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Performance Improvement of Propane Refrigeration Unit in a Liquefied Natural Gas Plant

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ABSTRACT

The aim of this study is to improve the performance of the propane refrigeration unit for precooling in a liquefied natural gas plant. Energy and exergy methods were used to analyze the performance of the unit. Aspen HYSYS v10 and Microsoft excel v2016 were used to carry out the simulation and analysis. Results obtained show that the base scenario has a mass flow rate of 17509kg/h. However, sensitivity analysis results show that decrease in pressure from the expansion valves causes a decrease in temperature of the propane refrigerant which in turn causes an increase in the mass flow rate of the propane refrigerant from 17509kg/h to 17814kg/h. Further results obtained considering the base scenario on the effect of the operating parameter on the compressor power show that the compressor power obtained is 761.3 KW. However, sensitivity and graphical analysis show that decrease in pressure from the expansion valves causes a decrease in temperature of the refrigerant which in turn causes an increase in the mass flow rate of the refrigerant. This increase in the mass flow rate of the refrigerant makes the compressor to do less work thereby reducing its power consumption from 761.3 KW to 678.3 KW, the cooler duty from 135KW to 125KW, exergy loss in the air cooler increases from 12.74KW to 11.77KW, the exergy loss in the compressor increases from 27.45KW to 25.35KW, the exergy loss in the heat exchanger decreases from 15.1KW to 14.3KW, and the expansion valve's energy loss goes from 28.66KW to 33.52KW. The results of exergy efficiency for the propane pre-cooling cycle reveal that Scenario 5 have the lowest exergy efficiency (34.67%), which suggests greater irreversibilities within the process. The process has a strong potential for improvement since Scenario 1 has the best exergy efficiency (40.03%) and COP rises by 20% compared to Scenario 5. The findings indicate that altering the propane evaporator's working parameters causes a larger creation of entropy, which lowers the exergy efficiency.

KEYWORDS: Exergy, propane, precooling, performance, refrigeration.

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1.0 INTRODUCTION

The speed of change and development in LNG liquefaction technology is changing quicker than ever before since the demand for LNG is sharply rising and new gas resources are being discovered across the world. According to Wood (2012), LNG output was anticipated to reach 320 million tons per year (MTPA) by 2015 and 450 MTPA by 2020. Many existing and modern LNG plant facility engineers are carrying out research to improve production efficiencies and developing more LNG trains for positive economic effects in fulfilling the rising demand. Poor power supply constraints and insufficient process water utilization also contributes to operational upset and shutdown of process equipment such as pumps, compressors, and automated valves, if they are not properly examined (Ujile & Amesi, 2014).

Figure 1 displays the schematic diagram of the standard propane refrigeration facility.

The propane refrigeration unit of Liquefied Natural Gas (LNG) plant requires temperature of -30°C in the chiller to precool a high temperature Natural Gas (NG) entering it. This has not been achieved because, the temperature in the chiller is less than -30°C. However, investigation has shown that the insufficient precooling of the Natural Gas (NG) is because of low heat transfer in the chiller during operation. This is because the liquid propane refrigerant level is set at 30% of the calibrated volume of the propane chiller vessel (E100) to immerse the tubes through which the (NG) is flowing for pre-cooling. However, this level is not achieved due to the low rate of liquid propane refrigerant recovery from the economizer. This has resulted in high propane refrigerant temperature in the equipment and subsequently leads to poor operational performance of the equipment.

In the earlier liquefied natural gas process, the pre-cooling step was not used in single mixed refrigerant (SMR) facilities; instead, the natural gas was chilled immediately to -160°C using a single mixed refrigerant. The first cycle of the LNG production process involves the pre-cooling of the natural gas which removes heat from it to a temperature range between -30°C and -55°C. The pre-cooling cycle may now be created using either pure or mixed refrigerant, thanks to technical advancements: Richardson and Butterworth (1995) investigated the performance of propane/isobutane mixtures in a vapour-compression refrigeration system and confirmed that a drop is better replacement for refrigerant R12. Design and analysis of multi-stage expander refrigeration cycles were proposed and analyzed by Lim *et al.* (2014) to develop an efficient natural gas liquefaction process.

The proposed dual and cascade expander processes have high efficiency and the potential for larger liquefaction capacity and are suitable for small-scale and offshore natural gas liquefaction systems. While refrigeration cycles of conventional expander processes use pure nitrogen or methane as a refrigerant, the proposed refrigeration cycles use one or more mixtures as refrigerants. Since mixed refrigerants are used, the efficiency of the proposed multi-stage expander processes becomes higher than that of conventional expander processes. However, the proposed liquefaction processes are different from the single mixed refrigerant (SMR) and dual mixed refrigerant (DMR) processes. Mortazavi *et al.* (2012) carried out an investigation on LNG process using ASPEN software to increase the energy efficiency of LNG plants and explore potential improvements to the liquefaction cycle. Four expansion loss recovery strategies were simulated following the development of models for the LNG process. According to the simulation results, using expanders in place of traditional expansion procedures can reduce compressor power by up to 5%, recover expansion work by up to 7 % and boost LNG output by up to 1.24%, respectively. Numerous factors were assessed while two modeling and optimization strategies were investigated by Mehrpooya *et al*. (2006).

