that will produce shaft work. From the turbine outlet, the stream will exchange heat with the LNG to increase the LNG temperature while turning the steam back into water. Producing as much power as the Kellogg's process, the Rankine cycle part of the system has already found to produce about 11 MW of power through the steam turbine.

As for the Brayton cycle, it starts by compressing about 30 bar of helium to the heat exchanger to increase its temperature to 1000°C. The heated helium will then go to the gas turbine for power generation and then back to the heat exchanger. This is where the balance LNG will be heated with hot helium. The high temperature flue gas from the fire heater will also be used to heat up the cold LNG turning it to completely change it from liquid to gas. Besides that, cooling down helium through heat exchanger (E-101) decreases the specific volume of the gas, and simultaneously reduces the compression work. This results in an increase in the net power of the cycle. As for the Brayton cycle part of the process, it produces about 10 MW of power through the gas turbine.

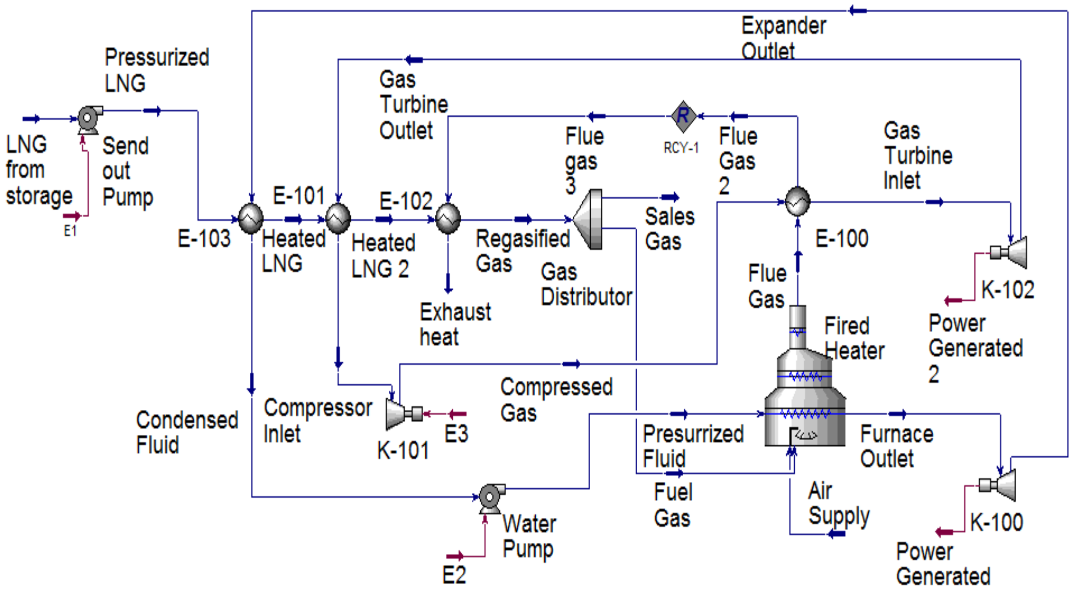


Figure 15: Hysys simulation overall process

Power generated from both Rankine and Brayton cycle, the system performance is evaluated based on their thermal and exergy efficiencies (Gomez et al, 2014) as per explained in chapter 3. Based on the analysis, the thermal and exergy efficiency of the proposed simulation is 35.4% and 64% respectively. This has an increase of 11% for thermal efficiency and 4% for exergy efficiency comparing to the Kelloggs process. The main conditions of the simulations are tabulated in Table 3.

Table 3: Main parameters used in the simulation

System	Parameters	Value
LNG	Storage temperature (°C)	-165
	Storage pressure (kPa)	108.1
	Pump efficiency (%)	90
Rankine cycle	Turbine efficiency (%)	70
	Pump Efficiency (%)	80
	Turbine inlet temperature (°C)	540
	Turbine inlet pressure (kPa)	5516
Brayton cycle	Turbine efficiency (%)	91

	Compressor efficiency (%)	89
	Compressor inlet pressure (kPa)	500
	Turbine inlet pressure (kPa)	2975
	Turbine inlet temperature (°C)	1000

4.3 Parameters Manipulation

The effects of some crucial parameters such as pressure ratio, turbine inlet temperature, working fluid, inlet and exhaust pressure are examined to analyze the performance of the proposed model. The magnitudes of the involved parameters are varied while other parameters are maintained the same. Thus, the effects on the thermal and exergy efficiencies can be investigated. By doing so, we are able to obtain the optimum parameters of the plant that will generate the biggest amount of power.

4.3.1 Effect of Pressure Ratio on Efficiency

The effect of pressure ratio is investigated on the proposed model. Pressure ratio is the ratio of pressure at the inlet and at outlet of the compressor. Generally, with higher a pressure ratio, the power generated will be higher. This is because thermodynamically, as the pressure ratio goes higher, there is a bigger change in enthalpy and entropy between the inlet and outlet of the compressor. This will cause a higher power generated by the turbine. There is an optimum pressure ratio that the system can produce highest amount of power.

As per the designed process, the compressed gas will pass through the heat exchanger (E-100) before going into the turbine. Based on the literature review (Gomez et al, 2014), the pressure of the compressor inlet is set to be 500 kPa. On the other hand, the outlet pressure of the compressor is varied from 2000 (4:1 pressure ratio) to 4000 kPa (8:1 pressure ratio) with the increment of 200 kPa.

Based on figure 16, it can be seen that the thermal efficiency increases and gradually decrease after 5.0 pressure ratio. This can be as explained by as Goktun & Yavuz (1999) suggested, for a close loop Brayton cycle, the thermal efficiency depends on the pressure ratio where the case is the opposite with open cycle. For a Brayton close cycle, the lower the pressure ratio, the higher the thermal efficiency as the regeneration process is most effective at a lower pressure ratio On the other hand, for exergy efficiency, it increases as the pressure ratio increases. The trend for exergy efficiency increases until 8.0 pressure ratio. Thus, the best pressure ratio is 6.6 with 31.3% and 61.3% of thermal and exergy efficiency respectively. This is because the pressure ratio 6.6 is the point that satisfies both lines, thermal and exergy efficiency where they are not too low and not too high.

