A case study: Application of energy and exergy analysis for enhancing the process efficiency of a three stage propane pre-cooling cycle of the Cascade LNG   process

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ABSTRACT
The propane pre-cooling cycle has been widely used in most LNG plants as the first cooling cycle in

the natural gas liquefaction process. As LNG plants consume high amounts of energy, enhancements in the process design and plant operation will minimize the overall energy consumption of the plant. The aim of this study is to enhance the process efficiency of.J! three stage propane pre-cooling cycle of the

Cascade LNG process for the large-scale LNG train by determining the optimal operating conditions of

the propane evaporator that will minimize the overall energy consumption. Energy and   exergy analysis

methods are adopted to evaluate the process efficiency of the propane pre-cooling cycle. Six case studies were presented to determine the optimal operating conditions of the propane evaporator that gives maximum energy reduction. The propane pre-cooling cycle is modelled and simulated using Aspen

HYSYS with detailed thermodynamic information obtained to calculate the exergy loss. The results of

the energy and exergy analysis indicate that Case 6 gives the highest coefficient of performance (COP)

and the maximum exergy efficiency compared to the baseline case, which are 15.51% and 18.76% respectively . The results indicate that by reducing the cooling duty at the intermediate stage§. of propane evaporator about 13.5% energy saving can be achieved compared to the baseline case.

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1. Introduction

As the demand on LNG is drastically increasing and the discovery of new large gas fields is continuously taking place worldwide, the pace of change and development in LNG liquefaction technology is becoming more rapid than ever before. LNG production is estimated to hit 320 million tonnes per annum (MTPA) by 2015 and to 450 MTPA by 2020 as reported by Wood [l]. In order to meet this escalating demand, most of the existing and new LNG plants are looking for opportunities to make a further increase in their LNG capacity and building larger LNG trains which will provide economic benefits and be process efficient. Since the 1970s, when the kick started for the LNG plant and until the present day, three main LNG processes have been applied in the LNG plants viz. Single

mixed refrigerant (SMR), Propane precooled mixed refrigerant (C3MR) and cascade liquefaction process [2]. In the last 10 to 15 years, the innovations of LNG technologies have drastically progressed whereby new LNG processes have been introduced such as Mixed fluid cascade (MFC), Air Products (AP-X™), Dual mixed refrigerant (DMR) and Parallel mixed refrigerant (PMR) [3]. Most of the existing LNG plants have three main cooling cycles, namely the pre-cooling, liquefying and sub-cooling

cycle. Earlier LNG plants that employed the SMR process did not have the pre-cooling cycle, and instead the natural gas was cooled directly to -160°C using a single mixed refrigerant. The pre-cooling cycle is the first cycle in an LNG process which removes the heat from natural gas to a temperature range between -30°C to -55°C depending on the pre-cooling technology applied. As a result of technological advancement, the pre-cooling cycle can now be designed using either pure refrigerant or

mixed refrigerant. Castillo, et al. [4] reported that 95% of the current LNG plants employ the pre-cooling cycle; 85% of which are dominated by propane refrigerant compared to mixed refrigerant.

Thermodynamic analysis has been widely used in the LNG plants to determine the sources and locations of the main process irreversibilities that occur within the process or are due to ill!_individual unit operation. Energy analysis or the first law of thermodynamic method only indicates the energy

conservation of the overall process which is measured using two parameters i.e. COP and specific power (SP). However, to locate the irreversibility that occurs within the unit operation of the process, the exergy analysis method is applied. These methods are widely applied by other scholars to evaluate the energy conversion process efficiency. Vatani, et al. [5], Kanoglu [6], Cipolato, et al. [7], Al-Otaibi, et al. [8] and Mehrpooya, et al. [9] applied the energy and exergy analysis methods for analysing the

process efficiency of various LNG processes. In a nutshell, these methods are also widely used in some power plants as mentioned in the following references [10-12].

Converting natural gas to liquid utilizes an extensive amount of energy. According to Alfadala et al. [13], a typical base load LNG plant consumes about 5.5-6 kWh of energy per kgmole of LNG produced . An energy-efficient refrigeration system will enhance the plant operation and provide economic benefits [14]. Several authors have discussed the area of enhancing the efficiency of the pre-cooling cycle.

Paradowski, et al. [15] discussed two operating parameters of the pre-cooling cycle in the C3MR process that can enhance the process efficiency plus debottleneck the existing LNG plant capacity to 5.5 MTPA. The pre-cooling temperature of the low pressure (LP) stage and the propane compressor speed were the operating parameters that were adjusted to meet the new capacity requirement.

| Nomenclature		Subscripts	
Ex	exergy **[MW]**	f	fluid
EXHX,loss	exergy loss of heat exchanger [MW]		inlet
ExcoMP,loss	exergy loss of compressor [MW]	o	outlet
Exv,ioss	exergy loss of valve [MW]	r	ratio
EXMIX,loss	exergy loss of mixer [MW]		
EXAC,loss	exergy loss of air cooler [MW]	Greek symbol	
e	specific exergy (MJ/kg)	*nex*	exergy efficiency
H	enthalpy (MJ/kg)		
n	mass flow rate [kg/s]	List of symbols	
p	pressure [bar]		
Q	refrigeration duty [MW]	C2H6 ethane	
S	entropy [MJ/kg K]	C2fu ethylene	
To	ambient temperature [K]	C3Hs propane	
W	compressor power [MW]	Cfu	
		methane N2	
		nitrogen	
Abbreviations			
AC	air cooler		
EOS	equation of state		
HP	high pressure		
HX	heat exchanger		
LNG	liquefied natural gas		
LP	low pressure		
MP	medium pressure		
MTPA	million tonnes per annum		
PR	Peng Robinson		
UA	product of overall heat transfer coefficient and heat exchanger area		
SP	specific power		

