Cold Energy Utilization from LNG Regasification

by

Mohd Zafri bin Mazri

Dissertation submitted in partial fulfilment of

the requirements for the

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(Chemical Engineering)

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CERTIFICATION OF APPROVAL

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Approved by,

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(Associate Professor Dr. Shuhaimi Mahadzir)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

MAY 2013

CERTIFICATION OF ORIGINALITY

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

MOHD ZAFRI BIN MAZRI

ABSTRACT

This report deals with Liquefied Natural Gas (LNG) cold energy utilization by integrating the system with gas power plant and Rankine cycle to generate electricity. There is a waste of cold energy available during the LNG regasification due to the use of seawater and returning it back to ocean. The cold energy available is better converted into useful energy such as electricity via Rankine power cycle implementation. Another aspect to look at is on the system performance of ordinary Rankine cycle utilizing water as working fluid which is quite low. The objectives of this research is to develop an integrated system to fully utilize cold energy available via Rankine cycle in existing gas power plant and to experiment with other working fluids to be utilized in the cycle taking pure ammonia as the basis. Two case studies are developed with first being an integration of the LNG regasification process with gas power plant to yield a targeted amount of power generation which is about 404 MW. The second case study integrates the Rankine cycle into previous system to utilize the LNG cold available. Simulation work is carried out using Aspen Hysys to check for the system's feasibility. The efficiency of the overall system is analysed based on thermal and exergy efficiency respectively for both case studies. The effect of the inlet and outlet pressure of the gas turbine on overall system efficiency is investigated which resulted in highest efficiency when the expansion ratio of the gas turbine is at the highest. It is found that the second case study improves the thermal and exergy efficiency by 5.1 % and 2.4 % respectively. Five working fluids are used to study their effects on system efficiency which are ammonia, water, ammonia-water mixture, ethane and propane. As expected from various literature reviews, ammonia yields the highest system's efficiency compared to other working fluids with improvement of about 0.64 % over pure water but with penalty of higher mass flow required approximately 5.2 tonnes h^{-1} to achieve operating specification as discussed in results section later in this paper. Based on the results obtained, it is proven that the efficiency of gas power plant can be further increased by integrating with Rankine power cycle and at the same time effectively utilizing the LNG cold energy available.

ACKNOWLEDGEMENT

First and foremost, I would like to express my deepest gratitude to **Associate Professor Dr. Shuhaimi Mahadzir** for his continuous support and excellent guidance during the period of this project's completion. I am also indebted to him for he gives morale support and encouragement to successfully finish this project. My gratitude also goes to Dr. Balasubramanian Periyasami for his support, valuable suggestions and comments to further improve undertaken project throughout his participation as internal examiner. Special thanks towards Universiti Teknologi PETRONAS (UTP) especially Chemical Engineering Department for their excellent management and organization for this course. I greatly appreciate the benefits of lifelong learning since embarking into this project as I gradually improve my report writing skills and gaining beneficial experiences over time. Last but not least, I would like to thank my family and friends as well as people whom directly and indirectly involved for their tireless support for the past eight months.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND OF STUDY

Transport of Natural Gas (NG) from a gas field by pipelines to consumers is often impossible to accomplish thus requiring the need of transporting NG to receiving terminal via ship. To accomplish this, NG is converted into Liquefied Natural Gas (LNG) at a very low temperature of around -162 ^{o}C and at atmospheric pressure after removing acid substances and water. During liquefaction also, the volume of LNG can be reduced about 600 times than that of NG therefore enable it to be transferred in LNG tanks via ship to the receiving terminal (Kumar, et al., 2011) . Large amount of mechanical energy is consumed to convert NG to LNG which is about 850 kWh of electric energy to produce one tonne of LNG (Shi & Che, 2009).

At the terminal, LNG needs to be evaporated into gaseous form at ambient temperature and at suitable elevated pressure prior to distribution system. This is usually done at regasification facility. LNG is a clear, non-toxic, non-corrosive, odourless, and in liquid form at atmospheric pressure. The density of LNG is around 400-500 kg m^{-3} depending on the given temperature, pressure and composition. Therefore, if spilled on water with approximately 1000 kg m^{-3} density, LNG will float on top and vaporizes faster and disperses leaving no residue. Such properties contribute to no cleaning up if LNG is accidentally spilled either on water or land. A conventional regasification facility is shown in Figure 1 which consists of subsystems including ship unloading, LNG storage, LNG low pressure pump, tank boil-off gas compression, vapour re-condensation, LNG high pressure pump and LNG regasification.

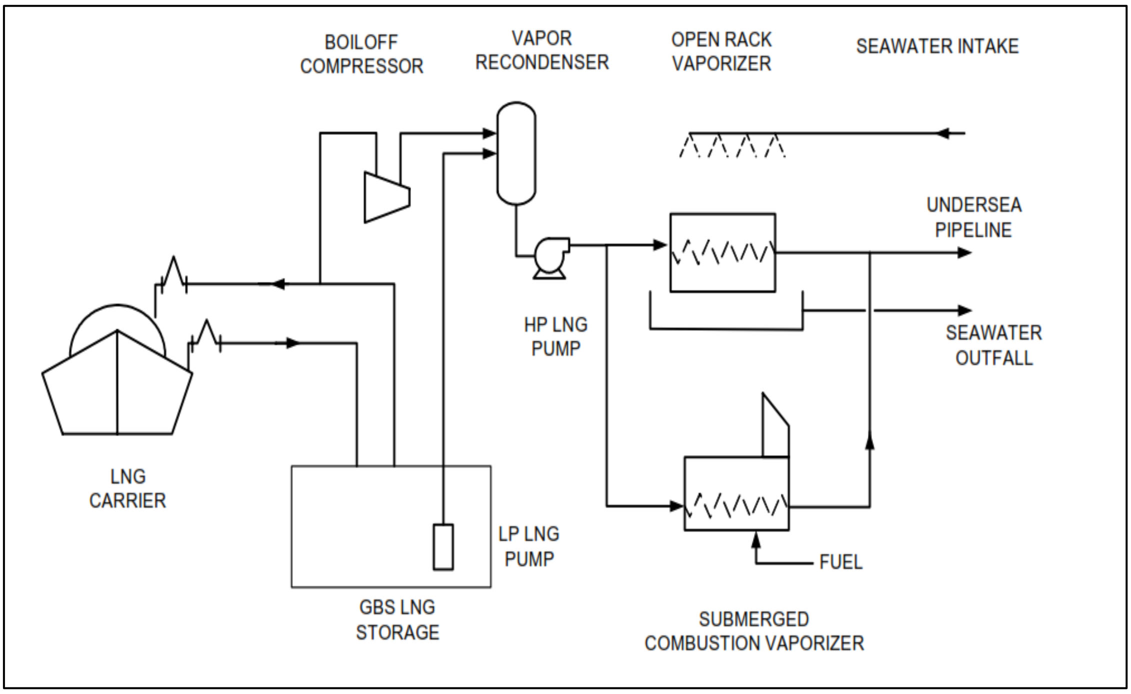


Figure 1 : Conventional LNG regasification plant.

Vapours generated from LNG ship unloading and during normal operation are compressed by the boil-off vapour compressor and then re-condensed by mixing with sub-cooled LNG send out. The condensed LNG is then pumped to pipeline pressure and heated to about 5 ^{o}C prior to the pipeline's distribution. In addition, latent heat of vaporization and any sensible heat required to superheat the vapour during the LNG regasification are termed 'cold energy', and this is usually supplied by seawater. Such process needs about 800 kJ kg^{-1} of energy (Liu & Guo, 2011). Liu and You (1999) stated that cold energy is a form of high quality energy in thermodynamic point of view. There is an opportunity to recover the cold energy rather than just taken off by seawater which shall contribute to environmental impact such as global warming. Past few decades have seen a lot of methods been developed to utilize cold energy from LNG. These include power generation, air separation, and intake air cooling. Among those, cryogenic power generation is the most effective one (Liu & Guo, 2011). One example of cold energy power generation system is propane organic Rankine cycle (ORC). La Rocca (2010) proposed a modular LNG regasification unit based on a power cycle utilizing ethane as the working fluid. Other effective LNG cold energy utilization to produce electricity is by employing cryogenic stream of LNG during regasification as a cold source in an improved Combined Heat and Power (CHP) plant.

This paper deals with the utilization of LNG cold energy from regasification process to produce electricity in a CHP plant utilizing Rankine cycle as the working cycle. The author shall study on the integration of LNG regasification plant, with a gas power plant via Rankine cycle in order to fully utilize LNG cold energy into electricity. A simulation work of complete set up of aforementioned integrated plant with suitable working fluid shall be carried out upon the completion of this project.

1.2 PROBLEM STATEMENTS

There is waste of cold energy available during LNG regasification due to the use of seawater as the heat source and returning it back to ocean. Such an act could also contribute to environmental impact such as global warming.

The performance of conventional Rankine cycle utilizing water as working fluid in CHP plant is quite low. There is an opportunity to further increase the system performance by using other working fluids.

1.3 OBJECTIVES

The objectives of this research include:

1. To develop an integrated plant consisting of a LNG regasification plant, a gas power plant and a Rankine power cycle.
2. To verify the feasibility of the project via simulation of complete plant design utilizing Aspen Hysys simulation software.
3. To determine optimum working fluid for Rankine power cycle for cold energy extraction

1.4 RELEVANCY OF PROJECT

Cold energy recovery during the LNG regasification is of vital importance in order to optimize the exergy thus resulting in lower operating cost. The LNG cold energy available is better converted into useful energy such as electricity rather than just releasing it to the open atmosphere. The author shall look into that kind of opportunity to recover and converting the cold energy available into electricity.

LNG has been commonly used in the energy industry aside from civil life nowadays. According to World Energy Outlook 2005 and 2006 (WEO 2005 and WEO 2006) of the International Agency (IEA), revealed that global energy demand is expected to rapidly increase by the year 2030. Thus, more and more LNG regasification plants capacity will further increase to meet the growing demand. However, with increasing capacity of regasification plants will also increase the environmental impact due to cold reject. Hence, effective recovery of cold energy available is important to tackle the above problem.

1.5 FEASIBILITY OF THE PROJECT

Many methods have been developed to utilize the LNG cold energy. The main focus of this project is for the author to embark into the practical utilization of Rankine power cycle as the working cycle in a combined regasification and a gas power plant.

This project is feasible enough for the author to carry out. The only constraint in this project is the total time allocated which is approximately nine (9) months. However, given the open literature sources and previous researches been carried out by other researchers shall provide the author insight on what to plan and benchmark based on their results. Furthermore, this project is a simulation based project, which means that the author shall not have difficulty to establish workstation to carry out the study. The need for setting up laboratory apparatus is neglected due to the fact that the main software based tool for this project is Aspen Hysys simulation software.

1.6 SCOPE OF STUDIES

The author shall cover the following scope of studies as shown in Figure 2:

Figure 2 : Scope of studies.

The cold energy available during LNG regasification process shall provide thermal heat to drive gas turbine in the power plant to produce electricity. The basis generation of electricity production in this project shall be based on production capacity per annum from well-known local company as the reference. Further research shall be carried out to obtain the optimal performance of Rankine power cycle via:

- Optimum working fluid based on the optimal thermodynamic condition.
- Optimal configuration of system units throughout the cycle for better thermal and exergy efficiency.

CHAPTER 2

LITERATURE REVIEW

2.1 LNG COLD ENERGY UTILIZATION

NG is widely available and renewable whether through the production of bio-gas or bio-methane (Kumar et al., 2011). NG is usually termed as 'green fuel' due to its higher energy density and environmental friendly advantages. Among the advantages of NG are it reduces greenhouse gas and produces lower emissions compared to other alternative fuels. Independent thermal cycle with NG direct expansion and closed-loop Rankine cycle is usually associated with electric power generations from LNG cold energy utilization (Lu & Wang, 2009).

Among the utilization of cold energy from LNG regasification include power generation, air separation, refrigeration, cold storage, inlet air cooling for gas turbine power generation, production of dry ice and seawater desalination. In fact, power generation system operated by utilizing cold exergy of LNG and propane organic Rankine cycle has been applied in Japan for approximately 40 years (Liu & Guo, 2011). Along the line, many other researches related to cold energy utilization to generate electricity have also been conducted. An ammonia-water Rankine cycle with refused incinerator has been proposed by Miyazaki et al. (2000) and was compared with typical Rankine cycle. In addition, Szargut and Szcygiel (2009) studied theoretical LNG cryogenic power cycle using two binary working fluids, and one with ethane as the working fluid.

Dang et al. (2009) proposed a cogeneration power system with two energy sources of fuel chemical energy and cryogenic LNG. Apart from that, two outputs of electrical and cooling power were also proposed. The system employed advanced integration of system and cascade utilization of cold energy thus yielded a great energy saving.

Dispenza, La Rocca and Panno (2009) proposed a CHP plant using Brayton cycle to produce electricity by recovering the exergy as a cold using the cryogenic stream of LNG. As shown in Figure 3, the cryogenic stream of LNG acted as the cold source whilst the working fluids used in the cycle were helium and nitrogen. Both helium and nitrogen were benchmarked in terms of overall percentage of electricity production. The proposed system improved the electric efficiency of an electric utility in power stations working with steam turbines by lowering the temperature of the condenser. One way to achieve this was by further utilizing cooled water reject by Open Rack (OR) unit.

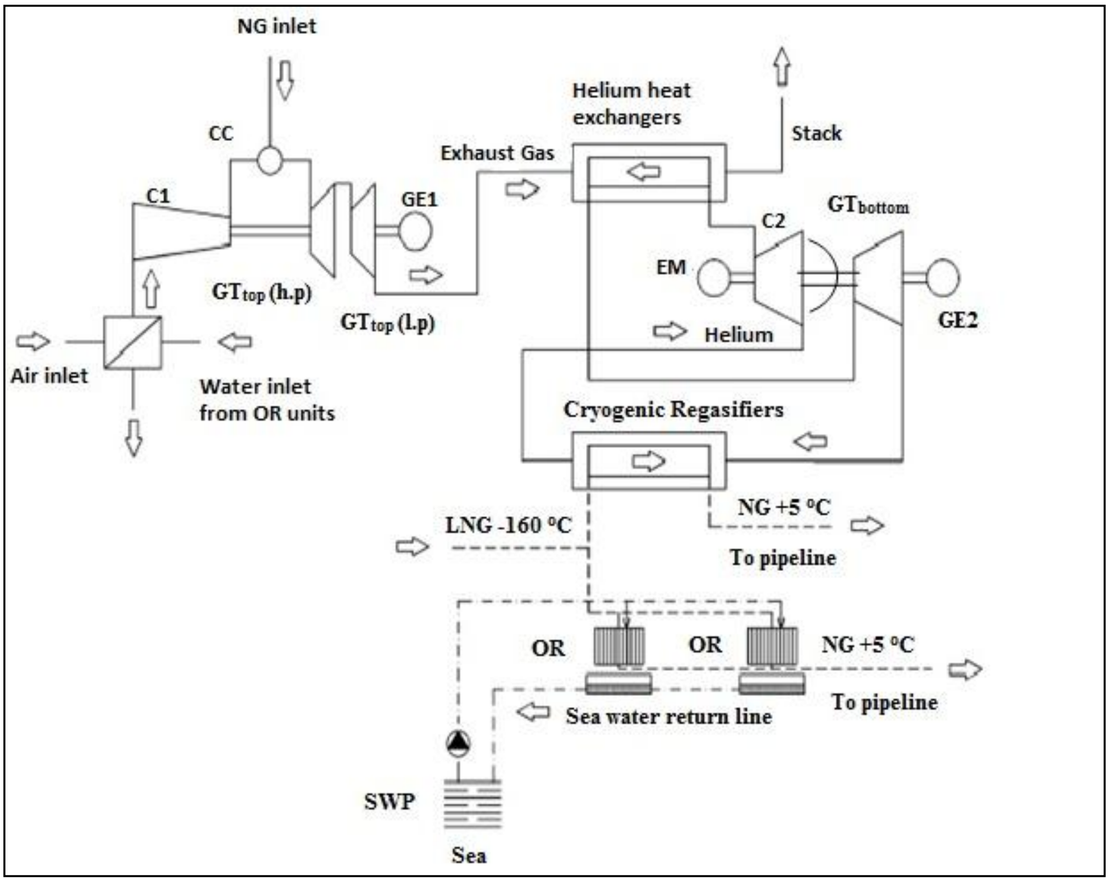


Figure 3 : The modular CHP cycle for LNG regasification. *Source*: Dispenza et.al (2009).

The system applied Brayton cycle with an open-loop top cycle with steam turbine whilst the bottom cycle was the closed-loop containing helium or nitrogen as the fluid. In order to further improve the thermal performance of heat transfer during regasification process, cryogenic regasifier had heat transfer matrix made with extended surface tubes. This allowed for working fluids to flow in from the shell side and LNG from the tube side. According to Dispenza et al. (2009), helium was more superior gaseous fluid compared to nitrogen. The nitrogen required higher operating pressure in the bottom cycle and its performance was lower compared to helium. Overall, modular plant working with helium produced 0.38 kW kg^{-1} of LNG regasified whilst nitrogen as the working fluid yielded about 0.29 kW kg^{-1} of LNG regasified. Nitrogen produced 0.09 kW less electricity compared to helium. A total of 24 % of higher performance in terms of electricity production could be obtained by using helium instead of nitrogen in the proposed modular plant.

Liu and Guo (2010), has proposed the use of novel cryogenic power cycle with binary working fluids and absorption process for LNG cold recovery. Their objective was to improve the energy recovery efficiency of an LNG cold power generation. The main components of the system include a generator, two turbines, a re-heater, a condenser, an absorber, a solution pump, an expander and a heat exchanger as shown in Figure 4. Tetraflouromethane (CF_4) and propane (C_3H_8) were used as the binary mixture of working fluid for the proposed system. CF_4 worked as the main expansion fluid in the turbine whilst C_3H_8 acted as the absorbent for CF_4.

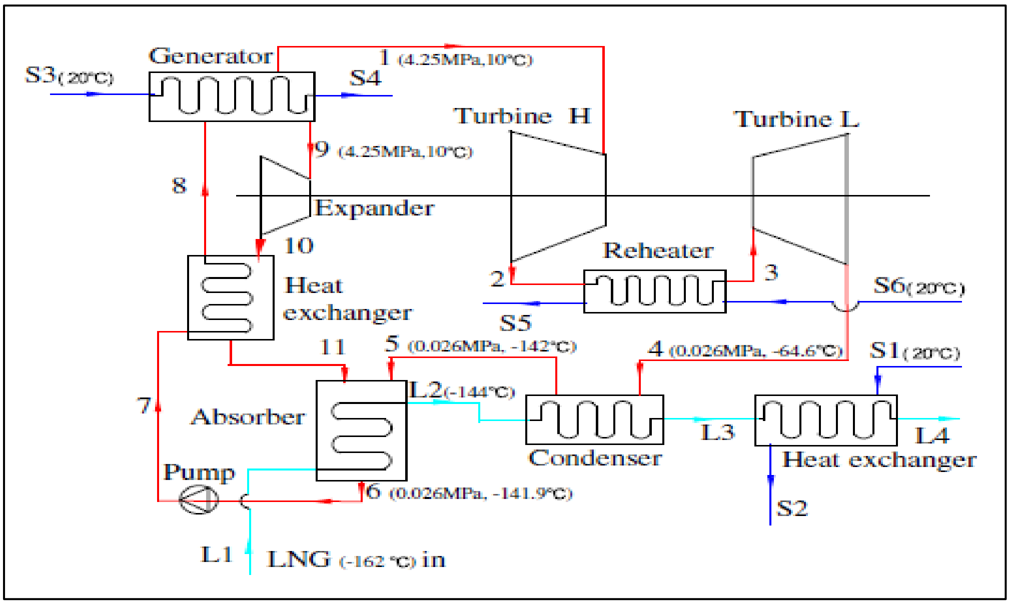


Figure 4 : A novel cryogenic power cycle. *Source*: Liu and Guo (2010).

