Optimization of UA of heat exchangers and BOG compressor exit pressure of LNG boil-off gas reliquefaction system using exergy analysis

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Abstract. Boil-off gas (BOG) generation and its handling are important issues in Liquefied natural gas (LNG) value chain because of economic, environment and safety reasons. Several variants of reliquefaction systems of BOG have been proposed by researchers. Thermodynamic analyses help to configure them and size their components for improving performance. In this paper, exergy analysis of reliquefaction system based on nitrogen-driven reverse Brayton cycle is carried out through simulation using Aspen Hysys 8.6®, a process simulator and the effects of heat exchanger size with and without related pressure drop and BOG compressor exit pressure are evaluated. Nondimensionalization of parameters with respect to the BOG load allows one to scale up or down the design. The process heat exchanger (PHX) requires much higher surface area than that of BOG condenser and it helps to reduce the quantity of methane vented out to atmosphere. As pressure drop destroys exergy, optimum UA of PHX decreases for highest system performance if pressure drop is taken into account. Again, for fixed sizes of heat exchangers, as there is a range of discharge pressures of BOG compressor at which the loss of methane in vent minimizes, the designer should consider choosing the pressure at lower value.

1. Introduction
LNG is kept as saturated liquid at atmospheric pressure during its storage and transportation. Due to heat in-leak into the storage tanks and pipelines, LNG evaporates continuously. The generated vapour, which is called boil-off gas (BOG), is removed from the tank by flaring or reliquefaction to avoid pressure-rise beyond safety level [1-3]. In offshore applications, the BOG is liquefied in a heat exchanger by colder nitrogen gas operating in reverse Brayton cycle (RBC) [1-4]. RBC is safe and compact for LNG carrier ships with nitrogen as working fluid and its high pressure and highly efficient turbine [4-7].

Exergy based parametric optimization is a useful thermodynamic tool that helps to locate thermodynamic irreversibility in the system and provide information on its root causes because it deals with energy quality and its degradation [8-10]. It generates additional information regarding the scope for improvement of the system beyond that is provided by First Law [11]. The input exergy to the reliquefaction system (the lowest temperature is near 100 K) is provided through compressors which consumes high-grade energy i.e. electricity. Reduction of loss of exergy (irreversibility) in the system gives higher values of output exergy that is proportional to a higher recovery of useful methane returning to the tank as liquid. Exergy efficiency helps to compare performance of different systems.
designed for the same purpose. So exergy analysis is useful for selection of appropriate operating and sizing parameters and configuration of LNG boil-off gas reliquefaction system and it can be optimized for chosen objective function (such as power consumption, volume, weight etc.). Sizing of heat exchanger involves determination of appropriate heat transfer surface area and, as it is increased, it is likely to reach the point of diminishing return without considering pressure drop. However, as pressure drop is taken into consideration, additional surface area will lead to higher pressure drop resulting at an increase of compressor power. Therefore effect of pressure drop of heat exchanger on the system performance needs to be studied.

LNG boil-off gas reliquefaction system may be analyzed by considering BOG as either 100% methane or as a mixture of methane and nitrogen [12, 13]. Presence of nitrogen in BOG results in a lower dew and bubble points and hence requires more power for reliquefaction of the mixture if venting is to be avoided. Therefore, it is economical to vent some nitrogen gas provided the quantity of methane lost in the process is acceptable.

Apart from nitrogen removal, sizes of heat exchangers, discharge pressures of nitrogen compressor and BOG compressors, flow rate of nitrogen working fluid in RBC etc. play vital roles for obtaining high exergetic efficiency.

The objective of this work is to perform exergy analysis on a simple RBC based LNG boil-off gas reliquefaction system to understand the effects of these parameters and determine their appropriate or optimum values. A larger objective of this work is to begin the process of evolving guidelines regarding the choice of appropriate configuration and operating and sizing parameters of reliquefaction system in general.

2. Methodology

2.1 Thermodynamic cycle

An RBC-based BOG reliquefaction system is one of the simplest systems that can accomplish the task of reliquefaction (figure 1). BOG from the tanks (stream A) is compressed in two stages by centrifugal compressor to a pressure of 5 bar(a) (B), and liquefied (C) in the BOG condenser by the cold N\textsubscript{2} gas. After flashing in a valve, the stream D enters a separator from where CH\textsubscript{4} and N\textsubscript{2} gas (E) is vented and the liquid (F) is sent back to the tank. In RBC, nitrogen (stream 1) is compressed in three stages where heat of compression is removed in intercoolers and after cooler. Stream 2 at near ambient temperature goes into a process heat exchanger (PHX), where it is cooled by low pressure nitrogen return stream. The cooled high pressure nitrogen (stream 3) is further cooled by expanding in a turbo-expander to about 100 K (stream 4), whose output power is consumed by N\textsubscript{2} compressor system. Low pressure cold nitrogen stream liquefies high pressure BOG in the BOG condenser and the nitrogen stream is warmed up (stream 5) in the process. It is further warmed in PHX and enters the N\textsubscript{2} compressor.