Castillo et al. [16] studied suitable choices of refrigerants that are applicable for pre-cooling cycle by analysing the effects of various refrigerants (i.e. N2, Cfu, C2H6, C3Hs) on the compressor power using the Linde-Hampson process. It was found that compared to other refrigerants, propane has a higher specific refrigerant effect which makes it the preferred refrigerant to be used in the pre-cooling cycle. Ransbarger [17] studied the comparison between three stage and four stage propane cycles for the cascade LNG process which resulted in a power reduction of 1%; nonetheless the economic evaluations

did not justify the increased cost associated with the additional stage. Evolution in the design of the propane pre-cooling cycle has emerged in recent decades. In this context, various studies have been presented that were related to the enhancement of the efficiency of the propane cycle with respect to significant changes made in the process configuration. Mortazavi, et al. [18] suggested the replacement of the conventional expansion valve in the C3MR process with expanders to improve the liquefaction efficiency. In another study, Mortazavi, et al. [19] investigated the usage of waste heat from gas turbines by installing absorption chillers in the propane cycle of the C3MR process. Kalinowski, et al. [20] proposed the replacement of the propane evaporator with an absorption refrigeration system utilizing waste heat from the electrical power generating gas turbines.

Although many studies have been conducted focusing on the efficiency enhancement of the LNG plants through modification of the process configuration [6, 18-21], there is only very scant information available which focuses on the operation perspective. In this study, we to analyse the impact of changing the operating conditions of the propane evaporator towards the energy consumption of the process. Six case studies are proposed with different operating conditions applied to the propane evaporator. The development of these case studies is discussed in section 2.2 of the manuscript. The sensitivity of the COP, the specific power (SP), and the exergy loss and exergy efficiency are analysed for all case studies presented.

1.1. Description of propane pre-cooling cycle process

Treated feed gas enters the three stage propane cycles at 29°C and 75 bar and is cooled to -40 °C. The propane evaporator (i.e. kettle type) also cools methane and condenses ethylene. Cooling of the process stream is achieved by the evaporation of propane in the pool on the shell side with the process streams

flowing inside the immersed tubes. The propane compressor (i.e. centrifugal type) with side streams recovers the evaporated propane and compresses the vapour to 18 bar. Propane is finally condensed at 49°C using the air cooler. The condensed propane is then recycled back to the propane evaporator. Figure 1 shows the simplified process scheme of propane pre-cooling cycle.
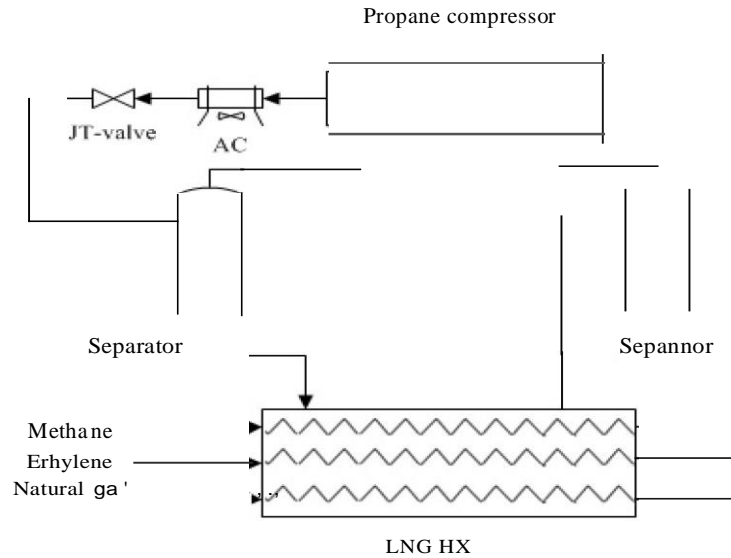
Fig. 1. Simplified process scheme of propane pre-cooling cycle. Only one stage is shown for simplicity.

2. Simulation method and modelling assumptions

Aspen HYSYS which is a steady state process modelling software was employed to model the three stage propane pre-cooling refrigeration cycles. Peng Robinson equation of state (PR-EOS) fluid package was used for modelling the property of the substances. This process simulator is well known for modelling the LNG processes and has been widely used by others [22-25]. The feed gas composition and modelling assumptions are summarized in Table 1 and 2 respectively .

Table 1

Feed gas composition after sweetening.

Component•	Mole fraction
Nitrogen	0.0028
Methane	0.8974
Ethane	0.0496
Propane	0.0343
i-Butane	0.0079
n-Butane	0.0073
i-Pentane	0.0005
n-Pentane	0.0002
n-Hexane	0.0000
Total	1.0000

ꞌFeed gas composition obtained from Western Australia gas plant

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Table 2
Modelling assumptions

Natural gas temperature	29°c
Natural gas pressure	75 bar
Natural gas flow rate	41,700 kgmole/h
Compressor polytropic efficiency	80%
Pressure drop in LNG heat exchanger	1.2 bar
Pressure drop in air cool heat exchanger	1.3 bar
Minimum temperature approach in heat exchanger	> 2°C
Ambient temperature	27°c
Air cooler exit temperature	40°c

The following are the constraints applied in modelling the propane pre-cooling cycle:

1. Temperature approach (Tapp) in the LNG heat exchanger should be above 2°C to prevent temperature cross.

2. The inlet temperature of propane compressor should be above its dew point temperature to safeguard the operation of the compressor.