The purpose of this study was to conduct performance improvement of the propane refrigeration unit for precooling in a liquefied natural gas plant. This research was to investigate and to improve on the performance of the propane refrigeration equipment, mitigating on operations concern of low level of liquid propane refrigerant in the chiller, by inserting an expansion valve between the compressor upstream and chiller downstream.

2.0 MATERIALS AND METHODS

The information used in this investigation came from an operational propane refrigeration unit of an LNG Plant in Pan Ocean Ovade/Ogharaefe Gas Plant in the Niger Delta region of Nigeria. The log sheet over a period of five months was utilized to acquire values for the input parameters that were employed in this investigation. Data was likewise obtained every day as the plant operated for six weeks.

2.1 Data Analysis Methods

The MATLAB v2015a and Microsoft Excel v2016 software packages were used to calculate the modeled equations of the propane refrigeration unit, while ASPEN HYSYS software (software for process simulation) was used to simulate the propane refrigeration unit.

2.1.1 Description of the Propane Refrigeration Unit

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The description of the propane refrigeration unit (shown in Figure 1) is as follows.

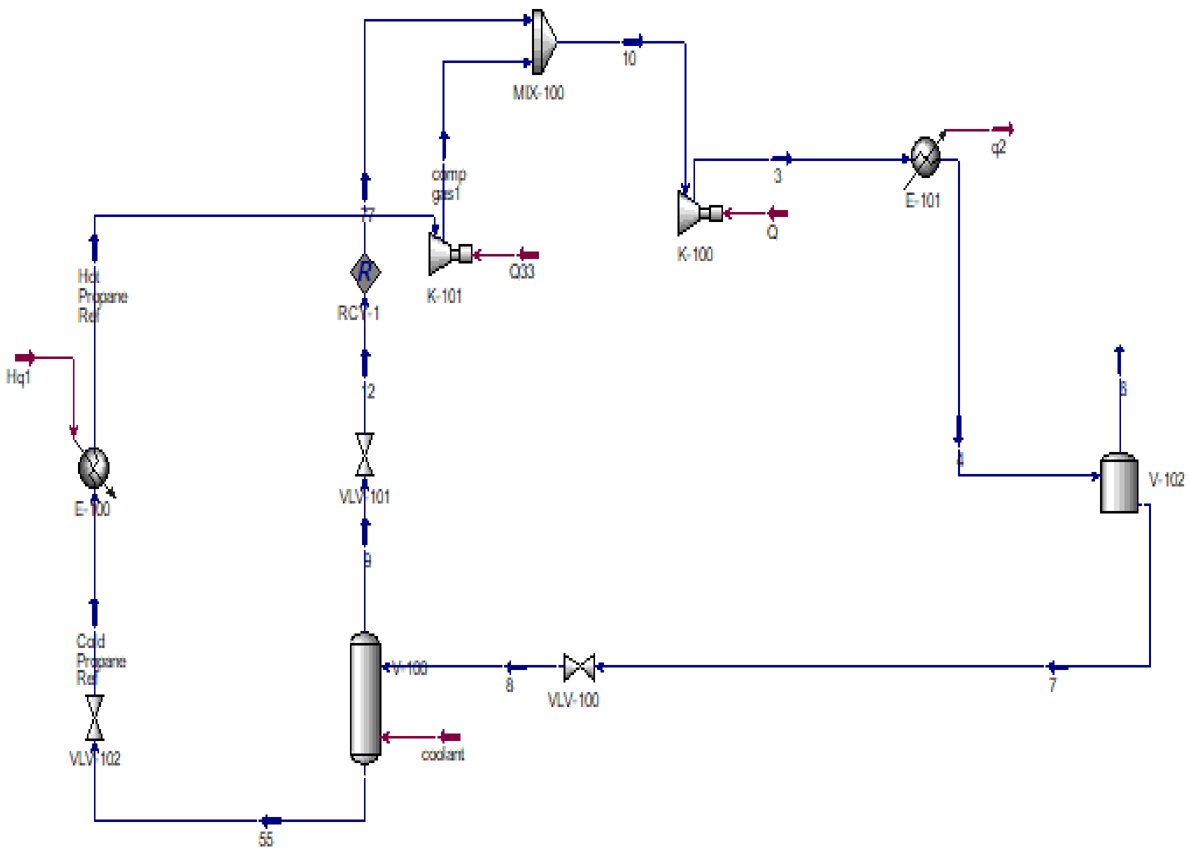

Figure 1: Propane Refrigeration Unit

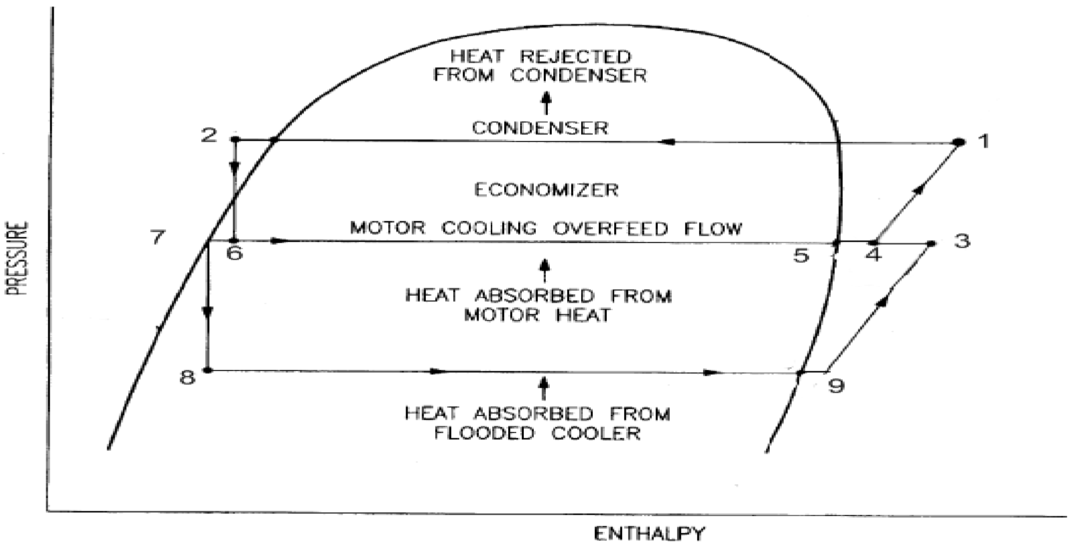

Figure 2: The Pressure-Enthalpy Diagram of Propane Refrigeration cycle

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i. The refrigerant passes through the propane heat exchanger (E-100 chiller), where it is heated and turns into a vapour after absorbing heat from the natural gas.
ii. The vapour with low pressure enters the compressor (K-100), where it is compressed via a two-stage compression to high pressure.
iii. The heat is generated as high pressure vapour passes through the condenser (E-101) and is rejected into the atmosphere and the refrigerant becomes condensed.
iv. The condensed propane refrigerant enters an accumulator (storage) and then enters the economizer or flash tank V-100, where it is expanded.
v. Flash vapour is gathered in the economizer's top section and sent through the return line to the compressor. Then, the flash liquid enters the heat exchanger (propane chiller), and the process continues in a loop.