A simulation of the above cycle was performed to model the real working application. The thermodynamic of the binary fluids were determined using Peng-Robinson Equation of State (EOS). The proposed novel cycle had 66.3 % of improvement in term of power generation and availability efficiency compared to ORC. In addition, there were 15.4 ^{o}C improvements in term of LNG utilization temperature. This was caused by the introduction of cryogenic absorber thus reducing turbine back pressure. The authors also proposed to use power generated per unit vaporized LNG as the performance indicator rather than using the power per unit heat discharged from seawater.

Use of LNG cold energy for power generation has also been reported by Shi and Che (2009). A combined power cycle utilizing low-temperature waste heat and LNG cold recovery was their main focus in the research. The basic concept of the combined power cycle is for LNG cold energy could be fully utilized as well as the recovery of low-temperature waste heat. Figure 5 reveals their schematic of integrated Rankine cycle and LNG power generation cycle with ammonia-water mixture as the binary working fluid.

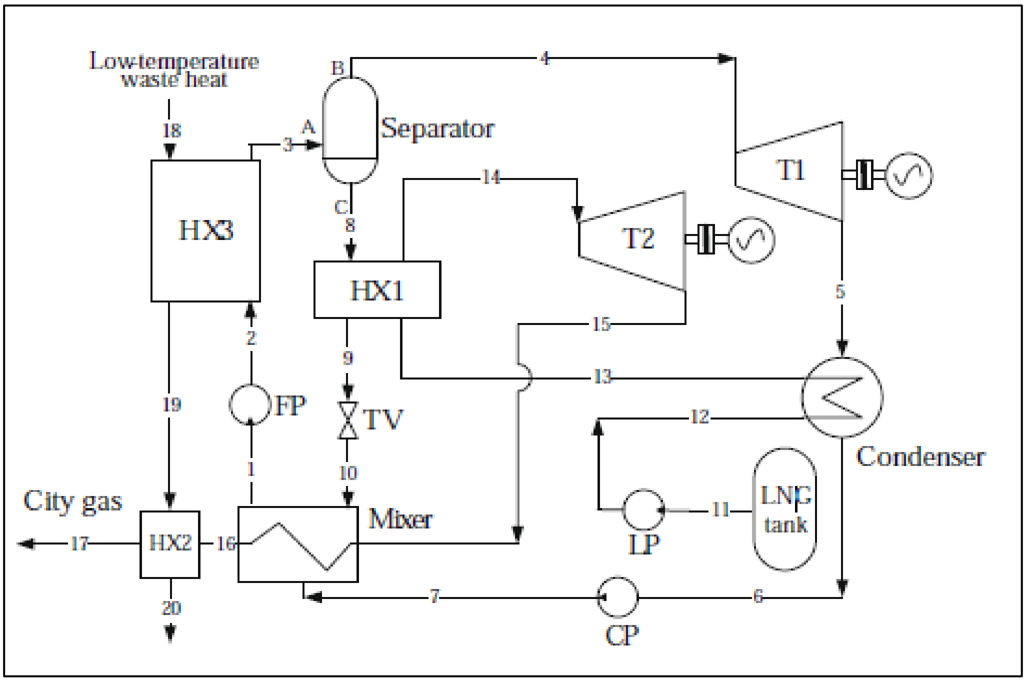


Figure 5 : A schematic diagram of proposed combined power cycle. *Source*: Shi and Che (2007).

For ammonia and LNG turbine to work, partial vaporization of the working fluid was implemented. The partial vaporized ammonia would drive ammonia turbine and the weak solution coming out from the bottom of separator would heated LNG to drive LNG turbine. LNG cycle with directly expanding NG utilized latent heat of spent ammonia vapour. In contrary, the heat of weak solution of ammonia-water returning to mixer acted as the heat source for the power generation. To fully recover both low-temperature waste heat and heat exchanged within the system components, the heat exchangers in the system were arranged in the manner of that according to temperature of both hot and cold streams. The proposed system yielded net electrical efficiency of 33 % with power generation of 8.3 MW. In addition, the exergy efficiency recorded was 48 %. Another important finding using this system was that 0.2 MW of electrical power for operating seawater pumps could be saved.

Shi, Agnew, Che and Gao (2010) proposed a thermal power system integrated with inlet air cooling, compressor inter-cooling and LNG cold energy utilization to further improve LNG power plant. The proposed system was targeted to enhance the performance of conventional power cycles by using heat of spent steam from the steam turbine. The system proposed by Shi et al. (2010) worked by recovering latent heat of spent steam from steam turbine and latent heat of vaporization of water vapour contained in the flue gas. This could be achieved by using novel recovery and utilization system for LNG fuelled conventional combined power plant. According to the authors also, cold energy generated during the vaporization of LNG was used to condense the spent steam from the steam turbine thus saving more electric power. Their results yield an increase of 76.8 MW in terms of power output compared to conventional power plant. The net electrical efficiency also increased by 2.8 %. Another important finding was that 0.9 MW of electric power for operating seawater pumps could be further reduced due to the elimination of 10000 tonnes h^{-1} of seawater during the regasification process. Overall, the proposed systems yield an increase in power output as well at the same time reducing the shaft work needed to operate the seawater pump.

According to S. Mahadzir and V. Gopinadhan, a further recovery opportunity of LNG cold energy could be maximized from the scrub column bottom during the NG liquefaction process. Their main objective was to focus on cold energy recovery from optimization of heat integration point of view. The modified system proposed an addition of a new heat exchanger into the existing configuration as shown in the Figure 5.

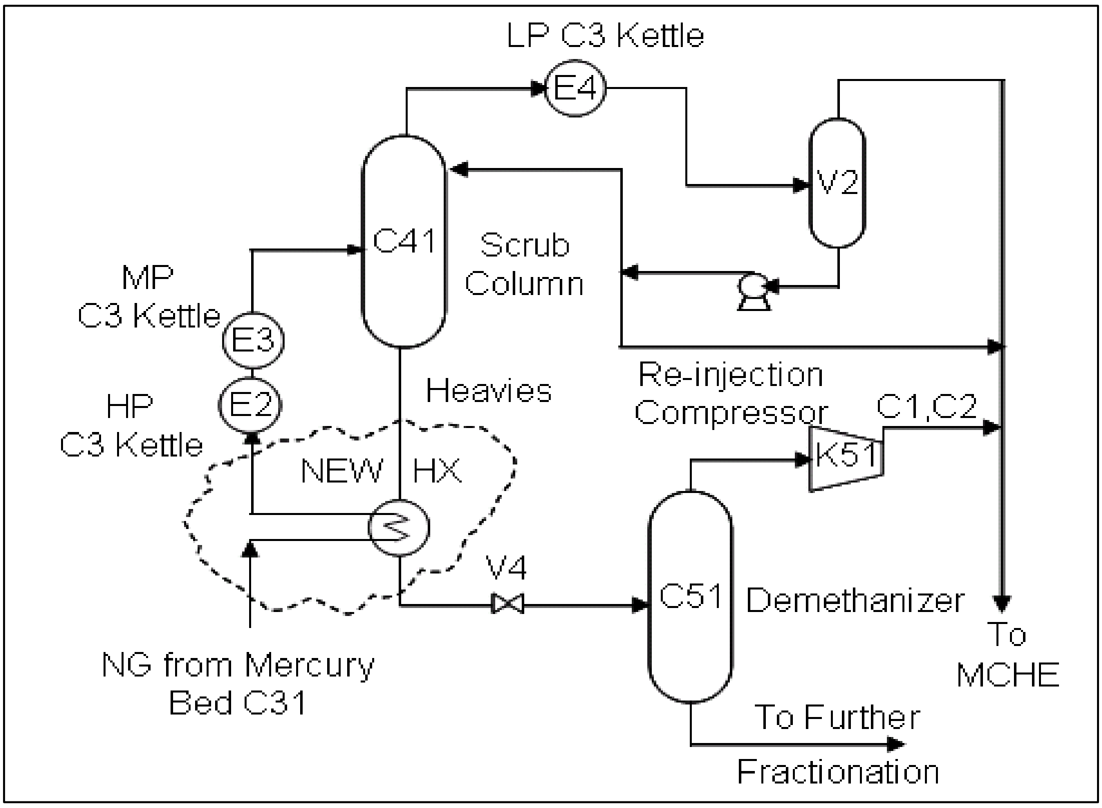


Figure 6 : Schematic diagram for the scrub column unit line-up (proposed modification in dotted lines). *Source*: S. Mahadzir and V. Gopinadhan.

The study focused on Propane-to-NG (C3-NG) circuit and Propane-to-Mixed Refrigerant (C3-MR) circuit. Heat integration analysis and process simulation was developed using SMIRK property package. The proposed modification performance was in terms of compression power saved from C3-NG and C3-MR circuits in four stages. It was found that there was an increase of power savings for all the stages with the introduction of the new heat exchanger. The total savings of C3 compressor power for all the 4 stages was 0.413 MW. Meanwhile, the total power saving for the MR compressor was 0.686 MW. The power savings yielded contribute to 0.82 % increment for daily production led to a total of RM 43 Million per year increased in revenue. The only drawback of the proposed modification was on the initial investment of approximately RM 0.72 Million for the extra heat exchanger.

Sharrat from Foster Wheeler discussed LNG terminal cold energy integration opportunities offered by contractors. In his study, he discussed major cold energy utilization methods which are Inlet Air Cooling (IAC) to Gas Turbine Generators (GTG) in which he claimed increased the power output of the turbine. For IAC to GTG application, suitable working fluid such as glycol-water mixture was used as the working fluid to extract the cold energy. The cold extracted during the vaporization process was then transferred to the GTG inlet air. According to him also, the reason why there was an increased in power output was mainly due to density difference between the working fluid and inlet air where the cooler inlet air has higher density. This resulted in larger mass entered the air compressor of the gas turbine given a fixed volumetric flow. The research yielded significant results where the power output from the GTG was increased by 0.5 % for every degree Celsius (^oC) of lowering the inlet air temperature. However, there is an addition of penalty to his proposed system where it generated electrical power but in exchange of higher capital cost for both the LNG import terminal and power plant.

Querol et al. (2011) has proposed power generation cycle utilizing pure ammonia as the working fluid to utilize LNG cold energy available. In their study, ammonia was chosen because Rankine cycle with pure component was the simplest option available yet has proven to be very effective. In order to obtain the desired result, the authors proposed that the concentration of ammonia must be as high as possible in the turbine and vice versa in the condenser. The other sense of using pure ammonia was because ammonia-water mixture required large exchange surfaces, thus increment in capital cost. The author carried out the study based on two alternatives. The first alternative was simulated on gas engine with an ammonia cycle (GE 1pNH_3) with one pressure step and the other one with two pressure steps. For the first alternative, ammonia was directly expanded to 110 bar over turbine. For the second alternative, an intermediate pressure of 12 bar was applied before expanded to 110 bar. It was found that alternative 2 provide lower cost of mean power generation which was about 61.29 euro h^{-1} compared to alternative one, 65.5 euro h^{-1}. The calculation includes all the available power generated by the turbines as well as from NH_3 cycle and then divided by the total power produced. Another important finding was the authors were able to use pure ammonia due to the integration of the ammonia condensation process with the vaporization of LNG.

Khan, Rangaiah, and Lee (2011) focused on Knowledge Based Optimization (KBO) of Mixed Refrigerant (MR) system. Their main focus was on the determination of the optimal composition of MR based on MR boiling points for each component as well as introduction of KBO method. The KBO method was implemented on single-stage MR (SMR) system and C_3MR system. To increase energy efficiency of compressor, the optimal composition of MR was determined at the lowest possible system pressure. According to the authors also, MR was more advantageous than pure refrigerants due to the fact of able to undergo isobaric phase change through a range of temperature contained within the dew and bubble point temperature. Their method employed minimization of shaft work generated with MR component flow rate and system pressure as the decision variable. They also suggested MR to be employed must be a mix of high and low boiling point components in order to provide high refrigeration effect at low pressure. Thus, sequentially decreasing flow fraction of higher boiling components to determine the optimal effect on compression energy was employed. Their research revealed that highest improvement in terms of exergy efficiency could be realized by decreasing the nitrogen flow rate as it had the lowest boiling point among other refrigerants. The optimal MR mixtures determined were nitrogen, methane, ethane and propane.

Gao, Lin and Gu (2011) has also presented their findings on LNG cold energy utilization. In their research, they proposed methods of separating other light hydrocarbons in the LNG aside from methane by effectively utilizing the LNG cryogenic energy. The authors proposed two novel separation methods for the light hydrocarbon. The first method employed demethanizer operating at high pressure of about 4.5 MPa and known as 'high pressure process'. On the other hand, the second method employed low demethanizer operating pressure of about 2.4 MPa and known as the 'low pressure process'. The LNG was first pumped to a pressure of 4.5 MPa and 1.5 MPa in the high and low pressure process respectively. High pressure process employed the use of deethanizer after the demethanizer to recover liquefied ethane and LPG. For low pressure process, the LNG was first heated to gas-liquid two-phase fluid before being fed to demethanizer. The subsequent steps to recover the light hydrocarbon were by utilizing the sensible cooling capacity of LNG to liquefy methane gas after being flashed in the vapour-liquid separator. Low pressure process utilized more of the LNG cold available compared to high pressure process. Finally, the performance of the two methods was benchmarked based on overall economic analysis. It was found that the low pressure process has better performance than high pressure process in profit per Million CNY. [1]However both of the methods had advantages and disadvantages. High pressure process was suitable for limited space environment due to its simplicity and compactness whereby low pressure process required accurate temperature matching. Overall, both of the processes able to effectively utilized the LNG cold available.

Liu and You (1999) investigated the characteristics and applications of the cold exergy of LNG. The authors developed a mathematical model using Soave - Redlich-Kwong (SRK) EOS to predict the various effects of LNG such as the ambient temperature and compositions on its thermal properties. The results obtained revealed that pressure exergy and total cold heat exergy increased when the ambient temperature was increased as shown in Figure 6.

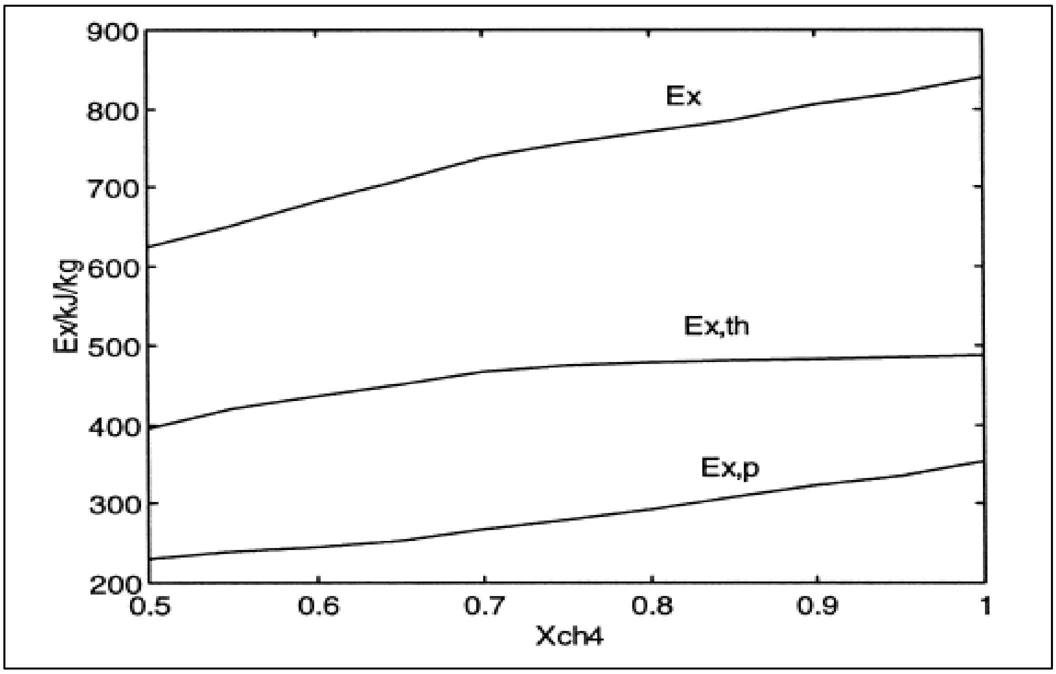


Figure 7 : The variation of exergy of LNG with the methane concentration (T=298 K).
*Source:*Liu and You (2011).

[1] CNY=Chinese Yuan Renminbi

From Figure 7, the total cold heat exergy increased with the concentration of methane. The overall results obtained could be deduced as the low temperature exergy of LNG decreases, the pressure exergy decreases as well as resulting in the decrease of total cold heat exergy. The tendency was however decreasing when pressure is set more than 2 MPa. The authors also suggested that the application of using low temperature exergy could be integrated with Rankine cycle using low boiling temperature of working substances thus increasing further the exergy of the overall system.

2.2 RANKINE CYCLE

Rankine cycle is a mathematical model that is used to predict the performance of steam engines. Rankine cycle converts heat into mechanical work or electric power .Typical Rankine cycle is composed of a pump, turbine, a boiler, and a condenser as shown in Figure 7. The media vapour is condensed by LNG producing a low temperature media. The low temperature media is then pumped into evaporator whereas the seawater provides the supply heat to the evaporator. The high temperature and pressure media is then passed through the turbine to produce power by driving the generator and completing the closed-loop cycle.

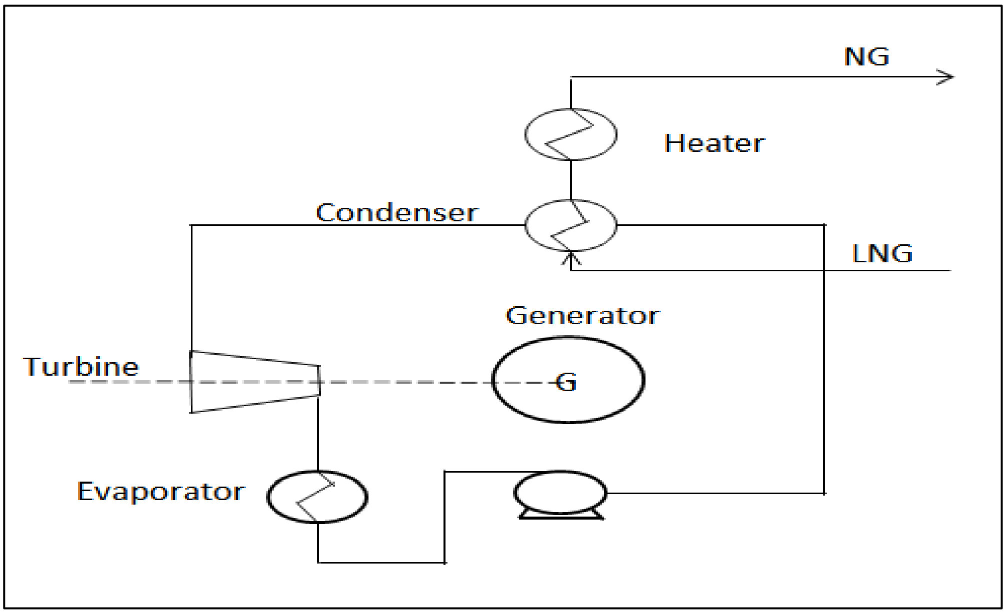


Figure 8 : A closed-loop Rankine cycle.