2.1 Process simulation description

In this study, six case studies as shown in Table 3 have been studied with different operating conditions applied at each evaporator stage to analyse the performance of the propane pre-cooling cycle. The operating conditions of the propane evaporator is changed through an expansion valve that is located upstream of each evaporator (i.e. CV-1: HP stage; CV-2: MP stage and CV-3: LP stage) as depicted in Figure 2. The expansion valve pressure is the key manipulated variable that is adjusted to obtain the desired cooling duty for each stage propane evaporator and also to maintain the temperature approach above 2°C. Discharge pressure of LP and MP stage propane compressors is connected to the MP and HP propane evaporator stage respectively to obtain the resultant compressor power as shown in Figure 2 and 3. Propane pre-cooling cycle configuration for the baseline case and case 6 are also shown in Figure 2 and 3 respectively and changes made on the operating parameters are marked with dotted lines on these figures.

Table 3
Propane evaporator operating conditions for all case studies.

Case studies	Propane evaporator operating conditions
Case 1•	-2s·c, -3o·c, -4o·c
Case 2	-1s·c, -2o·c, -4o·c
Case 3	-10·c, -1s·c, -4o·c
Case 3	SC, -lO°c, -4o·c
Case s	o·c, SC, -4o·c
Case 6	5°C, o·c, -4o·c

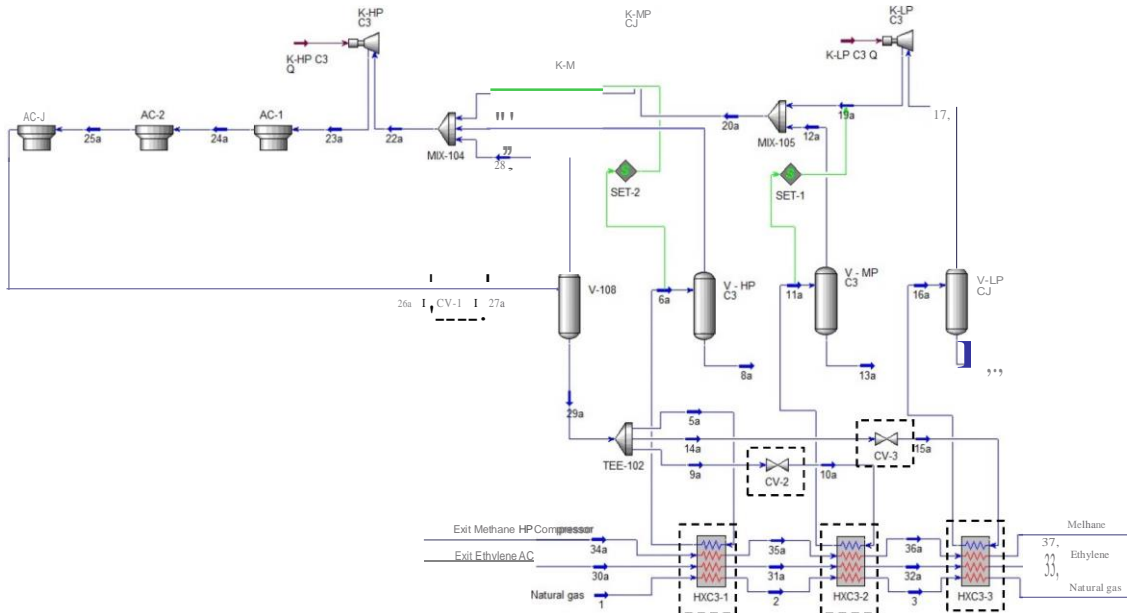•(-25°C, -30"C, -40°C means process exit temperature at HP, MP and LP stage respectively)

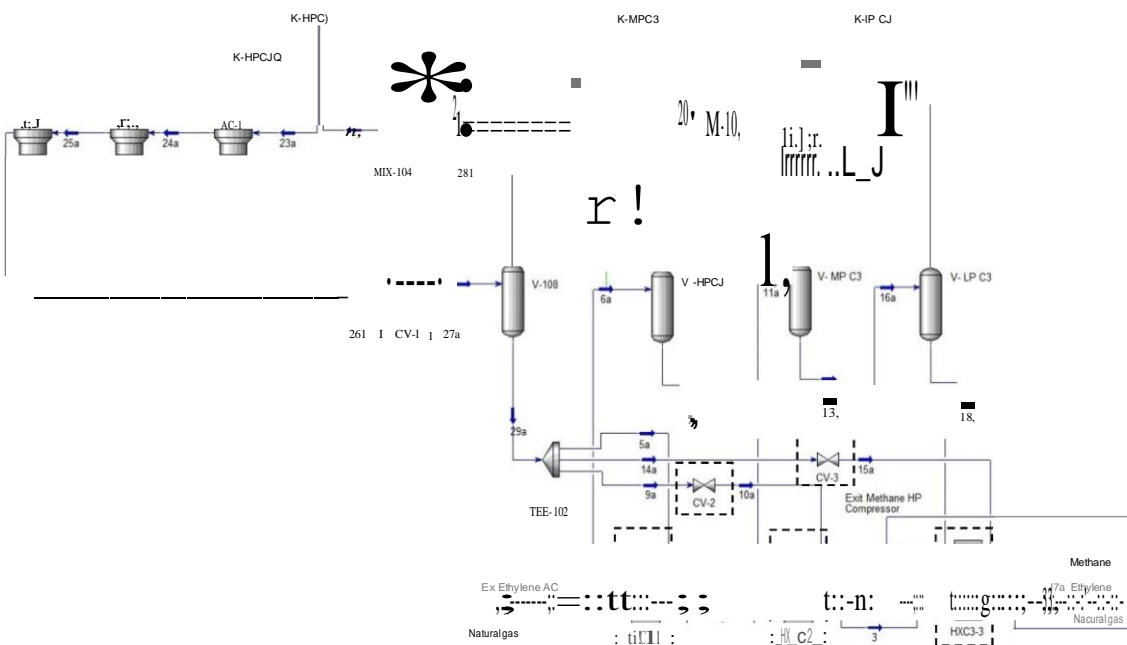Fig. 2. Propane pre-cooling cycle Case 1 configuration.