Figure 2 represents a Pressure-Enthalpy (P-h) diagram of a propane refrigeration system, which depicts every thermodynamic cycle step. Therefore, the procedure (1-2) depicts a drop in heat exchanger temperature (condenser). As soon as the vapour reaches point (2)'s temperature, it turns into liquid. The liquid is then expanded by an electronic expansion valve to the point's medium pressure (6). Here, the motor-driven compressor heats the vapour by separating it in the flash tank before sending it there (5). Through partial evaporation in the flash tank, liquid at point (6) cools to temperature (7).

This is defined as the ratio of the heat absorbed (Qin) by the refrigerant while passing through the chiller to the work input (Win) required compressing the refrigerant in the compressor. This is mathematically expressed as in Eastop and McConkey (2005).

a. Coefficient of Performance

The ratio of the absorbed heat (Qin) by the refrigerant as it passes through the chiller to the work input by the compressor (Win) is called coefficient of performance (Eastop & McConkey, 2005):

$$COP = \frac{Q_{in}}{W_{in}} \qquad (1a)$$

but

$$COP = \frac{(1 - X_6)\, M_5\, (h_9 - h_8)}{M_5(1 - X_6)\,(h_3 - h_9) + M_5\, \left(h_1 - h_{4,MIX}\right)} \qquad (1b)$$

where COP is the coefficient of performance of the refrigerator system, X_6 is the mole fraction of vapor leaving the economizer unit and M_5 is the refrigerant circulation rate, h_9, h_8 h_3, h_1 and $h_{4,mix}$ are the enthalpies of refrigerant at compressor inlet, inlet to the refrigerant chiller, compressor outlet (kJ/kg), condenser inlet, and the vapor mixture coming from compressor, economizer units respectively.

b. Refrigeration Capacity

According to Ujile and Amesi (2014), the following formulae helps to determine the capacity of refrigeration:

$$R_C = \frac{M_5\,(h_9 - h_8)}{210} \qquad (2)$$

c. Mass Flow Rate of Cooling Water

Also, Ujile and Amesi (2014) posted that the calculation of the cooling water mass flow rate is as follows:

$$M_{CW} = \frac{Q}{CP_{CW}\Delta T}. \qquad (3)$$

where M_{CW} is the mass flow rate of cooling water, Q is the amount of heat removed in the intercoolers, Cp_{CW} is the specific heat capacity of water, ΔT is the difference in temperature.