Usually, water is used as the working fluid in typical Rankine cycle. ORC employs the use of organic fluids such as refrigerants and hydrocarbons instead of typical water. There are two variations of Rankine cycle which are reheat and regenerative power cycle. For reheat power cycle, fully expansion of working fluid to the condenser pressure does not occur in single stage. Instead, the partially expanded working fluid is fed back to the evaporator to be reheated followed by subsequent second expansion resulting in the working fluid ending at the condenser pressure. The advantage of such type of cycle is that it increases the quality of expander exhaust. In regenerative power cycle, a portion of partially expanded working fluid preheats the condensed liquid before it enters the boiler. The advantage of such preheating process include a decrease in the amount of heat added at low temperature, increase of mean effective temperature of heat addition and increase of cycle efficiency. These two variants only apply to ordinary Rankine cycle and ORC is not included.

Liu et al. (2012) studied two stage Rankine cycle for electric power generation. In their research, a modified steam cycle combined with ORC bottoming cycle was applied. In traditional ORC, the steam expands to low pressure whilst their proposed system vapour left the turbine at a very high pressure. About ten pre-selected organic working fluids and ammonia had been tested in the system. The performance of the system was evaluated using the respective working fluids. In their findings, no single fluid had been identified as the optimum working fluid to be used in Rankine cycle mainly due to the strong interdependence between the fluids, as well as the working condition and the cycle architecture itself.

A study on application of ORC in low waste heat conversion into electric power has been carried out by Tchanche, Lambrinos, Frangoudakis and Papadakis (2011). Tchanche et al. (2011) had conducted a study across seven different applications utilizing organic Rankine cycle to produce electric power. These included binary geothermal plants, solar thermal plants, solar ORC-RO desalination systems, Duplex-Rankine cooling systems, ocean thermal energy conversion systems, waste heat recovery plants and last but not least the biomass power plants. Important findings include ORC-RO was still at research stage, solar thermal power plants even though has already proven but still not widely applied, ocean thermal energy conversion was in rapid research, whilst biomass and binary geothermal were already matured and widely adopted. The most rapid growing business among the mentioned applications was the low-grade waste heat recovery plants. The authors conclude that, ”Environmental concern over climate change and rising of oil prices are reasons supporting the explosive growth of this efficient, clean and reliable way of producing electricity”, (Tchanche et al.,2011, pg.14).

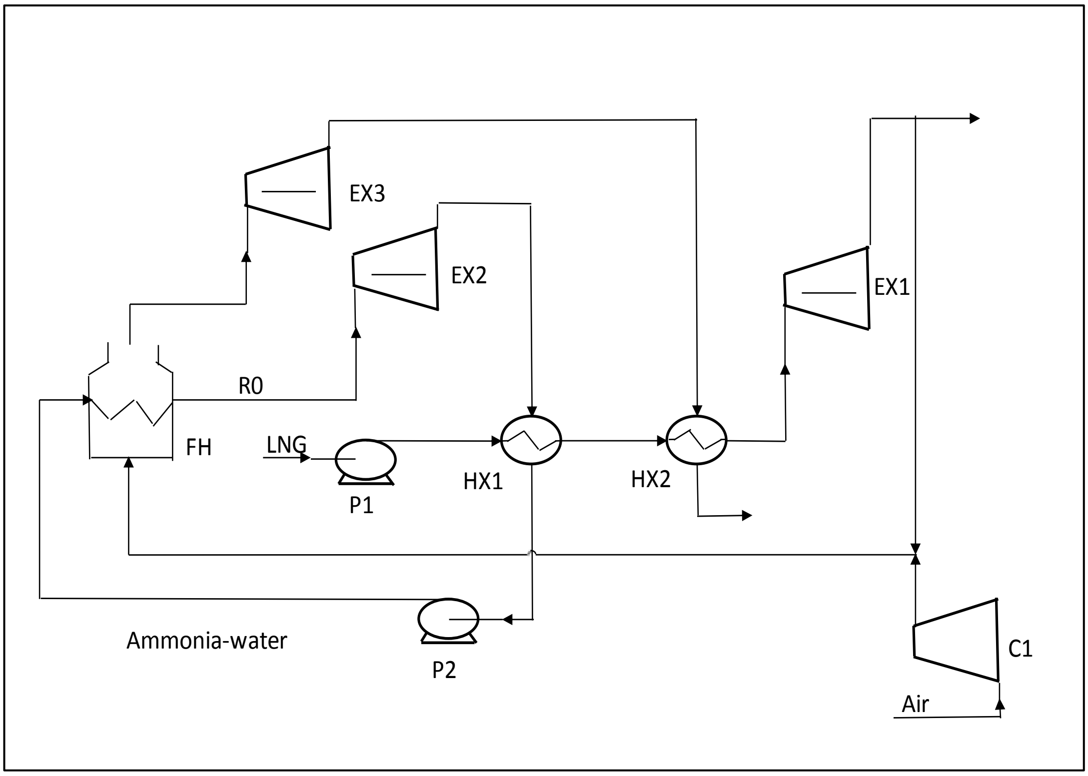


Figure 9 : Flow sheet of cascading power cycle to recover LNG cold availability. *Source*: Lu and Wang (2009).

Lu and Wang (2009) studied the application of Rankine cycle in power generation by proposing a gas electricity generation combined power system for LNG cold energy utilization using Rankine cycle with ammonia-water as the working fluid and Brayton power cycle of combustion gas. The schematic diagram of the proposed system is shown in Figure 8. In the system, a part of heat released by combustion was absorbed by ammonia-water whilst the other part was absorbed by combustion gas itself. The binary fluid of ammonia-water heated in the fired heater

(FH) and converted to high temperature gas then driving the expansion work in the turbine (EX2). The heat sink of the Rankine cycle provided latent heat and sensible heat to LNG. It was found that the thermal and exergy efficiency of the system increased with the condensation temperature of Rankine cycle. Overall, the system performance measured was based on thermal exergy efficiency of the cascading power cycle by manipulating several key parameters such as the condensation temperature and inlet-outlet pressure of the Rankine cycle.

According to Yan et al. (2013), performance of solid oxide fuel cell (SOFC) could be further increased by integrating the system with Rankine cycle. Hence, the author proposed an integrated SOFC-GT and ORC using LNG as the heat sink to recover cold exergy of LNG. LNG was used as the heat sink to condense the exhaust of turbine, to cool down the suction of air compressor as well as be fed to the SOFC as the fuel. The performance criteria were evaluated based on SOFC electrical efficiency and overall electrical efficiency. The study with regard to such integration system has also been carried out by Akkaya and Sahin (2009) whereas a SOFC-ORC system was proposed. Results shown an increase of 14-25 % of efficiency obtained by recovering SOFC waste heat via ORC. Another study by Al-Sulaiman et al. (2010), proposed a tri-generation plant based on SOFC-ORC. 22 % efficiency gain obtained using the proposed combined system compared to just one SOFC system.

Smith (2005) provided a thorough analysis of LNG cold utilization by implementing Rankine cycle to produce electricity. In his research, electricity was generated by utilizing LNG as the heat sink. The basis behind this was there was an increased in terms of power generation efficiency when LNG was used as the heat sink. The idea was that when LNG was used as the heat sink (T_1-T_2) was increased, hence shall yield greater power generation efficiency. According to the author also, the optimal selection of the working fluid in the cycle must be thermally stable at very high temperature and condense at low temperature with no freezing issues. The proposed system is shown in Figure 9. The system working principal was that working fluid was pumped to supercritical pressure at 15000 psig before first heated via the hot expander discharge in the E-2 Recuperator. The fluid then further heated to 600 ^{o}F before depressurized into atmospheric condition in the turbine thus generating work output. The vaporization process of the LNG is simply by utilizing the condensation duty from the working fluid at E-1 Condenser.

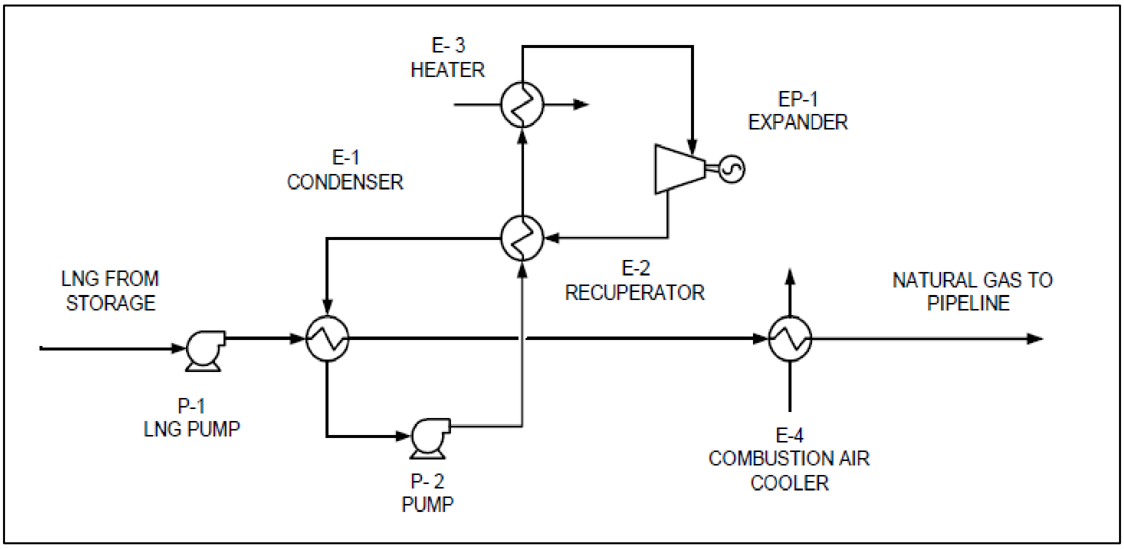


Figure 10 : Rankine cycle for LNG power generation. *Source*: Smith (2005).

The results yielded different power generation efficiency for both the pure and the mixed fluid. The power generation efficiency of butane, propane and mixed fluid were 24.1%, 33.5% and 37.8% respectively. Mixed fluid yielded the most favourable result due to its higher thermal efficiency performance. The performance of butane and propane could be tied back to the condensation temperature for each where butane has the lower condensation temperature compared to Propane with 30^{o}F and 44^{o}F respectively. The general rule of thumb was that the fluid chosen must have sufficient condensation temperature to heat the LNG to the required pipeline specification.

CHAPTER 3

METHODOLOGY

3.1 RESEARCH PROCESS MODEL

The flow of this project shall be assisted by proper step-wise research methodology for effective execution and management. The point being is for the project to be completed according to time schedule allocated, and within the feasibility region. Opportunity to recover LNG cold energy available is evaluated and analysed based on open literature reviews and validated via simulation works. Operating data and parameters necessary for simulation are gathered from the aforementioned sources. These include operating parameters for LNG regasification, expander, working fluid, compressors and other system units that are available prior to simulation works. The basis for operating parameters gathered above shall include stream flow, temperature, pressure and composition. The simulation shall be performed via Aspen Hysys modelling software by inputting the gathered parameters.

Thermodynamic analysis shall be carried out afterwards to examine the efficiency and the exergy efficiency of the system. Operating parameters are then to be manipulated to study their effects on the efficiency of the system. Selection of working fluids shall be mainly based on their critical temperature and pressure. Furthermore, selection of working fluid shall be varying in order to investigate their effects on system efficiency and their economics based on the flow needed to achieve the targeted power output.

Based on the results of simulation, the optimum parameters such as the minimum temperature approach of the hot stream and cold stream in the heat exchangers shall also be investigated. In addition, the optimal pressure ratio of the compressors, pump and expanders shall be determined to obtain the best overall results. Finally, the results obtained shall be compiled and presented in the form of tables and figures to best displayed the overall system's performance. For better understanding and simplicity, research process model of this project is shown in Figure 10.

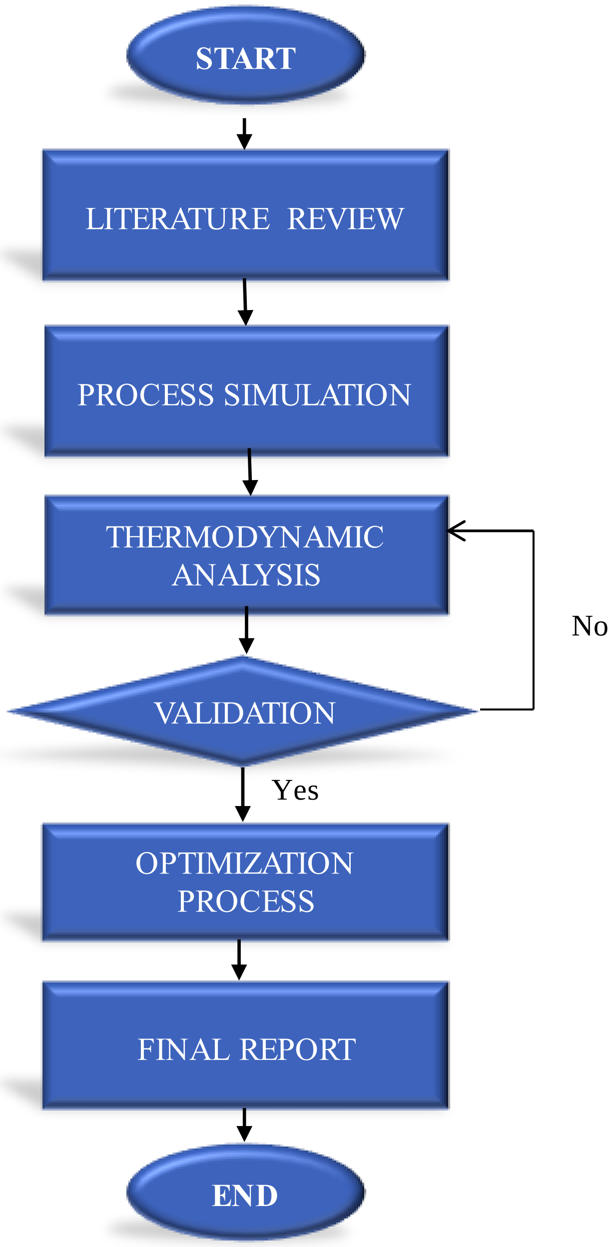


Figure 11 : Step-wise research methodology.

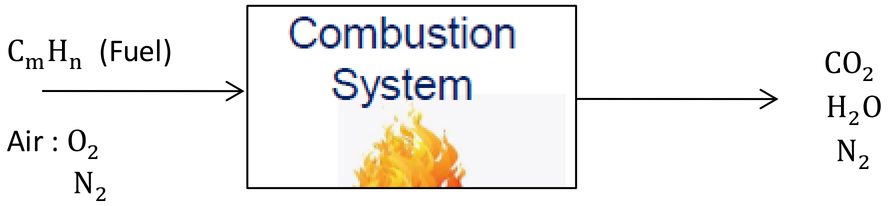
3.2 MATHEMATICAL MODEL FOR PERFORMANCE ANALYSIS

The following reveals basic thermodynamic equations associated with this particular project. These include the combustion chemistry as occurred in the combustor, Theoretical Flame Temperature (TFT) calculation as well as the energy and exergy efficiency of the overall system.

3.2.1 Combustion Chemistry

The combustion reaction occurs in the combustor is simply following the below stoichiometric:

$$C_mH_n + \left(\frac{4m+n}{4}\right) O_2 \rightarrow mCO_2 + \frac{n}{2} H_2O \qquad (1)$$



Excess Air - Products $(CO_2, \ H_2O, N_2)$

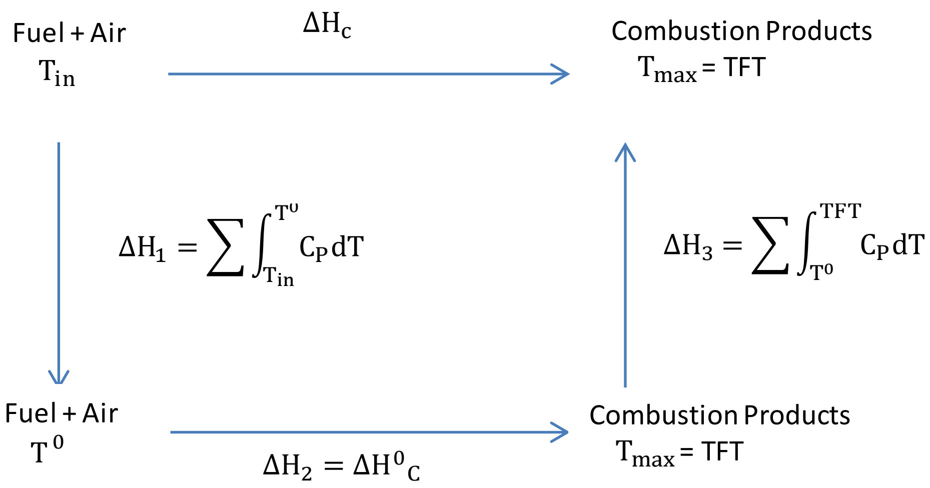
Incomplete combustion – Products $(C_mH_n, C, CO_2, CO, \ H_2O, N_2)$

Whereby air-to-fuel ratio is defined as:

$$AFR = m_{air}/m_{fuel} \qquad (2)$$

3.2.2 Theoretical Flame Temperature (TFT)

Hess's Law is used to calculate the TFT for the combustion reaction. The reaction occurs is assumed to be adiabatic, $\Delta H_c = 0$. Hess's Law states that enthalpy is a state property and the change of enthalpy is independent of path.



$$\Delta H_c = \Delta H_1 + \Delta H_2 + \Delta H_3 = 0 \tag{3}$$

3.2.3 Energy Balance Equation and Thermal Efficiency

A few assumptions is made prior to simulation which include the simulation is carried out in steady-state environments and there is no pressure drop in each system unit and pipeline. Thermodynamic equations for each unit are per following:

For Expanders / Turbines:

$$W_j = m_{j,inlet}\left(h_{j,outlet} - h_{j,inlet}\right) = \eta_j m_{j,inlet}\left(h_{j,outlet} - h'_{j,inlet}\right) \tag{4}$$

$$\text{where Work} = \text{kW} \qquad j = \text{EX1, EP2}$$

For Compressor and Pumps:

$$W_i = m_{i,inlet}\left(h_{i,outlet} - h_{i,inlet}\right) = m_{i,inlet}\left(h_{i,outlet} - h_{i,inlet}\right)/\eta_i \tag{5}$$

$$\mathrm{Work} = \mathrm{kW} \qquad i = C1, C2, P1, P2$$

For Heat Exchangers:

$$Q_k = m_{k,inlet} C_p (T_2 - T_1) = m_{k,inlet}\left(h_{k,outlet} - h_{k,inlet}\right) \tag{6}$$

$$\text{where } Q = \mathrm{kW} \quad k = HX1, HX2, HX3$$

For Combustor:

$$Q_{in} = m_{fuel} HHV_{fuel} \tag{7}$$

$$LHV_{fuel} = HHV_{fuel} - \left(\frac{m_{H_2O}}{m_{fuel}}\right) \Delta H_{vap,H_2O} \tag{8}$$

where LHV = Lower Heating Value (kJ/kg)
HHV = Higher Heating Value (kJ/kg) $= -\Delta H_c$

For isentropic compression and expansion:

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = \left[\frac{P_2}{P_1}\right]^{(\gamma-1)/\gamma} = \left[\frac{1}{r}\right]^{(\gamma-1)/\gamma} \tag{9}$$

where $r = P_2/P_1$, pressure ratio
$\gamma = C_p/C_v$

Thus, the ideal gas turbine cycle:

$$\eta_T = 1 - \left[\frac{1}{r}\right]^{(\gamma-1)/\gamma} \tag{10}$$

The thermal efficiency of the gas turbine power plant can be estimated using the following equation.

$$\eta_{TH} = \frac{(\Sigma W_{EX1} - \Sigma W_{C1})}{Q_{in}} \tag{11}$$

On the other hand, the thermal efficiency of the power cycle system is:

$$\eta_{TH} = \frac{(\Sigma W_j - \Sigma W_i)}{Q_{in} + \Sigma Q_k} \tag{12}$$

3.2.4 Exergy Balance Equation and Exergy Efficiency

From thermodynamic point of view, exergy also called as availability or work potential is the maximum useful work that can be obtained from a system at a given state in a given environment. Unlike energy which follows the First Law of thermodynamic, energy cannot be created nor destroyed, exergy accounts for the irreversibility of the process due to increase in entropy. Second Law of thermodynamic states that entropy of a system never decreases. Exergy is always destroyed when involving a temperature change in the process and this destruction is proportional to the entropy increase of the system. Thermodynamic performance of a process better evaluated using exergy to provide more useful improvement in efficiency efforts compared to energy analysis.

