COLD ENERGY UTILIZATION FROM LNG REGASIFICATION PROCESS

By

Khuong Minh Cam Tu

Dissertation submitted in partial fulfillment of
the requirements for the
Bachelor of Chemical Engineering (Honours)

December 2012

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CERTIFICATION OF APPROVAL

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Approved by,

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(Assoc. Prof. Dr. Shuhaimi B Mahadzir)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

December 2012

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 reference and acknowledgements, and that the original work contained herein has not been undertaken or done by unspecified sources or persons.

KHUONG MINH CAM TU

ABSTRACT

Energy savings becomes one of the important effects to reduce the effect of global warming. Effective utilization of the cryogenic energy associated with LNG regasification gains more and more important to this energy issue. LNG regasification plant operates in combined cycle mode comprising ammonia Rankine cycle and a Brayton power cycle of combustion gas, open LNG cycle, interconnected by the heat transfer process in the recuperation system. LNG at low temperature of -162^{o}C and atmosphere pressure is gasified by absorbing heat from hot fluid of Rankine cycle. Typically ammonia is used as the working fluid for Rankine cycle. Then the LNG will be superheated by exhaust gas of Brayton cycle evaporation system as the cycle cold sink, the cycle condensation process can be achieved at a temperature much lower than ambient without consuming additional power**.** By using this thermal sink in a combined cycle plant that produces both power and gas, it is possible to recover cold energy from vaporization of LNG.

Energy equilibrium equations and exergy equilibrium equation of each equipment in the cascading power cycle are established. Taken some operating parameters as key parameters, influences of these parameters on thermal efficiency and exergy efficiency of the cascading power cycle were analyzed. The net power overall after optimization is obtained at 22099.45kw compared to initial value which is only at 20175.41kW. The project objectives have been achieved. The research shows that it is able to identify the opportunity for cold energy recover from LNG regasification process by electricity generation and the optimization of heat recovery system is successfully achieved.

ACKNOWLEDGEMENT

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ABBREVIATIONS AND NOMENCLATURES

C cost of power ($/kWh)

e exergy (kJ/kg)

E exergy (kJ)

EB economy benefit ($/h)

H enthalpy (kJ/kg)

LHV lower heating value (kJ/kg)

M mass flow rate (kg/s)

P pressure (MPa)

S entropy (kJ/kg-°C)

T temperature (°C)

W work (W)

efficiency η

CHAPTER 1
INTRODUCTION

1.1 BACKGROUND STUDY

Liquefied natural gas is primarily composed of methane, which has been converted to liquid form for ease of storage and transport. In liquefied form,LNG volume is reduced to 1/600 of the volume of natural gas. Liquefaction describes the process of cooling natural gas to 162°C at atmospheric pressure. . LNG must be turned back into gas for commercial use at regasification plants.

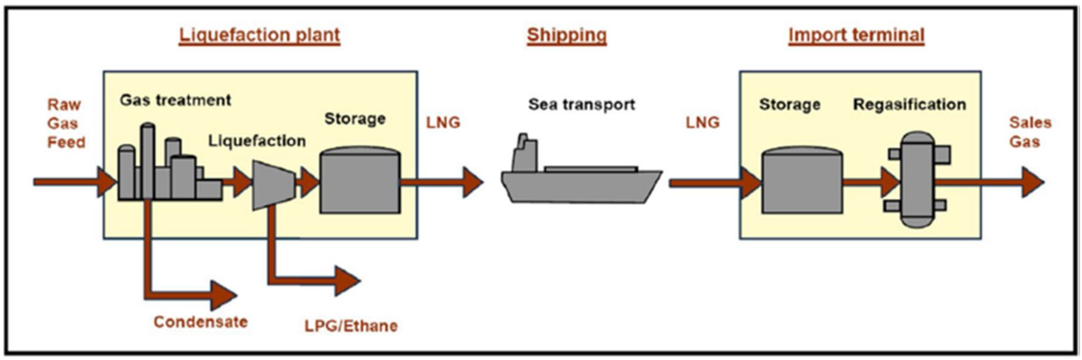


Figure 1.The LNG Process chain- from Extraction, Processing and Transport to Consumption

LNG load arriving at receiving terminals is off-loaded into storage tank and then pumped from storage tank at the required pressure and vaporized for final transmission to the consumers. In practice the regasification is performed in gradually warming the gas back up to ambient temperature.

It is done under high pressures of 60 to 100bar. A series of heat exchangers running on seawater heats up the LNG stream. During the vaporization process, latent heat and any sensible heat required to superheat the vapor are supplied to the LNG. During the liquefying process, a large amount of mechanical energy is consumed in refrigeration process, so LNG contains much cold energy (cryogenic exergy). If LNG is used as a fuel in a combined system, the waste heat of exhaust gases and the cold energy of LNG can be utilized at the same time. From a thermodynamic viewpoint, re-heating represents a net loss of available energy, which causes degradation of overall energy efficiency of the conversion chain. Accordingly, utilization of LNG cold energy proves to be an interesting area of study.

The potential of cold energy utilization includes power generation, refrigeration, air liquefaction and separation, reduction of CO_2 emission, cryogenic thermoelectric generator, and similar applications to cold usage.

In other word, LNG cold energy utilization may be divided into two major approaches: cooling power supply and electric power generation. For cooling power supply, LNG is commonly used in freezing foods, making dry ice, air conditioning, low-temperature crushing, etc. However, its performance is usually not good enough because the users only need relatively high-temperature of about -500C. For electric power generation, there are two kinds of LNG cold energy utilization: (1) independent thermal cycle with natural gas direct expansion and closed-loop Rankine cycle. For natural gas then vaporized to superheating natural gas, and finally expands in a turbo-expander to a certain pressure for supplying to users. For closed-loop Rankine cycle type, the surrounding is a heat source and LNG is a cold sink.

1.2 PROBLEM STATEMENT

In assessing an extent of power saving in existing systems, overall power recovery rate is estimated at around 8% of the total potential value. This percentage suggests that the remaining 92% of the cold energy is still being wasted and more cold utilization will attain more energy saving (M. Sugiyama, et. al, 1998).There are many ways of LNG energy utilization, which use the LNG coldness as the heat sink in closed-loop Rankine cycles, Brayton cycles and combinations thereof. However the energy efficiency is still not high. Therefore, it would be good if a study to optimize this cold energy utilization in higher efficiency can be implemented. In the other word, the effective utilization of the huge cryogenic energy associated with LNG vaporization is very important.

1.3 OBJECTIVE

The objectives of this study are:

1) To identify further opportunities of recovering cold energy from the LNG regasification process.

2) To simulate the cold energy utilization process during LNG regasification process.

3) To optimize the heat recovery system from regasification process by Hysys software.

1.4 SCOPE OF STUDY

This project will focus on external parties that may use the cold energy potential of LNG from the regasification process at LNG import terminal. These external processes that may require the cold available from the LNG regasification are referred to as cold users.

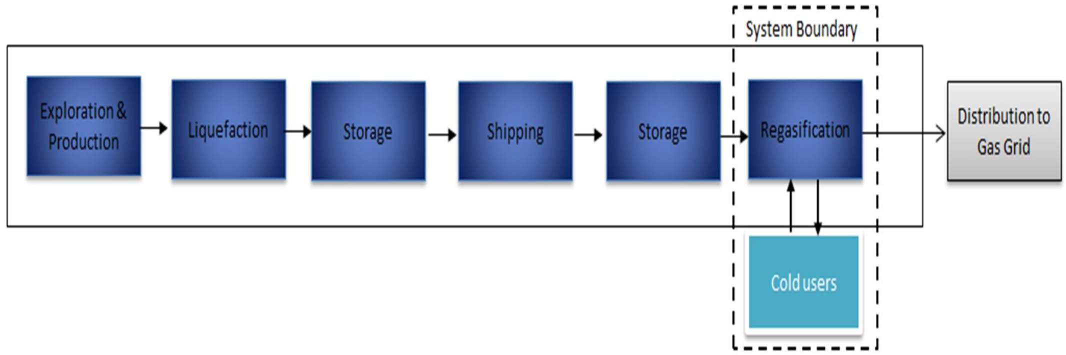


Figure 1.2: LNG regasification process

1.5 RELEVANCY AND FEASIBILITY OF THE PROJECT

This project is relevant to the author's field of study since it focuses in one of the areas in Chemical Engineering. The project is feasible since it is within the scope and time frame. The author has planned to complete the research and literature review by the mid-semester break. Besides that, this project requires only Aspen Hysys software to simulate which are readily available, no more equipment or chemicals required.

CHAPTER 2
LITERATURE REVIEW

LNG cold energy is one type of waste energy released from the re-gasification of liquefied natural gas (LNG). The ‘cold energy’ refers to the heat absorption effect from the ambient surrounding when LNG is re-gasified at the LNG terminals. LNG cold energy is recognized as one high quality energy source to cool media. The cold energy stored by LNG could be recovered rather than directly taken off be seawater.

Use of the cryogenic exergy of LNG for power generation includes methods which use the LNG as the working fluid in natural gas direct expansion cycles, or its coldness as the heat sink in closed-loop Rankine cycles, Brayton cycles and combinations thereof. Other methods use the LNG coldness to improve the performance of conventional thermal power cycles (Na Zhang and Noam Lior, 2006). Below are some methods that almost the studies used for their research

2.1. CLOSED-LOOP RANKINE CYCLE

In steam turbine system, Rankine cycle is often used. As LNG’s temperature is very low (-162^{o}C), ammoniac water solution or organic fluid instead of steam which works as intervening media is used in Rankine cycle.

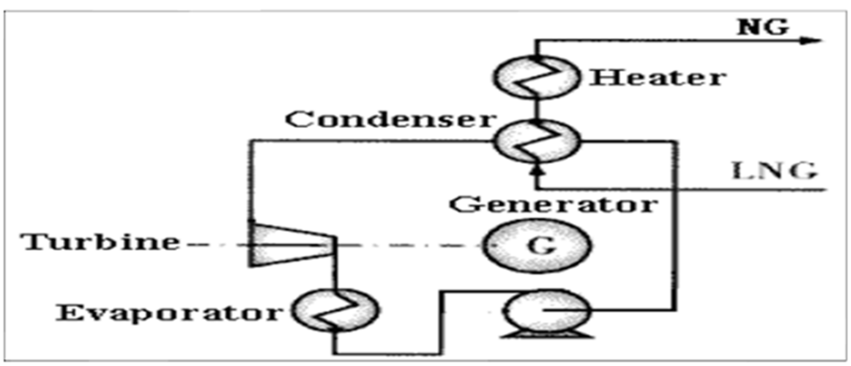


Figure 2.1: Close-Loop Rankine Cycle

The media vapor is condensed by LNG in condenser and the low temperature media is pump to evaporator heated by seawater then the high temperature and pressure media expansion in turbine to drive generator to complete whole cycle. As LNG need not expand to work, the regasification LNG has high pressure. The practical LNG cold energy power generation systems have been operated by using the propane

ORC (organic Rankine cycle) in Japan for about 40 years. However, the recovery rate of LNG cold availability with the ordinary ORC is usually around 14% (Liu Y and Guo K, 2010). Szargut and Szczygiel(2009) studied the Rankine LNG cold power cycles with three variants, two with binary working fluids and one with single ethane working fluid. La Rocca (2010) proposed a modular LNG regasification unit based on a power cycle working with ethane, which allows the cold energy to be used for multipurpose to reduce the irreversibility of the regasification process.

Miyazaki et.al (2000) proposed an ammoniae water Rankine cycle with refuse incinerator and LNG cold energy, and compared it with the conventional steam Rankine cycle. It was found that the thermal and the exergy efficiencies of the combined cycle were 1.53 and 1.43 times higher than those of the conventional cycle, respectively.