The lost productivity or the rate at which a refrigerant is irreversible in the compressor unit as shown in Figure 2

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d. Compressor unit

$$L_W = M_5\,[(h_9 - T_0\,s_9)] - (h_3 - T_0\,s_3) - W_E \quad (4)$$

where L_W is the wasted work or the rate of refrigerant irreversibility, T_o is the dead state temperature, S_9, S_3 are the refrigerant entropies at the compressor's input and outflow respectively and W_E is the amount of electricity needed for the compressor (Ujile & Amesi, 2014)

e. Condenser Unit

The lost work of irreversibility of refrigerant in the condenser unit, shown in Figure 2 is as follows (Ujile and Amesi, 2014):

$$L_W = T_0\,[M_5(s_4 - s_3)] + \left(M_{CW} s_{CW(out)} - s_{CW(in)}\right) \quad (5)$$

where s_4, s_3 are the entropies of refrigerant leaving the economizer and compressor outlet respectively, $s_{CW(out)}$, $s_{CW(in)}$ are the entropies of the outlet cooling water and inlet cooling water respectively.

$$M_{CW} = M_5\left[\frac{h_1 - h_2}{h_{CW(out)} - h_{CW(in)}}\right] \quad (6)$$

where h_2 is the enthalpy of refrigerant leaving the condenser (kJ/kg), $h_{CW(out)}$, $h_{CW(in)}$ are the enthalpies of the outlet cooling water and inlet cooling water respectively.

f. Expansion Valve

$$L_W = M_{CW} T_0 (s_3 - s_5) \quad (7)$$

where s_5 is the entropy of refrigerant leaving the economizer (Ujile & Amesi, 2014).

g. Refrigerant Chiller

When heat is evacuated from a refrigerated environment or absorbed by refrigerant, the refrigerating effect is stated as:

$$Q_{in} = (1 - X_6) M_5\,(h_9 - h_8) \quad (8)$$

h. The Fraction of Refrigerant

From enthalpy balance, the amount of vaporized refrigerant in the chiller may be calculated as follows:

$$M_C (H_7)_L = M_5 X_V (h_8) + M_5 (1 - X_V) \quad (9)$$

where X_V is the percentage of refrigerant in a mole that evaporates when it is throttled to the economizer unit (Ujile & Amesi, 2014).

2.1.2 Simulation Tool

There are several simulation packages that are available, but ASPEN HYSYS provides one of the best process modeling environments for conceptual design and operations improvement of petroleum oil and gas process.

2.2 Equipment

The model is a cascade liquefaction/refrigeration approach which employs a compressor, condenser, valve, and an LNG unit operation. Downstream equipment includes fractionating columns, reboilers, pumps and valves.

2.3 Feed Stream Parameters

The plant takes in a single feed stream of pre-treated natural gas from the gas recovery section of a gas flare system. The flare condition and composition (feed) are obtained based on available laboratory data and is shown in Tables 1 and 2 respectively.

Table 1: Feed Gas Condition from Chromatography

Property	Value
Temperature (°C)	**30**
Pressure (kPa)	**2275**
Flowrate(SCFD)	32460000

Source: Audubon Engineering (2008)

Table 2: Feed Gas Composition

Composition	Mole fraction
Nitrogen	0.0025
Carbon dioxide	0.0048
Methane	0.7041
Ethane	0.1921
Propane	0.0706
Isobutane	0.0112

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Composition	Mole fraction
Normal Butane	0.0085
Iso-Pentane	0.0036
Normal Pentane	0.0020
Hexane	0.0003
Heptane	0.0002
Octane	0.0001

Source: Audubon Engineering (2008)

2.4 Exergy Analysis

The quantity of reversible work completed by a system when its components are placed into a condition of thermodynamic equilibrium with their surroundings is known as exergy, which is derived from the second law of thermodynamics. Two key factors, enthalpy, and entropy, determine how much energy a system changes. Exergy change (ΔE_x) between a system's initial and final states is represented as:

$$\Delta E_x = (H_o - H_i) - T_o(S_o - S_i) \qquad (10)$$

where H_o is the enthalpy outlet stream, H_i is the enthalpy inlet stream, T_o is the ambient temperature, S_o is the entropy of the outlet stream, and S_i is the entropy of the inlet stream. The exergy efficiency is written as follows:

$$\eta_{ex} = \frac{C_{wreq} - Ex_{loss}}{S_{wreq}} \qquad (11)$$

where C_{wreq} is the needed total compressor power, Ex_{loss} is the sum of the energy lost via each unit-operation and S_{wreq} is the total power that the system needs.

a. Compressor

The compressor as shown in Figure 3 is a mechanical tool that delivers energy to a gaseous fluid to increase the fluid's pressure or cause a flow. It is mostly driven by a prime mover (engine or electric motor).