Fig. 3. Propane pre-cooling cycle Case 6 configuration.

2.2 Case studies development of propane pre-cooling cycle

These six case studies were derived by analysing various operating conditions of current LNG   plants located in Southeast Asia region, Australia and also based on the information available from   the literature [6, 15-21, 26]. Figure 4 illustrates the development of the propane pre-cooling cycle case studies.

Case studies presented are defined as follows:

Case 1: Baseline case (i.e. higher cooling duty at the intermediate stage§. (i.e. **HP** and MP stage)

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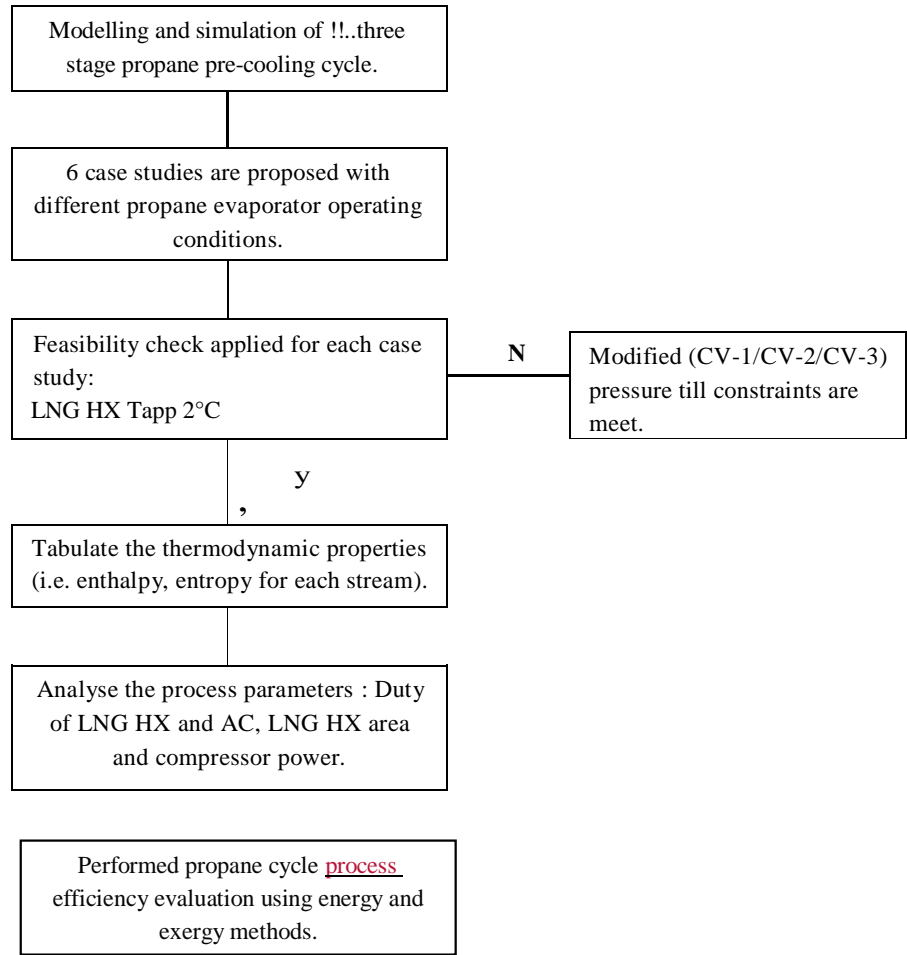Case 6:(i.e. lower cooling duty at intermediate stage§.)

Modelling and simulation of !!..three stage propane pre-cooling cycle.

6 case studies are proposed with different propane evaporator operating conditions.

Feasibility check applied for each case study:
LNG HX Tapp 2°C

N → Modified (CV-1/CV-2/CV-3) pressure till constraints are meet.

y
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Tabulate the thermodynamic properties (i.e. enthalpy, entropy for each stream).

Analyse the process parameters : Duty of LNG HX and AC, LNG HX area and compressor power.

Performed propane cycle process efficiency evaluation using energy and exergy methods.

Fig. 4. Development of propane pre-cooling cycle case studies.

3. Energy analysis

Energy consumption and process efficiency of the overall process are determined through energy analysis. This method has two quantitative parameters; COP and SP. COP is a standard criterion applied in evaluating the efficiency of a cryogenic system. It is defined as the ratio of total heat removed by refrigerant to the amount of power required by the system (Eq. 1) while SP is defined as total power consumption per unit mass of LNG (Eq. 2).

$$COP = Q / W \tag{1}$$

$$SP \,(MWh/tonne\ LNG) = L_{W,eq} \,/\, \dot{m}_{LNG} \tag{2}$$

where the nomenclature for the above equations are as follows: Q is refrigeration duty (MW); W is

compressor power (MW), L_{Wreq} is the total compressor power required (MW) and $\dot{m}_{LNG}$ is the amount of LNG produced in tonne/h.