Yanni and Kaihua (2011) suggested improving the energy recovery efficiency of an LNG cold power generation. The authors used the simulation method for the research. The cycled was simulated with seawater as the heat source and LNG as heat sink. The authors found that the efficiency is increased by 66.3% and the optimized LNG recovery temperature is around -60^{o}C

The system proposed by Shi & Che (2007) uses this LNG vaporization as a low temperature thermal sink to the Rankine cycle. The outlet steam from the turbine is condensed by utilizing the cold energy generated during LNG vaporization. Therefore, the steam condenser pressure can be reduced to a lower value for increasing the output and efficiency of the steam turbine. Within the condenser pressure range of 0.040 – 0.010 bar, the calculated fuel efficiency of a gas turbine combined cycle is improved from 62 to 64.5%.

2.2. BRAYTON CYCLE

Brayton cycle is composed by compressor, nitrogen turbine and heat exchangers. Nitrogen is cold by LNG before it enters compressor which makes compressor consume less energy to get the same compression ratio. The high pressure nitrogen is heated and sent to turbine to drive generator. According to theory calculation(Gianfranco &Costante, 2009) if the temperatures of compressor inlet and turbine inlet are -130^{o}C and 72^{o}C , the average temperature difference of heat exchanger is 15C, the efficiency of heater is 90%, the whole efficiency of Brayton cycle can rise up to 53%. Obviously, heater or very high temperature exhaust air is needed in this method too.

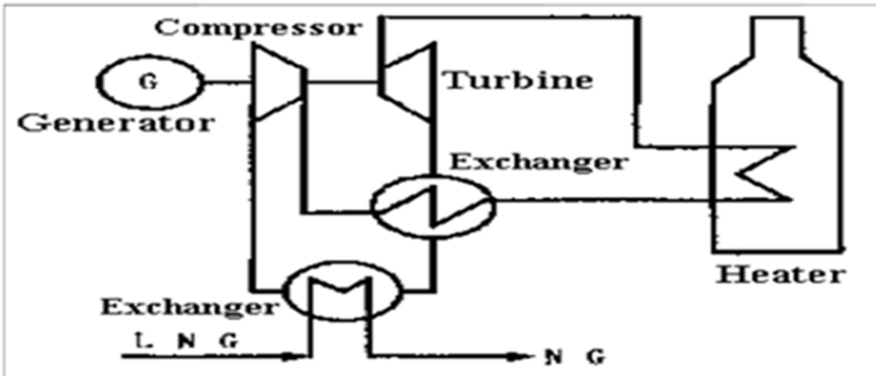


Figure 2.2: Brayton cycle

Ken'ichi , Kiyoshi, Yoshiharu and Shoichu (2004) discussed about the recovery of the energy consumed in liquefaction using the mirror gas-turbine. The optimum characteristics have been calculated. The result showed that 7-20% of exhaust energy can be converted to useful work; thermal efficiency of the TG can be improved more than 25% and 60% of the total exergy efficiency in the case of 15000C TIT.

Celidonio, Giorgio, Vincenzo and Giuseppe (2004) discussed about the exergy recovery during regasification. The authors proposed a combine-cycle system of a gas turbine as a topping cycle and inverted Brayton cycle with three intercooling as a bottoming cycle. The research showed the advantages and disadvantages of CHP plants working with helium, nitrogen and the comparison between CHP modular plants working with helium or nitrogen.

2.3. COMBINED CYCLE

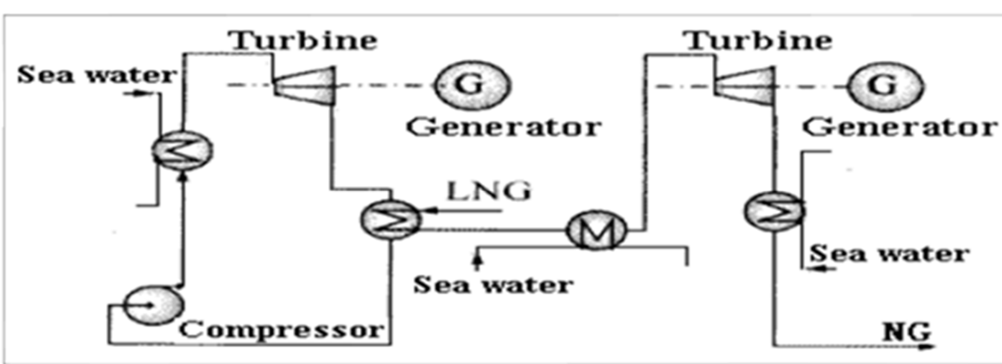


Figure 2.3: Combined Cycle

Combined Cycle is combination of Rankine cycle and Brayton cycle. The right part shows Braton cycle. The left part shows Rankine cycle with intervening media. In order to rise effect of cold energy recovery in this part, regeneration is often used, and about 50% cold energy can be recovered (T.S.Kim& S.T.Ro, 2000). But in this method, sea water still carries away some cold energy. And some more heat exchangers are used. Zhang et al. (2006) presented a mode of the super/sub-critical CO2 Rankine-like cycle combined with Brayton cycle by using LNG as the heat sink, and the liquefied CO2 ready for disposal can be withdrawn without consuming additional power.

Through the result of optimization, T.Lu and K.S.Wang (2009) show that the economy benefit increase by increasing of work generated by expander in open LNG cycle and the decrease of work consumed by compressor and pump in Brayton and Rankine cycle

2.4. LNG AND GAS TURBINE COMBINED CYCLE

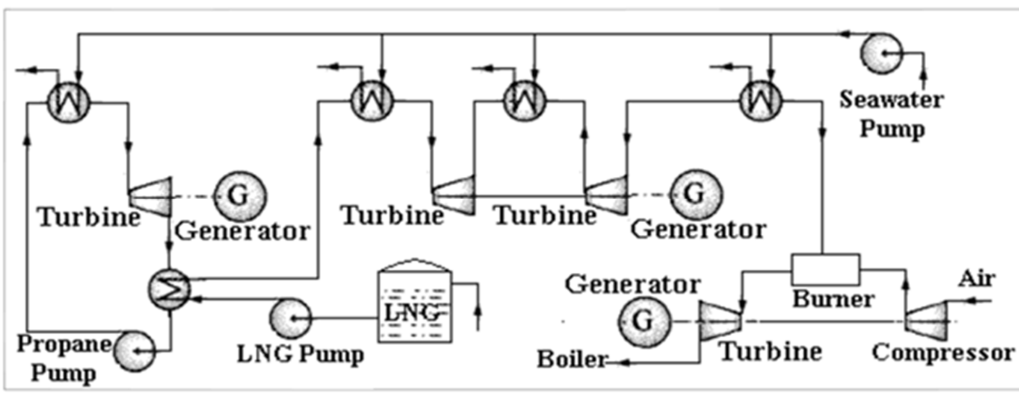


Figure 2.4: LNG and Gas Turbine Combined Cycle

LNG and gas turbine combined cycle is a comprehensive energy utilization cycle. One Rankine cycle and two or more direct cycles are used for LNG cold energy recovery and generation power. Meanwhile, the LNG is regasified, heated and sent to combustion chamber. Then turbine is driven by high temperature gas to get the generator work. The heat of exhaust gas is recovered in exhaust-heat boiler to produce steam to drive steam turbine generator.

The maximum efficiency of LNG and gas turbine combined cycle is 55%. It is much higher than the efficiency of steam turbine and gas turbine which is 38~41% and 35% respectively. It is suitable for large amount of LNG regasification and power generation. (Fan Zhang& Xiao-min Yu).

Y. Hisazumi and et.al (1998) proposed a system consists of a Rankine cycle using a Freon mixture, natural-gas Rankine cycle and a combined cycle with gas and steam turbines. The heat sources for this system are the latent heat from the steam-turbine's condenser and the sensible heat of exhaust gas from the waste-heat recovery boiler. The results of these studies show that in the total system, about 400 kWh can be generated by vaporizing 1 ton of LNG, including about 60 kWh/LNG ton recovered from the LNG cold energy when supplying NG in 3.6 MPa.. About 8.2 MWh can be produced by using 1 ton of LNG as fuel, compared with about 7 MWh by the conventional combined system.

Xiaojun Shi and Defu Che(2009) also has proposed a combined power system, in which low temperature waste heat can be efficiently recovered and cold energy of liquefied natural gas (LNG) can be fully utilized as well. The results show that the proposed combined cycle has good performance, with net electrical efficiency and exergy efficiency of 33% and 48%, respectively, for a typical operating condition. The power output is equal to 1.25 MWh per kg of ammonia–water mixture. About 0.2MW of electrical power for operating sea water pumps can be saved

2.5. LNG AND GAS TURBINE COMBINED CYCLE WITH CO2 RECOVERY

LNG cold energy and carbon dioxide recovery system is based on LNG and gas turbine combined cycle

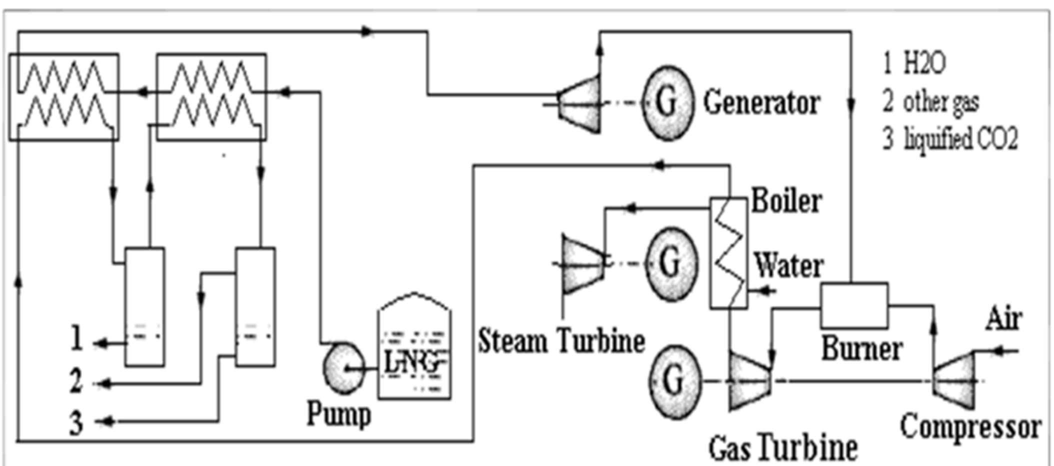


Figure 2.5: LNG and Gas Turbine Combined Cycle with CO_2 Recovery

After the exhaust from gas turbine makes water change into steam in the boiler, it is sent to the two-stage heat exchangers and flow countercurrent against cold LNG.At first stage water is condensate and separated from exhaust gas. Then the water free exhaust gas is further cold at second stage and carbon dioxide is liquefied and removed. After those the remaining is other noncondensable gas. During that process the LNG is heated and sent to direct expansion cycle to generate power. Na Zhang and Noam Lior (2006) proposed a novel near-zero CO2 emission thermal cycle with LNG cryogenic exergy utilization. The plant operates in a quasi-combined cycle mode with a supercritical CO2 Rankine-like cycle and a CO2 Brayton cycle by coupling with the LNG evaporation system as the cycle cold sink. The cycle condensate process can be achieved at a temperature much lower than ambient and the net energy and exergy efficiencies are found to be 65 and 50%, respectively. According to Shimin Deng et.al 2004, due to the advanced integration of system and cascade utilization of LNG cryogenic energy, the system has excellent energy saving: chemical energy of fuel and LNG cryogenic energy are saved by 7.5–12.2% and 13.2–14.3%, respectively. As CO2 is selected as working fluid and oxygen as fuel oxidizer, CO2 is easily recovered as a liquid with LNG vaporization.

Vincenzo La Rocca, et. al, (2009) deal with facilities delivering cold released during LNG regasification and related pipeline facilities to transfer cold at far end users. The author has used thermodynamic analysis during his studies. In this paper it is proposed using Carbon dioxide circulating in an innovative service loop to supply for a cluster of Agro food factories and a Hypermarket which 2km far away from the regasification site.

2.6. WORKING FLUID FOR RANKINE CYCLE

Jan Szargut et.al (2009) said, to apply the Rankine cycle in a cold power plant, the working fluid should meet the following conditions:(1) the critical temperature should be higher than the environmental one,(2) the saturation pressure in the condenser should be higher than the environmental one, in order to avoid problems with the vacuum in the condenser.