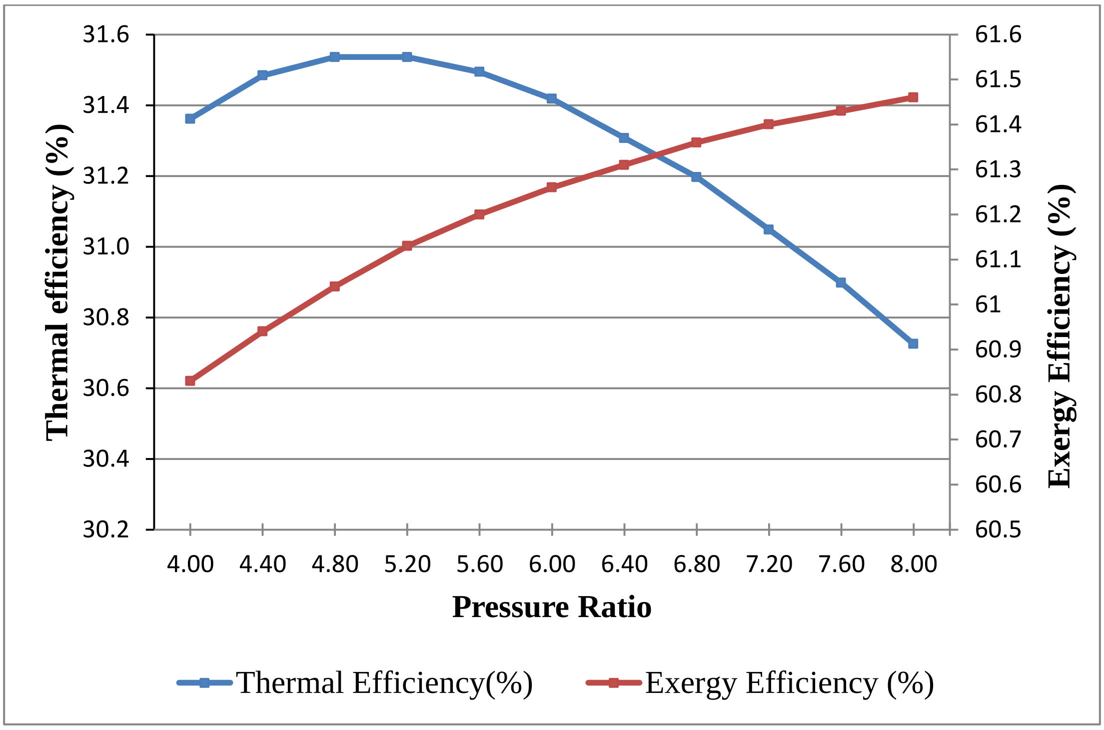


Figure 16: Effect of Pressure Ratio on Thermal and Exergy Efficiency

4.3.2 Effect of Turbine Inlet Pressure on Efficiency

The next parameter to be studied is the inlet pressure of the gas turbine. The inlet pressure is varied from 575 to 3275 kPa with step size of 200 kPa while other parameters maintained the same. According to figure 17, the efficiencies of the system increase as the inlet pressure of the gas turbine increase. As the inlet pressure increase, this will cause a bigger change between the enthalpy and entropy of the stream going in and out of the turbine. As shown in equation (1) and (6) in chapter 3, the bigger the change of enthalpy and entropy will give positive impact on power generation. Thus, the optimum turbine inlet pressure for the system is 3275 kPa as this is the maximum pressure of the gas stream with 6.6 pressure ratio. In general, there is no specific maximum pressure limit for gas turbines inlet pressure as each turbine is custom designed according to their specific power generation capacity.