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3.1. Exergy analysis

In this study, exergy analysis is applied to locate the irreversibility that occurs within the unit operations of the propane pre-cooling cycle. This method identifies the individual unit operation that exhibits a higher amount of lost work which gives the process engineer valuable information for improving the

process from the equipment and process design point of view. Exergy which is derived from the second law of thermodynamics is defined as the amount of reversible work achieved by a system when the system components are brought into thermodynamic equilibrium state with its environment in a reversible process [27]. The exergy change of a system is a function of two main parameters which are the enthalpy and entropy. Change in exergy ΔE_x) between the initial and the final state of a system is

expressed as:

$$\Delta E_x = (H_o - H_i) - T_o(S_o - S_i) \tag{3}$$

where T_0 is the ambient temperature, H_0 and S_0 represent the enthalpy and entropy of the outlet stream and Hi and Si represent the enthalpy and entropy of the inlet stream respectively. The difference of this property will define whether the processing system requires or produces work as the systems moves

from initial state to final state. If the exergy difference (Lllix) is greater than 0, this indicates that the processing system produces work, whereas if the exergy difference is less than 0, this indicates the processing system requires work from the outer system for the state change [28].
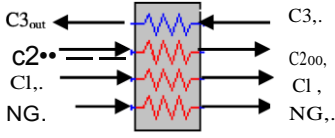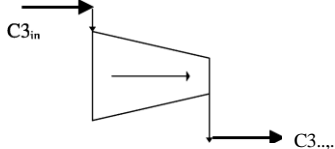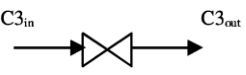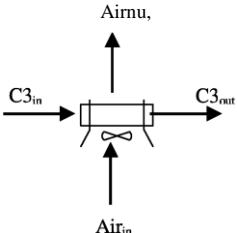
The exergy efficiency of the process is defined as the ratio of the difference between the total compressor power required and the total exergy loss to the total amount of power required by the system (Eq.4). The exergy efficiency is expressed as:

$$\eta_{ex}\ (\%) = \frac{\sum Wreq - \sum Wloss)}{\sum Wreq} \times 100 \tag{6} \tag{4}$$

where $\sum Wloss$ is the total exergy loss work from each unit operation.

The expressions to determine the exergy loss for all the unit operations in this study are summarized in Table 4.

Table 4

Exergy loss calculation of various unit operations in propane cycle.

LNG heat		$Ex_{HX, loss} = \dot{m} \sum ex_i - ex_o$
Compressor		$Ex_{COMP, loss} = \dot{m} (ex_i - ex_o) - W$
Valves		$Ex_{V, loss} = \dot{m} T_o (S_o - S_i), h_i = h_o$
Mixer		$Ex_{MIX, loss} = \dot{m}_1 e_1 + \dot{m}_2 e_2 - \dot{m}_3 e_3$
Air cooler		$Ex_{AC, loss} = (\dot{m}_f e_f + \dot{m}_a e_a)_i - (\dot{m}_f e_f + \dot{m}_a e_a)_o$

In the above equations, th is the mass flow rate of propane at the inlet stream (kg/s), W is the compressor power (MW), S is the entropy (MJ/kg.K), th $_a$ is the mass flow rate of air in (kg/s) and e is the specific exergy for the respective stream in (MJ/kg).

Thermodynamic analysis comprising of the first and second law is used in this study to overcome the

limitations of the deterministic optimization method i.e. a numerical approach which excludes the uncertainty changes involved in the design process, hence it is not considered as the best option for many actual engineering problems [29]. Additionally, the optimization result obtained using this method causes ambiguity because it is not embedded with any process knowledge (i.e. enthalpy and entropy) [23]. Knowing these important process parameters for each process stream provides a better

understanding of the changes occurring within the process. Hence, necessary adjustment can be done on the operating parameter to improve the process performance.

4. Results and discussion

Operating conditions of the baseline (Case 1) and Case 6 are summarized in Table 5. The effect of different operating conditions at each evaporator stage on the heat exchangers, compressors and air coolers are discussed in the section 4.1.

Table 5
Operating conditions of each stream in propane refrigeration cycle.

Stream no	(Case 1)				Case 6			
	x	T (OC)	P(bar)	S(kJ/kg K)	x	T (OC)	P(bar)	S(kJ/kg K)
1	1	29	75	7.98	1	29	75	7.98
2	0.99	-25	74.80	7.34	1	5	74.80	7.74
3	0.98	-30	74.60	7.25	1	0	74.60	7.69
4	0.92	-40	74.40	7.02	0.92	-40	74.40	7.02
Sa	0	-27.49	1.85	1.56	0	2.12	5.05	1.84
6a	1	-23	1.85	3.26	1	7	5.05	3.23
7a	1	-23	1.85	3.26	1	7	5.05	3.23
Sa	0	-23	1.85	1.60	0	7	5.05	1.88
9a	0	-27.49	1.85	1.56	0	2.12	5.05	1.84
10a	0.02	-32.34	1.53	1.56	0.03	-2.96	4.32	1.84
lla	1	-27	1.53	3.27	1	2	4.32	3.23
12a	1	-27	1.53	3.27	1	2	4.32	3.23
13a	0	-27	1.53	1.57	0	2	4.32	1.84
14a	0	-27.49	1.85	1.56	0	2.12	5.05	1.84
15a	0.07	-42.19	1.01	1.57	0.24	-42.19	1.01	1.88
16a	1	-33	1.01	3.32	1	-25	1.01	3.37
17a	1	-33	1.01	3.32	1	-25	1.01	3.37
18a	0	-33	1.01	1.57	0	-25	1.01	1.59
19a	1	-16.34	1.53	3.34	1	35.95	4.32	3.43
20a	1	-19.77	1.53	3.32	1	34.91	4.32	3.43
21a	1	-11.98	1.85	3.33	1	41.83	5.05	3.43
22a	1	-24.25	1.85	3.25	1	24.18	5.05	3.33
23a	1	79.88	18	3.35	1	85.16	18	3.38
24a	1	70	17.70	3.29	1	70	17.70	3.29
25a	1	60	17.40	3.23	1	60	17.40	3.23
26a	0	49	17.10	2.28	0	49	17.10	2.28
27a	0.50	-27.48	1.85	2.40	0.35	2.12	5.05	2.32
28a	1	-27.48	1.85	3.23	1	2.12	5.05	3.20
29a	0	-27.48	1.85	1.56	0	2.12	5.05	1.84
30a	1	49	24.10	5.38	1	49	24.10	5.38
31a	0	-25	23.90	3.67	1	5	23.90	5.11
32a	0	-30	23.70	3.60	1	0	23.70	5.08
33a	0	-40	23.50	3.46	0	-40	23.50	3.46
34a[s]	1	-10.01	36.50	9.17	-	-	-	-
3Sah	1	-25	36.30	9.02	-	-	-	-
36a	1	-30	36.10	8.97	1.00	-9.97	36.50	9.17
37a	1	-40	35.90	8.86	1	-40	36.30	8.86

•• There is no stream data for case 6 due to the exit temperature of methane stream is -9.97℃ and the cooling range for case 6 is (S"C, O"C, -40"C). Therefore methane stream enters HXC3-3. Refer to Fig. 4 for the configuration.