According to Athanasios I.Papadopoulos et.al (2010), there are numerous properties that should be considered for the design and selection of working fluids for Rankine cycle processes . Such properties are as follows:(1)The density (ρ) of the working fluid must be high either in the liquid or vapor phase. High liquid or vapor density results to increased mass flowrate and equipment of reduced size. (2)The latent heat of vaporization (Hv) of the working fluid must be high for many reasons. Hv enables most of the available heat to be added during the phase change operation, hence avoiding the need to regulate the superheating and expansion of the vapor through regenerative feed heating in order to enable higher efficiency. High latent heat also help in reducing moisture during the expansion as well as avoids the necessity to condense a superheated fluid .(3)The liquid heat capacity (Cpl) of the working fluid must be low.(4)The viscosity (μ) of the working fluid should be maintained low in both liquid and vapor phases in order to achieve a high heat transfer coefficient with reduced power consumption .(5)The thermal conductivity (λ) must be high in order to achieve high heat transfer coefficients in both the employed condensers and vaporizers . (6)The melting point temperature (Tm) should be lower than the lowest ambient operating temperature in order to ensure that the working fluid will remain in the liquid phase. (7)The critical temperature (Tc) should be higher than the maximum cycle operating temperature, as we only consider sub-critical operations in this work. (8)The ozone depletion potential (ODP) is an index that determines the relative ability of chemical substances to destroy ozone molecules in the stratosphere, hence working fluids with low or zero ODP are required.(9)The global warming potential (GWP) is an index that determines the potential contribution of a chemical substance to global warming. (10)The determination of the toxicity (C) of the designed working fluids is important for human safety reasons. (11)The flammability (F) is an index used to assess the flammability characteristics of the designed working fluids. (12) The critical pressure (Pc) of the working fluid should be higher that Pmax, as only sub-critical operations are considered in this work. Working fluid is usually divided into two groups: one is organic working fluids and one is natural working fluids. For organic working fluids in Rankine cycle(ORC), Liu BT et.al (2004) said that for practical LNG cold power generation, ORC is most commonly used. However, the phase change temperature of ORC usually needs to be kept at constant, and hence cannot match well with the temperature variation of the sensible heat sink formed by LNG vaporization, and may cause a large irreversibility .

Natural working fluids are substances, naturally existing in the biosphere. They generally have negligible global environmental drawbacks (zero or near-zero ODP and GWP). They are therefore long-term alternatives to the CFCs. Examples of natural working fluids are ammonia (NH3), hydrocarbons (e.g. propane), carbon dioxide (CO_2), air and water. Some of the natural working fluids are flammable or toxic.

■Ammonia (NH_3) is in many countries the leading working fluid in medium- and large refrigeration and cold storage plants. Codes, regulations and legislation have been developed mainly to deal with the toxic and to some extent, the flammable characteristics of ammonia. Thermodynamically and economically ammonia is an excellent alternative to CFCs and HCFC-22 in new heat pump equipment. It has so far only been used in large heat pump systems, and high-pressure compressors have raised the maximum achievable condensing temperature from 58°C to 78°C.

■Hydrocarbons (HCs) are well known flammable working fluids with favorable thermodynamic properties and material compatibility. Presently, propane, propylene and blends of propane, butane, iso-butane and ethane are regarded as the most promising hydrocarbon working fluids in heat pumping systems. HCs are widely used in the petroleum industry, sporadically applied in transport refrigeration, domestic refrigerators/freezers and residential heat pumps (notably in Europe). Due to the high flammability, hydrocarbons should only be retrofitted and applied in systems with low working fluid charge.

■Water is an excellent working fluid for high-temperature industrial heat pumps due to its favourable thermodynamic properties and the fact that it is neither flammable nor toxic. Water has mainly been used as a working fluid in open and semi-open MVR systems, but there are also a few closed-cycle compression heat pumps with water as working fluid. Typical operating temperatures are in the range from 80°C to 150°C. 300°C has been achieved in a test plant in Japan, and there is a growing interest in utilising water as a working fluid, especially for high- temperature applications. The major disadvantage with water as a working fluid is that the low volumetric heat capacity (kJ/m^3) of water. This requires large and expensive compressors, especially at low temperatures.

■CO_2 is a potentially strong refrigerant that is attracting growing attention from all over the world. CO_2 is non-toxic, non-flammable and is compatible to normal lubricants and common construction materials. The volumetric refrigeration capacity is high and the pressure ratio is greatly reduced. However, the theoretical COP of a conventional heat pumping cycle with CO_2 is rather poor, and effective application of this fluid depends on the development of suitable methods to achieve a competitively low power consumption during operation near and above the critical point. CO_2 products are still under development, and research continues to improve systems and components. A prototype heat pump water heater has already been developed in Norway. CO_2 is now being used as a secondary refrigerant in cascade systems for commercial refrigeration.

Among those properties of ORC and natural working fluids, it is realized that natural working fluid is the best choice for Rankine cycle. And here, the combination of Amoniac water working fluid is the best choice for the process so far. As the research from Xiaojun

Shi, Defu Che(2009), Ammonia–water mixture is used as working fluid to recover low temperature waste heat because multi-component working fluid is suitable for sensible heat source. The boiling temperature of the ammonia–water mixture increases during the boiling process, so that a better thermal matching between the heat source and working fluid is obtained and exergy destruction is decreased. It's also meet the criteria of an efficient working fluid for a Rankine cycle.

CHAPTER 3
METHODOLOGY

3.1 RESEARCH PROCESS MODEL

The overall methodology for this research is shown in figure 3.1

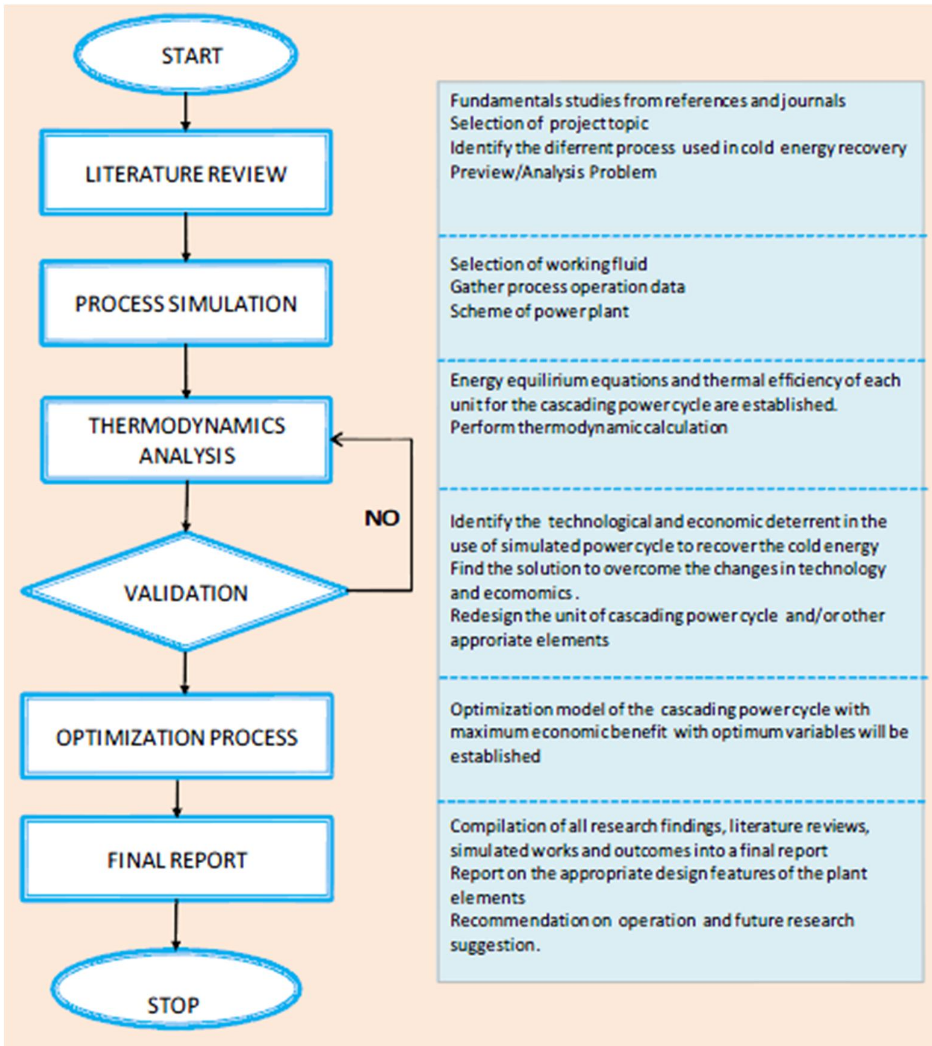


Figure 3.1: Methodology of Project

Evaluation of opportunities for cold energy utilization from LNG regasification process is performed using basic thermodynamic analysis and process simulation works. Information is gathered and obtained from open literatures and process operations data. Data on operating parameters around the compressor, turbine, fire heater, heat exchanger and pump unit are obtained as inputs for simulation of the regasification process section. The operating parameters above include basic parameters such as flow, pressure, temperature and composition. The simulations are developed using a Hysys software. A basic thermodynamic analysis is carried out to study the energy/exergy efficiency for cold utilization in the LNG regasification process. The study focuses on the Brayton cycle of combustion gas, Rankine cycle and open LNG cycle in the LNG regasification process. The selection of working fluid for the Rankine cycle will be chosen based on critical temperature and saturation temperature.

Based on the flow simulation, optimum variables such as condensation temperature of heat exchanger in Rankine cycle, minimum temperature difference of heat exchanger between outlet temperature of hot stream and inlet temperature of cold stream in power cycle of combustion gas, pump pressure ratio both of open LNG cycle and Rankine cycle, and expander pressure ratio of open LNG cycle will be established. Finally, Evaluation of the installed power and the pay-back of the cold power plant will be carried out to see whether the new process works efficiently as expected.

3.2. MATHEMATIC MODEL FOR PERFORMANCE ANALYSIS

Energy equilibrium equations and exergy equilibrium equations of each unit for the cascading power cycle are established with neglect of pressure drop in fired heater, heat exchangers, and pipelines.

3.2.1 **Energy balance equations and thermal efficiency**

For air compressor C1, pumps P1 and P2, the energy equation is

$W_i = m_{i,inlet}(h_{i,outlet} – h_{i,inlet}) = m_{i,inlet}(h'_{i,outlet} – h_{i,inlet})/\eta_i$

i= C1, P1, P2

For turbine EX1, EX2 and EX3, the energy equation is

$W_j = m_{j,inlet}(hj_{,outlet} – h_{j,inlet})$

$\quad = \eta_j m_{j,inlet}(h_{j,outlet} – h'_{j,inlet})$

j= EX1, EX2, EX3

For heat exchanger HX1 and HX2, the energy equation is

$(mc(hc_{,outlet}\text{-}hc_{,inlet})=mh(hh_{,inlet}\text{-}hh_{,outlet})$

For the fired heater FH, the energy equation is

$\eta_{CMB} mL_6 \mathrm{LHV}+mS_2hS_2+mR_3hR_3=mS_3hS_3+mR_0hR_0$

The efficiency of this power cycle is

$$\eta_{TH} = \frac{\left(\sum W_j - \sum W_i\right)}{\eta_{CMB} m_{L6} LHV}$$
$$i = C1, P1, P2;$$
$$j = EX1, EX2, EX3$$

3.2.2 Exergy balance equation and exergy efficiency

From the thermodynamic point of view, exergy is defined as the maximum of work which can be produced by a system or a flow of a matter or energy as it comes to the equilibrium with a reference environment. Unlike energy, exergy is not subject to a conservation law. Rather, exergy is consumed or destroyed due to irreversibility in the real process. Thermodynamic performance of a process is best evaluated by performing an exergy analysis because exergy analysis appears to provide more insights and to be more useful in efficiency improvement efforts than energy analysis.

