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Case Studies of Single Nitrogen Expander Liquefaction for FLNG

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ABSTRACT

Nowadays, global carbon dioxide emissions into the atmosphere have reached a historically high level due to carbon dioxide emissions. The current effort to change the alternative energy source of LNG fuel was found to be one of the cleanest fossil fuels due to lower carbon emissions to ease the rapid growth of carbon emissions. LNG marine industry forecasts that the increasing demand for FLNG vessels will accelerate the development of gas resources, research work on the system process analysis and optimisation of the Liquefaction system. The rapid growth in equipment and processes of FLNG development is the response to the challenges due to weight and space limits.

One of the critical objectives onboard FLNG is to build the combined plant models of clean energy. The research boundary covers processes of LNG liquefaction, nitrogen separation and boil-off gas handling. BOG re-liquefaction will be used as a fuel gas supply system. BOG will also be converted to clean energy fuel such as hydrogen or ammonia for the main and auxiliary engines onboard.

A small-scale liquefaction process with a refrigerant cycle is proposed in this study to meet these FLNG challenges. The Brayton refrigeration (BR) cycle is found to be most suitable for FLNG vessels, among other refrigerant cycles. The BR cycle using nitrogen as a refrigerant and a single expander is the focus of this study [8].

Keywords: FLNG; LNG; exergy; efficiency; liquefaction; cycle; regasification; pre-cooling;

NOMENCLATURE

Abbreviations	
NG	Natural Gas
LNG	Liquified Natural Gas
FLNG	Floating liquefied natural gas
HX	Heat Exchanger
BR	Brayton refrigeration
LMTD	Logarithmic mean Temperature Difference
Symbols:	
ΔT	Temperature difference
η_s	Isentropic Efficiency
Cp	Isobaric specific heat
C_v	Isochoric specific heat
H	Specific enthalpy
h_1	Suction enthalpy at P1, T1
h_2	Discharge enthalpy at P2, T2
$\dot{m}$	Mass flow rate
p	Pressure
S	Specific entropy
T	Temperature
$\dot{W}$	Power
U	Heat transfer Co-efficient
A	Heat Transfer Surface Area

1. INTRODUCTION

The single nitrogen expander model will be based on the Brayton refrigerant cycle and nitrogen as cryogen refrigerant. The characteristics of nitrogen are also a practical cryogen for most low-temperature applications because of its extremely low boiling temperature (–195.8°C), which is kept in the gas phase throughout the cycle.

The refrigeration capacity is high at atmospheric pressure, indicating nitrogen's critical pressure is around 33.9 bar. In contrast, the critical temperature is about 126.2 K (−146.9 °C), and nitrogen behaviour is close to ideal gas under the same conditions.

For this reason, assuming perfect gas as nitrogen refrigerant, simplified process models may be applied to optimise nitrogen expander processes for natural gas liquefaction. Also, one of the reasons is that the use of nitrogen is accessible and available onboard. The assumption is that the ideal gas law does not apply to real gases. However, ideal gas behaviour may provide decent accuracy for modelling. A simplified process model is easier to optimise. Hence, a perfect gas model for case studies was performed on simulation and optimisation of the nitrogen expander process model of a single nitrogen expander LNG process and performed optimisation using the built-in optimiser in Aspen HYSYS® [10].

2. BRAYTON CYCLES FOR LNG LIQUEFICATION

The working principle of Most LNG processes is based on vapour-compression refrigeration systems such as reverse Brayton processes (also known as expander processes). Brayton liquefier consists of a high-pressure compressor, one or more heat exchangers, an expander and a Joule- Thomson (isenthalpic) expansion valve, as illustrated in Fig 1. The basic principle of the cycle consists of compressed isentropically to HP and cooled at constant pressure to reject heat. Then the low-temperature stream is then expanded isentropically low pressure stream to absorb heat as a cold refrigerant stream.

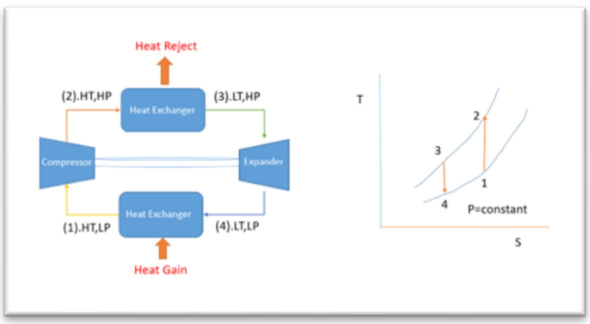

Fig. 1 Brayton Cycle for Ideal Gas

Another key aspect of the BR cycle is that the refrigeration temperature approaches ambient temperature, benefiting natural gas liquefaction. The foremost step for Liquefaction is to modify the cycle as natural gas goes through the refrigerant heat exchanger before the liquid heat exchanger uses as recuperative to reduce the significant temperature difference between natural gas and refrigerant. [3]

3. MODEL ANALYSIS FOR THE SINGLE NITROGEN EXPANDER PROCESS

A simplified process model will likely be easier to optimise, such as modelling the Nitrogen pure refrigerant cycle differently from other refrigeration cycles. This cycle involves no phase change in the exciting design of nitrogen expander processes for natural gas.