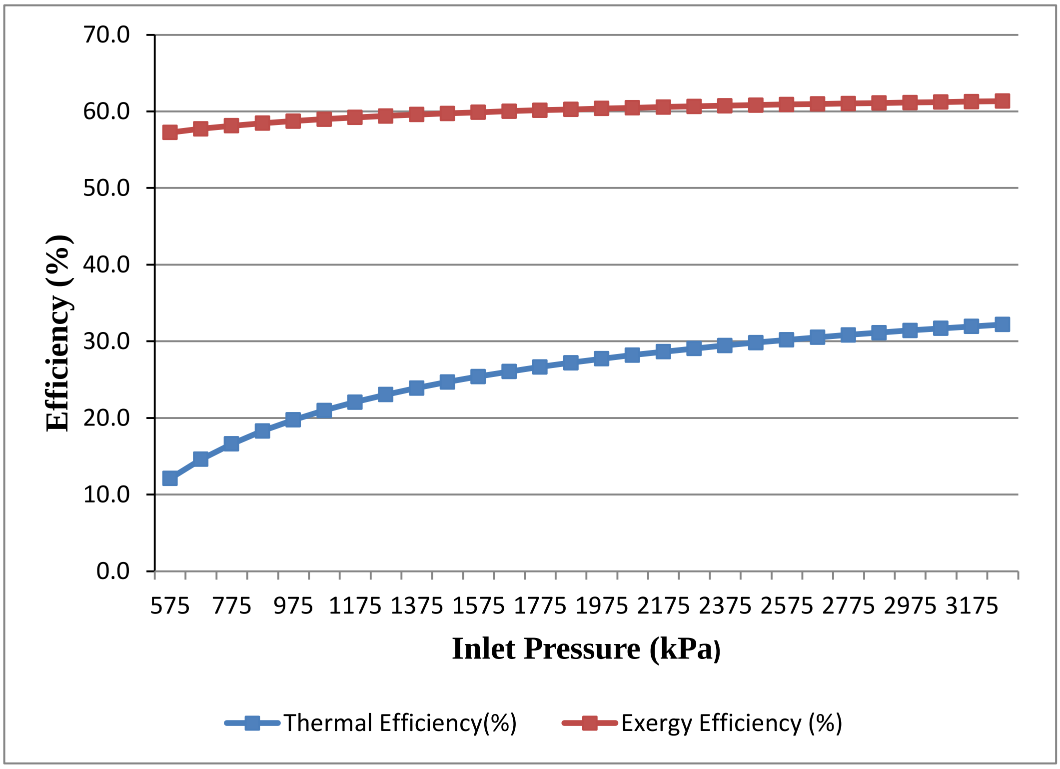


Figure 17: Effect of Turbine Inlet Pressure on Thermal and Exergy Efficiency

4.3.3 Effect of Turbine Exhaust Pressure on Efficiency

Investigating the effect of turbine exhaust pressure is quite similar as the parameters before. The exhaust pressure varies from 550 to 1550 kPa while other parameters remain the same. Due the upstream of heat exchanger E-101 having a pressure of 500 kPa as per described in the literature review, and by allowing the pressure drop of 50 kPa in the heat exchanger, the turbine outlet pressure is limited to only 550 kPa.

It can be seen from figure 18 as the exhaust pressure increases, both thermal and exergy efficiencies also decreases. This can be explained as the exhaust pressure increase, the efficiencies dropped due to a lower pressure change between the inlet and exhaust of the turbine. When the pressure difference between the turbine inlet and outlet is small, the change of enthalpy and entropy are also small. This leads to a smaller power generation by the gas turbine. Therefore, the optimum exhaust pressure is 550 kPa.

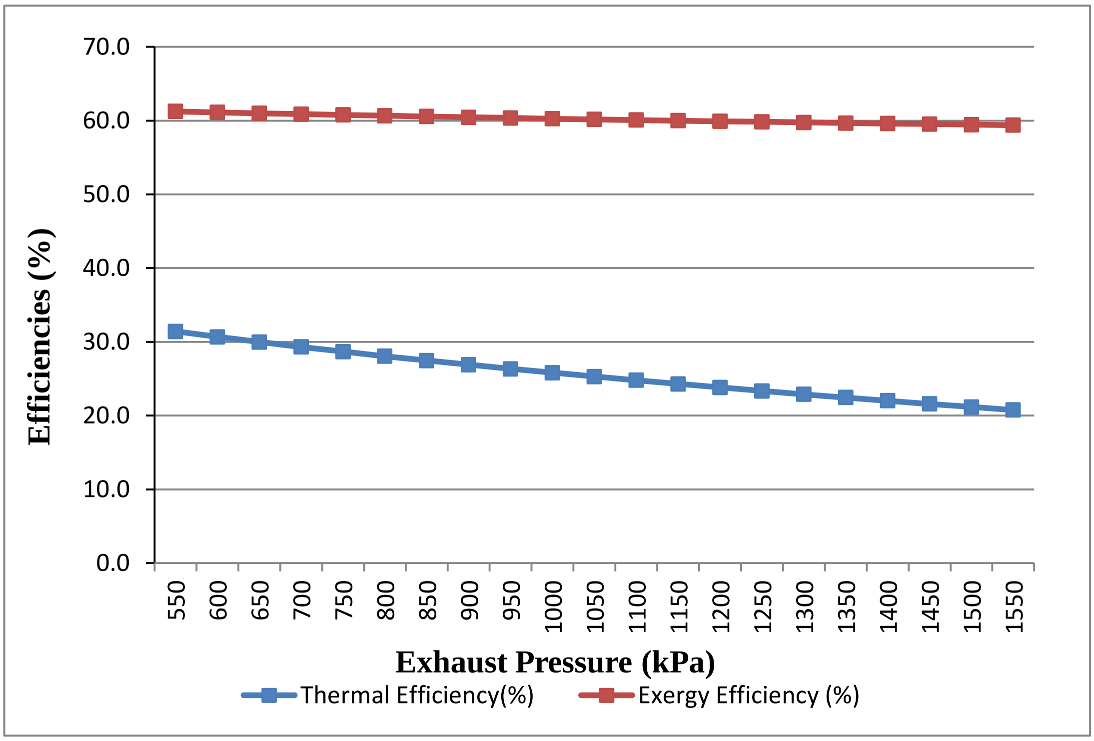


Figure 18: Effect of Turbine Exhaust Pressure on Thermal and Exergy Efficiency

4.3.4 Effect of Turbine Inlet Temperature on Efficiency

Turbine inlet temperature also plays an important role for an efficient operation of a turbine where the higher the temperature, the higher the efficiency of a turbine. This happens because as the turbine inlet temperature is higher, the generation of entropy in the combustion chamber will be lower. As show in Equation (6), this can cause an increase of exergy as exergy is inversely proportional to entropy. However, normally the maximum temperature is limited by the thermal properties of the material used for the equipment (Wartsila, 2015). Therefore, ceramic heat exchangers are assumed to be used in this study where according to Schulte-Fischedick et al (2007), a ceramic high temperature heat exchanger can withstand temperature up to 1100 °C which is much higher than a conventional alloy type.

The data from table 3 is used to find the effect of turbine inlet temperature ranging from 400 to 1000°C on the efficiencies of the system. It is safe to say that the temperature 1000°C is still allowed in a process as according to Ishikawa et al (2008), due to technology advancement gas turbine nowadays can even reach up to 1700°C. The result can be seen on figure 19, where there is about 10% of change in thermal efficiency about 3% of increase in exergy efficiencies as the temperature increases from 400 to 1000°C. Therefore, it can be concluded when the temperature increases, the efficiency also increases and therefore the optimum turbine inlet temperature is set to be 1000°C.