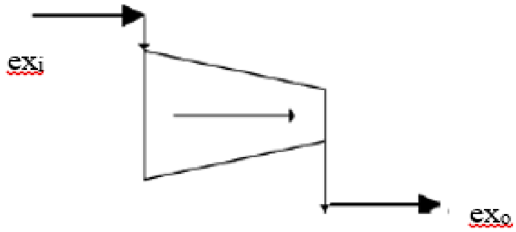

Figure 3: Compressor

$$Ex_{Comp,loss} = \dot{m}(ex_i - ex_o) - W \qquad (12)$$

where $Ex_{Comp,loss}$ is the exergy loss in compressor, $\dot{m}$ is the mass flow rate of propane refrigeration, ex_i is the specific exergy in, ex_o is the specific exergy out and W is the compressor power.

b. Aerial Condensers

Aerial condenser as shown in Figure 4 is a simple exchanger with tubes exposed to a stream of moving air across them. The tubes usually have aluminum fins pressed onto the outer wall of the tubes to increase the heat transfer area. Air is blown across the tubes with a fan driven by an electric motor or engine. It is frequently called fin-fan cooler units. The tube fluid usually makes at least six to eight passes, while the air flow is a single pass.

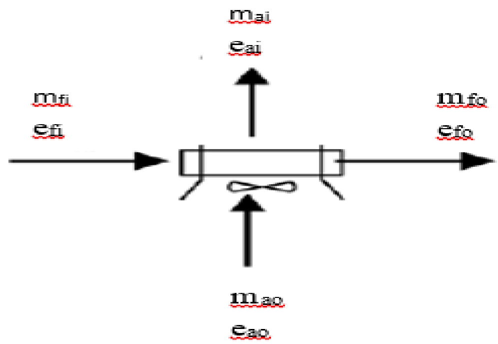

Figure 4: Aerial condensers (AC108)

$$Ex_{AC,loss} = \left(\dot{m}_f e_f - \dot{m}_a e_a\right)_i - (\dot{m}_f e_f - \dot{m}_a e_a)_o \qquad (13)$$

where $Ex_{AC,loss}$ is the exergy loss in aerial condenser, m_f is the mass flow rate of fuel, e_f is the specific exergy of fuel, m_a is the mass flow rate of air, and e_a is the specific exergy of air.

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c. Propane Chiller

A chiller is a heat exchange device used in lowering the temperature of a gas to condense some hydrocarbons. Gas passes through the tube side of the chiller and the cold liquid refrigerant is on the shell side of the chiller. Heat transfers from the warm gas inside the tube to the cold refrigerant on the shell side. The refrigerant boils and vaporizes.

$$Ex_{HX,loss} = \dot{m}\,\sum(ex_i - ex_o) \qquad (14)$$

where ex_i is the exergy in and ex_o is the exergy out.

d. Expansion Valve

The expansion valve as shown in Figure 5 is used to drop the pressure of the propane refrigerant which in turn drops the temperature.

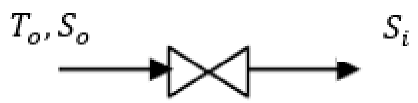

Figure 5: Expansion valve

$$Ex_{V,loss} = \dot{m}\,T_o(S_o - S_i) \qquad (15)$$

2.5 Scenario Study

In this plant scenario study, five (5) scenarios have been studied with different operating conditions as shown in Table 2 applied at the propane chiller to evaluate the propane pre-cooling cycle's effectiveness. An expansion valve (CV-1, CV-2, or CV-3) that is situated upstream of the chiller can be used to alter the working conditions of the propane evaporator. The primary variable that is altered to achieve the required cooling function for the propane chiller is the expansion valve pressure. Scenario 3 (T=-30 and P = 223kPa) is the base scenario, and a sensitivity analysis was done by varying the expansion valve pressure which in turn varied the chiller outlet temperature as shown in Table 3.

Table 3: Propane Evaporator Operating Conditions for all Scenario Studies

Scenario	Propane Chiller temperature (ºC)	Expansion valve pressure (kPa)
1	-40	168
2	-35	195
3	-30	223
4	-25	250
5	-20	270

3.0 RESULTS AND DISCUSSION

The results for this study were obtained using data such as temperatures, pressures, mass flow and composition as determined by a functioning propane refrigeration system unit of the LNG Plant in Pan Ocean Ovade/Ogharaefe Gas Plant in the Niger Delta region of Nigeria.