3.1 Single Nitrogen Expander with single-stage heat exchanger cooling (Model A)

As shown in Fig. 2, the single nitrogen expander process is the most straightforward N2 expander cycle. In this process, the pure N2 refrigerant is compressed and expanded before and after the heat exchanger.

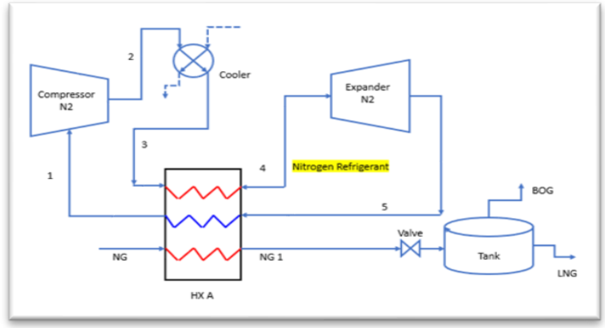

Fig. 2 Simple Nitrogen Expander (Model A)

The expansion of a high-pressure gaseous refrigerant through a turbine expander produces work, and lower pressure at the exit results in a temperature reduction. Ideally, the N2 refrigerant maintains a gaseous state throughout the process, providing the required refrigeration, including the Precooling, Liquefaction and Subcooling sections.

Fig. 1 shows the schematic plot of the temperature-entropy diagram of the cycle. The compression adiabatic process (1–2) is high pressure, the isobaric heat rejection (2–3) to the surroundings, the adiabatic expansion (3–4) in the turbine) to low-pressure and the isobaric heat removal (4–1) from hot gas. The turbine workload can share a fraction of the necessary work input for the compressor.

3.2 Single Nitrogen Expander with multistage heat exchanger cooling (Model B)

The heat extraction from cryogenic processes can be increased by adding the exchanger to closer the temperature gap in stages to make sense for a refrigeration process where the main objective of the cryogenic process is to reach a temperature level suitable for cooling between heat exchangers, as shown in Fig 3.

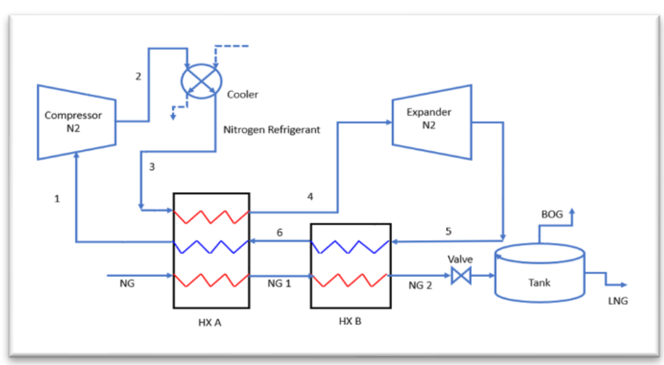

Fig. 3 Single expander with two HX (model B)

3.3 *Single Nitrogen Expander with multistage heat exchanger cooling (Model C)*

The more heat exchanger train for the Model C refrigeration process is shown in Fig 4.

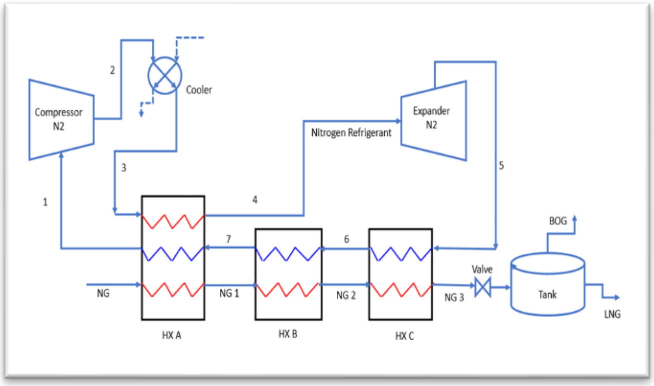

Fig. 4 Single expander with HX (Model C)

3.4 *Single Nitrogen Expander with multistage cooling (Model D)*

Fig. 5 shows that the subcooling process introducing a Joule-Thomson (JT) valve to branch out the high-pressure N2 can be further cooled liquefied to get a more efficient process based on the model reference [1]. The vaporising liquid N2 expanded for the subcooling step rather than warming up low temperature N2 vapour.