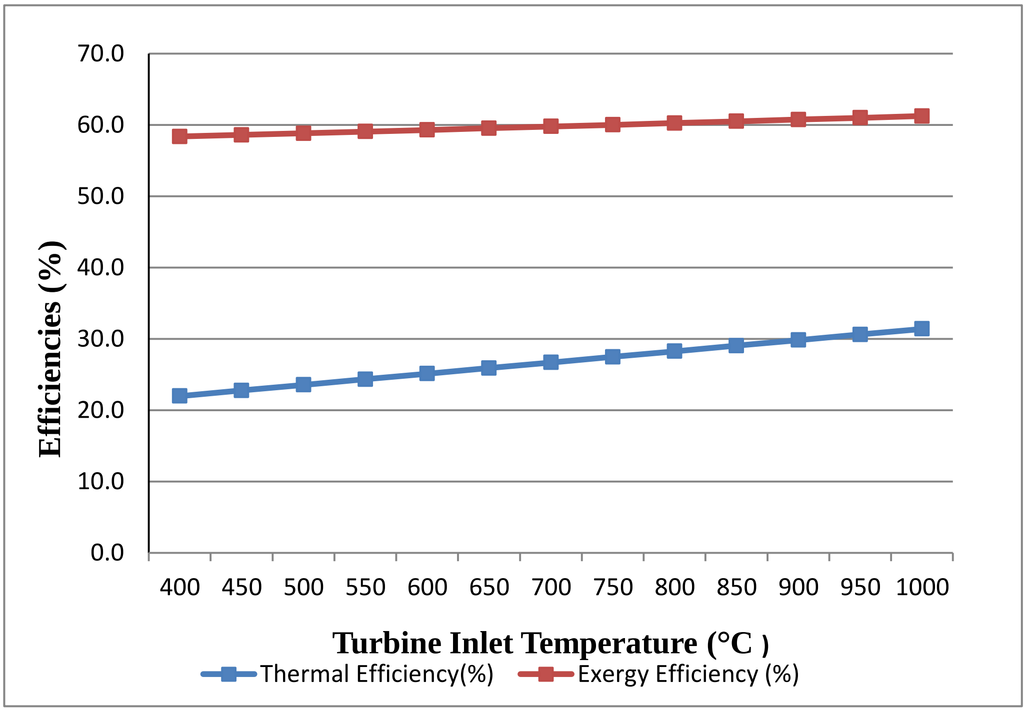


Figure 19: Effect of Turbine Inlet Temperature on Thermal and Exergy Efficiency

The final parameter to be investigated is the effect of working fluid on the thermal and exergy efficiencies. As per discussed in the earlier part of this chapter, working fluid plays a major role in the performance and the economics of a plant. There are few working fluids recommended by literature for example helium (Gomez et al, 2014), carbon dioxide (Chen, 2011), refrigerant R-128 or octafluoropropane (Rovira et al, 2013), and binary mixture of helium and nitrogen (El-Genk & Tournier, 2009). Other than that, commonly used working fluids like nitrogen, propane, ammonia, and air will also be tested.

The results are shown in figure 20 where the highest thermal and exergy efficiencies are achieved when using the refrigerant R-218 with 44.3% and 73.4% respectively. Next, the trend is followed by propane, ammonia, carbon dioxide, air, nitrogen, ammonia-water mixture and lastly helium. However, although R-218 yields the highest efficiency, using this working fluid will result in a mixture formation of gas and liquid form of LNG in the heat exchanger E-101. According to Panchal & Ljubicic (2007), two phase flow in a heat exchanger has a high tendency that will lead to fouling and corrosion due to the uneven distribution of the vapor and liquid. Therefore, the working fluid R-218 is not suitable to be used. Some changes in the parameters like the turbine inlet temperature stream exiting heat exchanger E-100 needs to be made in order to avoid the mixture formation.

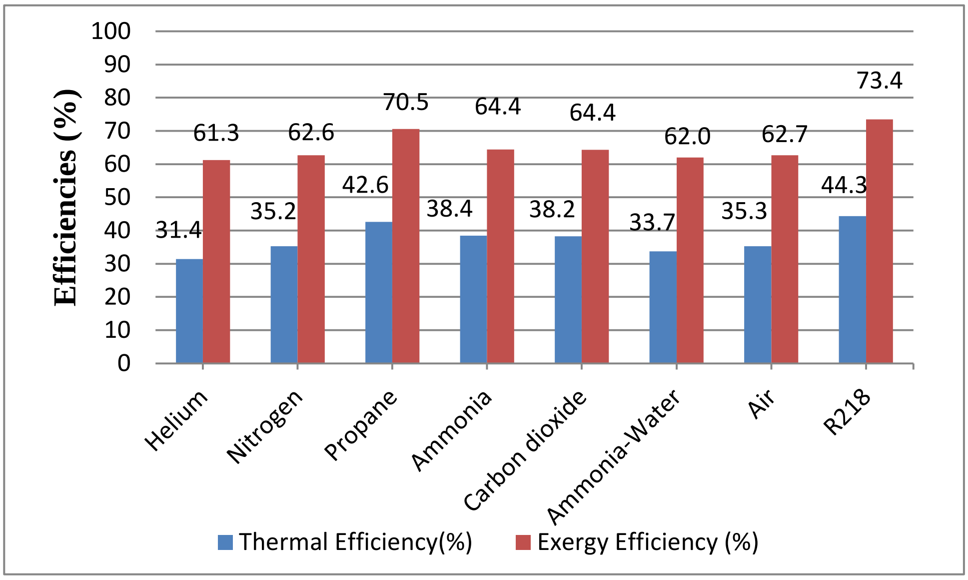


Figure 20: Effect of Working fluids on Thermal and Exergy Efficiency

The disadvantage of using propane is that according to the MSDS of propane, it has the autoignition temperature of 504°C. This will make propane flammable and an unsuitable working fluid as there high temperature reaching up to 1000°C. As for ammonia, according to the MSDS, it cannot be exposed to temperature greater than 426°C and has an autoignition temperature of 651°C. This makes ammonia not a suitable working fluid.