3.1 Effect of Pressure on the Mass Flow Rate of Propane

Figure 6 shows the effect of the operating parameter on the mass flow rate of propane in propane refrigeration unit for the precooling of LNG, and considering the base scenario (Scenario 3: T= -30ºC /and P=223kPa), the mass flow rate of the refrigerant obtained is 17814kg/h.

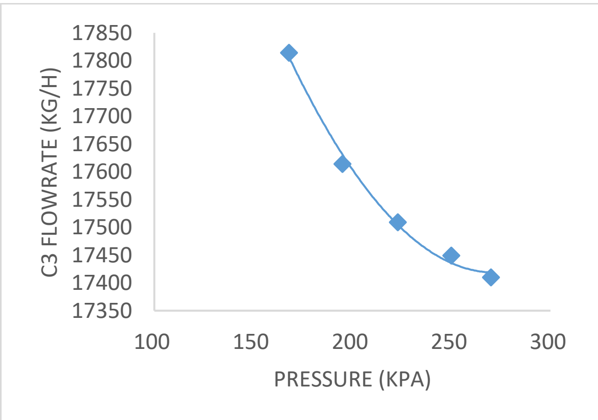

Figure 6: Effect of Pressure on the propane flowrate

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However, graphical analysis and trend show that decrease in pressure from the expansion valve upstream of the cooler and downstream of the chiller causes a decrease in temperature of the propane refrigerant from -30ºC to -40ºC which in turn causes the mass flow rate of the refrigerant to increase from 17509kg/h to 17814kg/h.

3.2 Effect of Pressure on the Compressor Power

Figure 7 shows the pressure-enthalpy (P-h) diagram with their respective data points for the propane refrigeration loop, as explained in Figure 2.