![Figure 1. Schematic of LNG boil-off gas reliquefaction by Reverse Brayton system.](image-url)
2.2 Assumptions

- The system is at steady state.
- The boil-off gas from the tank is superheated due to heat inleak at the long pipeline before it reaches BOG compressor suction port. Heat inleak is neglected elsewhere in the cycle.
- There is no pressure drop in pipelines.

2.3 Nondimensionalization of heat exchangers UA

For given LNG density and heat in-leak (which is based on quality of insulation), BOG load is proportional to surface area of LNG tanks, which the ship carries. The size of a BOG reliquefaction system is represented by the flow rate of BOG. Any parameter in a reliquefaction system should be nondimensionalized with respect to the BOG load as it is the only way a designer can scale up or down the system. Specific heat of BOG at 300 K and 1 atm ($c_p$) has been multiplied to $m_{BOG}$ at the denominator as a way to arrive at nondimensionalized parameter (matching the dimension of UA).

Thus, for both the heat exchangers, UAs are nondimensionalized as

$$\text{Nondimensional UA} = \frac{(UA)_{\text{effective}}}{(m_{BOG}c_p,BOG \ (300 \ K \ & 1 \ atm))}$$

$(UA)_{\text{effective}}$ is calculated from the experimentally-determined performance (inlet and outlet temperatures) of the heat exchanger. These parameters have been defined and explained in detail by Thomas et al. [14], who have used the same procedure for nondimensionalization of a helium liquefaction system. It may be pertinent to point out that this kind of nondimensionalization should not be confused with $N_u$ (Number of Transfer Units), where UA is divided by $(m_{cp})_{\text{average}}$ of the minimum heat capacity stream of the heat exchanger. $c_p,\text{average}$ is the mean value of specific heat of the fluid calculated at an average temperature (which may be the arithmetic or harmonic mean of temperatures at number of points along the length of heat exchanger). Table 1 illustrates how nondimensionalized UA helps to scale a reliquefaction system up or down. It may be seen that when the BOG flow rate reduces by a factor of 4, the effective UA also decreases by the same factor, though the nondimensional UA remains the same in both the cases.

Table 1. Application of Nondimensionalized UA in design of BOG Condenser and PHX.

| Mass flow rate of BOG (kg/s) | $c_p$ of BOG at 300 K and 1 atm (a) (kJ/kgK) | Nondimensionalized UA | Effective UA (kW/K) |
|----------------------------|--------------------------------------------|-----------------------|---------------------|
|                            |                                            | PHX                   | BOG Condenser       |
| 2                          | 2.074                                      | 200                   | 20                  | 829.6 | 82.9 |
| 0.5                        | 2.074                                      | 200                   | 20                  | 207.4 | 20.7 |

Unless otherwise specifically mentioned, UA in the remaining text means nondimensional UA.

2.4 Conditions for modeling and simulation

Typical process parameters for onboard reliquefaction system have been given in table 2. The last row of the table shows fixed simulation conditions which were used to study the effects of UA of heat exchangers. After determining the appropriate UA, the effects of BOG compressor exit pressure have been analyzed. At that time, other parameters are kept at same values as given in table 2.

Process heat exchanger (PHX) is a gas to gas heat exchanger and BOG condenser is gas to condensing fluid heat exchanger. Heat transfer coefficients on both sides of PHX are very close, while it is at least one or two order of magnitude higher on condensing side than that of the gas side in the BOG condenser. Consequently, for equal surface area, the process heat exchanger would usually have a UA about half that of the BOG condenser. This is explained through a sample calculation. Let the
heat transfer coefficient on the gas side be 150 W/m$^2$K and that of condensing side is 4000 W/m$^2$K. Neglecting the thermal resistance of the wall, the overall heat transfer coefficient is calculated using the following equation for the same surface area on both sides:

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2}$$  \hspace{1cm} (2)

For PHX, both $h_1$ and $h_2$ will be 150 W/m$^2$K. BOG condenser $h_1$ and $h_2$ will be 150 W/m$^2$K and 4000 W/m$^2$K respectively. So, $U = 144$ W/m$^2$K.