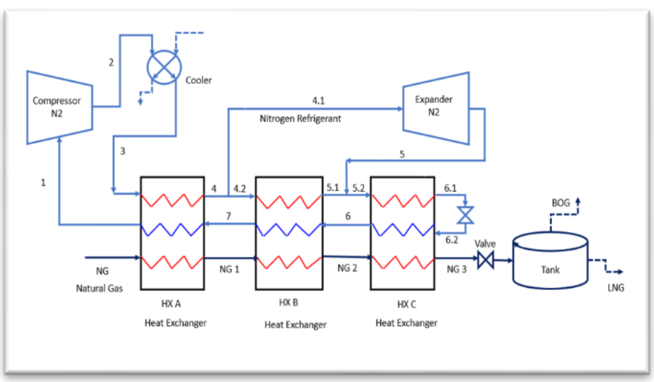

Fig. 5. Single expander with three HX and JT valve (Model D)

3.5 *Simulation for Nitrogen Expander Process*

3.5.1 Methodology

Modelling and steady-state simulation were performed using the commercial tool Aspen HYSYS for different processes. The main objective is to evaluate the various process designs and find the favourable design for single nitrogen expander Liquefaction based on specific constraints.

As for the single expander process, the temperature difference between both ends is minimum when assuming a perfect gas model. However, to avoid losses, the mixing of streams of different temperature entries is designed such that the temperature of streams 5 and 5.2 is the same. In addition, the temperature difference in the hot end of HX-C is set to the minimum temperature difference.

Therefore, it is necessary to simulate the system to find the proper process parameter, such as the pressure, temperature, and mole fraction, including the dependency of the components' parameters, to create an optimal design. The composition of LNG and Nitrogen refrigerant adopted here for the studies is given in Table 1.

Refrigerant Composition	
Refrigerant	Mole fraction
Nitrogen N_2	0.980
Oxygen O_2	0.002
Natural Gas composition	
Component	Mole fraction
Nitrogen	0.0229
Methane	0.9004
Ethane	0.0732
Propane	0.0035
i-Butane	0.0000
n-Butane	0.0000
i-Pentane	0.0000
n-Pentane	0.0000
CO2	0.0000
Oxygen	0.0000

Table 1. LNG and Nitrogen refrigerant composition

3.5.2 *Aspen HYSYS Model*

Several simple process simulations are compared to the single expander nitrogen BR cycle. Aspen HYSYS simulation built the LNG systems as steady-state models using the fluid package "Peng-Robinson" to evaluate and optimise the basic design concepts of LNG systems.

In addition, this paper focused on optimisation and modelling of cryogenic heat exchangers, compressors, and expanders in Aspen Hysys as Model A, B,C and D. Model C and Model D.

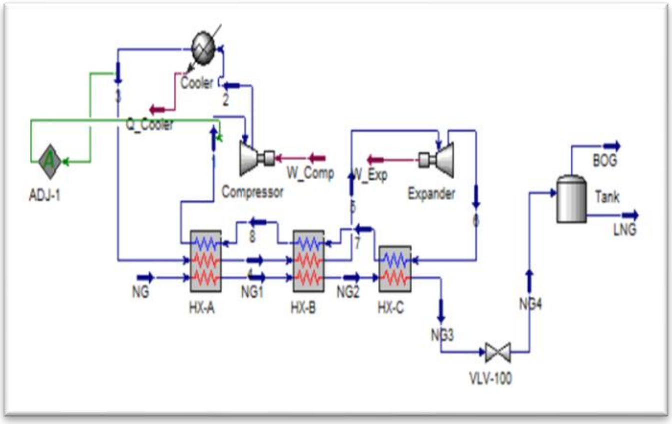

Fig. 6 Aspen HYSYS model of Single Nitrogen Expander (Model C)

The main Key variables of the single nitrogen expander process are the pressure ratio, the polytropic efficiency, and the isentropic efficiency of the compressor and the expander. Also, there will be three discrete ranges divided by the temperature ranges of natural gas. In the simulation, each step of cooling, such as the feed natural pre-cooling process from 20°C to –30°C, the Liquefaction process from –30°C to –120°C, the subcooling process from –120°C to –160°C to LNG tank.