Exergy can be defined as:

$$e = (h - h_o) - T_0 (s - s_0) \tag{13}$$

Where T_0 is taken at it reference condition $= 298.15$ K.

The overall exergy balance equation is:

$$E_{in} = E_{out} + E_{loss} \tag{14}$$

The input exergy of the system is given as:

$$E_{in} = m_{S3} HHV + m_{S0} e_{S0} \tag{15}$$

The output exergy of the system is given as:

$$E_{out} = \sum E_j - \sum E_i \tag{16}$$

Where $E_j = m_{j,inlet}(e_{j,inlet} - e_{j,outlet}) \quad j = EX1, EX2$ (17)

$E_i = m_{i,inlet}(e_{i,outlet} - e_{i,inlet}) \quad i = C1, P1, P2$ (18)

Notation j represents expanders/turbines whilst i represents compressors and pumps respectively. Finally, the exergy efficiency of the power cycle can be written as:

$$\eta_E = E_{out} / E_{in} \tag{19}$$

3.3 SIMULATION WORKS

This section explains a brief procedure to successfully verify the case study by performing simulation works. The simulation is performed via Aspen Hysys version 7.3 in steady - state environment. There are two basic case studies to be simulated with optimizing efforts shall be carried out after the completion of second basic case study. The first case study is combined LNG regasification and Gas Turbine power plant whilst the second one is the combination of the aforementioned with Rankine cycle.

3.3.1 Selection of Property Package and Working Fluid

The thermodynamic fluid package selected for the simulation is Peng-Robinson EOS. Peng-Robinson is chosen because of its compatibility with the components used in the simulation. As initial start-up, ammonia is chosen as the working fluid for the Rankine cycle as per analysed from most of the literature reviews initially. This is because ammonia can satisfy many properties for Rankine cycle to work effectively. Among the selection criteria of the working fluid is tabulated in Table 1.

Table 1 : Selection criteria for working fluid.

Selection Criteria	
1. Critical Temperature	5. Toxicity
2. Critical Pressure	6. Flammability
3. Density	7. Global warming potential
4. Latent Heat	8. Ozone depletion potential

The most important criteria perhaps are the critical temperature and the latent heat of the working fluid. As working fluid reaches its critical temperature, the properties of its gas and liquid phase converge, thus resulting in only one phase at this critical point. Above the critical temperature, the working fluid cannot exist in liquid form by an increase in pressure. In addition, as the latent heat of the working fluid increases, the flow rate required to condense or vaporize the fluid is decreasing. This is due to an increase of heat duty for same amount of temperature change as proven by the following equation:

$$\text{Heat Duty } Q = \lambda \Delta T \tag{20}$$

$$\begin{aligned} \text{where } Q &= \text{heat duty, kW} \\ \lambda &= \text{latent heat of vaporization/condensation} \\ \Delta T &= \text{temperature change, °C} \end{aligned}$$

3.3.2 Aspen Hysys Simulation

After necessary process scheme have been established, the overall system feasibility is verified via Aspen Hysys modelling software. Aspen Hysys is very reliable software for engineers to model and predict the actual process behaviour prior to real-life implementation. However, precautions must also be taken into account due to the inability of the software to model the exact gas turbine application. Thus certain modifications must be made to account for such uncertainty. PFD for simulation using Aspen Hysys is attached in appendix section. Some of the operating parameters for the equipment used in the simulation are tabulated in Table 2.

Table 2: Initial operating parameters.

Cycle	Operating Parameter	Value
LNG Regasification	LNG T_{inlet} before LNG Exchanger HX1	-165°C
	LNG mass flow	25.9 kg/s
	Adiabatic efficiency of Pump P1	99%
Gas Turbine System	Polytropic efficiency of Compressor C1	89%
	Polytropic efficiency of Expander EX1	89%
	Pressure ratio of Compressor C1	24
	Pressure ratio of Expander EX1	24
	Mass flow of fuel	13.1 kg/s
	Mass flow of air	263.2 kg/s
Rankine cycle of Ammonia fluid	Polytropic efficiency of Expander EX2	89%
	Pressure ratio of Expander EX2	4.4
	Adiabatic efficiency of Pump P2	90%
	Mass flow of Ammonia	6.6 kg/s

The above parameters are used as the initial input to kick-start the simulation in order to yield targeted power output of 404 MW from expander EX1. Next, process optimization shall be carried out in order to get the best overall system efficiency. Other parameters like the use of other working fluids for Rankine cycle shall be taken into consideration, as well as pressure ratio for pumps, compressor, and expanders.

3.4 PROJECT ACTIVITIES

To ensure smooth project flow, activities planned for this project are segregated into three stages which are early, middle and final project stage. For early project stage, the main activities shall be on the selection of suitable project title and preliminary research work. Topic selected must be within the scope of study and related to LNG cold energy utilization. After topic selection, literature reviews is next and perhaps one of the most important parts to provide strong overview on scope of projects to be undertaken. For middle project stage, the focus is now shifting to process simulation, thermodynamic analysis and interim report's preparation. Simulation works are to be carried out based on the developed PFD and good thermodynamic analysis shall provide good results at the end of project's period. At the end of FYP 1, author is required to submit an interim report to supervisor for marking. Last but not least is the final project stage whereby the simulation works are validated to produce the intended results. After that, author can move on to prepare for project dissertation as the final step as per discussed in research process model in section 3.1. In summary, the activities that shall be carried out throughout this project are summarized in Table 3.

Table 3: Project activities.

No	Project Activities
1	Early Project Stage ▪ Project topic selection ▪ Preliminary research work
2	Middle Project Stage ▪ Process Simulation ▪ Thermodynamic Analysis ▪ Preparation of interim report
3	Final Project Stage ▪ Validation Stage ▪ Results analysis and optimization process ▪ Preparation of draft report ▪ Preparation of project dissertation

3.5 KEY MILESTONE

Apart from project activities involved, another important aspect to look at is the key milestone. Key milestone can best be described as an event that receives special attention. It usually marks the completion of work package or phase or in this case project stage. For early project stage, key milestones are submission of extended proposal and proposal defence. In addition to submitting project's proposal, author is required to defend previously submitted proposal before been approved to proceed with the chosen project title. For middle project stage, interim report and progress report shall be the milestones. For final project stage, author shall go through oral presentation where there will be an internal and external examiner to evaluate the project's overall performance followed by submission of project dissertation and technical report. Key milestones for this project are summarized in Table 4.

Table 4: Key milestones.

Key Milestone	**Week**
Early Project Stage ▪ Extended proposal submission ▪ Proposal defence	 7 9
Middle Project Stage ▪ Submission of interim report ▪ Submission of progress report	 14 22
Final Project Stage ▪ Submission of technical report ▪ Oral presentation ▪ Submission of project dissertation	 28 30 30

3.6 GANTT CHART

A project schedule has been developed which best summarizes all activities and key milestones throughout this project period and shown in Figure 11.

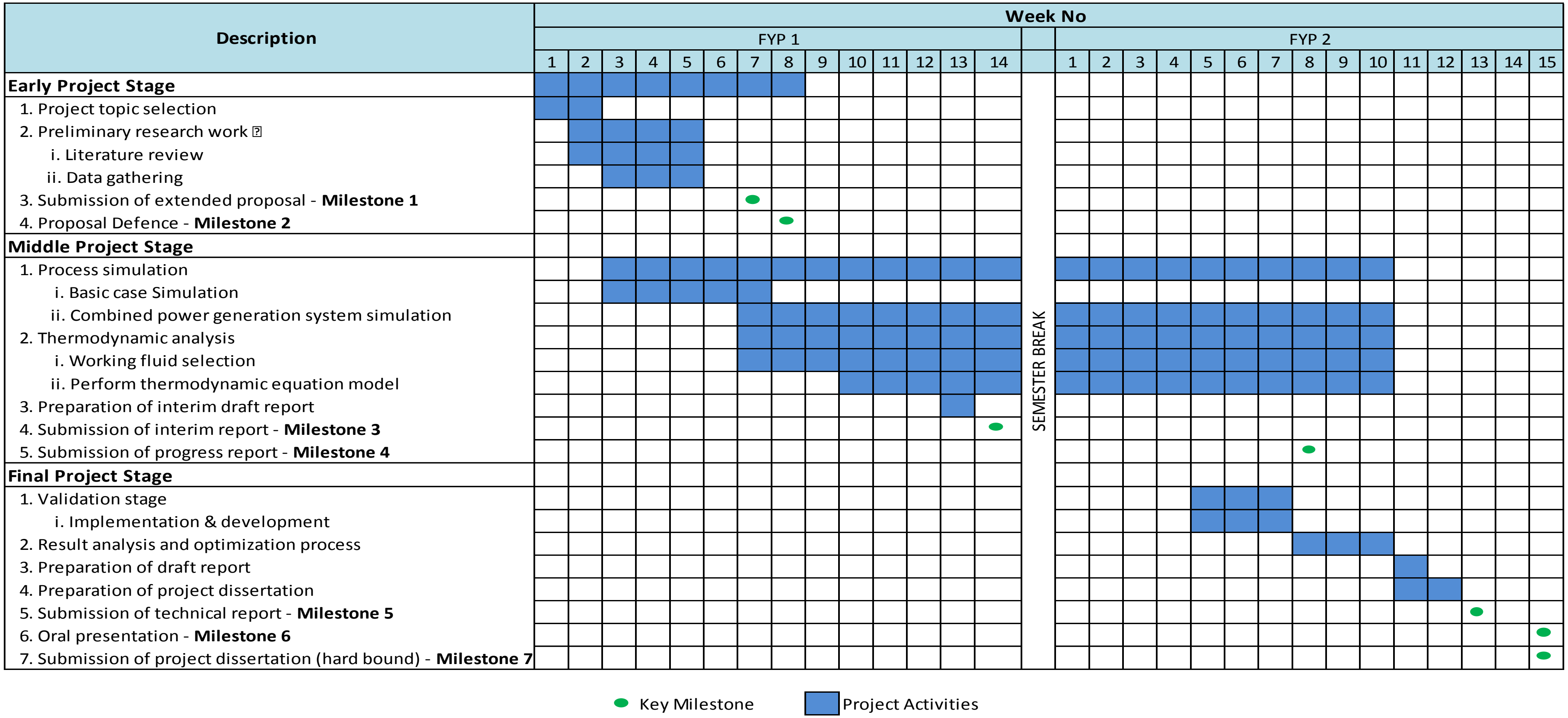


Figure 12 : Gantt chart

3.7 SOFTWARE AND TOOLS

Among the software based tools that are used to complete this project include Microsoft Office, AutoCad and Aspen Hysys. Microsoft Office includes Excel, Word, Power Point and Visio. Microsoft Office is a compulsory software as it is the most well-known office computing software whereby it provides the means of report writing, slides presentation, large scale calculation using excel as well as PFD construction through Visio. AutoCad on the other hand is the software to design Two-Dimensional (2D), Three-Dimensional (3D) designs, speed documentation and a very useful tool to use for constructing shapes and PFD for the proposed system design. Aspen Hysys is a comprehensive process modelling software developed by AspenTech. It is intended to optimize process designs and operations by performing simulation works first hand prior to real life implementation. For this project, it is safe to note that Aspen Hysys gives the highest contribution due to the nature of the project itself which is simulation based. Tools that are used to complete this project are listed below.

1. Aspen Hysys
2. Microsoft Excel
3. Microsoft Power Point
4. Microsoft Word
5. Microsoft Visio
6. AutoCad

CHAPTER 4

RESULTS AND DISCUSSION

4.1 PROCESS SCHEMES

Process scheme can be divided into two case studies as mentioned earlier. The schematic diagram is segregated into three parts which are simple LNG regasification (S0-P1-S1-HX1-S2), gas power plant (S3-FH-S7-C1-S4-S5-EX1-S6) and Rankine cycle (S9-HX2-S10-EX2-S11-HX1-S12-P2). Case study 1 applies the integration of LNG regasification with the gas power plant generation whilst case study 2 applies combination of those three systems. PFD for case study 1 and 2 are shown in Figure 12 and Figure 13 respectively.

4.1.1 Case Study 1: Integration of LNG Regasification with gas power plant

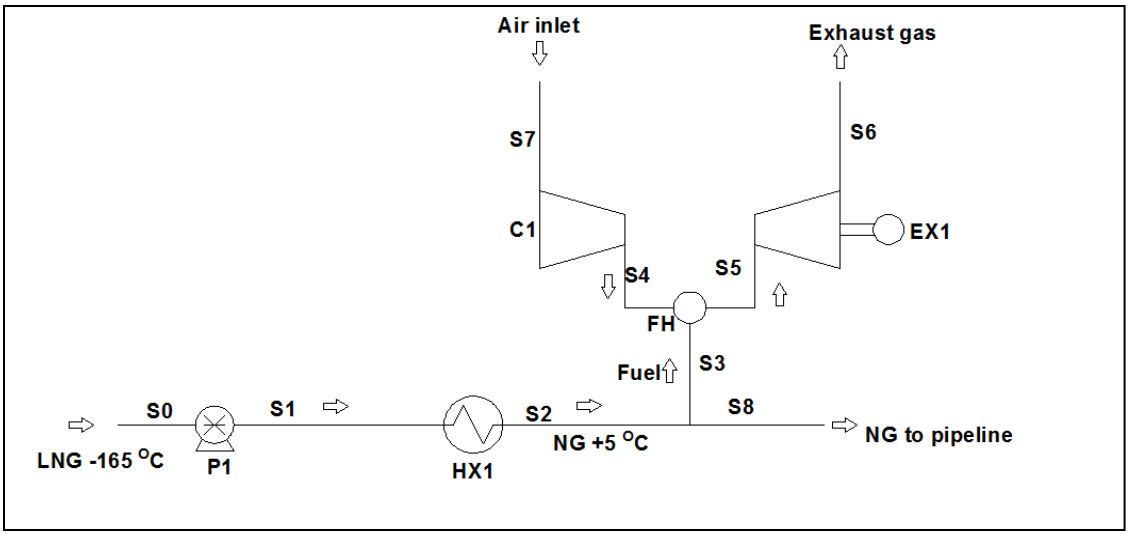


Figure 13 : Integration of LNG regasification and gas power plant.

The LNG from the ship unloading (S0) is first pressurized (S1) before is sent to LNG exchanger HX1 to be vaporized into gaseous state, NG. Then, some portion of NG is fed into the combustor together with pressurized air (S4) to allow for combustion to take place and the rest (S8) is sent to pipeline system for distribution to customers. The exhaust from the combustor FH (S5) is then expanded in the expander EX1 to produce work needed. The air is fed in 20% in excess into the combustor. The amount of air and fuel required for the whole cycle is dependent on the amount of power generation targeted which is 404 MW in capacity.

4.1.2 Case Study 2: Proposed LNG Cold Energy Utilization Schematic Diagram

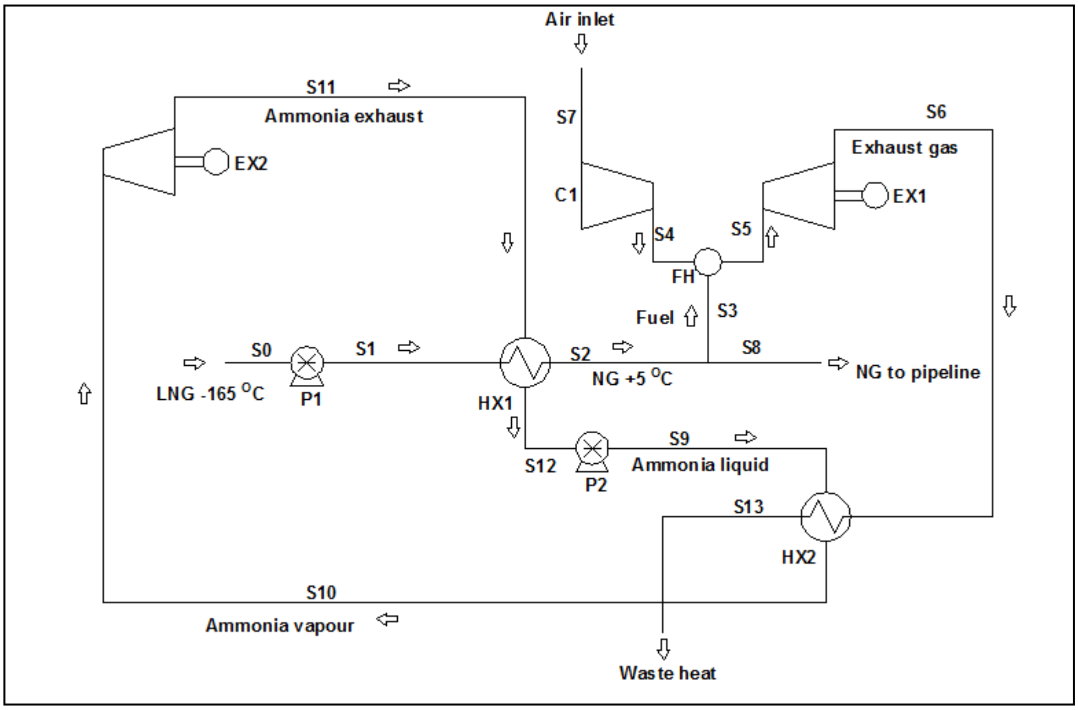


Figure 14 : Integration of LNG regasification plant with gas power plant and Rankine cycle.