4.1 Results of different operating conditions of propane evaporator on the process parameters

A simple and practical way of minimizing the energy consumption of a process is by adjusting the operating conditions of g. Based on the case studies presented, different propane evaporator operating conditions affect the overall energy consumption of the process. As depicted in Figure 5, Case 6 consume§. the lowest compression power and air cooler duty which are 129.36 MW and 342.60 MW respectively compared to the baseline case. This can be translated into an energy saving of 13.5% and 5.57 % for the compressor power and air cooler duty respectively . Meanwhile, as can be seen in Figure 6, the total propane flow rate of Case 6 is also reduced by 8.6% compared to the baseline case. Though, the overall duty of the propane evaporator remains constant, having a greater cooling duty at the intermediate stage§. of propane evaporator result§. in the increased of power consumption, air cooler duty and propane flow rate which reduces the process efficiency.
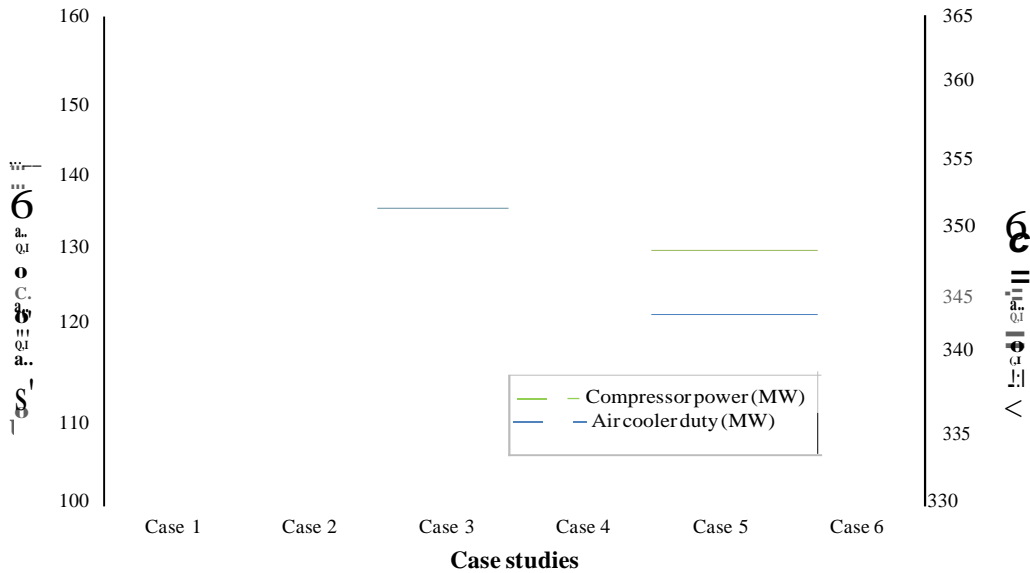

Fig. 5. Effect of different operating conditions of propane evaporator on compressor power and air cooler duty.
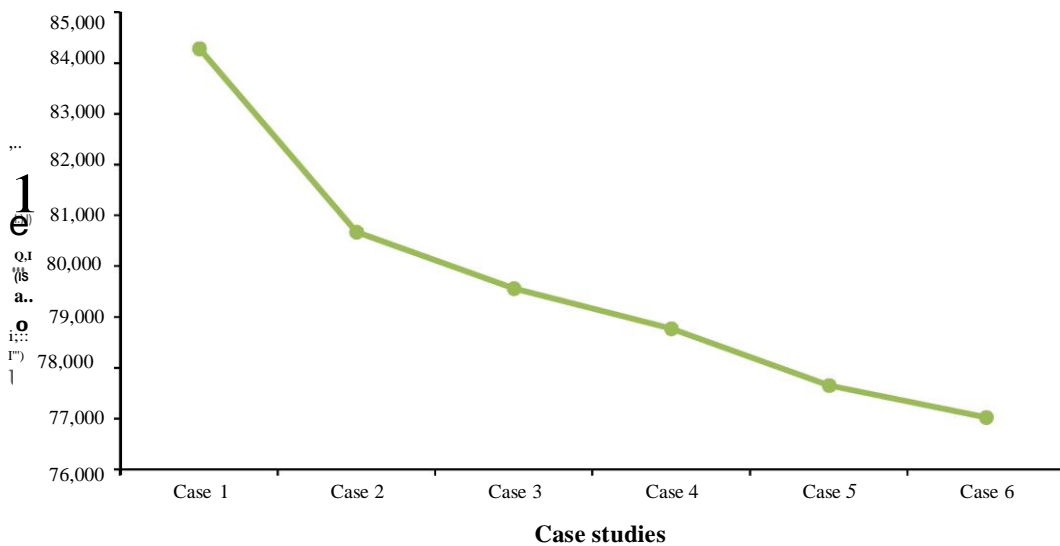

Fig. 6. Effect of different operating conditions of propane evaporator on propane flow rate.