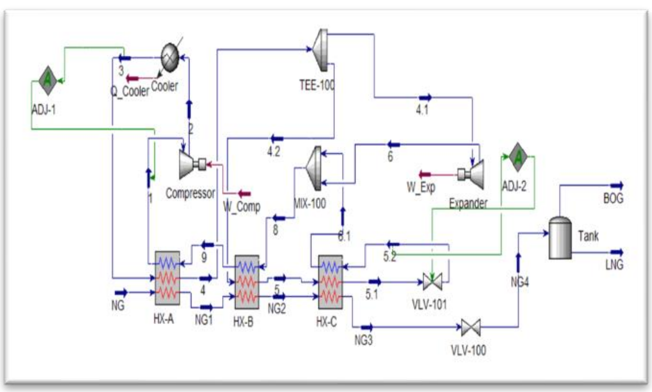

Fig. 7 Single Expander with three HX with JT valve (Model D)

The molar flow rates of the refrigerant and feed gas are constant for the whole process. The natural gas flow rate assumption is 7471 kgmole/h, and the nitrogen refrigerant is 19320 Kgmole/h. The HYSYS calculated the "UA" values of the LNG heat exchanger during natural gas and nitrogen refrigerant temperature variation.

Finally, the decision variable chosen for the simplified model is the temperature of NG between the heat exchangers HX-A, HX-B and HX-C; the degree of freedom is zero balance for design optimisation. The stream of NG assumption data is 15 Deg C and 60 bar for all models for fair testing comparison.

For model C, the stream Nitrogen (3) entry to exchanger HX-A at 15 degrees C and 80 bar is cooled in LNG heat exchanger HX-A to -67.47 Degrees C (stream 4). It was further cooled at heat exchanger HX-B, and Its pressure decreased from 80 to 10 bar by expansion in Expander; its temperature dropped to -153.5 Deg C (stream 6) as per the cycle loops described above Fig 6.

For model D, the stream Nitrogen (3) entry to exchanger HX-A at 15 Deg C and 160 bar is cooled into LNG heat exchanger HX-A, outlet stream (4) at -60 Deg C (stream 4). Then, split to stream 4.1 to the expander and stream 4.2 to further cool at heat exchanger HX-B to -95 Deg C and pass through HX-C to -115 Deg C, subcooling through JT valve to -157.3 Deg C stream 5.2 feedback to HX C and outlet stream 6.1 combined with expender outlet stream in the mixer, then go through to HX B and HX A, then to the compressor as stream 1.

3.6 Theory/calculation

The liquefaction system consists of a heat exchanger, compressor, aftercooler, and expansion units such as an expander or throttle JT valve [7].

There are two ways to determine the compression work, isentropic and polytropic approaches. In the isentropic approach, the entropy of the fluid is ideally constant before and after the compression process. The polytropic approach uses the ideal gas equation to determine the efficiency.

The compressor power can be calculated by

$$W_{Comp}= (\dot{m})(h_2\text{-}h_1) \quad (1)$$

While Isentropic efficiency calculation is shown in equations (2)(3)(4),

For a perfect gas compression reference to [2], which shows the equation with constant heat capacity, the relationship between enthalpy, pressures, and temperatures is:

$$\Delta h= C_p\,(T_2\text{-}\ T_1) \quad (2)$$

Because, for an isentropic compression, the discharge temperature is determined by the pressure ratio (with k = cp/cv):

$$T_2= T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} +T_1 \quad (3)$$

For an isentropic compression of a perfect gas, relate the isentropic head, temperature, and pressures by

$$\Delta h_s= C_pT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} -1\right] \quad (4)$$

For real gases (for which k and cp in the above equations become functions of temperature and pressure), the enthalpy of gas of known composition allows the calculation of relationships that, if any two of its pressure, its temperature, or its entropy are known.

The actual head can calculate the compression by

$$\Delta h = h(p_2, T_2) - h(p_1, T_1) \quad (5)$$

The performance of a compressor can be calculated by comparing the actual head (which directly relates to the amount of power we need to spend for the compression) with the head that the ideal isentropic compression would require.

This defines the isentropic efficiency:

$$\eta_s = \frac{\text{Isentropic Enthalpy Change}}{\text{Actual Enthalpy Change}} = \frac{\Delta h_s}{\Delta h} \quad (6)$$

After compression, the aftercooler lowers the refrigerant's temperature by rejecting its heat. The amount of heat released is calculated with equation (7).

$$Q = \dot{m}_{ref}\,(h_{in} - h_{out}) \quad (7)$$

The calculation for the Heat exchanger as per energy conservation law is shown in equation (8).

$$\sum(\dot{m}_{HE\ in} * h_{HE\ In}) = \sum(\dot{m}_{HE\ out} * \dot{m}_{HE\ out}) \quad (8)$$

To determine the heat transfer process, 'the log mean temperature difference' (LMTD) method is used to assume the specific heat capacities and the constant heat transfer coefficient [3].

The rate of heat transfer formula is

$$Q = U * A * LMTD \quad (9)$$

Where "U" would be the overall heat transfer coefficient for the process, "A" would be the surface area, and "LMTD" is the log mean temperature difference.