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

$e = (h- h_0) - T_0 (s-s_0)$

Overall exergy balance equation is: $E_{in}=E_{out}+E_{loss}$

Where the overall input exergy of the system E_{in} is:

$E_{in}=mL_6 LHV+mL_0 eL_0$

The output exergy of the system E_{out} is

$E_{out} = \Sigma E_j - \Sigma E_i$ i= C1, P1, P2; j= EX1, EX2, EX3

The output exergy Ej for turbine EX1, EX2 and EX3 is

$E_j = m_{j, inlet} (e_{j, inlet} - e_{j, outlet})$ j= EX1, EX2, EX3

The consumed exergy Ei for air compressor C1, pump P1 and P2 is

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

The exergy efficiency of the cascading power cycle is: $\eta = \frac{E_{out}}{E_{in}}$

3.3 PROCESS SIMULATION STEPS:

This section explains the simulation of the LNG regasification with the combined cycle power plant. The simulation is carried out in Aspen Hysys version 2006 environment

3.3.1. Selection of working fluid:

First step in process simulation is the selection of working fluid for Rankine cycle. As indicated in literature review, Amoniac-water is chosen to be the working fluid for the process since it can satisfy many properties required for a Rankine cycle working fluid.

Table 3.1: Required Properties for Selection of Working Fluid

Properties considered as performance measures for the design of Rankine cycle working fluids.			
Thermodynamic	Environmental	Safety	Process-related
1. Density	8. Ozone depletion potential (ODP)	10. Toxicity (C)	12. Efficiency (%)
2. Latent heat of vaporization(Hv)	9. Global warming potential (GWP)	11. Flammability (F)	13. Maximum operating pressure (Pmax)
3. Liquid heat capacity (Cpl)			14. Mass flowrate (mf)
4. Viscosity (μ)			15. Critical pressure (Pc)
5. Thermal conductivity (λ)			
6. Melting point temperature (Tm)			
7. Critical temperature (Tc)			

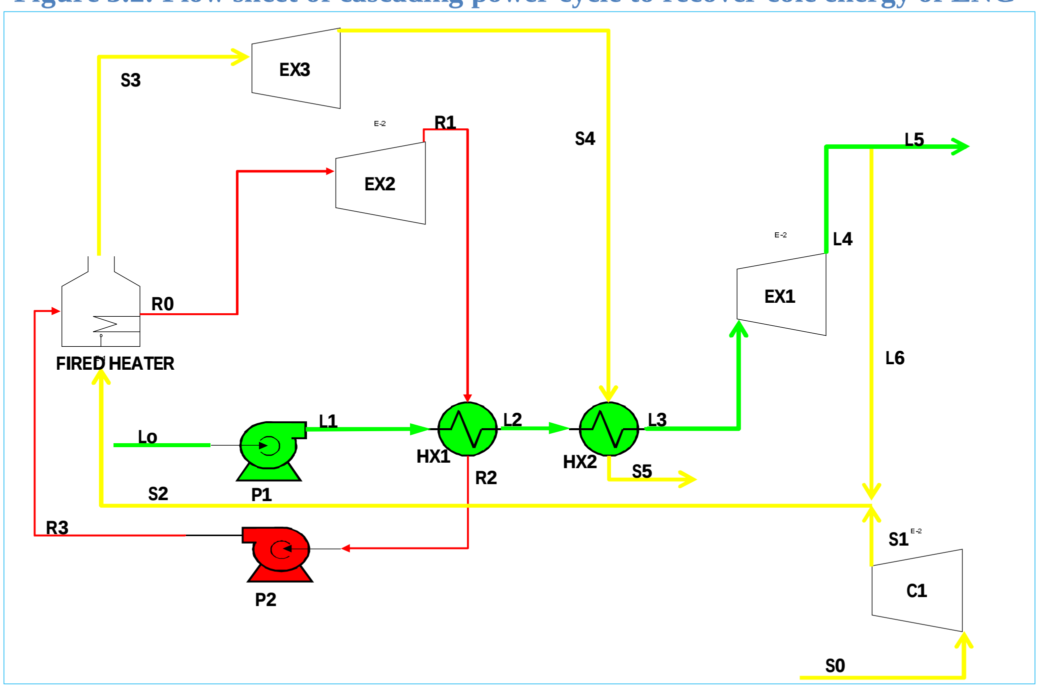
3.3.2 Process scheme and Description:

Second step is construction the process scheme. The cascading power cycle which constructed consists of three cycles: open LNG cycle (L0-P1-L1-HX1-L2-HX2-L3-EX1-L4-L5), Brayton cycle of combustion gas (S0-C1-S1-S2-FH-S3-EX3-S4-HX2-S5), Rankine cycle of ammonia–water (R0-EX2-R1-HX1-R2-P2-R3-FH-R0). Three of these cycles are illustrated in figure 3.2.

Open LNG cycle: LNG with low-temperature of −162°C at atmospheric pressure enters LNG pump P1 to pressurize, and next goes heat exchanger HX1, which is condenser of Rankine cycle of ammonia–water, to be gasified by absorbing heat from outlet gas of turbine EX2, and then comes into heat exchanger HX2 to be superheated by exhaust gas of Brayton cycle, and finally enters turbine EX1 to expand, so natural gas is produced with required pressure according to variable usages of natural gas. A majority part of NG L4 from outlet of turbine EX1 as gas-supplying resource L5 supplies to consumers, and the rest one enters L6 as the fuel of Brayton cycle of combustion gas.

Brayton cycle of combustion gas: air S0 is compressed by compressor C1 and then mixed with NG L6 as fuel S2 to combust in fired heater FH. A part of heat released by combustion is absorbed by ammonia–water of Rankine cycle, and the other one is absorbed by combustion gas itself. Combustion gas S3 with higher temperature and higher pressure goes combustion gas turbine EX3 to do expansion work. Exhaust gas S4 of turbine EX3 with relatively high-temperature entrances heat exchanger HX2 to superheat NG L2. S5 with lower temperature is discharged into the ambient.

Figure 3.2: Flow sheet of cascading power cycle to recover cole energy of LNG



Rankine cycle of ammonia–water: ammonia–water absorbs heat in fired heater FH and converts to high-temperature gas R0, and next does expansion work in turbine EX2. Ammonia–water gas R1 enters heat exchanger HX1 for condensing to preheat LNG L1. Ammonia–water R2 is pumped to fired heater FH to produce high-temperature gas after ammonia–water R1 being cooled to bubble point below in heat exchanger HX1. In the present cascading power cycle, LHV of NG through combustion is heat source both of Brayton cycle of combustion gas and Rankine cycle of ammonia–water. The heat sinks of Rankine cycle of ammonia–water is latent heat and sensible heat of LNG.

3.3.3 Aspen Hysys as System Modeling Tool

After the work done for first and second step, finally Aspen Hysys was chosen to be the tool for simulate the overall process. Aspen Hysys is a Process modeling tool for steady-state simulation, design, performance monitoring, optimization and business planning for chemicals, specialty chemicals, petrochemicals and metallurgy industries. The process simulation capabilities of Aspen Hysys enables engineers to predict the behavior of a process using basic engineering relationships such as mass and energy balances, phase and chemical equilibrium, and reaction kinetics. With reliable thermodynamic data, realistic operating conditions and the rigorous Aspen Hysys equipment models, they can simulate actual plant behavior.

For the present study an attempt has made to simulate LNG cold energy recovery process. The details of process are discussed below.

Problem Specifications

The simulation LNG cold energy recovery as shown in Figure 3.2,using ASPEN HYSYS is prepared according to specification listed in table 3.2

Table 3.2: Calculation parameter of cascading power cycle

Cycle	Items	Unit	Value
Brayton cycle of combustion gas	Pressure ratio of compressor C1	100%	20
	Isentropic efficiency of turbine EX	100%	0.8
	Isentropic efficiency of compressor C	100%	0.8
	Combustion efficiency of fired heater FH	100%	0.99
	Mass flow rate of fuel L6	kg/s	2.792
Rankine cycle of ammonia–water	Isentropic efficiency of turbine EX2 100% 0.8	100%	0.8
	Efficiency of pump P2 100% 0.7	100%	0.7
	Ammonia–water concentration in molar	100%	0.7
	Mass flow rate of working fluid R0	kg/s	31.187
Open LNG Cycle	Open LNG cycle Inlet temperature of LNG pump P1	oC	-162

	Inlet pressure of LNG pump P1	Mpa	0.1
	Efficiency of Pump P1	100%	0.7
	Isentropic efficiency of turbine EX1	100%	0.8
	Mass flowrate of LNG	kg/s	96.262
Others	Pinch point temperature difference	oC	10
	Ambient temperature	oC	20
	Ambient pressure	Mpa	0.1
	Mass flow rate of air S0	kg/s	46.737

Aspen Hysys Process Flow Diagram

To represent above LNG cold energy recover process in Aspen Hysys the first step is to make a process flow diagram (PFD). In Simulation Basic Manager a fluid package is to be selected along with the fluid which is to be cycled in the process. Peng Robinson and UNIQUAC are chosen to be fluid packages. Now using an option "Enter to simulation Environment" PFD screen is started. An object palette will appear at right hand side of the screen.

In the object palette a number of components available some are given below.

i. Streams (Material/Energy streams)

ii. Vessels (Separator and Tanks)

iii. Heat Transfer Equipments (Heat exchanger, Valves)

iv. Rotating Equipments.

v. Piping Equipments.

vi. Solid Handling.

vii. Reactor.

viii. Logical.

ix. Sub Flow sheet.

x. Short Cut Column.

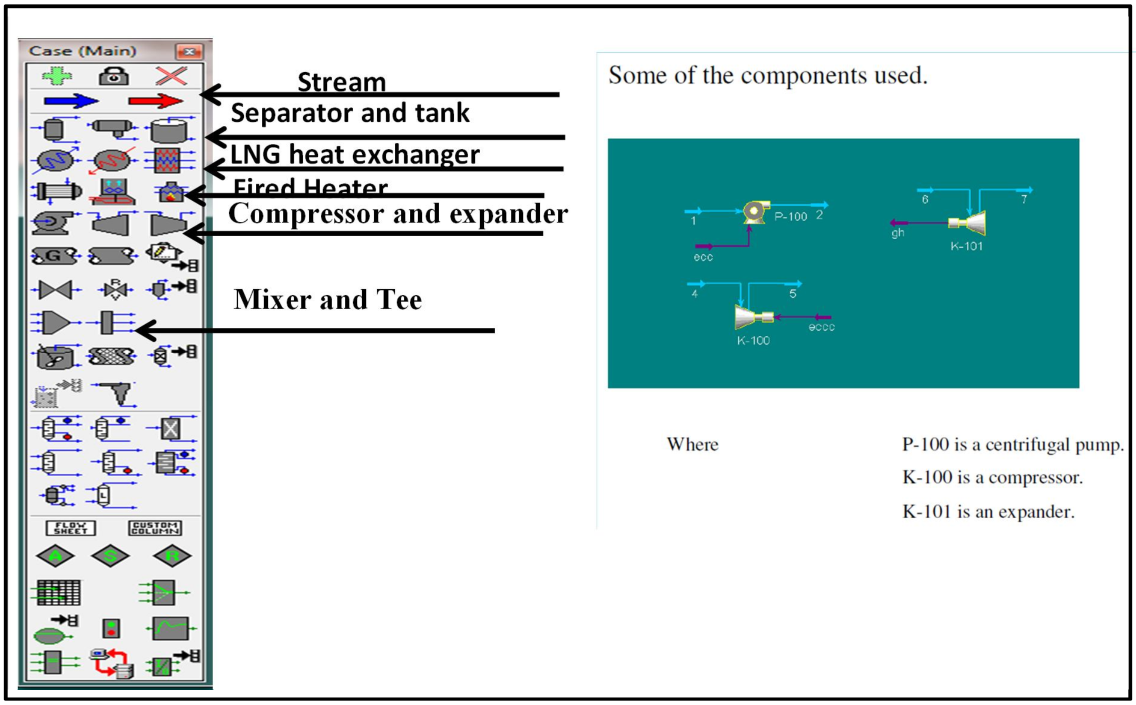


Figure 3.3: Object Palette and Hysys component used example

For every component certain inputs are supplied as constraints for the component operation. Some of the specifications are:

1. Streams mass flow rate, temperature, and pressure.

2.Compressor compression ratio, inlet and outlet streams, duty factor, adiabatic efficiency.