For this case study, Rankine cycle is integrated into the previous system by implementing pressurized ammonia (S9) as the working fluid to exchange heat with the exhaust gas from the expander EX1. The ammonia is then vaporized in the vaporizer HX2 before fed into expander EX2 to be expanded to generate power output. The ammonia outlet of vaporizer EX2 is then fed into the LNG exchanger HX1 to exchange heat with LNG thus vaporizing the LNG and the same time condensing back to liquid form. The ammonia cycle is completed as it is once again pressurized before being fed back to exchange heat with the exhaust gas of expander EX1. In addition, the temperature approach between the ammonia liquid outlet and the LNG inlet at LNG exchanger HX1 is kept at minimum of $10^{o}C$.

4.2 COMBUSTION CHEMISTRY

The combustion of NG take place in the combustor FH before exhaust gas produced is expanded in the expander to generate power output. In order to allow the combustion to take place at constant pressure, LNG from the ship is pressurized to a pressure of 24 bar whilst air is compressed to the same amount of pressure. The reason why the LNG is pressurized before being vaporized into NG is to avoid exceeding the flash point of the NG. If, NG is compressed after the regasification, there is a possibility of exceeding the flash point according to ideal gas law which is $PV=nRT$, clearly the temperature of the NG will also increase. Hence, pressurizing the LNG prior to regasification is the better choice to avoid any unnecessary complication.

For complete combustion:

Table 5 : Combustion stoichiometry.

				No of Moles Reactants				No of Moles Products			
NG	**mol fraction**	**m**	**n**	**HC**	**O_2**	**N_2**	**$C0_2$**	**$C0_2$**	**$H2_0$**	**O_2**	**N_2**
Methane	0.9122	1	4	0.9122	1.8244	6.8806	0.0000	0.9122	1.8244	0.0000	6.8806
Ethane	0.0496	2	6	0.0496	0.1736	0.6547	0.0000	0.0992	0.1488	0.0000	0.6547
Propane	0.0148	3	8	0.0148	0.0740	0.2791	0.0000	0.0444	0.0592	0.0000	0.2791
i-butane	0.0026	4	10	0.0026	0.0169	0.0637	0.0000	0.0104	0.0130	0.0000	0.0637
n-Butane	0.0020	4	10	0.0020	0.0130	0.0490	0.0000	0.0080	0.0100	0.0000	0.0490
i-Pentane	0.0010	5	12	0.0010	0.0080	0.0302	0.0000	0.0050	0.0060	0.0000	0.0302
n-Pentane	0.0006	5	12	0.0006	0.0048	0.0181	0.0000	0.0030	0.0036	0.0000	0.0181
Hexane	0.0003	6	14	0.0003	0.0029	0.0107	0.0000	0.0018	0.0021	0.0000	0.0107
CO_2	0.0020	0	0	0.0000	0.0000	0.0000	0.0020	0.0020	0.0000	0.0000	0.0000
N_2	0.0149	0	0	0.0000	0.0000	0.0149	0.0149	0.0000	0.0000	0.0000	0.0149
				0.9831	**2.1176**	**8.0011**	**0.0169**	**1.0860**	**2.0671**	**0.0000**	**8.0011**

For 1 mole of fuel as shown in Table 5, the overall stoichiometric of the reaction is calculated as follows:

$$0.9122CH_4 + 0.0496C_2H_6 + 0.0148C_3H_8 + 0.0046C_4H_{10} + 0.0016C_5H_{12} + 0.0003C_6H_{14} + 2.1176O_2 + 8.0011N_2 + 0.0169CO_2 \longrightarrow 1.0860CO_2 + 2.0671H_2O + 8.0011N_2 \quad (21)$$

For this project, an excess of 20% air is supplied for the reaction. Thus the overall stoichiometric balance with 20% excess is shown in Table 6:

Table 6 : Combustion stoichiometry with excess air.

				No of Moles Reactants				No of Moles Products			
NG	**mol fraction**	**m**	**n**	**HC**	**O2**	**N2**	**C02**	**C02**	**H20**	**O2**	**N**
Methane	0.9122	1	4	0.9122	2.1893	8.2567	0.0000	0.9122	1.8244	0.3649	8.2567
Ethane	0.0496	2	6	0.0496	0.2083	0.7857	0.0000	0.0992	0.1488	0.0347	0.7857
Propane	0.0148	3	8	0.0148	0.0888	0.3349	0.0000	0.0444	0.0592	0.0148	0.3349
i-butane	0.0026	4	10	0.0026	0.0203	0.0765	0.0000	0.0104	0.0130	0.0034	0.0765
n-Butane	0.0020	4	10	0.0020	0.0156	0.0588	0.0000	0.0080	0.0100	0.0026	0.0588
i-Pentane	0.0010	5	12	0.0010	0.0096	0.0362	0.0000	0.0050	0.0060	0.0016	0.0362
n-Pentane	0.0006	5	12	0.0006	0.0058	0.0217	0.0000	0.0030	0.0036	0.0010	0.0217
Hexane	0.0003	6	14	0.0003	0.0034	0.0129	0.0000	0.0018	0.0021	0.0006	0.0129
CO_2	0.0020	0	0	0.0000	0.0000	0.0000	0.0020	0.0020	0.0000	0.0000	0.0000
N_2	0.0149	0	0	0.0000	0.0000	0.0149	0.0149	0.0000	0.0000	0.0000	0.0149
				0.9831	**2.5411**	**9.5983**	**0.0169**	**1.0860**	**2.0671**	**0.4235**	**9.5983**

With 20% air excess, the reaction stoichiometry is as follows:

$$0.9122CH_4 + 0.0496C_2H_6 + 0.0148C_3H_8 + 0.0046C_4H_{10} + 0.0016C_5H_{12} + 0.0003C_6H_{14} + 2.5411O_2 + 9.5983N_2 + 0.0169CO_2 \longrightarrow 1.0860CO_2 + 2.0671H_2O + 0.4235O_2 + 9.5983N_2 \quad (22)$$

With overall stoichiometric of the combustion reaction known, the air-to-fuel ratio can then be calculated as given by Equation (2).

AFR:

$$= \frac{2.5411(32) + 9.5983(28)}{0.9122(16) + 0.0496(30) + 0.0148(44) + 0.0046(58) + 0.0016(72) + 0.0003(86)}$$

$$= 20.4 \text{ kg/kg}$$

From above calculation, to have a complete combustion with 20% excess oxygen, 20.4 kg of air is needed for each kg of NG.

4.3 THEORETICAL FLAME TEMPERATURE (TFT)

To estimate TFT for the combustion, Hess' Law is taken as the reference for calculation. The TFT estimated is for the initial inlet temperature and pressure of the air and fuel inlet to combustor. For different values of operating pressure and temperature, the TFT has to be recalculated due to different specific heat values for the components involved in the combustion reaction. The inlet conditions of air and fuel to combustor is given by Table 7 .

Table 7: Inlet operating pressure and temperature.

Operating Parameter	Air	Fuel
Temperature (°C)	486.1	5
Pressure (bar)	24	24

Assume adiabatic condition, $\Delta H_c = 0$

$$\Delta H_1 = \sum n_{i,fuel} C_{p,i,fuel} \int_{T_{in}}^{T^o} dT + \sum n_{i,air} C_{p,i,air} \int_{T_{in}}^{T^o} dT \quad (23)$$

$$\Delta H_1 = 41.63(25 - 5) + 388.06(25 - 486.1) = -178099.71 kJ$$

$$\Delta H_1 = \Delta H^o{}_c = -851958 kJ$$

$$\text{Assume TFT} = 2100\,^\circ C; \quad \Delta H_3 = \sum n_i C_{p,i} \int_{T^o}^{T^{TFT}} dT = 486.03(TFT - 25) \quad (24)$$

$$\Delta H_c = -178099.71 - 851958 + 486.03(TFT - 25) = 0$$

$$486.03(TFT - 25) = 1030057.71$$

$$TFT = 2144\,^\circ C$$

$$\%\text{Deviation} = \frac{2144 - 2100}{2144} * 100\% = 2.1\% \quad (25)$$

The deviation is below 5%, thus the estimated TFT=2144^{o}C can be accepted. The TFT calculated is also estimated against process simulation. The percentage deviation with simulation results yield:

$$\%\text{ Deviation} = \frac{2144 - 2054}{2144} * 100\% = 4.2\% \quad (26)$$

There is 4.2% deviation with the simulation results. This deviation mainly contributed by the use of fluid package in the simulation which is Peng-Robinson EOS, that might contribute in the difference of specific heat enthalpy calculation as well as the stoichiometric result of the combustion. Therefore, the previously calculated TFT = 2144^{o}C is valid.

4.4 RESULTS OF ENERGY ANALYSIS

Based on the mentioned modules, a computer simulation using Aspen Hysys modelling software is developed. The initial input parameters are summarized in Table 2 with the heat source temperature of 750 ^{o}C, the NG supplying pressure of 24 bar, the expander EX1 inlet pressure of 24 bar, and the ammonia inlet expander pressure EX2 of 10 bar. The performance of the system for both case studies is summarized in Table 8.

Table 8: Calculation results for the proposed combined cycle.

	W_{EX1} (MW)	W_{EX2} (MW)	W_{C1} (MW)	η_{TH}	E_{in} (MW)	E_{out} (MW)	η_E
CASE STUDY 1	404	0	129.3	38.8	667.1	276	0.413
CASE STUDY 2	404	5.9	129.3	0.409	667.1	281.8	0.423

This system is able to generate power output of 404 MW and the net thermal efficiency is estimated as 38.8 % in case study 1 and increases to 40.9% in case study 2 as calculated based on Equation (12). This happens due to introduction of Rankine cycle in case study 2 which eliminates the use of seawater as the heat source for LNG regasification. In addition, extra power output is realized by an addition of expander EX2. There is an increase of 4.6% in thermal efficiency from case study 1 to case study 2. About 93.3 tonne h^{-1} of LNG can be heated up to 5 ^{o}C at the same time. The exergy efficiency of the case study 1 is 41.3% and then increases about 2% to 42.3% with the introduction of Rankine cycle in the system as calculated using Equation (19).

4.5 EFFECT OF EXPANDER EX1 INLET PRESSURE

Effect of expander EX1 inlet pressure is investigated for both case studies. For case study 1, the combustion is allowed to take place at constant pressure of 24 bar, then increases to 30 and 40 bar respectively. The outlet pressure however is set to the lowest possible values hence at 1 bar. The reason is that to have the highest expansion ratio regardless of the inlet pressure.

For case study 2, due to introduction of Rankine cycle with ammonia as the working fluid, there are a few constraints must be satisfied for the system to effectively converge. Among the constraints include maintaining the state of ammonia after vaporizer HX2 in the vapour state and condensing it back to liquid in LNG exchanger HX1. Another constraint is to avoid temperature cross between the hot and cold stream at LNG exchanger HX1. Therefore, basic guideline of minimum temperature approach in the heat exchanger is set at 10 ^{o}C minimum. Taking that as the basis, temperature of ammonia liquid outlet after LNG exchanger HX1 is fixed at -150 ^{o}C hence resulted in 15 ^{o}C difference with LNG inlet temperature. Apart from that, the inlet and outlet pressure of expander EX2 is set to 10 bar and 1 bar respectively whilst the ammonia vapour outlet of vaporizer HX2 is fixed at 750 ^{o}C. The results obtained by varying the inlet pressure of expander EX1 for both case studies are tabulated in Figure 15 and Figure 16.

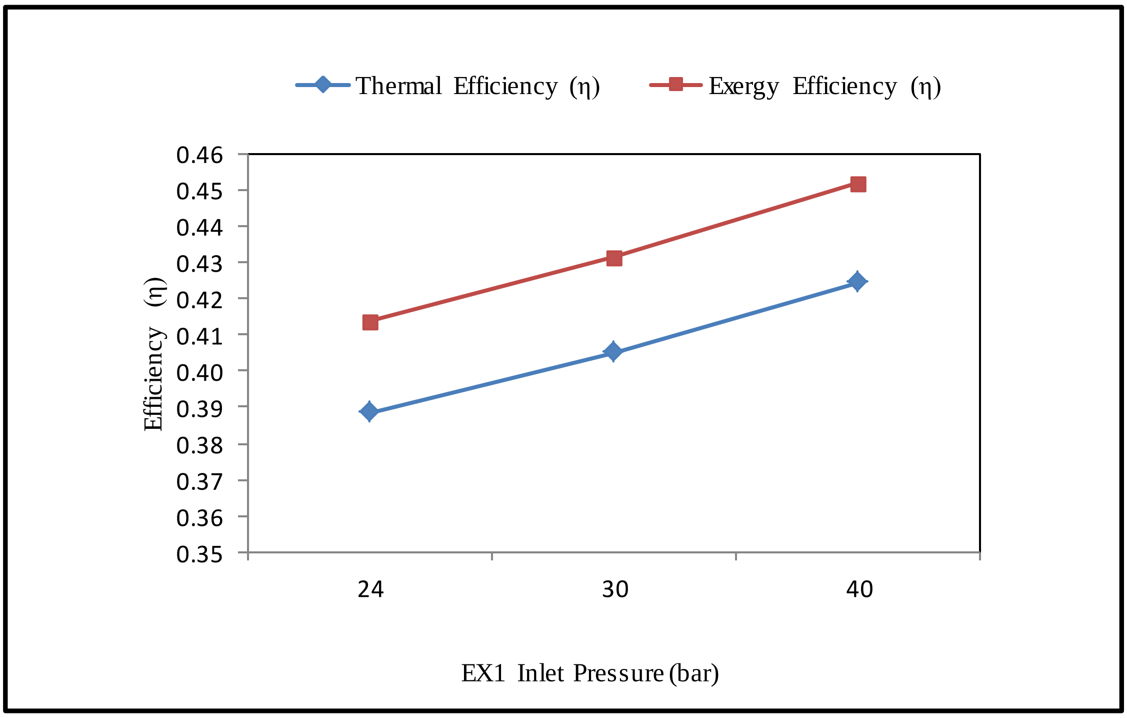


Figure 15: System efficiency by varying inlet pressure of expander EX1 for case study 1.

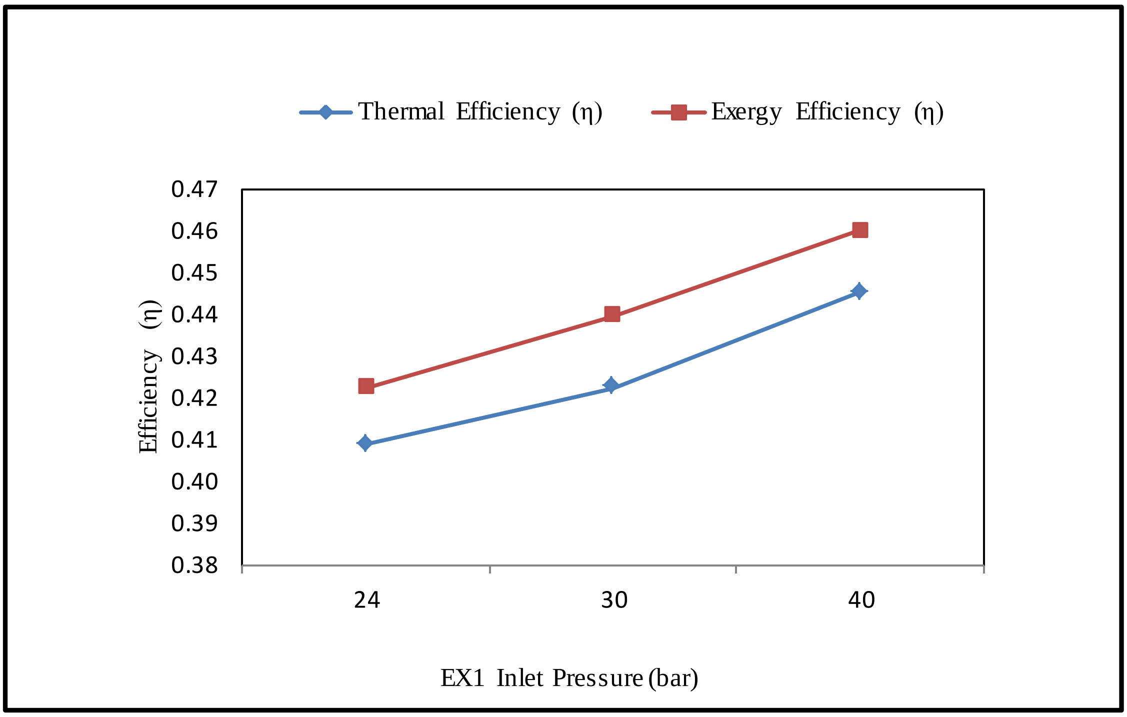


Figure 16: System efficiency by varying inlet pressure of expander EX1 for case study2.

From Figure 15 and 16, it is found that as the inlet pressure of the expander EX1 is increased, the system efficiency of the system also increases. This is due to the increase of enthalpy and entropy change between the inlet and outlet stream of expander EX1. In addition, expansion ratio of the expander also increases which directly impact the gas turbine system efficiency. Other factor to note is that the overall exergy efficiency of the system is higher for all there scenarios. This is contributed by the involvement of entropy changes in the exergy thermodynamic calculation as shown by Equation (13). Hence, from thermodynamic point of view, exergy analysis is more useful as it provides further insight in system efficiency effort compared to thermal analysis.

4.6 EFFECT OF EXPANDER EX1 OUTLET PRESSURE

Using the same specifications as discussed in section 3.3.2, the effect of outlet pressure of expander EX1 on overall net system efficiency is also investigated by varying the outlet pressure to 3, and 5 bar respectively. The results obtained are plotted in Figure 17, and 18.

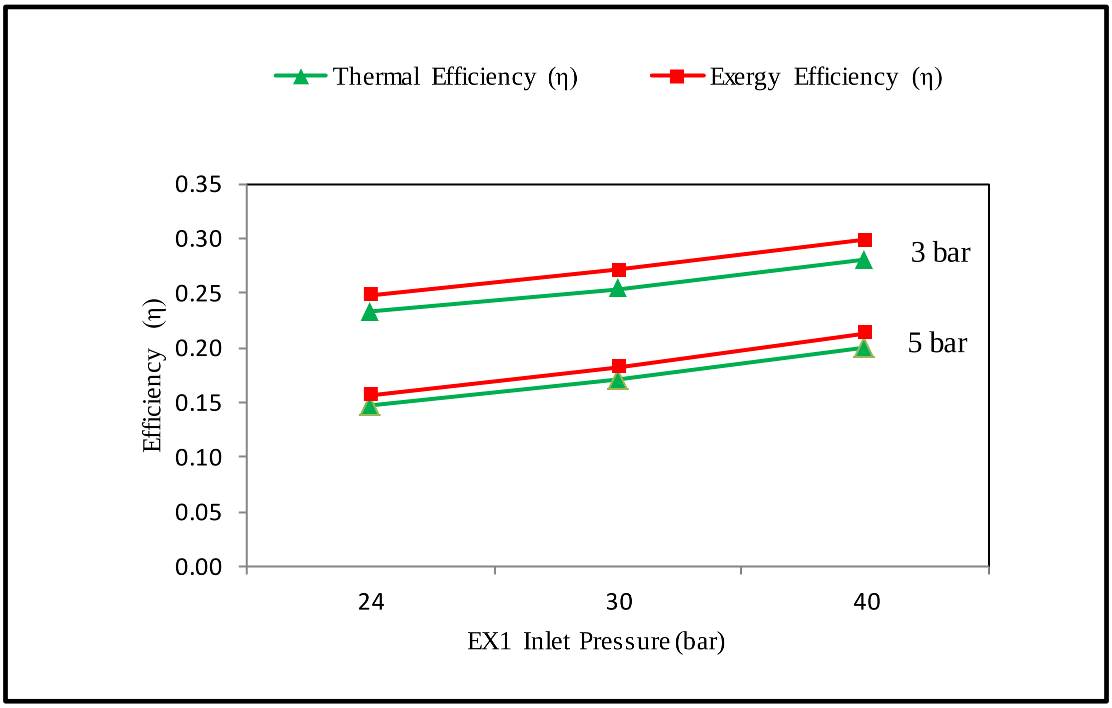


Figure 17: System efficiency by varying outlet pressure of expander EX1 for case study 1.