LMTD is a comparison of the two fluid's temperatures throughout the exchanger. LNG exchangers are mostly counter-current flow. Comparing either end of the exchanger gives a simplified example of the LMTD. This would be the hot outlet and cold inlet, or the cold outlet and hot inlet.

When there is insufficient information to calculate the Log-Mean Temperature Difference (LMTD), the number of Transfer Units (NTU) Method is used to calculate the heat transfer rate in heat exchangers (especially counter current exchangers). But NTU method will not be used for this paper.

Temperature - heat transfer Profile of LNG heat exchanger HX-B of the model (B) is shown in Fig.8.

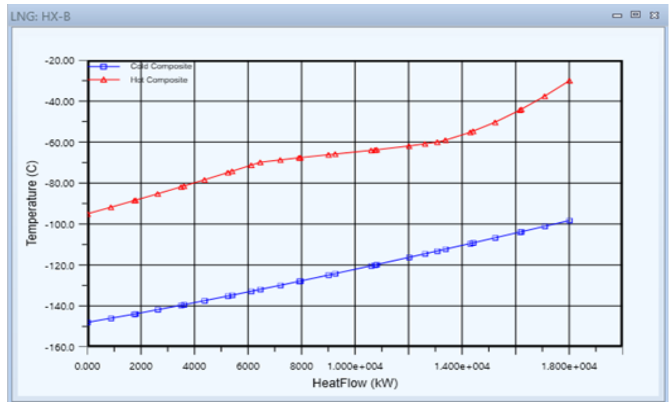

Fig. 8 Temperature – heat transfer profile of Model (B)

The temperature gap between the hot and cold fluid inside the LNG heat exchanger HX-B causes irreversibility due to the heat transfer between finite temperature differences.

Consistent with the thermodynamic second and third law, heat flow to infinite temperature difference is reversible, meaning that entropy change is zero. Whereas in finite temperature difference, the entropy change is positive so that it is irreversible. Heat transfer through finite temperature differences is irreversible because heat cannot be transferred from cold to hot temperature without additional work [d].

Infinite temperature (or negative temperature) can only be achieved with the input of more energy than comes out, and infinite temperature is fundamentally unobtainable, as is the negative absolute temperature. The heat conduction from source to sink is spontaneous, whether it is a finite or infinite temperature difference.

The effective mean temperature difference (MTD) of the heat exchanger depends on the terminal temperatures of the two streams, the distributions of flows over the transfer area with the associated local mixing effects and, most importantly, the relative directions of flow of the two streams, especially in counter flow.

Exergy measures the maximum amount of available work or work potential that can be extracted from a process in terms of the stream enthalpy and entropy relative to the surroundings when it is at steady-state thermodynamic conditions and neglecting kinetic and potential energy [5][7][9].

In the counterflow recuperative heat exchanger, maximum exergy destruction also occurs. In order to calculate the exergy balance, the general equation for the total heat exchanger area is set, shown in equation (10).

$$\Delta e_{hx} = (e_{hot\ in} - e_{hot\ out}) + (e_{cold\ in} - e_{cold\ out}) \quad (10)$$

Exergy for process calculation can be done by the equation for the state to state as equation (11)

$$e = (h - T_0 s)_{T,P} - (h - T_0 s)_{T_0 P_0} \quad (11)$$

where T_0 and P_0 are at ambient conditions, and e is exergy.

When taken from state 1 to state 2, the change in exergy is given by:

$$\Delta e = (h - T_0 s)_{T_2,P_2} - (h - T_0 s)_{T_1 P_1} \quad (12)$$

In reality, the process is nonreversible, and the actual work required is more than in an ideal case. As per the second law of thermodynamics, actual work for compression can be defined as the combination of work loss and the change in exergy:

$$W_{actual} = W_{lost} + \Delta e \quad (13)$$

and actual work for expansion can be defined as the difference between the change in exergy and work loss:

$$W_{actual} = \Delta e - W_{lost} \quad (14)$$

Exergy efficiency can be calculated as the relation between the exergy change of natural gas to be liquefied and the power consumption.

$$\eta_{exergy} = \frac{\text{Output Stream Exergy}}{\text{inlet Stream Exergy+ Energy}} \quad (15)$$

3.7 Simulation results

3.7.1 Single Nitrogen Expander

The HYSYS simulation result of a single nitrogen expander process was based on process simulation of key parameters calculated by the system. LNG heat exchangers "UA" values and "LMTD" values found in Model A are LMTD values 6.647 Deg C and UA 9329000 W/C, exchanger cold duty 62010 KW. And Other models, B, C, and D, are shown in the tables below.