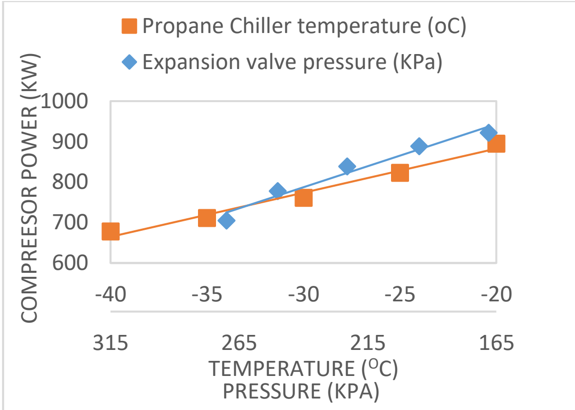

Figure 8: Effect of Pressure on the Compressor Power

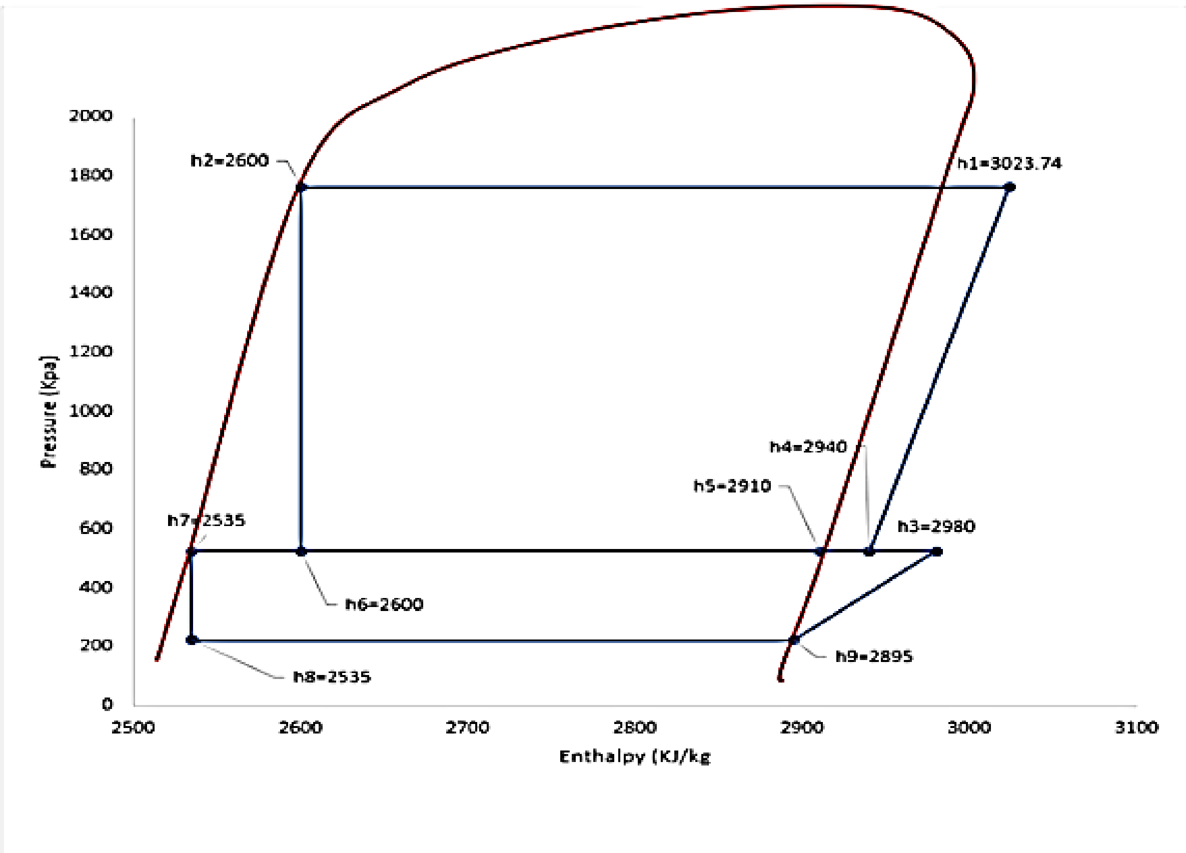

Figure 7: Pressure-Enthalpy (P-h) Diagram for the Propane Refrigeration Loop

Figure 8 shows the effect of the operating pressure on the propane refrigeration system's compressor power unit and considering the base scenario (Scenario 3: T= -30ºC and P=223kPa). Higher pressure often corresponds to higher temperatures, which requires more work to compress the refrigerant to the desired level. The compressor power obtained at the base case is 761.3kW.

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However, as the mass flow rate of the propane refrigerant increases, the compressor does less work thereby reducing its power consumption from 761.3kW to 678.3kW.

3.3 Effect of Pressure on the Air Cooler Duty

Figure 9 shows the effect of the operating pressure on the air cooler duty in the propane refrigeration unit and considering the base scenario (Scenario 3: T= -30°C and P=223kPa). As the refrigerant enters the air cooler at a high pressure which corresponds to higher temperatures, the air cooler requires more cooling capacity to achieve the desired cooling effect. The air cooler duty obtained in this case is 2574kW.

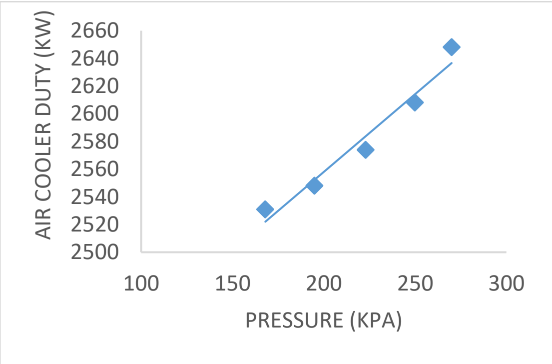

Figure 9: Effect of Pressure on the Air Cooler Duty

However, graphical analysis and trend result show that decrease in pressure from 223kPa to 168kPa causes a decrease in temperature from -30°Cto -40°C which in turn causes a decrease in the air cooler duty from 2574kW to 2531kW. This is because lower pressure, which often corresponds to lower temperatures, requires less cooling capacity of the air cooler to achieve the desired refrigeration effect.

3.4 Exergy Loss for Each Unit Operation in the Propane Pre-Cooling Cycle

Exergy loss for each unit operation in the propane pre-cooling cycle, which are the main sources of exergy loss in the propane cycle, is shown in Figure 10. The primary contributors to the exergy loss in the propane cycle were seen to be the valves, compressors, and heat exchangers.

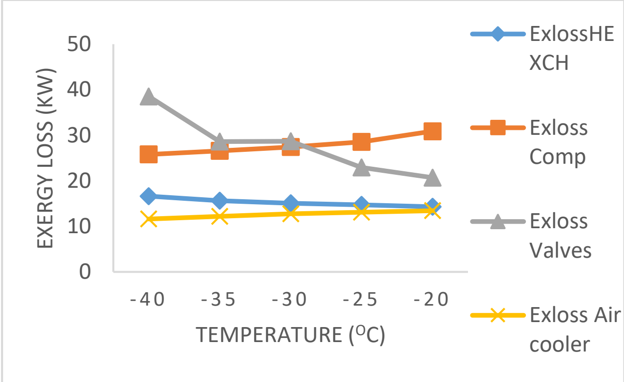

Figure 10: Exergy Loss for each Unit Operation in the Propane Pre-Cooling Cycle

The valves assigned the most energy loss for the base scenario (Scenario 3), at 33.52 kW, followed by the compressors, at 30.89 kW and then the propane chiller, 14.43kW. Also, results showed that the valve provided the highest exergy loss in the base scenario due to an increase in entropy generation when larger pressure drop is applied across the system.