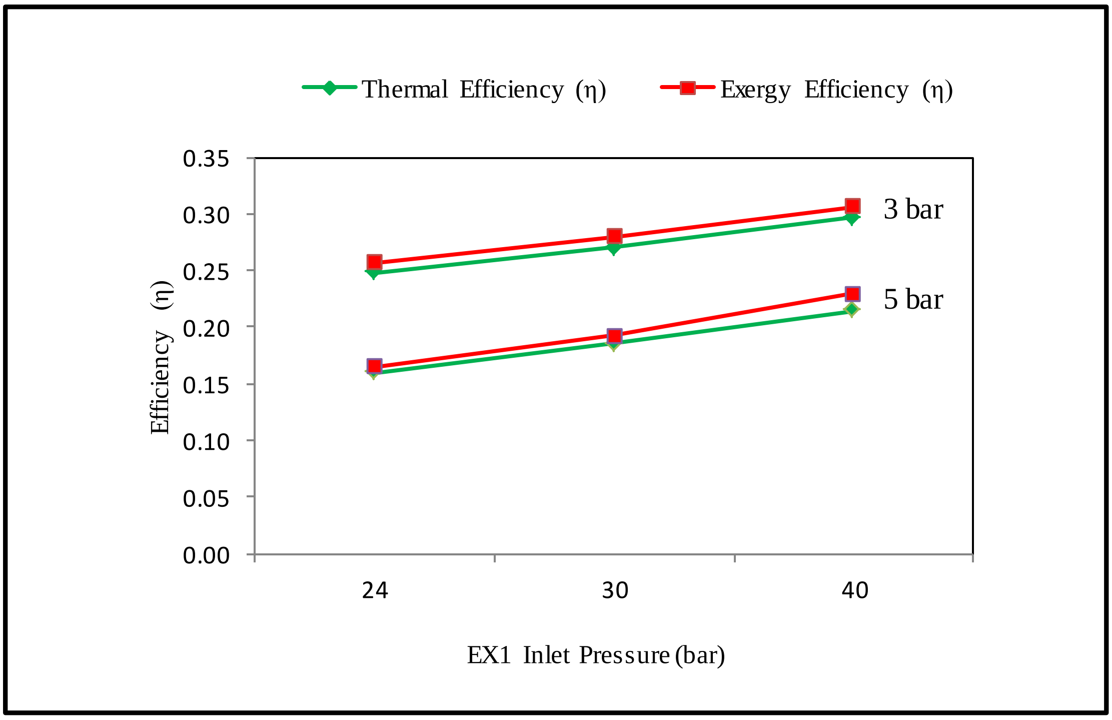


Figure 18 : System efficiency by varying outlet pressure of expander EX1 for case study 2.

It can be seen that by setting the outlet pressure of expander EX1 to 1 bar yield the highest system efficiency as shown in Figure 15 and Figure 16 compared to 3 and 5 bar. As the outlet pressure increases, the system efficiency decreases due to lower expansion ratio of expande EX1. Hence, it can be concluded that to have highest system performance available, highest expansion ratio of expander is preferable.. From thermodynamic point of view, the higher the expansion ratio, the higher is the change of enthalpy and entropy between the inlet feed and product outlet of the expander. This resulted in higher shaft work generated as shown by Equation (4) and Equation (17).

4.7 EFFECT OF WORKING FLUID ON SYSTEM EFFICIENCY

Investigation on effects of four other working fluids aside from ammonia is also conducted. The working fluids used are ammonia-water mixture with 0.5 mol fraction each, pure water, ethane and propane. The mass flow required to achieve aforementioned specifications is recorded for further analysis. The results on the effect of working fluid on system efficiency and mass flow required are shown in Figure 19. The results obtained are compared including pure ammonia which has been conducted initially and the results are already analysed.

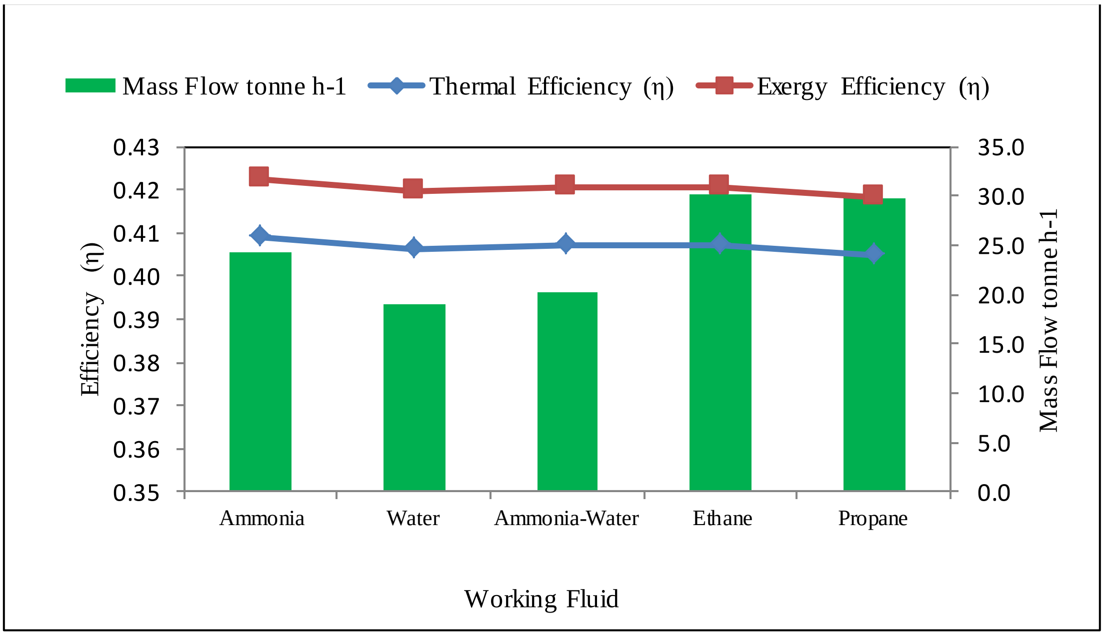


Figure 19: Effect of working fluid on system efficiency and mass flow required.

From Figure 19, ammonia yields the highest thermal and exergy efficiency followed by ammonia-water mixture, water, ethane and propane. The benchmarked results prove that the performance of typical Rankine cycle with water as working fluid can be increased by replacing it with ammonia as the working fluid. However, mass flow of water required to achieve the operating parameters at LNG exchanger HX1 and vaporizer HX2 is less than the ammonia by an amount of 5.2 tonnes h^{-1}. It is worth to note that even though water yields lower system efficiency, it can contribute to more cost saving in terms of lower initial purchase and operating cost. However, given the priority of this project which is to focus mainly on system performance and not costing, thus the cost benefit ratio is neglected.

CHAPTER 5

CONCLUSION AND FUTURE WORK

5.1 CONCLUSION

The basic case study of integration of LNG regasification, gas power plant and Rankine cycle is successfully developed and verified via Aspen Hysys. The targeted power production capacity of 404 MW is realized and efficiency of the system is increased by integrating Rankine cycle with ammonia as working fluid compared to typical Rankine cycle utilizing pure water. There is an improvement of 5.1 % and 2.4 % over thermal and exergy efficiency from case study 1 to case study 2. Setting outlet pressure of expander EX1 to 1 bar regardless of inlet pressure yields the highest system's efficiency due to highest expansion ratio available which directly affecting the turbine's efficiency. Effects of other working fluids which are pure water, ammonia-water mixture, ethane and propane on overall system's efficiency are also analysed. Pure ammonia gives the highest system's efficiency and propane produces the least favourable results. However, there is a trade-off between mass flows of ammonia and water whereby even though water yields lower system efficiency but required less flow rate of about 5.2 tonnes h^{-1} compared to ammonia. To conclude, objectives of this project are well achieved and overall project progress is within the project schedule as shown in Figure 12 with the exception of process optimization. Due to unforeseen circumstances, process optimization is skipped and recommended for future works.

5.2 FUTURE WORK

The author is to proceed with the next step as per discussed in research methodology. The main focus should be on optimizing the key operating parameters and arrangement of system units for better overall system's efficiency. Other constraints that might affect overall system's efficiency shall be identified and properly overcome. Apart from that, other feasible changes in the process flow diagram shall also be considered and keeping in track with the drafted gantt chart for smooth project flow.

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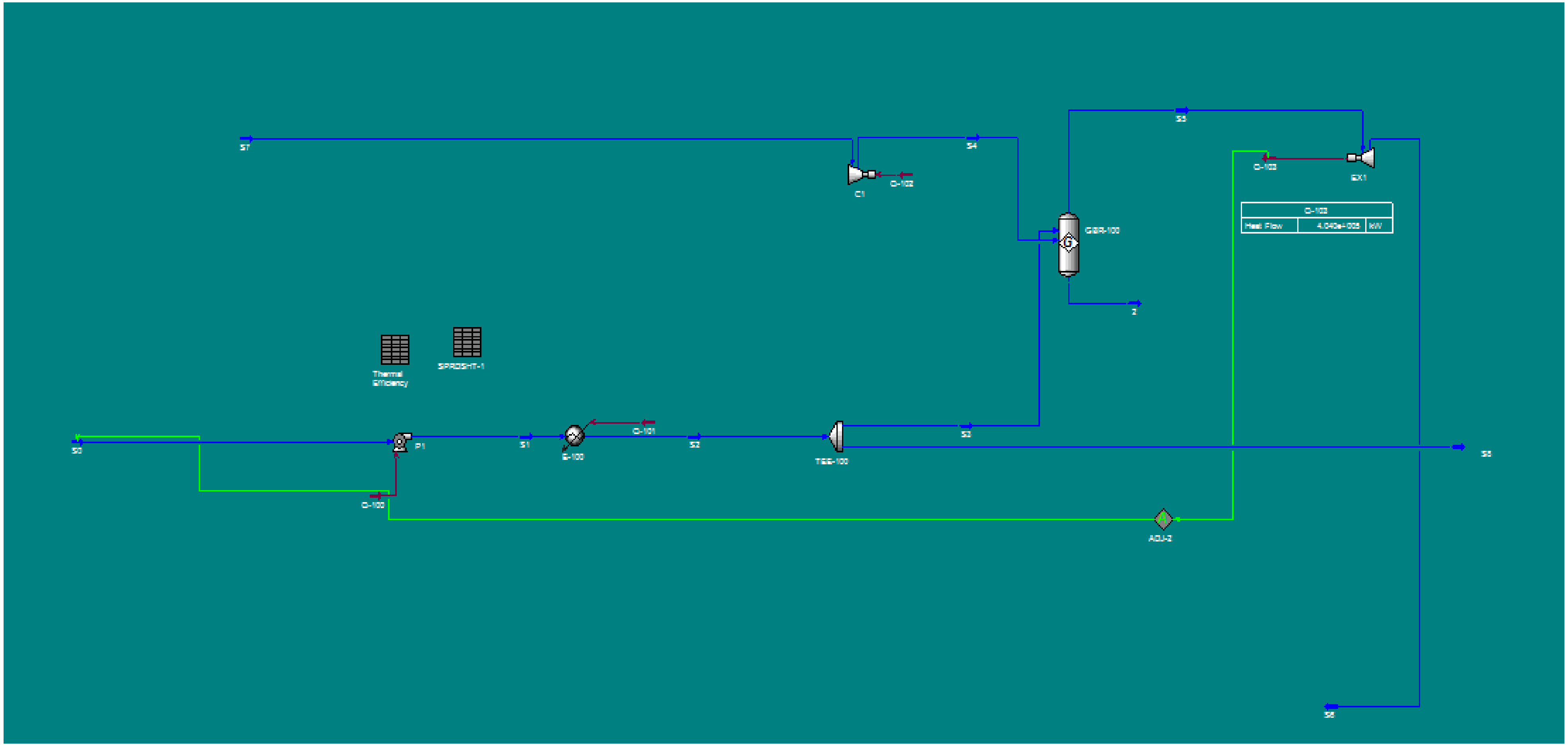
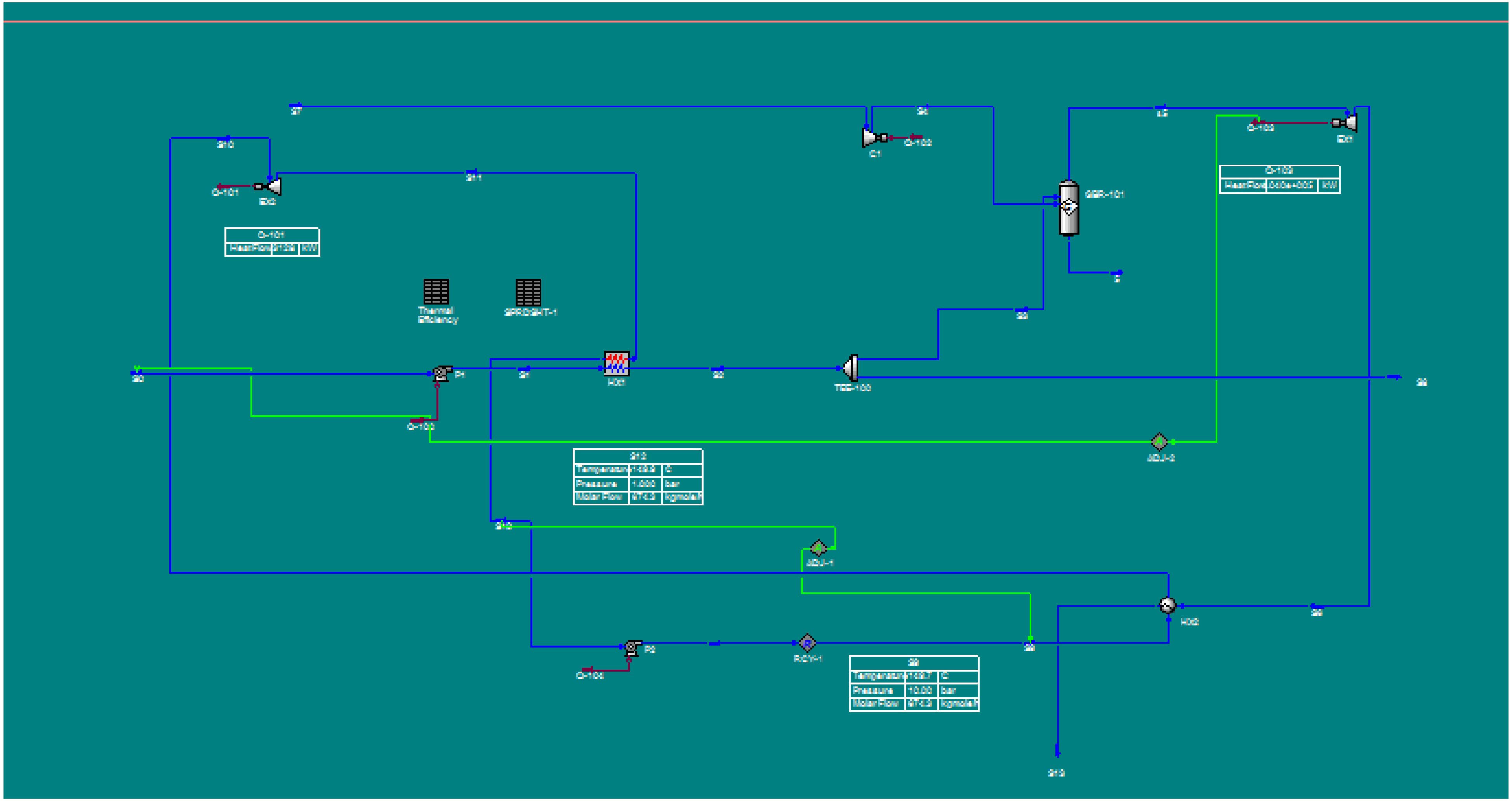
APPENDICES

Appendix I : Case Study 1 Hysys Process Flow Diagram



Appendix II : Case Study 2 Hysys Process Flow Diagram



Appendix III : Hysys Workbook Data

Name	S1	S0	S7	S4	S2	S3	S8	S13	S10	S9	S11	15
Vapour Fraction	0.0000	0.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	0.0000	1.0000	0.0000
Temperature [C]	-165.0	-165.8	25.00	486.1	5.000	5.000	5.000	973.3	750.0	-149.7	653.2	-149.7
Pressure [bar]	24.00	1.000	1.000	24.00	24.00	24.00	24.00	1.000	10.00	10.00	1.000	10.00
Molar Flow [kgmole/h]	5273	5273	3.298e+004	3.298e+004	5273	2636	2636	3.577e+004	674.3	674.3	674.3	674.3
Mass Flow [kg/h]	9.329e+004	9.329e+004	9.515e+005	9.515e+005	9.329e+004	4.664e+004	4.664e+004	9.982e+005	2.974e+004	2.974e+004	2.974e+004	2.974e+004
Liquid Volume Flow [m3/h]	293.6	293.6	1100	1100	293.6	146.8	146.8	1195	58.69	58.69	58.69	58.69
Heat Flow [kW]	-1.339e+005	-1.340e+005	-73.99	1.292e+005	-1.129e+005	-5.643e+004	-5.643e+004	-3.554e+005	-1408	-2.552e+004	-4536	-2.552e+004
Name	S6	S12	s5	5	** New **							
Vapour Fraction	1.0000	0.0000	1.0000	0.0000								
Temperature [C]	1038	-149.9	2041	2041								
Pressure [bar]	1.000	1.000	24.00	24.00								
Molar Flow [kgmole/h]	3.577e+004	674.3	3.577e+004	0.0000								
Mass Flow [kg/h]	9.982e+005	2.974e+004	9.982e+005	0.0000								
Liquid Volume Flow [m3/h]	1195	58.69	1195	0.0000								
Heat Flow [kW]	-3.312e+005	-2.553e+004	7.276e+004	0.0000								