Air on the other hand is not the best working fluid as it contains oxygen that has the potential to cause oxidation on equipment in a long run. Working fluids that contain Hydorgen bonds in certain molecules like water and ammonia, has the tendency to result in wet fluids due to the negative slope of the saturation vapor curve. Based on the working fluids left, which are carbon dioxide, nitrogen, helium and ammonia-water mixture, it is found that carbon dioxide has the highest thermal and exergy efficiencies.

As being suggested by Chen (2011), carbon dioxide has no ozone depleting potential (ODP) and low global warming potential (GWP). It is cheap, non-flammable, non-explosive and also easy to be obtained. Therefore, it is concluded that the best working fluid is carbon dioxide with 38.22% and 64.35% of thermal and exergy efficiencies respectively.

As an overall conclusion, based on the parameter manipulations and discussions about the influential parameters, maximum thermal and exergy efficiencies can be obtained with 6.6 pressure ratio, 3275 kPa turbine inlet pressure, 550 kPa turbine exhaust pressure, 1000 °C turbine inlet temperature, and carbon dioxide as the working fluid. By applying these data into the process simulation, the optimum thermal and exegy efficiencies are found to be 38.22% and 64.52% respectively.

4.3.6 Economic Impact

This process also uses about 3% lesser fuel gas comparing to the Kelloggs process where only about 1046 kmol/h of fuel gas is being used instead of 1078 kmol/h. This is done by changing the split percentage at the splitter in the HYSYS simulation. When translated into the monetary point of view, this saves about 262,800 kg of fuel gas/year or 224,428 MMBTU/year and saves the operating cost by RM 3.56 million/year.

CHAPTER 5: CONCLUSION AND RECOMMENDATION

5.1 Conclusion

In this era of where the demand for power is increasing for day to day, people are finding and searching for ways to improve the efficiency of the current power plant. The utilization of cold energy from LNG is one of the areas being exploited to generate more power due to having high power exergy. This study has found a new way of combining the Rankine and Brayton cycle that improved the current regasification process in terms of the energy. A new and better system is designed to use energy from the cold LNG and has been successfully simulated using HYSYS software. The new process which is the combination of Rankine and Brayton cycle has managed to improve about 16.9% from 21.9% to 38.8% for thermal efficiency and 5% from 59% to 64.5% for exergy efficiency. In terms of economic impact, this process uses about 3% lesser fuel gas comparing to the Kelloggs process. This saves about 262,800 kg of fuel gas/year or 224,428 MMBTU/year and if translated in terms of monetary, this cut down the operating cost by RM 3.56 million/year.

5.2 Recommendation

Based on the result obtained from the simulation, there may be some enhancements that can be done to further improve the result of this simulation. More influential parameters can be analysed to find a better operating conditions of this process.

Other than that, the heat exchanger (E-102) is supposed to represent the stream going into the convection section of the plant. This is done as in the initial simulation, when the stream goes through the convection section, the fired heater fails to heat up the stream and there is no increase in temperature. Thus, if this issue is able to be solved, then a higher accuracy result may be obtained for this simulation.

REFERENCES

Angelino, G., & Invernizzi, C. M. (2011). The role of real gas Brayton cycles for the use of liquid natural gas physical exergy. *Applied Thermal Engineering*, *31*, 827-833.

Bai, F., & Zhang, Z. (2008). Integration of Low-level Waste Heat Recovery and Liquefied Nature Gas Cold Energy Utilization.*Chinese Journal of Chemical Engineering*, *16*(1), 95-99.

Chen, Y. (2011). *Thermodynamic Cycles using Carbon Dioxide as Working Fluid* (Doctoral dissertation, Royal Institute of Technology, Stockholm, Sweden).

Choi, I., Lee, S., Seo, Y., & Chang, D. (2013). Analysis and optimization of cascade Rankine cycle for liquefied natural gas cold energy recovery. *Energy*, *61*, 179-195.

Dispenza, C., Dispenza, G., Rocca, V. L., & Panno, G. (2009). Exergy recovery during LNG regasification: Electric energy production – Part one. *Applied Thermal Engineering*, *29*, 380-387.

Dispenza, C., Dispenza, G., Rocca, V. L., & Panno, G. (2009). Exergy recovery during LNG regasification: Electric energy production – Part two. *Applied Thermal Engineering*, *29*, 388-399.

Dispenza, C., Dispenza, G., Rocca, V. L., & Panno, G. (2009). Exergy recovery in regasification facilities – Cold utilization: A modular unit. *Applied Thermal Engineering*, *29*, 3595-3608.

Eisentrout, B., Wintercorn, S., & Weber, B. (2006). Study focuses on six LNG regasification systems. *LNG journal*, *8*, 21-22.

El-Genk, M. S., & Tournier, J. (2009). Performance analyses of VHTR plants with direct and indirect closed Brayton cycles and different working fluids. *Progress in Nuclear Energy*, *51*, 556-572.

Ersayin, E., & Ozgener, L. (2014). Performance analysis of combined cycle power plants: A case study. *Renewable and Sustainable Energy Reviews*, *43,* 832-842.

Fengxia, L., Yuqiang, D., Wei, W., Jiupeng, Z., Che, Z., Chao, D., & Dapeng, H. (2013). Feasibility of Intermediate Fluid Vaporizer with Spiral Wound Tubes. *China Petroleum Processing and Petrochemical Technology*, *15*, 73-77.

Goktun, S., & Yavuz, H. (1999). Thermal efficiency of a regenerative Brayton cycle with isothermal heat addition. *Energy conversion & management,* 40, 1259-1266.

Gomez, M. R., Garcia, R. F., & Gomez, J. R. (2014). Thermodynamic analysis of a Brayton cycle and Rankine cycle arranged in series exploiting the cold exergy of LNG. *Energy*, *66*, 927-937.