3.5 Exergy Efficiency for the Propane Pre-Cooling Cycle

The exergy efficiency for the propane pre-cooling cycle is shown in Figure 11; scenario 5 has the lowest exergy efficiency (34.67%).

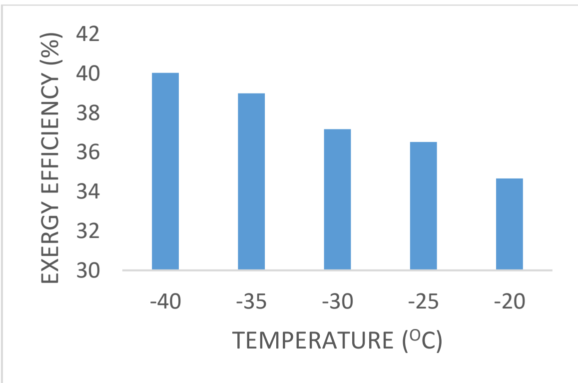

Figure 11: Exergy efficiency for the propane pre-cooling cycle

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The process has a large opportunity for improvement since Scenario 1 has the highest exergy efficiency (40.03%). The findings indicate that altering the propane evaporator's working parameters causes a larger creation of entropy, which reduces the exergy efficiency.

3.6 Effect of Pressure on Coefficient of Performance of the Propane Pre-Cooling Unit

Figure 12 shows the effect of pressure on the coefficient of performance (COP) of the propane precooling unit of the LNG plant.

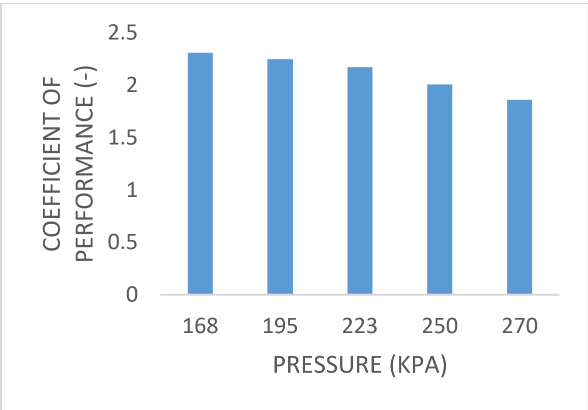

Figure 12: Effect of Pressure on Coefficient of Performance of the Propane Pre-Cooling Unit

The COP of Scenario 1 (168kPa) increases by 20% as compared to scenario 5 (270kPa). This is because a decrease in pressure causes a corresponding decrease in temperature which increases the mass flowrate of the propane refrigerant vapour which in turn causes a decrease in the compressor work. Also Figure 13 shows the effect of pressure on the refrigerating effect in the propane chiller. Results obtained show that a decrease in pressure which gives a corresponding decrease in temperature causes a decrease in the refrigerating effect.

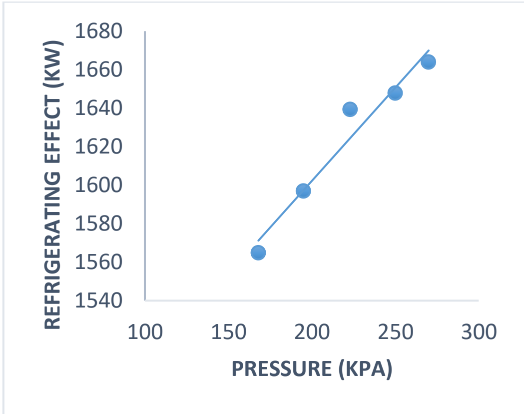

Figure 13: Effect of Pressure on Refrigerating Effect

4.0 CONCLUSION

This study develops an effective method of evaluating the performance of a propane refrigeration system for the precooling of LNG. It includes the study of the effect of pressure with corresponding temperature on the compressor power, mass flowrate of the propane refrigerant, air cooler duty, chiller duty, and exergy efficiency and loss across all the major components of the precooling refrigerating system. The findings of the study are summarized as follows:

i. The results obtained by a change in the pressure and temperature in the propane pre-cooling unit has significant effects on mass flow rate, compressor power, air cooler duty, exergy loss, exergy efficiency, and COP.

ii. Lower pressures generally correspond to increased mass flow rates, reduced compressor power consumption, decreased air cooler duty, and improved COP.

iii. However, these changes also result in increased exergy losses in different components of the system. These are optimal operating conditions that have determined which have enhanced the overall performance and efficiency of the propane pre-cooling cycle.

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