Appendix IV : Thermal and Exergy Efficiency Spreadsheet

W EX1	4.040e+005 kW		mfuel	3503 kgmole/h		For turbine		inlet	outlet
W C1	2.161e+005 kW		HHV	9.367e+005 kJ/kgr		EX1	Mass Flow	1.326e+006 kg/h	
W P1	322.0 kW		Eso	h	-5171 kJ/kg		h	381.8 kJ/kg	-714.7 kJ/kg
HX1	2.700e+004 kW			e	4.202 kJ/kg-C		s	7.310 kJ/kg-C	7.385 kJ/kg-C
				ho (kj/kg)	-4282		ho (kj/kg)	-2661	
Qin	HHV	9.367e+005 kJ/kgr		so (kj/kgC)	10.54		so (kj/kgC)	5.134	
	m fuel (kg/h)	3503 kgmole/h		s0	1.240e+005 kg/h		e	2988	1890
	Qin kW	9.116e+005		Exergy	-730.9		E (kW)	4.047e+005	
			Ein (kW)	8.864e+005		EX2		inlet	outlet
							Mass Flow		
Thermal Efficiency	0.1998						h		
Exergy Efficiency	0.2131						s		
							ho (kj/kg)	-2687	
							so (kj/kgC)	10.54	
							e	<empty>	<empty>
							E (kW)	<empty>	
For Compressor		inlet	outlet	For Pump		inlet	outlet		
C1	Mass Flow	1.264e+006 kg/h		P1	mass flow /s0	1.240e+005 kg/h			
	h	-0.2799 kJ/kg	615.1 kJ/kg		h	-5171 kJ/kg	-5162 kJ/kg		
	s	5.262 kJ/kg-C	5.335 kJ/kg-C		s	4.202 kJ/kg-C	4.219 kJ/kg-C		
	ho (kj/kg)	-0.2799			ho (kj/kg)	-4282			
	so (kj/kgC)	5.262			so (kj/kgC)	10.54			
	e	2.226e-014	613.6		e	-730.9	-721.9		
	E (kW)	2.155e+005			E (kW)	307.1			

Cold Energy Utilization from LNG Regasification

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Abstract—**This technical paper deals with Liquefied Natural Gas (LNG) cold energy utilization by integrating the system with gas power plant and Rankine cycle to generate electricity. There is a waste of cold energy available during the LNG regasification due to the use of seawater and returning it back to ocean. The cold energy available is better converted into useful energy such as electricity via Rankine power cycle implementation. Another aspect to look at is on the system performance of ordinary Rankine cycle utilizing water as working fluid which is quite low. The objectives of this research is to develop an integrated system to fully utilize cold energy available via Rankine cycle in existing gas power plant and to experiment with other working fluids to be utilized in the cycle taking pure ammonia as the basis. Two case studies are developed with first being an integration of the LNG regasification process with gas power plant to yield a targeted amount of power generation which is about 404 MW. The second case study integrates the Rankine cycle into previous system to utilize the LNG cold available. Simulation work is carried out using Aspen Hysys to check for the system's feasibility. The efficiency of the overall system is analysed based on thermal and exergy efficiency respectively for both case studies. The effect of the inlet and outlet pressure of the gas turbine on overall system efficiency is investigated which resulted in highest efficiency when the expansion ratio of the gas turbine is at the highest. It is found that the second case study improves the thermal and exergy efficiency by 5.1 % and 2.4 % respectively. Five working fluids are used to study their effects on system efficiency which are ammonia, water, ammonia-water mixture, ethane and propane. As expected from various literature reviews, ammonia yields the highest system's efficiency compared to other working fluids with improvement of about 0.64 % over pure water but with penalty of higher mass flow required approximately 5.2 tonnes h^{-1} to achieve operating specification as discussed in results section later in this paper. Based on the results obtained, it is proven that the efficiency of gas power plant can be further increased by integrating with Rankine power cycle and at the same time effectively utilizing the LNG cold energy available.**

Index Terms—cold energy; LNG; NG; Rankine cycle; thermal; exergy.

I. INTRODUCTION

Liquefied Natural Gas (LNG) has been commonly used in the energy industry aside from civil life nowadays. It is projected that global energy demand is expected to increase rapidly by the year 2030 [1-2]. Hence, more and more LNG regasification plants capacity will increase to meet the growing demand. Transport of Natural Gas (NG) from a gas field by pipelines to consumers is often impossible to accomplish thus requiring the need of transporting NG to receiving terminal via ship. To accomplish this, NG is liquefied into LNG at a very low temperature of around -162 ^{o}C and at atmospheric pressure after removing acid substances and water. During liquefaction also, the volume of LNG can be reduced about 600 times than that of NG therefore enable it to be transferred in LNG tanks via ship to the receiving terminal [3]. Large amount of mechanical energy is consumed to convert NG to LNG which is about 850 kWh of electric energy to produce one tonne of LNG [4]. At the terminal, LNG needs to be evaporated into gaseous form at ambient temperature and at suitable elevated pressure prior to distribution system. This is usually done at regasification facility. LNG is a clear, non-toxic, non-corrosive, odourless, and in liquid form at atmospheric pressure. The density of LNG is around 400-500 kg m^{-3} depending on the given temperature, pressure and composition. Thus, if spilled on water with approximately 1000 kg m^{-3} density, LNG will float on top and vaporizes faster and disperses leaving no residue. Such properties contribute to no cleaning up if LNG is accidentally spilled either on water or land. A conventional regasification facility consists of subsystems including ship unloading, LNG storage, LNG low pressure pump, tank boil-off gas compression, vapour re-condensation, LNG high pressure pump and LNG regasification. Vapours generated from LNG ship unloading and during normal operation are compressed by the boil-off vapour compressor and then re-condensed by mixing with sub-cooled LNG send out. The condensed LNG is then pumped to pipeline pressure and heated to about 5 ^{o}C prior to the pipeline's distribution. In addition, latent heat of vaporization and any sensible heat required to superheat the vapour during the LNG regasification are termed 'cold energy', and this is usually supplied by seawater. Such process needs about 800 kJ kg^{-1} of energy [5]. Liu and You [6] stated that cold energy is a form of high quality energy in the thermodynamic point of view. There is an opportunity of recovering the cold energy rather than just taken off by seawater which shall contribute to environmental impact such as global warming. Past few decades have seen a lot of methods have been developed to utilize cold energy from LNG. These include power generation, air separation, and intake air cooling. Among those, power generation is the most effective one. An example of cold energy power generation system is propane organic Rankine cycle (ORC). La Rocca [7] proposed a modular LNG regasification unit based on a power cycle utilizing ethane as the working fluid. Other effective LNG cold energy utilization to produce electricity is by employing cryogenic stream of LNG during regasification as a cold source in an improved Combined Heat and Power (CHP) plant. This paper deals with the utilization of LNG cold energy from regasification process to produce electricity in a CHP plant utilizing Rankine cycle as the working cycle. The author shall study on the integration of LNG regasification plant, with a gas power plant via Rankine cycle in order to fully utilize LNG cold energy to generate electricity. A simulation work of complete set up of aforementioned integrated plant with suitable working fluid shall be performed using Aspen Hysys modelling software.

2. LITERATURE REVIEW

NG is widely available and is usually termed as 'green fuel' due to its higher energy density and environmental friendly advantages. Among the advantages of NG are it reduces greenhouse gas and produces lower emissions compared to other alternative fuels. Independent thermal cycle with NG direct expansion and closed-loop Rankine cycle is usually associated with electric power generations from LNG cold energy utilization [8]. Along the line, many other researches related to cold energy utilization to produce electricity have also been conducted. An ammonia-water Rankine cycle with refused incinerator has been proposed by Miyazaki et al. [9] and was compared with typical Rankine cycle utilizing water as working fluid. In addition, theoretical LNG cryogenic power cycle using two binary working fluids and one with ethane as the working fluid is studied by Szargut and Szcygiel [10]. Liu and You [11] investigated the characteristics and applications of the cold exergy of LNG. The authors developed a mathematical model using Soave-Redlich-Kwong (SRK) Equation of State (EOS) to predict the various effects of LNG such as the ambient temperature and compositions on its thermal properties. The results obtained reveals that pressure exergy and total cold heat exergy is increased when the ambient temperature increases.

Dispenza, La Rocca and Panno [12-13] proposed a CHP plant using Brayton cycle to produce electricity by recovering the exergy as a cold using the cryogenic stream of LNG. As shown in Figure 1, the cryogenic stream of LNG acted as the cold source whilst the working fluids used in the cycle were helium and nitrogen. Both helium and nitrogen were benchmarked in terms of overall percentage of electricity production. The proposed system improved the electric efficiency of an electric utility in power stations working with steam turbines by lowering the temperature of the condenser. One way to achieve this was by further utilizing cooled water reject by Open Rack (OR) unit. The system applied Brayton cycle with an open-loop top cycle with steam turbine whilst the bottom cycle was the closed-loop containing helium or nitrogen as the fluid. In order to further improve the thermal performance of heat transfer during regasification process, cryogenic regasifier had heat transfer matrix made with extended surface tubes. This allowed for working fluids to flow in from the shell side and LNG from the tube side.

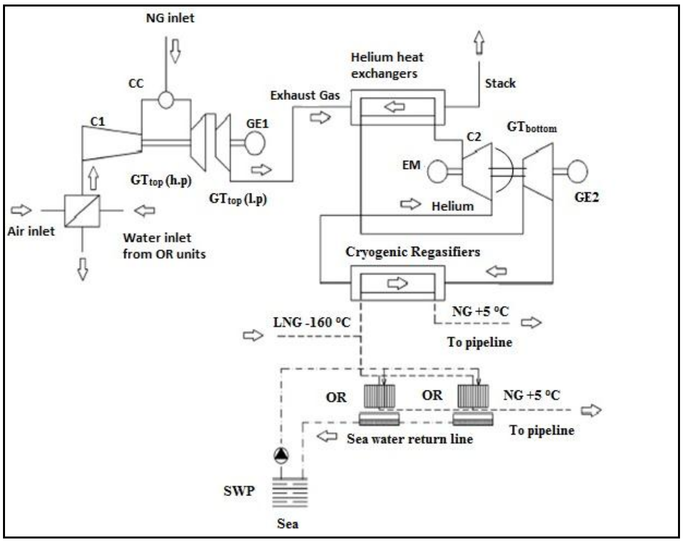

Figure 1: The modular Combined Heat and Power cycle for LNG regasification. *Source:* Dispenza et.al (2009).

The proposed system resulted in helium was more superior gaseous fluid compared to nitrogen. The nitrogen required higher operating pressure in the bottom cycle and its performance was lower compared to helium. Overall, modular plant working with helium produced 0.38 kW kg^{-1} of LNG regasified whilst nitrogen as the working fluid yielded about 0.29 kW kg^{-1} of LNG regasified. Nitrogen produced 0.09 kW less electricity compared to helium. A total of 24 % of higher performance in terms of electricity production could be obtained by using helium instead of nitrogen in the proposed modular plant.

Shi, Agnew, Che and Gao [14] proposed a thermal power system integrated with inlet air cooling, compressor inter-cooling and LNG cold energy utilization to further improve LNG power plant. The proposed system was targeted to enhance the performance of conventional power cycles by using heat of spent steam from the steam turbine. The system proposed worked by recovering latent heat of spent steam from steam turbine and latent heat of vaporization of water vapour contained in the flue gas. This could be achieved by using novel recovery and utilization system for LNG fuelled conventional combined power plant. According to the authors also, cold energy generated during the vaporization of LNG was used to condense the spent steam from the steam turbine thus saving more electric power. Their results yield an increase of 76.8 MW in terms of power output compared to conventional power plant. The net electrical efficiency also increased by 2.8 %. Another important finding was that 0.9 MW of electric power for operating seawater pumps could be further reduced due to the elimination of 10000 tonnes h^{-1} of seawater during the regasification process. Overall, the proposed systems yield an increase in power output and at the same time reducing the shaft work needed to operate the seawater pump.

Sharrat [15] from Foster Wheeler discussed LNG terminal cold energy integration opportunities offered by contractors. In his study, he discussed major cold energy utilization methods which are Inlet Air Cooling (IAC) to Gas Turbine Generators (GTG) in which he claimed increased in power output of the turbine. For IAC to GTG application, suitable working fluid such as glycol-water mixture was used as the working fluid to extract the cold energy. The cold extracted during the vaporization process was then transferred to the GTG inlet air. According to him also, the reason why there was an increased in power input was mainly due to density difference between the working fluid and inlet air where the cooler inlet air has higher density. This resulted in larger mass entered the air compressor of the gas turbine given a fixed volumetric flow. The research yielded significant results where the power output from the GTG was increased by 0.5 % for every degree Celsius (^{o}C) of lowering the inlet air temperature. However, there is an addition of penalty to his proposed system where it generated electrical power but in exchange of higher capital cost of both the LNG import terminal and power plant.

Querol et al. [16] has proposed power generation cycle utilizing pure ammonia as the working fluid to utilize LNG cold available. In their study, ammonia was chosen because Rankine cycle with pure component was the simplest option available yet has proved to be very effective. In order to obtain the desired result, the authors proposed that the concentration of ammonia must be as high as possible in the turbine and vice versa in the condenser. The other sense of using pure ammonia was because ammonia-water mixture required large exchange surfaces, thus increment in capital cost. The author carried out the study based on two alternatives. The first alternative was simulated on gas engine with an ammonia cycle (GE 1pNH_3) with one pressure step and the other one with 2 pressure steps. For the first alternative, ammonia was directly expanded to 110 bar over turbine. For the second alternative, an intermediate pressure of 12 bar was applied before expanded to 110 bar. It was found that alternative 2 provide lower cost of mean power generation which was about 61.29 euro h^{-1} compared to alternative one, 65.5 euro h^{-1}. The calculation includes all the available power generated by the turbines as well as from NH_3 cycle and then divided by the total power produced. Another important finding was the authors were able to use pure ammonia due to the integration of the ammonia condensation process with the vaporization of LNG.

Liu et al. [17] studied two stage Rankine cycle for electric power generation. In their research, a modified steam cycle combined with ORC bottoming cycle was applied. In traditional ORC, the steam expands to low pressure whilst their proposed system vapour left the turbine at a very high pressure. About ten pre-selected organic working fluids and ammonia had been tested in the system. The performance of the system was evaluated using the respective working fluids. In their findings, no single fluid had been identified as the optimum working fluid to be used in Rankine cycle mainly due to the strong interdependence between the fluids, as well as the working condition and the cycle architecture itself. Other researches pertaining to cold energy utilization with Rankine cycle are also analysed [18-23].

3. PROPOSED COMBINED POWER SYSTEM DESCRIPTION

3.1 Case Study 1: Integration of LNG regasification and gas power plant.

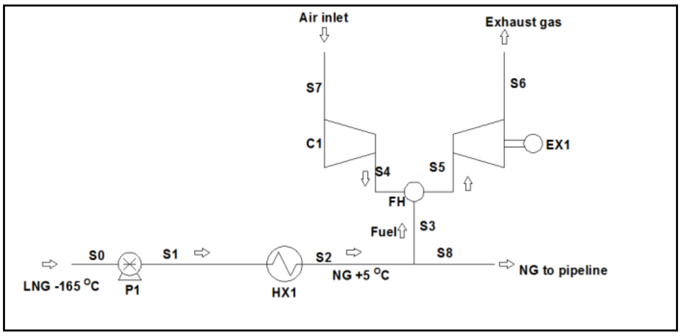

Figure 2: Schematic diagram of integrated LNG regasification and gas power plant. S0, LNG from ship unloading; S2, NG; S7, air feed; P1, LNG pump; HX1, LNG exchanger; FH, combustor; C1, air compressor; EX1; exhaust gas expander.

The LNG from the ship unloading (S0) is first pressurized into NG (S1) before is sent to LNG exchanger HX1 to be vaporized into gaseous state. Then, some portion of NG is fed into the combustor together with pressurized air (S4) to allow for combustion to take place and the rest (S8) is sent to pipeline system for distribution to customers. The exhaust from the combustor (S5) is then expanded in the expander EX1 to produce shaft work needed. The air is fed 20% in excess into the combustor. The amount of air and fuel required for the whole cycle is dependent on the amount of power generation targeted which is 404 MW in capacity.

3.2 Case Study 2: Integration of LNG regasification plant with gas power plant and Rankine cycle.

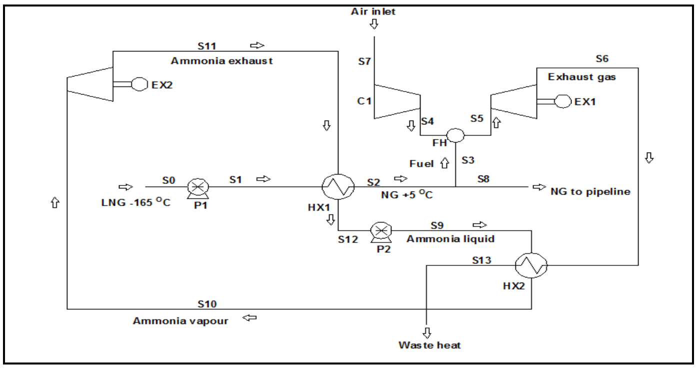

Figure 3: Schematic diagram of integrated LNG regasification and gas power plant via Rankine cycle. S0, LNG from ship unloading; S2, NG; S6, waste

heat recovery; S7, air feed; S9, liquid ammonia solution; S10, ammonia vapour; P1, LNG pump; P2, ammonia liquid pump; HX1, LNG exchanger; HX2, ammonia vaporizer; FH, combustor; C1, air compressor; EX1; exhaust gas expander; EX2, ammonia vapour expander.

For this case study, Rankine cycle is integrated into the previous system by implementing pressurized ammonia (S9) as the working fluid to exchange heat with the exhaust gas from the expander EX1. The ammonia is then vaporized in the vaporizer HX2 before fed into expander EX2 to be expanded to generate shaft work. The ammonia outlet of vaporizer EX2 is then fed into the LNG exchanger HX1 to exchange heat with LNG thus vaporizing the LNG and the same time condensing back to liquid form. The ammonia cycle is completed as it is once again pressurized before being fed back to exchange heat with the exhaust gas in vaporizer HX2. In addition, the temperature approach between the ammonia liquid outlet and the LNG inlet at HX1 is kept at minimum of 10^{o}C.

4. ANALYSIS

To determine the performance of the proposed combined systems, the steady-state components models are used. Every component is modelled in consideration of mass, energy and species balances. Main parameters of the proposed combined cycle for initial start-up simulation works and for the calculation are listed in Table 1. The values within the parentheses represent the variable range for thermodynamic analysis. The fluid package used in the simulation is Peng-Robinson EOS and the thermodynamic properties are calculated by Aspen Hysys.

The following assumptions are used in the proposed system analysis:

- Steady-state flow and the state of the working fluid at each specific location within the system does not change with time.
- All components are well insulated.
- Pressure drop and heat loss in pipelines are neglected.
- All components are well insulated.

Table 1: Main parameters for the calculations.