Specification	unit	HX-A	HX-B
UA	W/C	10240000	162800
LMTD	Deg C	4.129	75.78
Cold duty	KW	42310	12340

Table 2. UA, LMTD and Cold Duty of Model (B)

Specification	unit	HX-A	HX-B	HX-C
UA	W/C	9798000	503000	95180
LMTD	Deg C	3.596	27.81	33.84
Cold duty	KW	35240	13990	3221

Table 3. UA, LMTD and Cold Duty of Model (C)

Specification	unit	HX-A	HX-B	HX-C
UA	W/C	2865000	312500	129500
LMTD	Deg C	12.04	57.63	54.59
Cold duty	KW	34500	18010	7070

Table 4. UA, LMTD and Cold Duty of Model (D)

		Cold in	***Cold Out***	***Hot in***	***Hot Out***	***Hot in***	***Hot out***
		KJ/Kg	KJ/Kg	KJ/Kg	KJ/Kg	KJ/Kg	KJ/Kg
HX-A	Stream	(9)	(1)	(NG)	(NG1)	(3)	(4)
		302.4	262.7	560.5	577.0	444.2	466.0
HX-B	Stream	(8)	(9)	(NG1)	(NG2)	(4.2)	(5)
		368.2	302.4	577.0	672.8	466.0	499.3
HX-C	Stream	(5.2)	(6.1)	(NG2)	(NG3)	(5)	(5.1)
		446.1	368.2	672.8	689.4	499.3	531.6

Table 5. The exergy destruction for Model D

And using equation (10) for total exergy balance, Model D shows as HX-A is 1.4, HX-B is -63.5, and HX-C is 29, respectively. According to the first and second law of thermodynamics, exergy destruction is due to irreversibility within the exergy transfer accompanying process system.

Exergy destruction is positive in an irreversible process and vanishes in a reversible process. The change in the exergy of a system can be positive, negative, or zero. When the temperature of the process where heat transfer occurs is less than the environment's temperature, the transfer of heat and exergy is oppositely directed.

Also, work and the accompanying exergy transfer can be in the same direction or oppositely directed. The exergy of an isolated system during an irreversible process continuously decreases and remains constant for a reversible process only.

The HX-A is almost close to zero, and HX-C is positive. At the same time, HX-B shows the negative as a decrease of the exergy principle, which states that in line with the The increase of entropy principle can be regarded as an alternative statement of the second law.

For an isolated system, there is no exergy transfer between the system and its surroundings; hence, the exergy changes equal exergy destroyed.

The compressor pressure ratio was adjusted to get the polytropic efficiency of 80¬85%. The expander power generation can compensate back some amount of the plant's power consumption with a compressor and a cooler load.

Specification	unit	Model			
		A	B	C	D
Compressor Power	KW	9539	71150	76250	91580
Cooler load	KW	77030	77140	82250	99270
Expander power	KW	70670	11000	14940	7485
Net Power		15899	137290	143560	183372

Table 6 Power input to Compressor and Cooler, Power output from the expander

3.8 Discussion

Achieving modest refrigeration capacities for the reversed Brayton cycle requires relatively high-pressure ratios and sizeable volumetric flow rates. The refrigerant selected will depend on the temperature sought. Pure nitrogen can be easily used from ambient down to -170°C with the possibility of subcooling to the LNG.

Using a recuperative heat exchanger, which cools below the temperature of the surroundings, leads to a significant temperature decrease after the expansion, which means that the cooling effect is much more intense.

Thus, this paper examines refrigerant constraints on an LNG process having single expander nitrogen refrigerants. Reduced energy consumption of the liquefaction process was also observed using the total UA value with ΔT.

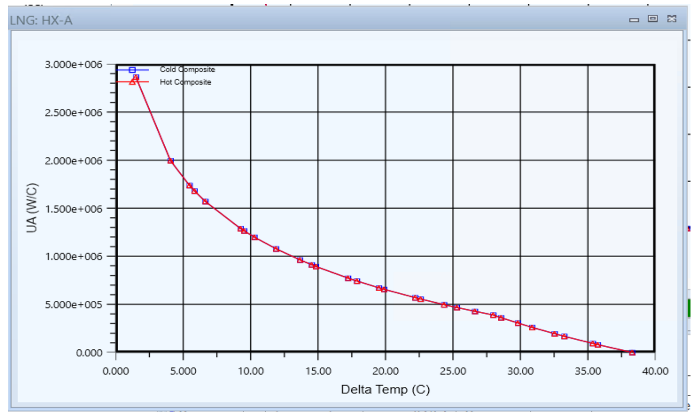

Fig. 8. Model D HX_A UA value vs ΔT

LMTD values of all heat exchangers give an idea about the exergy destruction associated with the process. If a process has shown a large LMTD (meaning the recently cooled HOT side is still at a much higher temperature than the recently heated COLD side), the heat is transferred more efficiently, and the surface area for the required heat load is reduced. If running a tight process and needing to cool very close to the target temperature, the LMTD would be very small and thus require a large amount of surface area to achieve the desired heat transfer.