3. Expander adiabatic efficiency, inlet and outlet streams, and work outputs.

After constraining every component properly the specific window gives a green signal and then Simulation is started.

Following figure shows a solved Process Flow Diagram by Aspen Hysys:

Detail information of each stream is showed in Appendix 1

CHAPTER 4
RESULTS AND DISCUSSION

4.1 RESULTS OF PARAMETER ANALYSIS

According to energy equilibrium equations and exergy equilibrium equations of the cascading power cycle, taken condensation inlet pressure pR0 and outlet pressure pR1 of turbine EX2 of Rankine cycle of ammonia–water, inlet pressure pL3 and outlet pressure pL4 of turbine EX1 of open LNG cycle as key parameters, influences of these parameters on thermal efficiency, exergy efficiency and power yield of the cascading power cycle are analyzed.

Changes of thermal and exergy efficiencies of the cascading power cycle with inlet pressure of turbine EX2 of Rankine cycle are shown in Fig4.1. Thermal and exergy efficiencies of the cascading power cycle increase with the increasing of inlet pressure of turbine EX2 resulting in increasing the overall power yield of the process as shown in fig.4.2. The increase in inlet pressure of turbine EX2 just has a small impact on the change in thermal and exergy efficiency and a fast increase in power yield. Although an increase of inlet pressure of turbine EX2 supplied by pump P2 needs consuming more pump work, an increase of power produced by turbine EX2 is more than that of work consumed by pump P2 resulting in an increase of thermal and exergy efficiencies of the cascading power cycle.

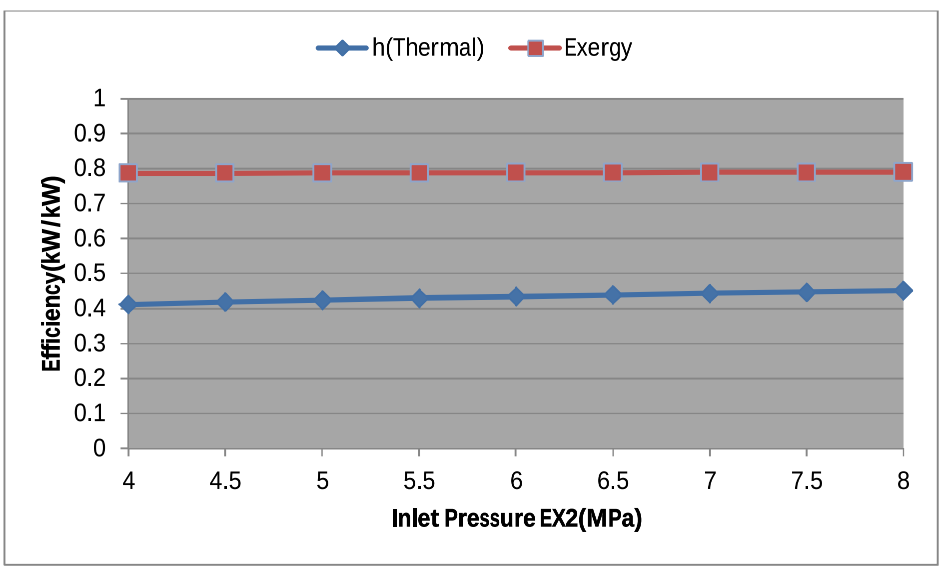


Figure 4.1: Influence of inlet pressure pR0 of turbine EX2 of Rankine cycle on thermal efficiency and exergy efficiency

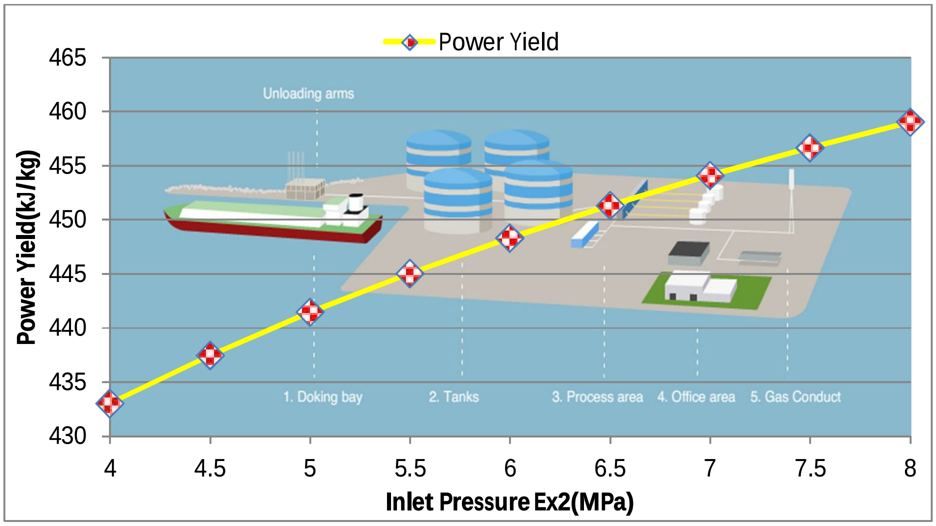


Figure 4.2: Influence of inlet pressure pR0 of turbine EX2 of Rankine cycle on power yield

Fig.4.3 shows that the changes of thermal and exergy efficiencies of the cascading cycle with outlet pressure of turbine EX2. Thermal and exergy efficiencies of the cascading power cycle decrease with the increasing of outlet pressure of turbine EX2 because higher outlet pressure of turbine EX2 is, lower output of power of turbine EX2 is. This leads to the decrease in overall power yield of the process as shown in fig.4.4.

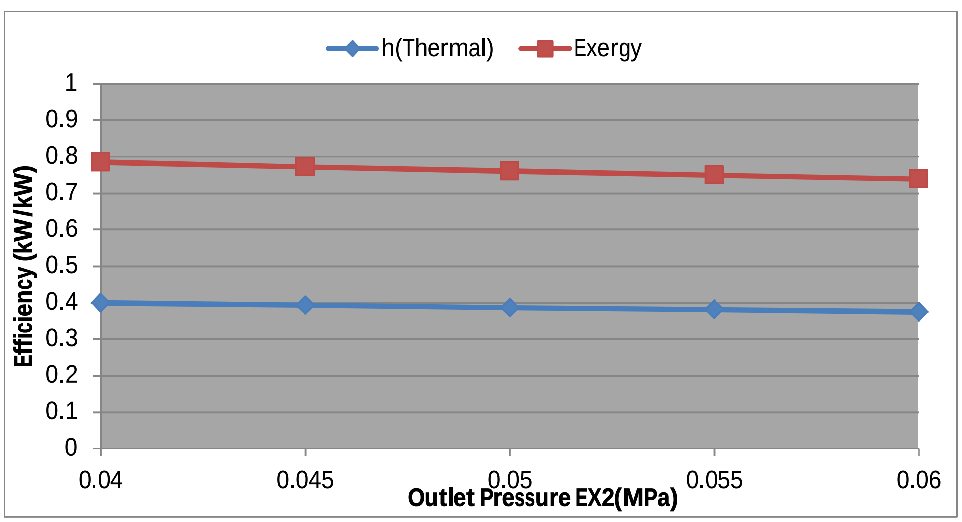


Figure 4.3: Influence of outlet pressure pR1 of turbine EX2 of Rankine cycle on thermal efficiency and exergy efficiency

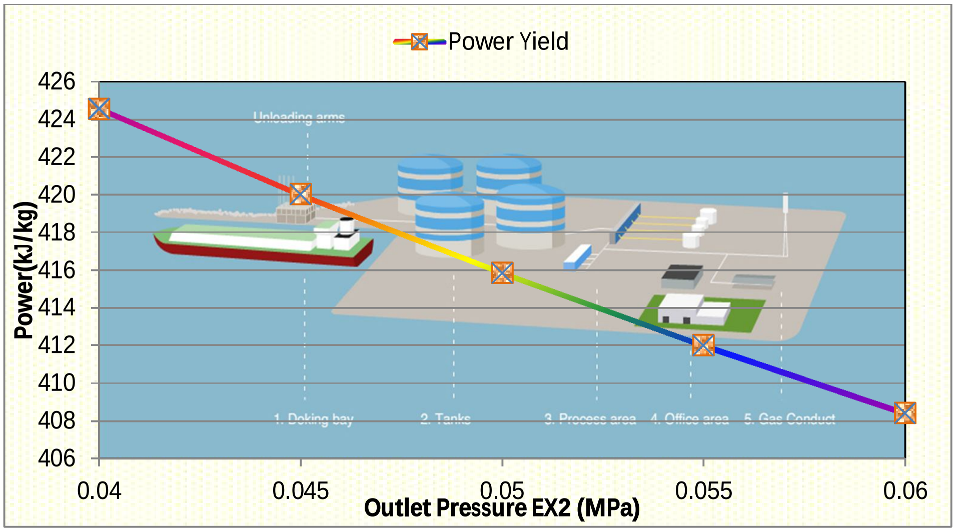


Figure 4.4: Influence of outlet pressure pR1 of turbine EX2 of Rankine cycle on Power Yield

The changes of thermal and exergy efficiencies of the cascading power cycle with inlet pressure of turbine EX1 of open LNG cycle are shown in Fig.4.5. Although an increase of inlet pressure of turbine EX1 supplied by pump P1 needs consuming more pump work, an increase of power produced by turbine EX1 is more than that of work consumed by pump P1 resulting in an increase of thermal and exergy efficiencies of the cascading power cycle as well as the overall power yield shown in figure Fig.4.6.

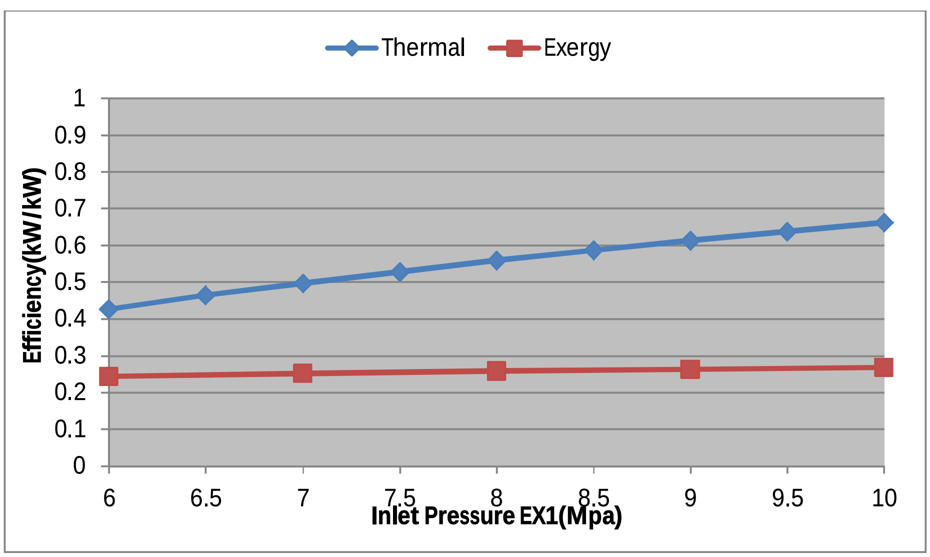


Figure 4.5: Influence of inlet pressure pL3 of turbine EX1 of open LNG cycle on thermal efficiency and exergy efficiency

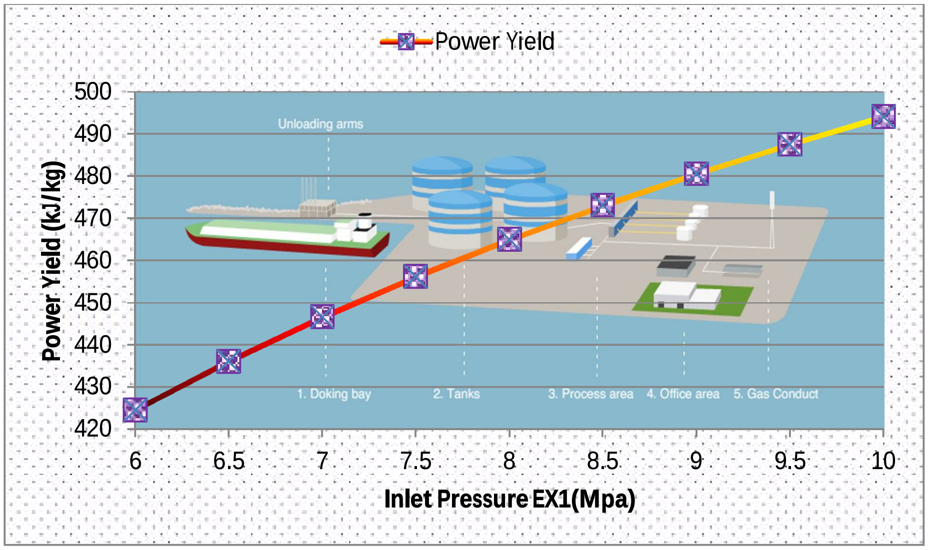


Figure 4.6: Influence of inlet pressure pL3 of turbine EX1 of open LNG cycle on power yield

Fig.4.7 describes the changes of thermal and exergy efficiencies of the cascading power cycle without pressure of turbine EX1 of open LNG cycle. The power generated by turbine EX1 decreases with the increasing of outlet pressure of turbine EX1 resulting in decreasing of thermal and exergy efficiencies of the cascading power cycle. Fig.4.8 below shows the decreasing in outlet pressure of EX1 also cause the decreasing in overall power yield of the process..