Parameters	Value
Heat source temperature (^{o}C)	750
Ammonia flowrate (kg/s)	6.6 kg/s
Inlet pressure of EX1 (bar)	24 (24-30)
Inlet pressure of EX2 (bar)	10
Outlet pressure of EXI (bar)	1 (1-5)
Outlet pressure of EX2 (bar)	1
Adiabatic efficiency of pumps	0.99
Polytropic efficiency of expanders	0.9
Polytropic efficiency of compressor	0.9

Thermodynamic equations for each system unit are given as follows:

4.1 LNG feed pump P1

The LNG from ship unloading is pumped to high pressure entering LNG exchanger HX1. The work output is given by Equation (1).

$$W_{P1} = m_{S0} (h_{S1} - h_{S0}) = m_{S0} (h_{S1} - h_{S0}) / \eta_{P1} \quad (1)$$

Where W_{P1} = work required (kW)
η_{P1} = efficiency of pump P1

4.2 Ammonia liquid pump P2

Ammonia liquid is pumped to high pressure before being fed into vaporizer HX2. The work required to do the pumping is given by Equation (2).

$$W_{P2} = m_{S12} (h_{S9} - h_{S12}) = m_{S12} (h_{S9} - h_{S12}) / \eta_{P2} \quad (2)$$

Where W_{P2} = work required (kW)
η_{P2} = efficiency of pump P2

4.3 Compressor C1

Air feed is compressed from atmospheric pressure to high pressure prior to entering the combustor FH. The shaft work required to perform the compression work is given by Equation (3).

$$W_{C1} = m_{S7} (h_{S4} - h_{S7}) = m_{S7} (h_{S4} - h_{S7}) / \eta_{C1} \quad (3)$$

Where W_{C1} = work required (kW)
η_{C1} = efficiency of compressor C1

4.4 Expander EX1

The exhaust gas from combustor passes through expander EX1 to produce shaft work. The exhaust pressure is allowed to drop at lowest possible value to generate more power output. Gross shaft work production by the expander is given by Equation (4).

$$W_{EX1} = m_{S5} (h_{S5} - h_{S6}) = \eta_{EX1} m_{S5} (h_{S5} - h_{S6}) \quad (4)$$

Where W_{EX1} = shaft work produced (kW)
H_{EX1} = efficiency of expander EX1

4.5 Expander EX2

Pressurized ammonia vapour passes through expander EX2 to produce additional shaft work. The equation used to calculate the generated shaft work is given by Equation (5).

$$W_{EX2} = m_{S10} (h_{S10} - h_{S11}) = \eta_{EX2} m_{S10} (h_{S10} - h_{S11}) \quad (5)$$

Where W_{EX2} = shaft work produced (kW)
H_{EX2} = efficiency of expander EX2

4.6 LNG exchanger HX1

Utilizing LNG cold energy, the ammonia at expander EX2 exhaust is condensed back into liquid state at a very low temperature of -165 ^oC. At the same time, LNG is vaporized into NG. The LNG exchanger HX1 heat duty is given by Equation (6).

$$Q_{HX1} = m_{S11} (h_{S12} - h_{S11}) = m_{S1} (h_{S2} - h_{S1}) \quad (6)$$

Where Q_{HX1} = heat duty (kW)

The mass flow of vaporized LNG is equal to Q_{HX1}/r_{LNG} where r_{LNG} is the latent heat of vaporization of LNG at the pressure of the LNG feed pump outlet.

4.7 Ammonia vaporizer (HX2)

Ammonia is vaporized into gaseous form before passes through expander EX2. A heat exchange between the exhaust gas from expander EX1 as the heat source and ammonia liquid as the heat sink occurs in vaporizer HX2. The heat duty required to vaporize the ammonia liquid is given by Equation (7).

$$Q_{HX2} = m_{S9} (h_{S120} - h_{S9}) = m_{S6} (h_{S13} - h_{S6}) \quad (7)$$

Where Q_{HX2} = heat duty (kW)

4.8 Combustor

Air and NG is mixed in the combustor to allow for combustion reaction to take place to produce high temperature of exhaust gas. The air is fed in excess of 20% compared to NG to allow for complete combustion. The heat generated during the combustion reaction is given by Equation (8).

$$Q_{FH} = m_{S3} (HHV_{fuel}) \quad (8)$$

Where Q_{FH} = heat duty (kW)
HHV_{fuel} = higher heating value of fuel

In this case, the HHV value is obtained from S3 that which is the supply fuel for combustion.

4.9 Efficiency

Power output of the proposed combined system is

$$W_{total} = W_{EX1} + W_{EX2} \quad (9)$$

The corresponding net thermal efficiency is defined as:

$$\eta_{TH} = (\sum W_j - \sum W_i) / (Q_{FH} + \sum Q_k) \quad (10)$$

Where $\sum W_j = W_{EX1} + W_{EX2}$ j = EX1, EX2 (11)

$$\sum W_i = W_{C1} + W_{P1} + W_{P2} \quad i = C1, P1, P2 \quad (12)$$

$$\sum Q_k = Q_{HX1} + Q_{HX2} \quad k = HX1, HX2 \quad (13)$$

Thermal efficiency and exergy efficiency are at different level. Therefore, to have further insight on energy performance, exergy analysis is also carried out. Exergy efficiency is defined as exergy output divided by the energy input to the system. From thermodynamic point of view, exergy also called as availability or work potential is the maximum useful work that can be obtained from a system at a given state in a given environment. The exergy input is taken as the exergy change of the heat source whilst exergy output is the exergy of the net work.

Exergy can be defined as:

$$e = h (h - h_0) - T_o (s - s_0) \quad (14)$$

Where T_0 is taken at its reference condition which is equal to 298K.

The overall exergy balance equation is:

$$E_{in} = E_{out} + E_{loss} \quad (15)$$

The input exergy of the system is given as:

$$E_{in} = m_{S3} HHV_{fuel} + m_{S0} e_{S0} \quad (16)$$

The output exergy of the system is given as:

$$E_{out} = \sum E_j - \sum E_i \quad (17)$$

Where $E_j = m_{j,inlet} (e_{j,inlet} - e_{j,outlet})$ j = EX1, EX2 (18)
$E_i = m_{i,inlet} (e_{i,outlet} - e_{i,inlet})$ i = C1,P1,P2 (19)

Notation j represents expanders whilst i represents compressor and pumps respectively. Finally, the exergy efficiency of the power cycle can be written as:

$$\eta_e = E_{ou}t/E_{in} \quad (20)$$

5. RESULTS AND DISCUSSION

5.1 Results of energy analysis

Based on the mentioned modules, a computer simulation using Aspen Hysys modelling software is developed. The initial input parameters are summarized in Table 1 with the heat source temperature of 750 ^oC, the NG supplying pressure of 24 bar, the expander EX1 inlet pressure of 24 bar, and the ammonia inlet expander pressure of 10 bar.

The performance of the system for both case studies is summarized in Table 2. This system is able to generate power output of 404 MW and the net thermal efficiency is estimated as 38.8 % in case study 1 and increases to 40.9% in case study 2 as calculated based on Equation (10). This happens due to introduction of Rankine cycle in case study 2 which eliminates the use of seawater as the heat source for LNG regasification. In addition, extra shaft work is realized by an addition of expander EX2. There is an increase of 5.1 % in thermal efficiency from case study 1 to case study 2. About 93.3 tonnes h^{-1} of LNG can be heated up to 5 ^{o}C at the same time. The exergy efficiency of the case study 1 is 41.3% and then increases about 2.4 % to 42.3% with the introduction of Rankine cycle in the system as calculated using Equation (20).

Table 2 : Calculation results for the proposed combined cycle.

	W_{EX1} (MW)	W_{EX2} (MW)	W_{C1} (MW)	η_{TH}	E_{in} (MW)	E_{out} (MW)	η_e
CASE STUDY 1	404	0	129.3	0.388	667.1	276	0.413
CASE STUDY 2	404	5.9	129.3	0.409	667.1	281.8	0.423

5.2 Effect of expander EX1 inlet pressure

Effect of expander EX1 inlet pressure is investigated for both case studies. For case study 1, the combustion is allowed to take place at constant pressure of 24 bar, then increases to 30 and 40 bar respectively. The outlet pressure however is set to the lowest possible values preferably which is at 1 bar. The reason is that to have the highest expansion ratio of expander regardless of the inlet pressure.

For case study 2, due to introduction of Rankine cycle with ammonia as the working fluid, there are a few constraints must be satisfied for the system to effectively converge. Among the constraints include maintaining the state of ammonia after vaporizer HX2 in the vapour state and condensing it back to liquid in LNG exchanger HX1. Another constraint is to avoid temperature cross between the hot and cold stream at LNG exchanger HX1. Therefore, basic guideline of minimum temperature approach in the heat exchanger is set at 10^{o}C minimum. Taking that as the basis, temperature of ammonia liquid outlet after LNG exchanger HX1 is fixed at -150 ^{o}C hence resulted in 15 ^{o}C differences with LNG inlet temperature. Apart from that, the inlet and outlet pressure of expander EX2 is set to 10 bar and 1 bar respectively whilst the ammonia vapour outlet of vaporizer HX2 is fixed at 750 ^{o}C. The results obtained by varying the inlet pressure of expander EX1 for both case studies are tabulated in Figure 4 and Figure 5.

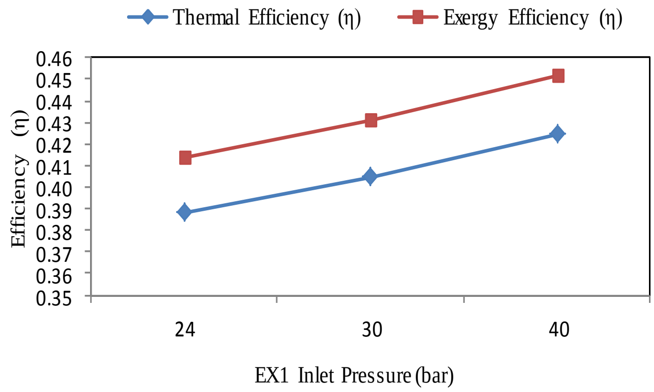

Figure 4 : System efficiency by varying inlet pressure of expander EX1 for case study 1.

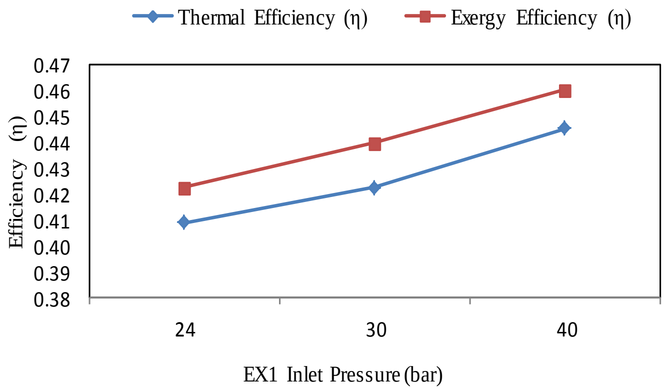

Figure 5 : System efficiency by varying inlet pressure of expander EX1 for case study 2.

From Figure 4 and 5, it is found that as the inlet pressure of the expander EX1 is increased, the system efficiency of the system also increases. This is due to the increases of enthalpy and entropy change between the inlet and outlet stream of expander EX1. In addition, expansion ratio of the expander also increases which directly impact the gas turbine system efficiency. Other factor to note is that the overall exergy efficiency of the system is higher for all there scenarios. This is contributed by the involvement of entropy changes in the exergy thermodynamic calculation as shown by Equation (14). Hence, from thermodynamic point of view, exergy analysis is more useful as it provides further insight in system efficiency effort compared to thermal analysis.

5.3 Effect of expander EX1 outlet pressure

Using the same specifications as discussed in section 4.2, the effect of outlet pressure of expander EX1 on overall net system efficiency is also investigated by varying the outlet pressure to 3, and 5 bar respectively. The results obtained are shown in Figure 6, Figure 7, Figure 8 and Figure 9 respectively.

5.3.1 Case Study 1: Integration of LNG regasification and gas power plant.

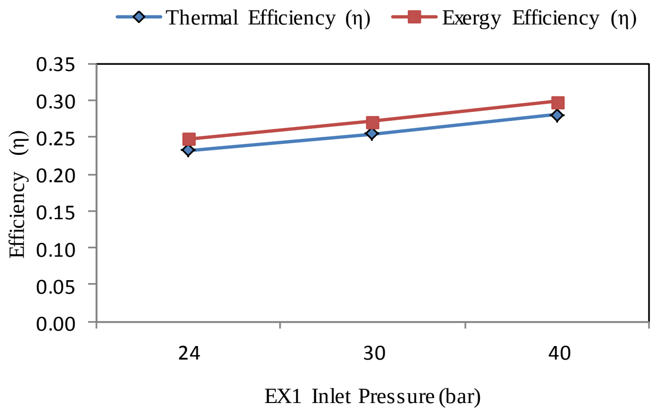

Figure 7: System efficiency by fixing the outlet pressure of expander EX1 to 3 bar.

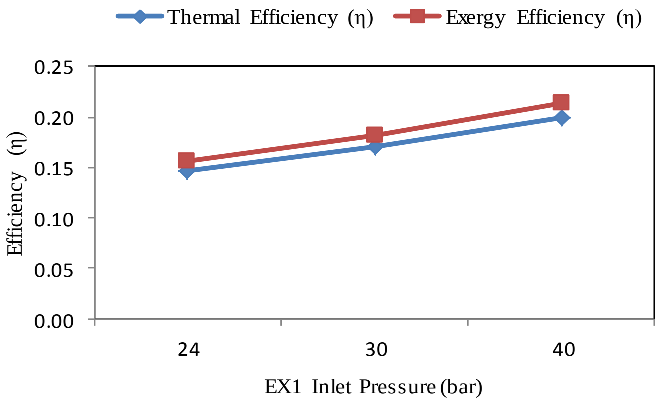

Figure 6 : System efficiency by fixing the outlet pressure of expander EX1 to 5 bar.

5.3.2 Case Study 2: Integration of LNG regasification plant with gas power plant and Rankine cycle

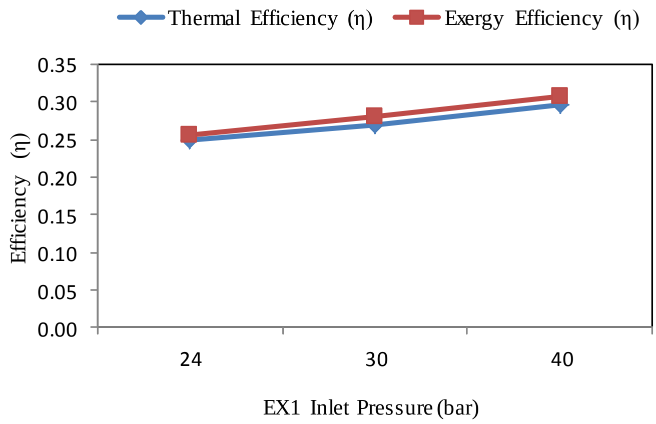

Figure 9 : System efficiency by fixing the outlet pressure of expander EX1 to 3 bar.

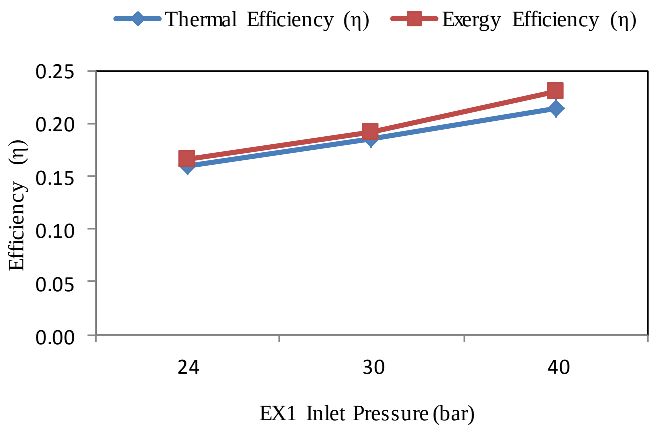

Figure 8: System efficiency by fixing the outlet pressure of expander EX1 to 4 bar.

It can be seen that by setting the outlet pressure of expander EX1 to 1 bar yield the highest system efficiency as shown in Figure 4 and Figure 5 compared to 3 and 5 bar. The results obtained are valid regardless of the inlet pressure of expander. This concludes that by setting it to lowest outlet pressure shall give the highest expansion ratio of expander. From thermodynamic point of view, the higher the expansion ratio, the higher is the change of enthalpy and entropy between the inlet feed and product outlet of the expander. This resulted in higher shaft work generated as shown by Equation (4) and Equation (18).

5.4 Effect of working fluid on system efficiency

Investigation on effects of four other working fluids aside from ammonia is also conducted. The working fluids used are ammonia-water mixture with 0.5 mole fraction each, pure water, ethane and propane. The mass flow required to achieve aforementioned specifications as discussed in section 4.2 is recorded for further analysis. The results on the effect of working fluid on system efficiency and mass flow required are shown in Figure 10. The results obtained are compared including pure ammonia which has been conducted initially and the results are already analysed.

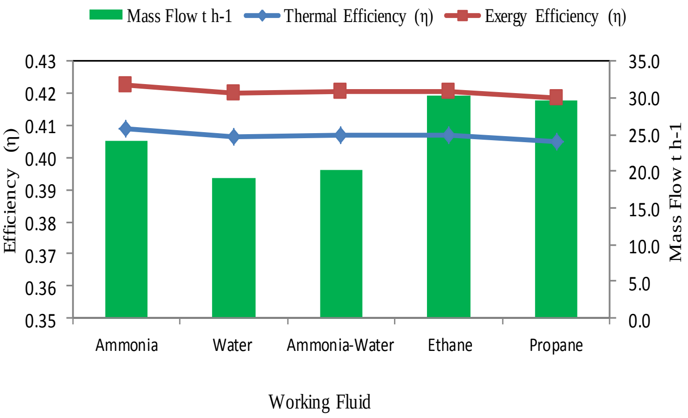

Figure 10 : Effect of working fluid on system efficiency and mass flow required.

From Figure 10, ammonia yields the highest thermal and exergy efficiency followed by ammonia-water mixture, water, ethane and propane. The benchmarked results prove that the performance of typical Rankine cycle with water as working fluid can be increased by replacing it with ammonia as the working fluid. However, mass flow of water required to achieve the operating parameters at LNG exchanger HX1 and vaporizer HX2 as discussed in section 5.2 is less than the ammonia by an amount of 5.2 t h-1. It is worth to note that even though water yields lower system efficiency, it can contribute to lower initial purchase and operating cost. However, given the scope of this project which is mainly based on system efficiency only and does not include economic analysis, thus the cost benefit ratio is neglected.

6. CONCLUSION

The basic case study of integration of LNG regasification, gas power plant and Rankine cycle is successfully developed and verified via Aspen Hysys. The targeted power production capacity of 404 MW is realized and efficiency of the system is increased by integrating Rankine cycle with ammonia as working fluid compared to typical Rankine cycle utilizing pure water. There is an improvement of 5.1 % and 2.4 % over thermal and exergy efficiency from case study 1 to case study 2. Setting outlet pressure of expander EX1 to 1 bar regardless of inlet pressure yields the highest system efficiency due to highest expansion ratio available in the expander which directly affecting the turbine's efficiency. Effects of other working fluids aside from ammonia such as water, ammonia-water mixture, ethane and propane on overall system's efficiency are also analysed. Pure ammonia gives the highest system's efficiency and propane produces the least favourable results. However, there is a trade-off between mass flows of ammonia and water even though water yields lower system efficiency but required less flow rate of about 5.2 tonnes h^{-1} compared to ammonia. To conclude, the main objectives of this project are achieved which are to develop an integrated plant consisting of LNG regasification, gas power plant and Rankine cycle to fully utilize LNG cold energy and effect of other working fluids on system performance is also investigated.

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