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Distribution of the propane evaporator duty at each stage and its total area required for all case studies are reported in Figure 7 and Table 6. As can be seen in Figure 7, the cooling duty is transferred from the HP stage to the LP stage of propane evaporator (i.e. Case 1 to Case 6). Duty of the propane evaporator is determined using (Eg. 5).Rearranging (Eg.5) gives (Eg. 6) which is used to determine the

propane evaporator area. The overall heat transfer coefficient (U) for propane refrigeration cycles was taken as an average value of 425 W/(m^2 .K) [30]. Based on this U value, the propane evaporator area for each case was determined. As shown in Table 6, Case 6 gives the lowest propane evaporator area for the same cooling duty which is 46.89% lower compared to the baseline case (Case 1). This indicate§. that increasing the cooling duty at the intermediate stage§. of the propane evaporator results in the

increased of the total propane evaporator area.

$Q= UAllT_{LMTD}$ (5)

whereby $A=Ql\{U \ .llhMTvJ$ 6

$Q= $ *Duty of heat exchanger (MW)*
$U=$*Overall heat transfer coefficient (MW/m^2 °C)*
llTIMTD= log min temperature difference (°C)
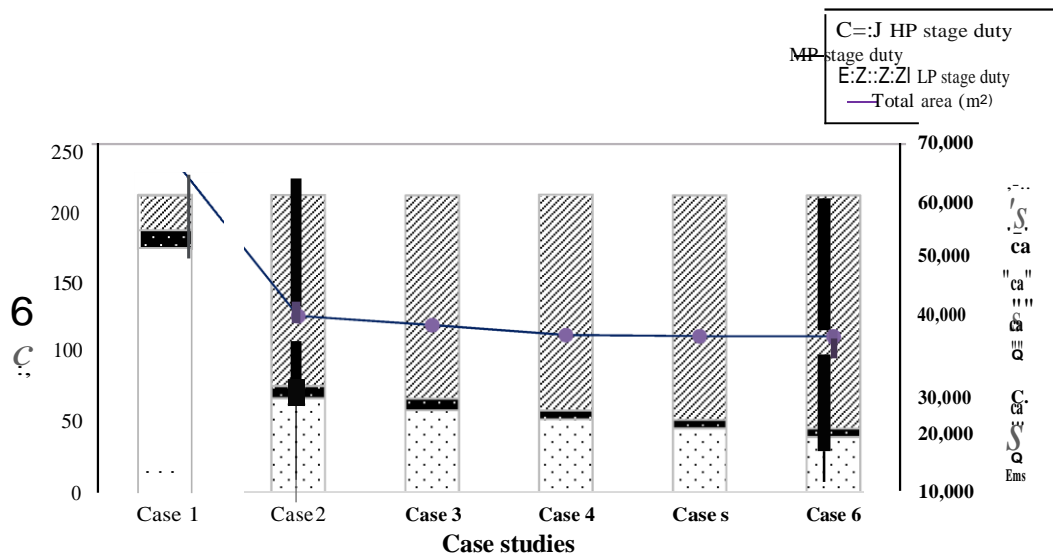
$A=$*heat exchanger area (m2)*

Fig. 7. Effect of different operating conditions of propane evaporator on its duty and total area.

Table 6
Propane evaporator duty for each stage and its total area for all case studies.

	Case 1	Case 2	Case 3	Case 4	Case s	Case 6
Duty (MW)						
HP	176.20	67.34	58.65	52.19	45.64	39.42
MP	12.21	8.72	8.36	6.33	6.13	5.98
LP	24.92	137.28	146.27	155.46	161.47	167.84
Total area (m2)	69,183.61	40,320.54	38,753.54	37,047.88	36,826.10	36,746.38

4.2 Sensitivity analysis of propane pre-cooling cycle

In this study, the sensitivity of COP, SP, exergy loss and exergy efficiency of the propane pre-cooling cycle were analysed with respect to different operating conditions of the propane evaporator. Effect of different evaporator operating conditions on COP and SP are presented in Figure 8. The exergy loss and exergy efficiency results are shown in Table 7 and Figure 9.
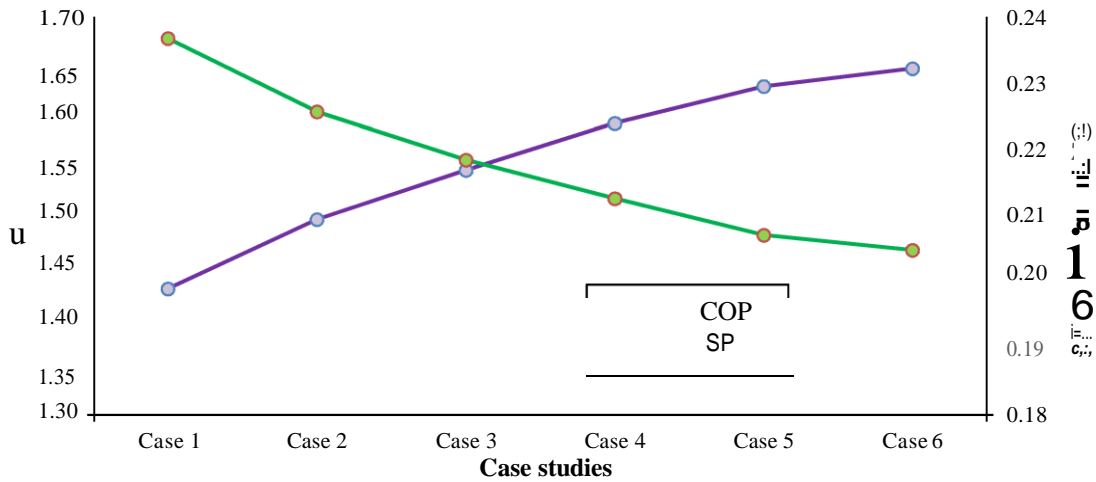

Fig. 8. Effect of different operating conditions of propane evaporator on COP and SP.