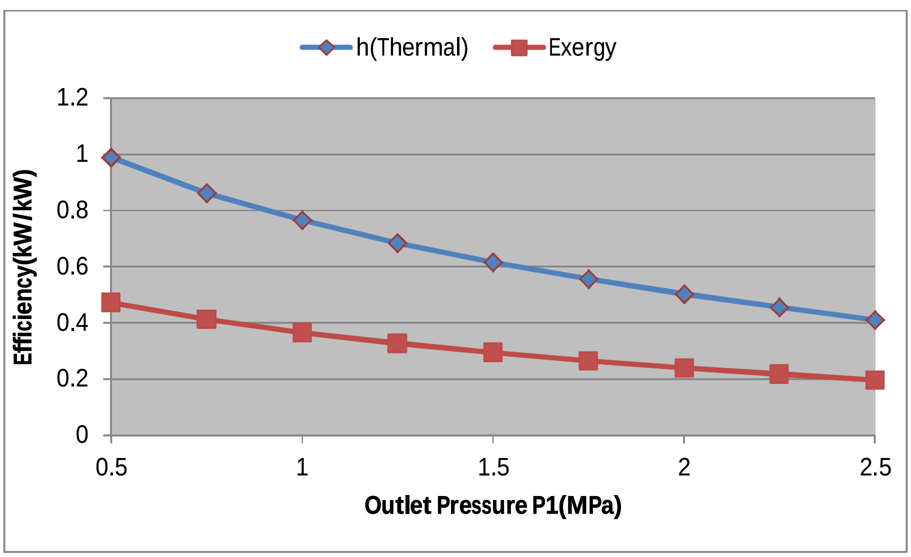


Figure 4.7: Influence of outlet pressure pL4 of turbine EX1 of open LNG cycle on thermal efficiency and exergy efficiency.

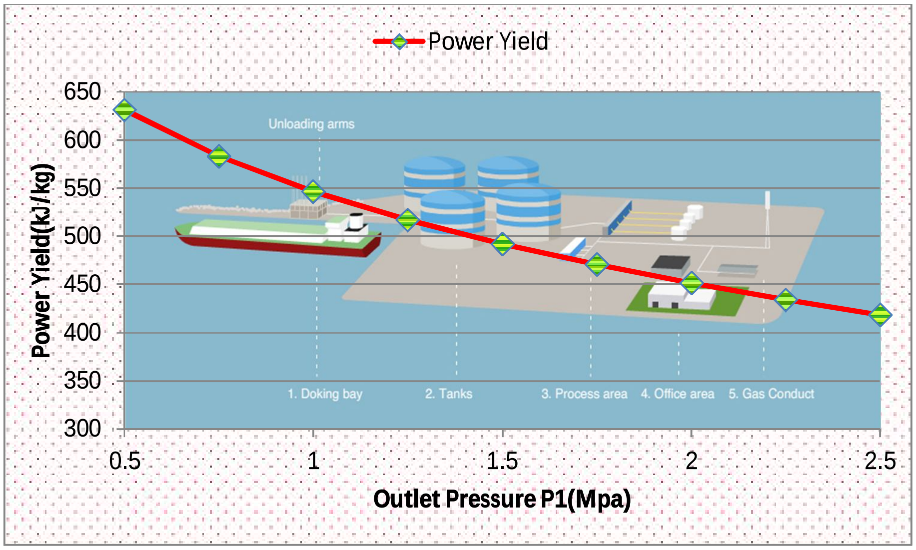


Figure 4.8: 1Influence of outlet pressure pL4 of turbine EX1 of open LNG cycle on power yield

However, higher thermal and exergy efficiencies could not be obtained by decreasing of outlet pressure of turbine EX1 due to the limitation of outlet pressure of turbine EX1 which is gas-supply pressure, e.g. 0.5 MPa for short-line gas-supply and 2.0 MPa for long-line gas-supply.

4.2 OPTIMISATION MODEL

The results of performance analysis show that the thermal and exergy efficiencies increase with the increasing of inlet pressure of turbine EX2 of Rankine cycle, inlet pressure of turbine EX1 of open LNG cycle, and with the decreasing of outlet pressure of turbine EX2 of Rankine cycle, outlet pressure of turbine EX1 of open LNG cycle. However, the result of the influence of one of these parameters on thermal and exergy efficiencies is achieved by fixing the other parameters. The increase of thermal and exergy efficiencies with some parameters increased and with the others decreased leads to an optimal solution. The maximum economy benefit is pursued for a real LNG terminal. In this section, based on the flow simulation of the cascading power cycle, optimization model of the cascading power cycle with maximum economic benefits as objective function together with optimum variables such as condensation temperature of heat exchanger in Rankine cycle, minimum temperature difference of heat exchanger between outlet temperature of hot stream and inlet temperature of cold stream in power cycle of combustion gas, pump pressure ratio both of open LNG cycle and Rankine cycle, and expander pressure ratio of open LNG cycle was established

4.2.1 Objective function

Maximum economy benefits as objective function is given by

Max EB = ($\Sigma W_j - \Sigma W_i$) x C_p

where C_p = 0.0896 \$/kWh

The objective function is obtained based on the principle maximization work of expanders EX1 and minimize cost of compressor C1 and pumps P1, P2.

For the compressor:

In industrial compressors, the compression path will be polytropic where Pv^n = constant (P=pressure, v=volume). The work required is given by a general expression;

$$-W = ZRT_1\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

where;

W = Compressor work (kJ/hr)
Z = Compressibility factor

R = Gas constant (kJ/kgmol.K)

T_1 = Inlet temperature of the stream (K)

P_1 = Initial pressure (bar)

P_2 = Final pressure (bar)

n = Cp/Cv

With all the necessary constant values such as Z, R and γ can easily obtain from Hysys, we can built the equation between W and P2/P1 easily.

For Pump and expander, due to the complicated equation for expression relationship between work consumed/generated, the author has chosen the graph method to construct the equation with can see from two of figures below:

For Expander 1(EX1):

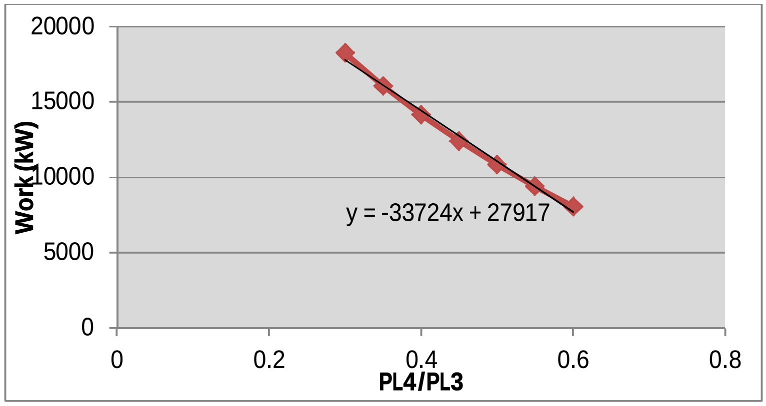


Figure 4.9: Expander Work.vs pressure outlet and inlet ratio

For Pump1 (P1):

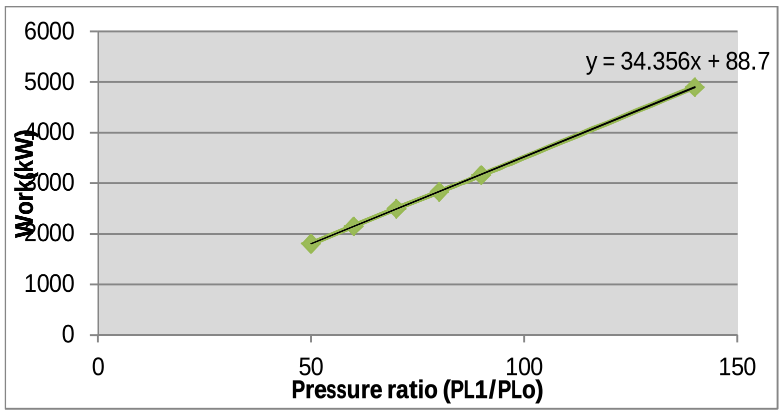


Figure 4.10: Pump1 Work.vs pressure outlet and inlet ratio

For Pump2 (P2):

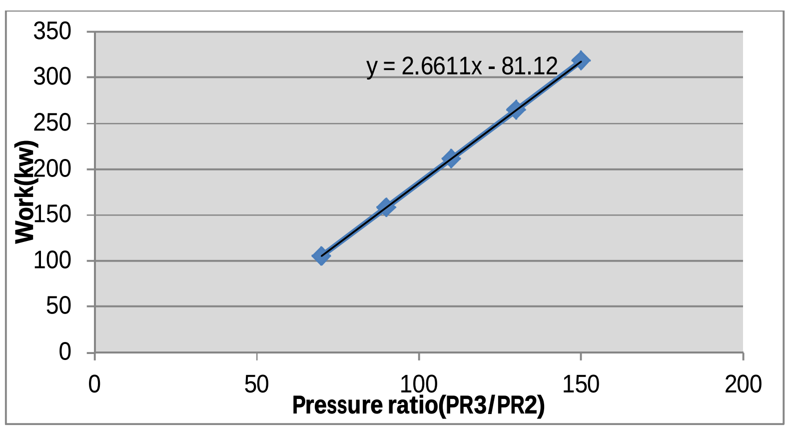


Figure 4.11: Pump2 Work. Vs. pressure outlet and inlet ratio

Finally, the obtained objective function is:

Max EB=-33724x_1 -2.6611x_2 - 34.356x_3 + 148983,

where x_1, x_2 and x_3 is the pressure ratio of Expander 1, Pump2 and Pump1 respectively.

4.2.2 Equality constraint conditions and inequality constraint conditions

The equality constraint conditions are mass and energy balance equations. The inequality constraint conditions by choosing the parameter of hot side temperature T_{R2} of heat exchanger HX1, minimum temperature approach between hot outlet and cold inlet of heat exchanger HX2, pressure ratio of pump P1, pressure ratio of pump P2, and pressure ratio of turbine EX1 as decision variables are given by

$2 \leqslant T_{R2}-T_{R2,BP} \leqslant 20$, this equation makes sure stream R2 in liquid state for pump transport. $T_{R2,BP}$ is bubble point temperature of ammonia–water.