Gomez, M. R., Garcia, R. F., Gomez, J. R., & Carril, J. C. (2014). Review of thermal cycles exploiting the exergy of liquefied natural gas in the regasification process. *Renewable and Sustainable Energy Reviews*, *38*, 781-795.

Hisazumi, Y., Yamasaki, Y., & Sugiyama, S. (1998). Proposal for a high efficiency LNG power-generation system utilizing waste heat from the combined cycle 1 Published in co-operation with the Japanese Society of Energy Resources. 1. *Applied Energy*, *60*, 169-182.

Hung, T. C., Wang, S. K., Kuo, C. H., Pei, B. S., & Tsai, K. F. (2010). A study of organic working fluids on system efficiency of an ORC using low-grade energy sources. *Energy*, *35*, 1403-1411.

Ishikawa, M., Terauchi, M., Komori, T., & Yausuraoka, J. (2008). Development of High Efficiency Gas Turbine Combined Cycle Power Plant. *Mitsubishi Heavy Industries ltd*, *45*(1), 15-17.

Jin, T., Wang, M., & Tang, K. (2014). Simulation and performance analysis of a heat transfer tube in SuperORV. *Cryogenics*, *61*, 127-132.

Kanoglu, M., Dincer, I., & Rosen, M. A. (2007). Understanding energy and exergy efficiencies for improved energy management in power plants. *Energy Policy*, *35*, 3967-3978.

Kawahara, A., Niikura, A., & Doko, T. (2003). Development of aluminum alloy fin stock for heat exchanger using twin roll continuous casting method. *Furukawa Review*, *24*, 81-87.

Kilicarsian, A., & Muller, N. (2005). A comparative study of water as a refrigerant with some current refrigerants. *International Journal of Energy Research*, 29, 947-959.

Kim, K. H., & Kim, K. C. (2014). Thermodynamic performance analysis of a combined power cycle using low grade heat source and LNG cold energy. *Applied Thermal Engineering*, *70*, 50-60.

Liu, Y., & Guo, K. (2011). A novel cryogenic power cycle for LNG cold energy recovery. *Energy*, *35*, 2828-2833.

Liu, B., Riviere, P., Coquelet, C., Gicquel, R., & David, F. (2012). Investigation of a two stage Rankine cycle for electric power plants. *Applied Energy*, *100*, 285-294.

Lu, T., & Wang, K. S. (2009). Analysis and optimization of a cascading power cycle with liquefied natural gas (LNG) cold energy recovery. *Applied Themal Engineering*, *29*, 1478-1484.

Martinez, F. R., Rueda, A. A., Velaquez, M. T., Diez, P. Q., Eslava, G. T., & Francis, J. A. (2011). Evaluation of the Gas Turbine Inlet Temperature with Relation to the Excess Air. *Energy and Power Engineering*, *3*, 517-524.

Miyazaki, T., Kang, Y. T., Akisawa, A., & Kashiwagi, T. (2000). A combined power cycle using refuse incineration and LNG cold energy. *Energy*, *25*(7), 639-655.

Panchal, C. B, & Ljubicic, B. (2007, July). Designing two-phase flow heat exchangers for mitigating fouling. *ECI Symposium Series.* Symposium conducted at 7th International Conference on Heat Exchanger Fouling and Cleaning Challenges and Opportunities, Tomar, Portugal.

Patel, D., Mak, J., Rivera, D., & Angtuaco, J. (2013, March 26). Training | GAS TECHNOLOGY INSTITUTE. Retrieved from http://www.gastechnology.org/Training/Pages/default.aspx

Pu, L., Qu, Z., Bai, Y., Qi, D., Song, K., & Yi, P. (2014). Thermal performance analysis of intermediate fluid vaporizer for liquefied natural gas. *Applied Thermal Engineering*, *65*, 564-574.

Qiang, W., Yanzhong, L., & Jiang, W. (2004). Analysis of power cycle based on cold energy of liquefied natural gas and low-grade heat source. *Applied Thermal Engineering*, *24*, 539-548.

Rocca, V. L. (2010). Cold recovery during regasification of LNG part one: Cold utilization far from the regasification facility. *Energy*, *35*, 2049-2058.

Rovira, A., Munoz-Anton, J., Montes, M. J., & Martinez-Val, J. M. (2013). Optimization of Brayton cycles for low-to-moderate grade thermal energy sources. *Energy*, *55*, 403-416.

Schulte-Fischedick, J., Dreißigacker, V., & Tamme, R. (2007). An innovative ceramic high temperature plate-fin heat exchanger for EFCC processes. *Applied Thermal Engineering*, *27*, 1285-1294.

Shah, R. K., & Sekulić, D. P. (2003). *Fundamentals of heat exchanger design*. Hoboken, NJ: John Wiley & Sons.

Shi, X., & Che, D. (2009). A combined power cycle utilizing low-temperature waste heat and LNG cold energy. *Energy Conversion and Management*, *50*, 567-575.

Shi, X., Agnew, B., Che, D., & Gao, J. (2010). Performance enhancement of conventional combined cycle power plant by inlet air cooling, inter-cooling and LNG cold energy utilization. *Applied Thermal Engineering*, *30*, 2003-2010.

Sigli, C., Herman, T., Graeve, I. D., Hugh, S., & Ryckeboer, M. (2010). aluMATTER | Aluminium | Heat Exchanger | Property Charts: Thermal Conductivity I. Retrieved from http://aluminium.matter.org.uk/content/html/ENG/default.asp?catid=166&pageid=2144416529

Stradioto, D. A., Seelig, M. F., & Schneider, P. S. (2015). Performance analysis of a CCGT power plant integrated to a LNG regasification process. *Journal of Natural Gas Science and Engineering*, *23*, 112-117.

Sun, H., Zhu, H., Liu, F., & Ding, H. (2014). Simulation and optimization of a novel Rankine power cycle for recovering cold energy from liquefied natural gas using a mixed working fluid. *Energy, 70*, 317-324.