For Model B, the first heat exchanger HX-A has the least exergy since the temperature gap between the working fluid is very high, LMTD is only 4.129, whereas HX-B is 75.78 Deg C, the required small surface area.

Tables 2 to 4 also illustrate that the heat exchanger results in higher LMTD values for lower operating temperatures for the heat exchangers. Thus, the UA have to pilot the temperature driving forces proportional to temperature. In contrast, the minimum temperature difference results in larger LMTD values for colder heat exchangers and smaller LMTD values for warmer heat exchangers.

An increase in the LMTD values means a more significant temperature difference between hot and cold composite curves in the heat exchangers, which results in more considerable entropy generation due to increased irreversibility.

In model D, trains are set up to liquify the Natural gas using the three stages of the heat exchanger. There can be observed that all the heat exchangers varied heat flow with Temperature. Precooling first stage HX_A shown in Fig.10

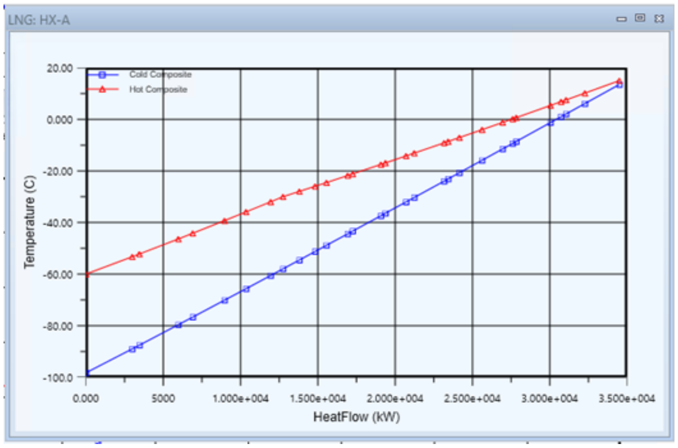

Fig.10 Model D HX_A Heat flow vs Temperature

The inlet stream 3 to the heat exchanger HX-A is 15 Deg C. The more heat in the evaporator, the higher the temperature and the larger the compressor inlet stream volume will increase compression power, which may exceed the increased refrigeration effect.

The power consumption gradually increased with the more extensive multistage heat exchanger train, as per Table 6.

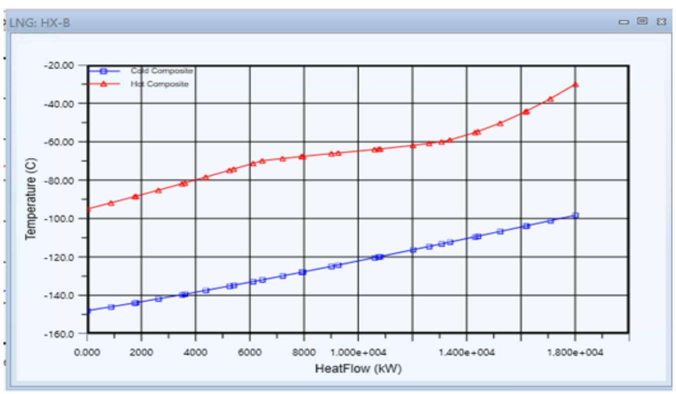

Fig.11 Model D HX_B Heat flow vs Temperature

Model D Temperature - heat transfer Profile of LNG heat exchanger HX-B is shown in Fig. 11, and HX-C is demonstrated in Fig.12. There can be seen irreversibility between the hot and cold fluid inside the LNG heat exchanger HX-B due to the heat transfer between finite temperature differences.

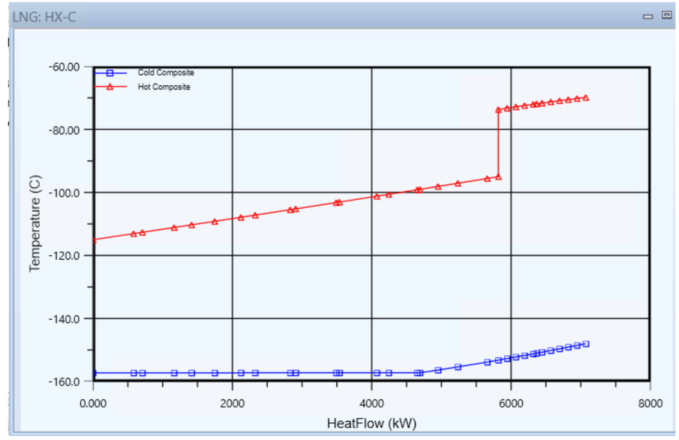

Fig.12 Model D HX_C Heat flow vs Temperature

This study's biggest concern is controlling the size of heat exchangers of FLNG to space constraints and cost factors. The size of heat exchangers can indirectly indicate the UA value, the product of the overall heat transfer coefficient (U), and the heat transfer surface area (A) shows the thermal size of the heat exchanger.