$2 \leqslant T_{S5}-T_{L2} \leqslant 10$, this equation makes insures the logic of heat transfer in heat exchanger correct.

$$50 \leq \frac{P_{L1}}{P_{Lo}} \leq 150\ ; \qquad 50 \leq \frac{P_{R3}}{P_{R2}} \leq 150; \qquad 50 \leq \frac{P_{L4}}{P_{L3}} \leq 150$$

4.2.3. Results of optimization

Table 4.1: Comparison of state parameter between initial value and optimum value

Cycle	Points	Temperature (°C)		Pressure (MPa)	
		Initial	Optimal	Initial	Optimal
Open LNG cycle	L0	-162	-162	0.1	0.1
	L1	-158.4	-158.9	6	5.25
	L2	12.66	12.64	6	5.25
	L3	167.5	173	6	5.25
	L4, L5,L6	104.6	96.7	2.4	1.7
Rankine cycle of ammoniac of water	R0	526.7	521.7	3.2	2.29
	R1	178.3	194.5	0.04	0.04
	R2	-49.3	-49.8	0.04	0.04
	R3	-48.9	-49.52	3.2	2.29
Brayton cycle of combustion gas	S0	20	20	0.1	0.1
	S1	517.5	517.5	2	2
	S2	458.6	457.7	2	1.7
	S3	990	990	2	1.7
	S4	499.3	491.4	0.1	0.1
	S5	22.3	18.2	0.1	0.1

Table 4.2: Comparison of initial value with optimal value

Variable	Initial value	Optimal value
$T_{R_2} - T_{R2,BP} /^{o} C$	10.25	10.51
$T_{S5} - T_{L_2} /^{o} C$	7.6	5.92
$\frac{P_{L1}}{P_{Lo}} / 100\%$	53.43	52.4
$\frac{P_{R3}}{P_{R2}} / 100\ \%$	60.29	57.2
$\frac{P_{L4}}{P_{L3}} / 100\ \%$	33.43	32.4

The optimization model is solved on the platform of Excel Solver the obtained objective function as mentioned above.

State parameters of cascading power cycle are listed in table 4.1 Most of state parameters change from initial value to optimal value. The changes of most of state parameter owe to the results of optimization of decision variables listed in table 4.2. As listed in table 4.3,the decrease of work consumed by pumps' P1 and P2 and the increase of work generated by expander EX1 contribute to the increase of economy benefit at the cost of the decrease of work generated by expanders' EX2 and EX2.

Table 4.3: Comparison of work consumed by compressor and pumps and generated by expanders

Name	**Work(kW)**	
	Initial	Optimal
C1	24720.54	24720.54
P1	1858.185	1621.975
P2	168.6521	120.0094
EX1	14011.37	17261.36
EX2	26086.32	24761.14
EX3	6825.102	6539.484
Net power	20175.41	22099.45

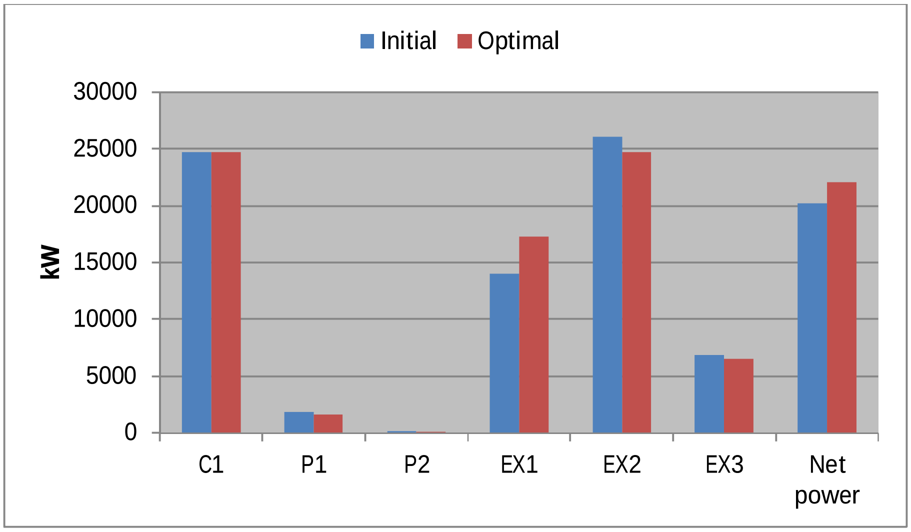


Figure 4.12: Comparison of work consumed by compressor and pumps and generated by expanders.

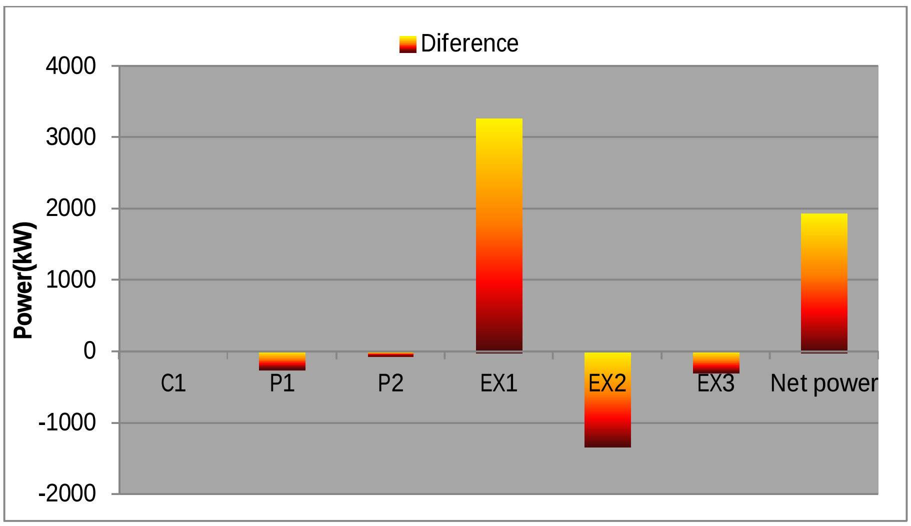


Figure 4.13: The power value differences between initial and optimal state of each component

The comparison of work consumed by compressor and pumps and generated by expanders between initial situation and optimal situation are completed in figure 4.12 and figure 4.13 above. As shown in figure 4.13, there is no change in the work consumed by compressor C1, the work consumed by pump P1 is less in optimal state. It's same as to Pump P2 but the difference of work consumed in optimal and initial value of pump P2 is not significant. Here, we can see that both work generated by expander EX2 and expander EX3 are decreased after optimization however due to the increase of work generated by expander EX1 in open LNG cycle, finally we still have the net power overall increases after optimization and it's the most expected result. From this, can say that expander EX1 plays most part of the whole contribution to the economy benefit.

CHAPTER 5
CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSION

A cascading power cycle with LNG cold energy recovery consisting of Rankine cycle using ammonia–water as working fluid, Brayton power cycle of combustion gas, open LNG cycle, was successfully simulated by Hysys software. The effect of key parameters on the energy and exergy efficiency are investigated are analyzed. Thermal and exergy efficiencies increase with the increasing inlet pressure of turbines of Rankine cycle and open LNG cycle, and fall with the increasing of outlet pressure of turbines of Rankine cycle and open LNG cycle. The optimization model with maximum economy benefits as objective function and hot side temperature of heat exchanger in Rankine cycle, minimum temperature approach between hot outlet and cold inlet of heat exchanger in Brayton cycle, pressure ratio of pump in open LNG cycle, pressure ratio of pump in Rankine cycle, and pressure ratio of turbine in Open LNG cycle as decision variables is proposed. The result of optimization shows that the economy benefit increase owes to the increasing of work generated by expander in open LNG cycle and the decrease of work consumed by compressor and pump in Brayton cycle and Rankine cycle. The increase of work generated by expander in Open LNG cycle plays most part of the whole contribution to the economy benefit.

5.2 FUTURE SCOPE

The results obtained from simulation will help to carry out experiments in lab at optimum condition. The system can also be enhanced by changing the data input into the system, operating condition as closed to reality plant as better. Also, current simulation does not use fire heater directly into process but separated into combustion and heat exchanger so the data accuracy obtained may be limited. For the future work, if this process can be simulated in dynamic mode, not static mode like current work due to limited time, the fire heater can run well in Hysys for higher accuracy which may meet the requirement of industry plant.

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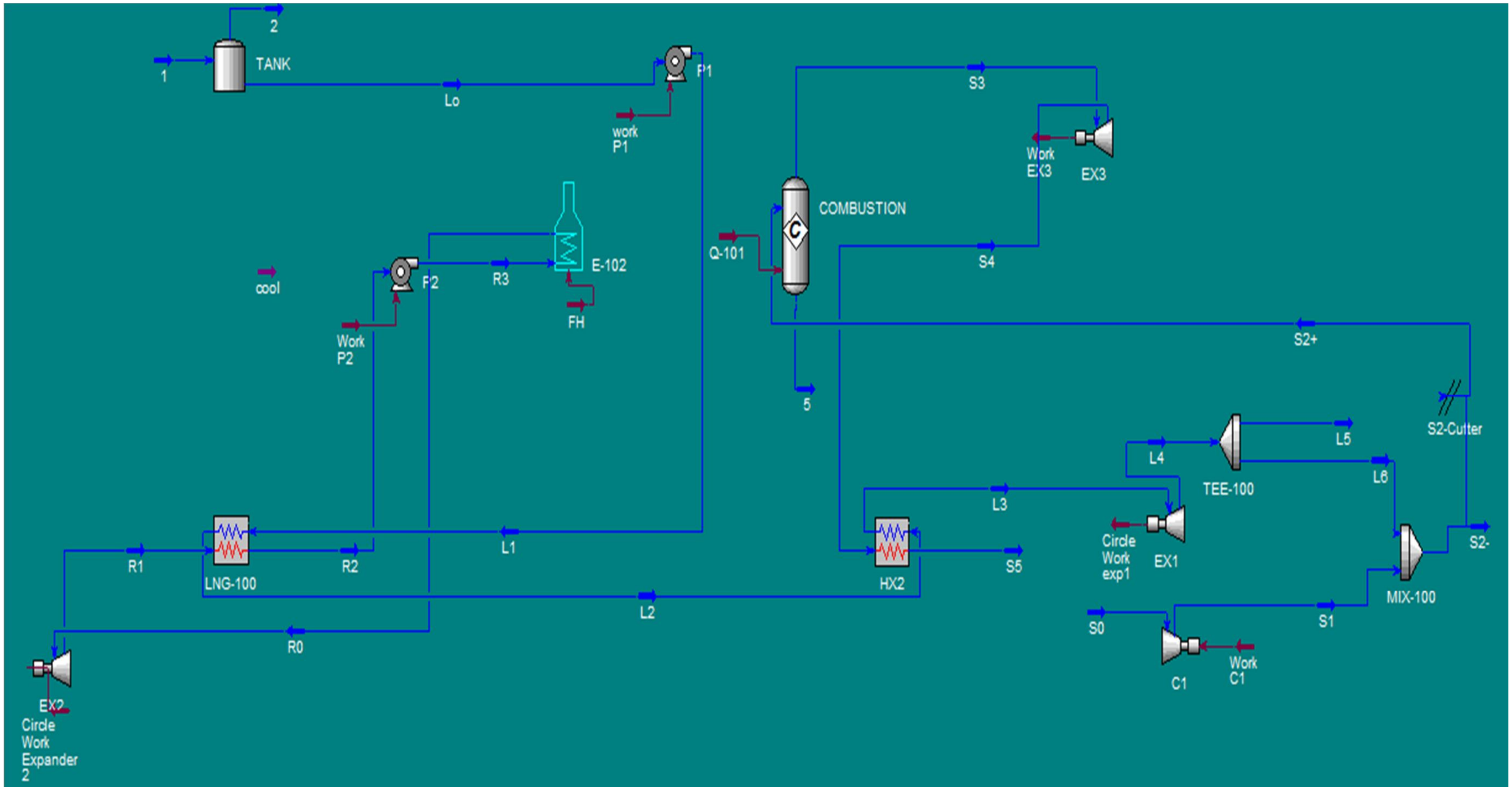
APPENDICES