Tchanche, B. F., Lambrinos, G., Frangoudakis, A., & Papadakis, G. (2011). Low-grade heat conversion into power using organic Rankine cycles – A review of various applications. *Renewable & Sustainable Energy Reviews*, *15,* 3963-3979.

Wang, E. H., Zhang, H. G., Fan, B. Y., Ouyang, M. G., Zhao, Y., & Mu, Q. H. (2011). Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. *Fuel and Energy Abstracts*, *36*, 3406-3418.

Wartsila. (2015). Gas Turbine for Power Generation: Introduction. Retrieved March 2, 2015, from http://www.wartsila.com/en/power-plants/learning-center/gas-turbine-for-power-generation

Yamazaki, K., Shimokawatoko, T., Yamasaki, Y., & Takata, M. (1998). *Development of new type of open rack LNG vaporizer*. Osaka Gas Co. Ltd.

Zhang, N., & Lior, N. (2008). A novel Brayton cycle with the integration of liquid hydrogen cryogenic exergy utilization. *International Journal of Hydrogen Energy*, *33*, 214-224

APPENDICES

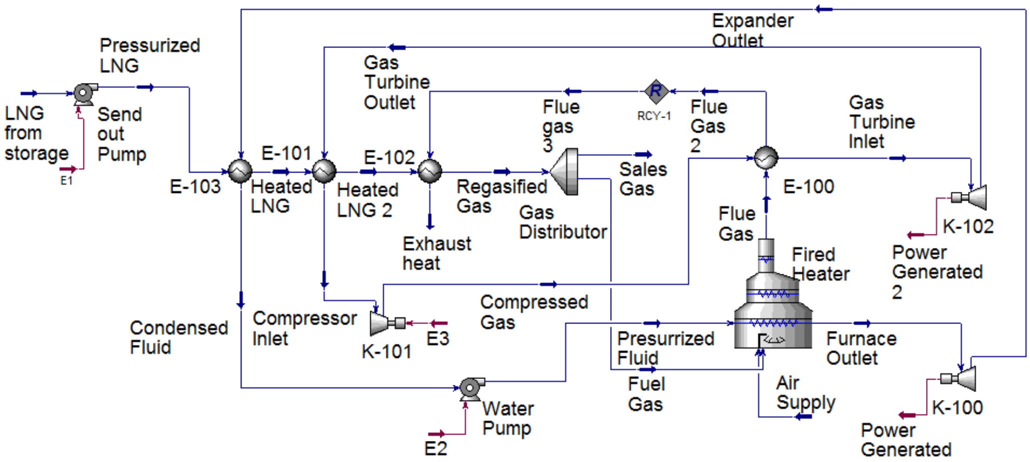
Appendix 1: Process Flow Diagram of proposed model.

Appendix 2: Hysys workbook stream data.

Appendix 3: Thermal efficiency spreadsheet.

Appendix 4: Exergy efficiency spreadsheet.

Appendix 1: Process Flow Diagram of proposed model.



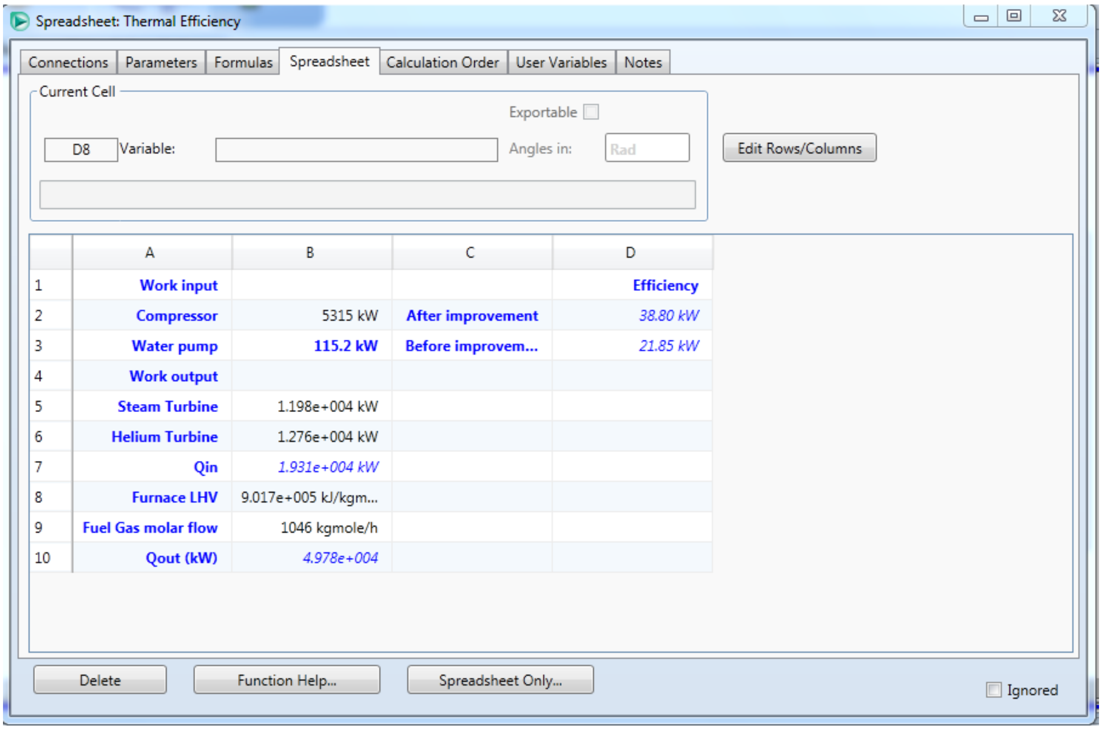
Appendix 2: Hysys workbook stream data.