UA results show smaller driving forces for the heat exchangers operating at lower temperatures and larger driving forces for heat exchangers working at higher temperatures, reducing total irreversibility. This agrees with the results of the case studies about the optimal use of the heat exchanger area.

Table 4 shows the UA requirements and distribution among the three heat exchangers in the cycle. It can be noted that UA values of HX-A and HX-B are comparatively lesser than that of the first HX-A. This may be due to a reduction in the flow rate of the working fluid since most of the gaseous nitrogen is diverted through the expander. There can see that the first two heat exchangers have the least exergy efficiency since the temperature gap between the working fluid is very high, whereas, in HX-C.

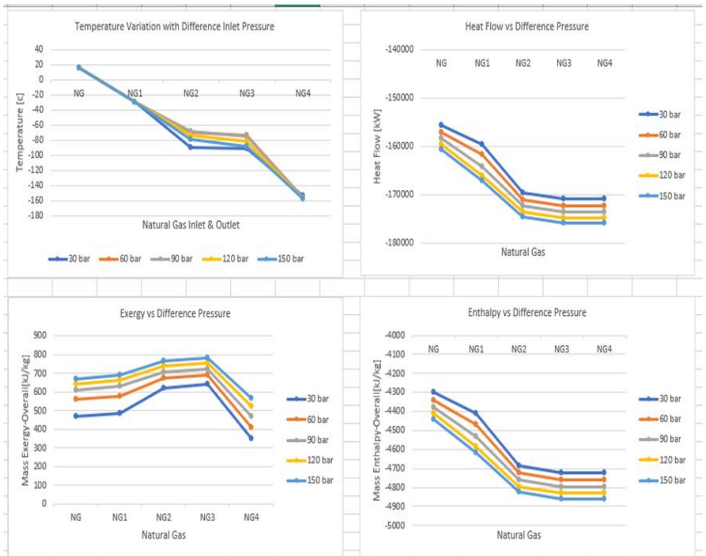

Fig.13 Temperature, Heat flow, Exergy, Enthalpy vs difference inlet pressure of natural Gas

Further complex research analysis on the difference in inlet pressure of natural gas entering the liquefaction process to meet the LNG specifications. Conventional process entry of NG at 30 to 60 bars compared with pressurised NG from 90, 120 to 150 bars as shown in Fig 13. The inlet pressure difference will effect the amount of exergy and enthalpy in such a way that more higher inlet pressure is preferred for the liquefaction process.

3.9 Conclusions

LNG processes can be classified according to the selection of refrigeration cycles used. Although, one refrigeration cycle can cover the entire range of cooling temperatures from ambient to -160 °C. However, LNG processes can be improved further by applying a more effective design of the heat exchangers.

Single Nitrogen expander evaluates process options for floating FLNG, and technology achieved a system efficiency close to cryogenic refrigeration.

Inherently, the gas expander processes displayed poor process efficiency. However, the alternative LNG liquefaction processes consume a considerable amount of power. The gas expander process is much safer than the other processes with different refrigerants since nitrogen gas is a refrigerant, and the refrigerants in the expander systems are always in the gas phase.

The system performance of liquefaction processes mainly depends on the gap between hot stream natural gas and cold stream nitrogen refrigerant composite curves in cryogenic heat exchangers. If the two curves match closely and have a small gap, entropy generation of the heat exchangers and the system will be minimised, increasing process efficiency. Thus, suggested different configurations to reduce the irreversible effect of cryogenic heat exchangers.

In the case of the multistage heat exchanger HX-C, the natural gas temperature will be slightly up at the exit due to the more significant gap between natural gas and cooling refrigerant.

Model D is slightly better for getting more down the LNG temperature. However, output power from the expander work is lower due to the split streams 4.1 and 4.2, and the LNG tank's subcooling process is still required.

The single nitrogen expander has a clear advantage for FLNG. One of the reasons is a simple system with no flammability and no need for refrigerant storage because of the gaseous phase throughout the process.

The single nitrogen expander operating in the gaseous stage without evaporation has constant specific heat. Less complex operation in a quick start-up time is an advantage of Nitrogen for FLNG consideration.

The disadvantage is that the simple nitrogen expansion cycle suffers from poor efficiency compared to the liquid refrigerant cycles. The heat dissipated by the compression loop significantly impacts the equipment's size.

However, the Nitrogen expansion cycle can enhance the flexibility of small-scale LNG and onboard FLNG. Future research will discuss the facts to consider in the case studies of dual nitrogen expander design for better efficiency.

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