Appendix 1: Hysys Simulation overall process

Name	1	2	Lo	L1	L2	S0	S1	R2	R3	R0	R1	L3
Vapour Fraction	0.0000	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000
Temperature [C]	-162.0	-162.0	-162.0	-158.4	12.66	20.00	517.5	-49.30	-48.90	526.7	178.3	167.5
Pressure [kPa]	100.0	100.0	100.0	6000	6000	100.0	2000	40.00	3200	3200	40.00	6000
Molar Flow [kgmole/h]	2.075e+004	0.0000	2.075e+004	2.075e+004	2.075e+004	5832	5832	6480	6480	6480	6480	2.075e+004
Mass Flow [kg/h]	3.500e+005	0.0000	3.500e+005	3.500e+005	3.500e+005	1.683e+005	1.683e+005	1.123e+005	1.123e+005	1.123e+005	1.123e+005	3.500e+005
Liquid Volume Flow [m3/h]	1146	0.0000	1146	1146	1146	194.5	194.5	160.5	160.5	160.5	160.5	1146
Heat Flow [kJ/h]	-1.883e+009	0.0000	-1.883e+009	-1.876e+009	-1.604e+009	-8.981e+005	8.810e+007	-9.136e+008	-9.129e+008	-5.472e+008	-6.412e+008	-1.459e+009
Name	L4	L5	L6	S4	S5	S2-	LNG Feed	L11	L22	REGAS FURN/	REGAS EXPAN	SALES GAS
Vapour Fraction	1.0000	1.0000	1.0000	1.0000	<empty>	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000	1.0000
Temperature [C]	104.6	104.6	104.6	499.3	<empty>	458.6	-162.0	-158.8	12.30	167.5	109.2	109.2
Pressure [kPa]	2400	2400	2400	100.0	100.0	2000	100.0	6000	6000	6000	2400	2400
Molar Flow [kgmole/h]	2.075e+004	2.016e+004	592.8	1213	1213	6425	2.075e+004	2.075e+004	2.075e+004	2.075e+004	2.075e+004	2.016e+004
Mass Flow [kg/h]	3.500e+005	3.400e+005	1.000e+004	2.190e+004	2.190e+004	1.783e+005	3.500e+005	3.500e+005	3.500e+005	3.500e+005	3.500e+005	3.400e+005
Liquid Volume Flow [m3/h]	1146	1113	32.75	21.98	21.98	227.2	1146	1146	1146	1146	1146	1113
Heat Flow [kJ/h]	-1.509e+009	-1.466e+009	-4.312e+007	-2.703e+008	-4.150e+008	4.497e+007	-1.883e+009	-1.876e+009	-1.604e+009	-1.459e+009	-1.505e+009	-1.462e+009
Name	FURNACE GAS	GAS TO FURN.	AIR	MIXED GAS	5-2	To EX3	S2+	5	S3	8	To HE2	ref stream
Vapour Fraction	1.0000	1.0000	1.0000	1.0000	0.0000	1.0000	1.0000	0.0000	1.0000	1.0000	1.0000	1.0000
Temperature [C]	109.2	105.0	494.0	438.8	966.1	966.1	458.6	990.0	990.0	443.8	542.0	25.00
Pressure [kPa]	2400	2327	1805	1805	2000	2000	2000	2000	2000	2000	100.0	101.3
Molar Flow [kgmole/h]	592.8	592.8	5832	6425	5899	1228	6425	5230	1213	7109	1228	2.075e+004
Mass Flow [kg/h]	1.000e+004	10000.	1.683e+005	1.783e+005	1.758e+005	2.220e+004	1.783e+005	1.564e+005	2.190e+004	1.980e+005	2.220e+004	3.500e+005
Liquid Volume Flow [m3/h]	32.75	32.75	194.5	227.2	215.7	22.29	227.2	193.3	21.98	250.1	22.29	1146
Heat Flow [kJ/h]	-4.301e+007	-4.311e+007	8.368e+007	4.058e+007	-6.173e+007	-2.496e+008	4.494e+007	-7.507e+007	-2.457e+008	5.037e+007	-2.711e+008	-1.567e+009
Name	ro	r11	r22	r33	amoniac ref stre	to atmosphere	** New **					
Vapour Fraction	1.0000	1.0000	0.0000	0.0000	0.6030	1.0000						
Temperature [C]	496.9	157.3	-59.89	-59.52	25.00	<empty>						
Pressure [kPa]	3200	40.00	40.00	3200	101.3	100.0						
Molar Flow [kgmole/h]	6480	6480	6480	6480	9711	1228						
Mass Flow [kg/h]	1.123e+005	1.123e+005	1.123e+005	1.123e+005	1.683e+005	2.220e+004						
Liquid Volume Flow [m3/h]	160.5	160.5	160.5	160.5	240.5	22.29						

Name	ro	r11	r22	r33	amoniac ref stre	to atmosphere	** New **					
Vapour Fraction	1.0000	1.0000	0.0000	0.0000	0.6030	1.0000						
Temperature [C]	496.9	157.3	-59.89	-59.52	25.00	<empty>						
Pressure [kPa]	3200	40.00	40.00	3200	101.3	100.0						
Molar Flow [kgmole/h]	6480	6480	6480	6480	9711	1228						
Mass Flow [kg/h]	1.123e+005	1.123e+005	1.123e+005	1.123e+005	1.683e+005	2.220e+004						
Liquid Volume Flow [m3/h]	160.5	160.5	160.5	160.5	240.5	22.29						
Heat Flow [kJ/h]	-5.565e+008	-6.464e+008	-9.189e+008	-9.183e+008	-1.172e+009	-4.161e+008						

Appendix 2: Stream data before optimization

Name	1	2	Lo	L1	L2	S0	S1	R2	R3	R0	R1	L3
Vapour Fraction	0.0000	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000
Temperature [C]	-162.0	-162.0	-162.0	-158.9	12.70	20.00	517.5	-49.80	-49.52	522.0	194.7	167.5
Pressure [kPa]	100.0	100.0	100.0	5250	5250	100.0	2000	40.00	2290	2290	40.00	5250
Molar Flow [kgmole/h]	2.075e+004	0.0000	2.075e+004	2.075e+004	2.075e+004	5832	5832	6480	6480	6480	6480	2.075e+004
Mass Flow [kg/h]	3.500e+005	0.0000	3.500e+005	3.500e+005	3.500e+005	1.683e+005	1.683e+005	1.123e+005	1.123e+005	1.123e+005	1.123e+005	3.500e+005
Liquid Volume Flow [m3/h]	1146	0.0000	1146	1146	1146	194.5	194.5	160.5	160.5	160.5	160.5	1146
Heat Flow [kJ/h]	-1.883e+009	0.0000	-1.883e+009	-1.877e+009	-1.600e+009	-8.981e+005	8.810e+007	-9.138e+008	-9.134e+008	-5.479e+008	-6.371e+008	-1.457e+009

Name	L4	L5	L6	S4	S5	S2-	LNG Feed	L11	L22	REGAS FURN/	REGAS EXPAN	SALES GAS
Vapour Fraction	1.0000	1.0000	1.0000	1.0000	<empty>	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000	1.0000
Temperature [C]	101.9	101.9	101.9	499.3	<empty>	458.4	-162.0	-158.8	12.30	167.5	109.2	109.2
Pressure [kPa]	2000	2000	2000	100.0	100.0	2000	100.0	6000	6000	6000	2400	2400
Molar Flow [kgmole/h]	2.075e+004	2.016e+004	592.8	1213	1213	6425	2.075e+004	2.075e+004	2.075e+004	2.075e+004	2.075e+004	2.016e+004
Mass Flow [kg/h]	3.500e+005	3.400e+005	1.000e+004	2.190e+004	2.190e+004	1.783e+005	3.500e+005	3.500e+005	3.500e+005	3.500e+005	3.500e+005	3.400e+005
Liquid Volume Flow [m3/h]	1146	1113	32.75	21.98	21.98	227.2	1146	1146	1146	1146	1146	1113
Heat Flow [kJ/h]	-1.511e+009	-1.467e+009	-4.316e+007	-2.703e+008	-4.129e+008	4.494e+007	-1.883e+009	-1.876e+009	-1.604e+009	-1.459e+009	-1.505e+009	-1.462e+009

Name	FURNACE GAS	GAS TO FURN.	AIR	MIXED GAS	5-2	To EX3	S2+	5	S3	8	To HE2	ref stream
Vapour Fraction	1.0000	1.0000	1.0000	1.0000	0.0000	1.0000	1.0000	0.0000	1.0000	1.0000	1.0000	1.0000
Temperature [C]	109.2	105.0	494.0	438.8	966.1	966.1	458.4	990.0	990.0	443.8	542.0	25.00
Pressure [kPa]	2400	2327	1805	1805	2000	2000	2000	2000	2000	2000	100.0	101.3
Molar Flow [kgmole/h]	592.8	592.8	5832	6425	5899	1228	6425	5230	1213	7109	1228	2.075e+004
Mass Flow [kg/h]	1.000e+004	10000.	1.683e+005	1.783e+005	1.758e+005	2.220e+004	1.783e+005	1.564e+005	2.190e+004	1.980e+005	2.220e+004	3.500e+005
Liquid Volume Flow [m3/h]	32.75	32.75	194.5	227.2	215.7	22.29	227.2	193.3	21.98	250.1	22.29	1146
Heat Flow [kJ/h]	-4.301e+007	-4.311e+007	8.368e+007	4.058e+007	-6.173e+007	-2.496e+008	4.490e+007	-7.507e+007	-2.457e+008	5.037e+007	-2.711e+008	-1.567e+009

Name	ro	r11	r22	r33	amoniac ref stre	to atmosphere	** New **					
Vapour Fraction	1.0000	1.0000	0.0000	0.0000	0.6030	1.0000						
Temperature [C]	496.9	157.3	-59.89	-59.52	25.00	<empty>						
Pressure [kPa]	3200	40.00	40.00	3200	101.3	100.0						
Molar Flow [kgmole/h]	6480	6480	6480	6480	9711	1228						
Mass Flow [kg/h]	1.123e+005	1.123e+005	1.123e+005	1.123e+005	1.683e+005	2.220e+004						
Liquid Volume Flow [m3/h]	160.5	160.5	160.5	160.5	240.5	22.29						

Name	ro	r11	r22	r33	amoniac ref stre	to atmosphere	** New **					
Vapour Fraction	1.0000	1.0000	0.0000	0.0000	0.6030	1.0000						
Temperature [C]	496.9	157.3	-59.89	-59.52	25.00	<empty>						
Pressure [kPa]	3200	40.00	40.00	3200	101.3	100.0						
Molar Flow [kgmole/h]	6480	6480	6480	6480	9711	1228						
Mass Flow [kg/h]	1.123e+005	1.123e+005	1.123e+005	1.123e+005	1.683e+005	2.220e+004						
Liquid Volume Flow [m3/h]	160.5	160.5	160.5	160.5	240.5	22.29						
Heat Flow [kJ/h]	-5.565e+008	-6.464e+008	-9.189e+008	-9.183e+008	-1.172e+009	-4.161e+008						

Appendix 3: Stream data after optimization

Appendix 4: GANT CHART and key milestone

The detailed Gantt chart for both FYP1 and FYP2 are shown in **Table 1** and **Table 2** below.

Table 1: Gantt chart for FYP1

No	Details/Week	1	2	3	4	5	6	Midterm Break	7	8	9	10	11	12	13	14
1	Selection of Project Topic															
2	Literature Review															
3	Project Work															
	3.1 Selection of working fluid															
	3.2 Operation process data Gathering															
4	Results/Analysis															
	4.1 Process Simulation															
	4.2 Perform thermodynamic equation model															
5	Reporting															
	5.1 Preliminary Report															
	5.2 Extended Proposal															
	5.3 Seminar															
	5.4 Interim report															

Table 2: Gantt chart for FYP2

No.	Detail/Week	1	2	3	4	5	6	Midterm Brea	7	8	9	10	11	12	13	14
1	Implementation & Development															
2	Result Analysis and optimization process															
3	Submission of Progress Report															
4	Pre-SEDEX															
5	Dissertation															
6	Viva: Oral Presentation															
7	Submission of Interim Report															

Table 3: Key Milestone for FYP2

Week	FYP2 Activities	Date
1-10	Implementation & System Development	-
7	Submission of Progress Report	5th November
10	Pre-SEDX	26th November
11	Dissertation	10th December
13	Viva: Oral Presentation	20th December
14	Final Dissertation	11th January