Name	LNG from storage	Pressurized LNG	Heated LNG	Regasified Gas	Sales Gas	Fuel Gas	Presurrized Fluid	Expander Outlet
Pressure [kPa]	108.063	3344	3320	3294	3294	3294	5516	101.3
Temperature [C]	-160	-158.811	-113.808	4.44	4.44	4.44	38.89	159.844
Mass Flow [kg/h]	1.09463E+06	1.09463E+06	1.09463E+06	1.09463E+06	1.07548E+06	19156.1	60116.4	60116.4
Std Ideal Liq Vol Flow [m3/h]	3419.33	3419.33	3419.33	3419.33	3359.49	59.8383	60.2377	60.2377
Vapor / Phase Fraction	0	0	0	1	1	1	0	1
Molar Enthalpy [kJ/kgmole]	-93004.2	-92862.4	-90191.7	-78624.4	-78624.4	-78624.4	-285050	-237279
Utility Type	-1	-1	-1	-1	-1	-1	-1	-1
Stream Price Factor								
Stream Price Basis	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow
Cost Flow [Cost/hr]								

Name	Furnace Outlet	Compressed Gas	Compressor Inlet	Gas Turbine Inlet	Gas Turbine Outlet	Condensed Fluid	Air Supply	Flue Gas
Pressure [kPa]	5516	3300	500	3275	550	7	101.3	101.3
Temperature [C]	540.047	193.315	27	1000	724.988	39.2365	25	1343.8
Mass Flow [kg/h]	60116.4	132029	132029	132029	132029	60116.4	428160	447316
Std Ideal Liq Vol Flow [m3/h]	60.2377	159.97	159.97	159.97	159.97	60.2377	498.499	538.109
Vapor / Phase Fraction	1	1	1	1	1	0	1	1
Molar Enthalpy [kJ/kgmole]	-224356	-387544	-393922	-345521	-360838	-285115	-8.14047	-17783.9
Utility Type	-1	-1	-1	-1	-1	-1	-1	-1
Stream Price Factor								
Stream Price Basis	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow
Cost Flow [Cost/hr]								

Name	Gas Turbine Outlet	Condensed Fluid	Air Supply	Flue Gas	Heated LNG 2	Exhaust heat	Flue gas 3	Flue Gas 2
Pressure [kPa]	550	7	101.3	101.3	3300	101.3	101.3	101.3
Temperature [C]	724.988	39.2365	25	1343.8	-90.9795	67.4747	1137.69	1137.69
Mass Flow [kg/h]	132029	60116.4	428160	447316	1.09463E+06	447316	447316	447316
Std Ideal Liq Vol Flow [m3/h]	159.97	60.2377	498.499	538.109	3419.33	538.109	538.109	538.109
Vapor / Phase Fraction	1	0	1	1	0	1	1	1
Molar Enthalpy [kJ/kgmole]	-360838	-285115	-8.14047	-17783.9	-88531.1	-62629.7	-25656.1	-25656.1
Utility Type	-1	-1	-1	-1	-1	-1	-1	-1
Stream Price Factor								
Stream Price Basis	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow	Molar Flow
Cost Flow [Cost/hr]								

Appendix 3: Thermal efficiency spreadsheet.



	A	B	C	D
1	Work input			Efficiency
2	Compressor	5315 kW	After improvement	38.80 kW
3	Water pump	115.2 kW	Before improvem...	21.85 kW
4	Work output			
5	Steam Turbine	1.198e+004 kW		
6	Helium Turbine	1.276e+004 kW		
7	Qin	1.931e+004 kW		
8	Furnace LHV	9.017e+005 kJ/kgm...		
9	Fuel Gas molar flow	1046 kgmole/h		
10	Qout (kW)	4.978e+004		

Appendix 4: Exergy efficiency spreadsheet.

	A	B	C	D	E
1	Unit	Exergy Source (kJ/...	Exergy Sink (kJ/h)	Exergy Lost (kJ/h)	Efficiency (%)
2	E-100	1.012e+008	8.011e+007	2.110e+007	79.15
3	E-101	5.059e+007	7.394e+007	-2.335e+007	146.2
4	E-102	3.632e+008	2.545e+008	1.087e+008	70.08
5	E-103	3.132e+007	1.900e+008	-1.586e+008	606.5
6	K-100	5.421e+007	4.312e+007 kJ/h	1.108e+007 kJ/h	79.55 kJ/h
7	K-101	1.913e+007 kJ/h	1.764e+007	1.494e+006 kJ/h	92.19
8	K-102	4.716e+007	4.595e+007 kJ/h	1.209e+006 kJ/h	97.44 kJ/h
9	Water Pump	4.146e+005 kJ/h	3.856e+005	2.901e+004 kJ/h	93.00
10	Fired Heater	2.640e+008	8.514e+007	1.788e+008	32.25
11	Overall Process	9.312e+008 kJ/h	6.008e+008 kJ/h	3.304e+008 kJ/h	64.52
12					
13	Compressed Gas	222.3 kJ/kg		Gas flowrate	1.320e+005 kg/h
14	Gas Turbine Inlet	829.0 kJ/kg		Steam flowrate	6.012e+004 kg/h
15	Compressor Inlet	88.68 kJ/kg		Flue gas flowrate	4.473e+005 kg/h
16	Gas Turbine Outlet	471.9 kJ/kg		LNG flowrate	1.095e+006 kg/h
17	Flue gas 1	1051 kJ/kg		Pressurized LNG	934.4 kJ/kg
18	Flue gas 2	824.7 kJ/kg		Heated LNG	760.8 kJ/kg
19	Pressurized Fluid	7.727 kJ/kg		Heated LNG 2	693.3 kJ/kg
20	Furnace Outlet	1424 kJ/kg		Regasified Gas	460.8 kJ/kg
21	Expander Outlet	522.3 kJ/kg		Exhaust Heat	12.76 kJ/kg
22	Condensed fluid	1.312 kJ/kg		Fuel Gas	460.8 kJ/kg
23					
24	Water Pump power	4.146e+005 kJ/h		Power generated 1	4.312e+007 kJ/h
25	Compressor power	1.913e+007 kJ/h		Power generated 2	4.595e+007 kJ/h
26					