As can be seen in Figure 8, the COP of Case 6 increase§. by 15.51% while the SP reduces by 13.5% in comparison to the baseline case (ase 1).Based on the above observation, the required shaft power can be minimized by reducing the cooling duty at the intermediate stage§. of the propane evaporator. Variation of the cooling duty in the propane evaporator is achieved by manipulation of the expansion valve which i§. located prior to each stage.

Table 7
Exergy loss of each unit operation in propane cycle and exergy efficiency (%).

Exergy loss (MW)	Case 1	Case 2	Case 3	Case 4	Case s	Case 6
Heat exchangers	16.63	20.34	20.01	20.02	19.78	20.00
Compressors	30.89	28.55	27.45	26.56	25.73	25.35
Valves	38.52	28.62	25.66	22.90	20.69	19.37
Air coolers	12.74	12.20	13.08	11.61	11.77	11.49
Mixer	0.08	0.76	0.85	0.96	1.04	1.11
1]ex(%)	33.87	36.52	36.89	38.90	39.58	40.22
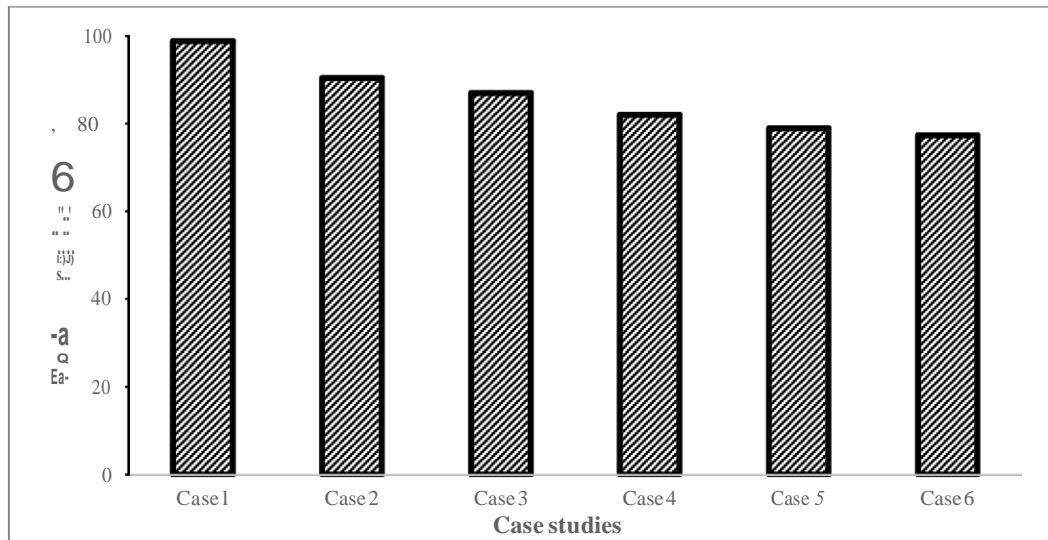
Fig. 9. Total exergy loss for each case studies.

Exergy loss for each unit operation in the propane pre-cooling cycle was determined using the equations as presented in Table 4. As shown in Table 7, the valves, compressors and heat exchangers were identified as the primary contributors to the exergy loss in the propane cycle. For the baseline case

(Case 1), the valves allocated the highest exergy loss 38.52%, followed by the compressors, 30.89% and the heat exchangers, 16.63%. Valves provided the highest exergy loss in the baseline case due to increase in entropy generation when larger pressure drop is applied across the system. Compressors are the second contributor in exergy loss due to the increase in power demand when increase in cooling duty occurred at the intermediate stage§. However, for all other case studies the exergy loss across the other components reduced marginally when a lower cooling duty was applied at the intermediate stage§. The exergy efficiency of Case 1 is the lowest {33.87%} which indicates that larger irreversibilities occur within the process. Case 6 show§the highest exergy efficiency {40.22%}which indicates high potential improvement of the process. Based on Figure 9, the total exergy loss decrease§by 21.78 % for Case 6 compared to the baseline case which improved the overall process efficiency. From the results it can be deduced that change in the operating conditions of the propane evaporator results in a lower entropy generation which reduces the exergy loss and increases the exergy efficiency. Exergy loss is influenced by larger temperature or pressure difference applied across the refrigerant stream and not by the number of equipment used in the propane cycle [31].

Based on the exergy analysis, it can be concluded that both the valves and the compressors are the main contributors of exergy loss. These unit operations can be improved by replacing the existing JT valves

with two phase expanders [18] or by using a higher efficiency compressor [32]. Nevertheless, the reduction in the energy consumption by installing these new components should be economically assessed to determine its feasibility.

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5. Conclusion

In this study, enhancement in the process efficiency of three stage propane pre-cooling cycle was studied using energy and exergy analysis. The results shows that Case 6 achieve§. the highest COP (l.65), lowest SP (0.205 MWh/tonne of LNG) and highest exergy efficiency .(40.22%). due to low

cooling duty distribution at the intermediate stage§.. Energy and exergy analysis can be a useful guide in enhancing the process efficiency of the existing LNG plant and also as a reference for future greenfield LNG projects . Changing the operating conditions of propane evaporator stage can be considered as an option to minimize energy consumption of the process which does not involve any additional cost. Additionally, this enhancement not only reduces the propane compression power but

also reduces the size of the heat exchanger as well as the refrigeration rate. Reducing energy consumption lead§. to smaller equipment sizes which generally reduce the capital and operating costs of the LNG plants.

6. Recommendations

In this study, no consideration is made from economic point of view. As this process is considered as licensed processed, cost related to the proprietary equipment such as compressor and heat exchanger are treated confidential. Thus, the detailed breakdowns on the equipment size, price, licensing fees are not available. Therefore, future work is to be considered which simultaneously provide the trade-off between the capital and operating cost of the plant with exergy analysis.

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