So, for equal UAs of PHX and BOG condenser, surface area of PHX will be double that of the BOG condenser. So, pressure drop in PHX will be double as that of BOG condenser on nitrogen side. A more precise calculation has been performed below for pressure drop and heat transfer coefficient in PHX having plate-fin heat exchanger configuration.

| Table 2. Data from literature and conditions assumed for simulation. |
|---------------------------------------------------------------|
| Reliquefaction system | $N_2$ cycle high pressure (bar(a)) | $N_2$ cycle low pressure (bar(a)) | Mass flow rate of nitrogen (kg/s) | Temperature (K) | Pressure (bar(a)) | Composition (mole %) | Mass flow rate (kg/s) | BOG compressor exit pressure (bar(a)) |
|-----------------------|------------------------------------|-----------------------------------|-------------------------------|----------------|-----------------|-------------------|-------------------------------|----------------------------------|
| Tractebel Gas Engineering [1, 3] | - | - | - | 133 | 1.073 | 91% CH$_4$, 9% N$_2$ | 1.74 | 3 to 6 |
| Hamworthy Gas System Mark 1 [3] | 58 | 14.5 | - | - | - | - | - | 1.67 | 4.5 |
| Ecorel, Cryostar [2, 3] | 47 | 9.5 | - | 153 | - | - | - | 1.94 | 4.8 |
| Shin et al [13] | 46 | 13 | 24.9 | 153 | 1.06 | 91.46% CH$_4$, 0.01% C$_2$H$_6$, 8.53% N$_2$ | 1.89 | 7.2 |
| Present work | 50 | 10 | 36 | 133 | 1.073 | 91% CH$_4$, 9% N$_2$ | 2 | 5 |

PHX type: gas to gas plate fin heat exchanger; $N_2$ gas on both sides. No. of channels: 36 for both fluids. Type of fins: Plain rectangular fin; thickness: 0.5 mm; spacing: 2 mm. Spacing between the plates: 14 mm and 8 mm for high and low pressure side respectively. Calculated pressure drop (Ref [15]): 0.19 bar and 0.20 bar in high and low pressure sides respectively. Fluid properties are taken at mean temperature on each side. Heat transfer coefficient (Ref [15]): 125 W/m$^2$K and 409 W/m$^2$K at high and low pressure side respectively for PHX. Overall UA = 274 kW/K. F-factor (deterioration factor): 0.75 is assumed considering configuration effect (e.g. crossflow pattern), flow mal-distribution, axial heat conduction, heat in-leak etc. So, Effective UA: 205 kW/K. Non-dimensional UA: (205 kW/K) / (2 kg/sec X 2.074 kJ/kg.K) $\approx$ 50 Thus, for nondimensional UA=50, a pressure drop of 0.2 bar is selected for both the channels in PHX. Table 3 shows that pressure drop assumed by other researchers are close to our calculation.

The system is simulated with the aid of Aspen Hysys 8.6®, a commercial process simulator. Peng-Robinson equation is used in simulation for determining thermo-physical properties of both BOG and nitrogen [16]. Adiabatic efficiencies of nitrogen compressors, BOG compressors and turbine are assumed as 70%, 70% and 75% respectively.

2.5 Exergy Analysis

Exergy analysis is used to estimate the extent of thermodynamic irreversibility and performance of the reliquefaction system. A higher exergy efficiency of any system may be attributed to either improvement of exergy output (reliquefaction rate) or reduction of input power or both. In BOG reliquefaction system, a higher exergy output means a higher recovery of BOG (as liquid) with higher methane content in it. The following expressions are used for calculating exergy [4, 11]:

$$\text{Exergy} = \int_{T_1}^{T_2} \left( h - T s \right) dt$$

where $h$ is enthalpy, $T$ is temperature, and $s$ is entropy.
Table 3. Pressure drop in BOG Condenser and Process Heat Exchangers.

| Reference               | Pressure drop in BOG Condenser (bar) | Pressure drop in Process Heat Exchangers (bar) |
|-------------------------|-------------------------------------|----------------------------------------------|
|                         | High pressure side (N₂ side)        | Low pressure side (BOG side)                 |
| Shin et al. [13]        | 0.3                                 | 0.1                                          |
| Sayyandi et al. [17]    | 0.1                                 | 0.03                                         |
| Present work            | 0.1                                 | 0.03                                         |

The specific physical exergy is given by

\[ e_{x,\text{ph},i} = [(h_i - h_0) - T_0 (s_i - s_0)] \]  

(3)

where \( h_i, s_i \), and \( h_0, s_0 \) are specific enthalpy and specific entropy for any state \( i \) and ambient conditions (298 K, 1 atm) respectively.

Net exergy output of the system,

\[ E_{x,\text{output}} = \dot{m}_F e_{x,F} - \dot{m}_A e_{x,A} \]  

(4)

where \( \dot{m}_A \) and \( \dot{m}_F \) are the mass flow rates of gaseous BOG (entering nitrogen compressors) and reliquefied BOG (returning to the tank) respectively.

Net exergy input to the system,

\[ E_{x,\text{input}} = W_{\text{NET}} = W_{N_2,\text{COMP}} + W_{\text{BOG,COMP}} - W_{\text{EXP}} \]  

(5)

\( W_{N_2,\text{COMP}} \) represent power input to nitrogen compressors, \( W_{\text{BOG,COMP}} \) represents power input to BOG compressors and \( W_{\text{EXP}} \) is the expander output which is consumed by nitrogen compressor system.

Physical exergy efficiency of the system,

\[ \eta_{e,x} = \frac{E_{x,\text{output}}}{E_{x,\text{input}}} = \frac{\dot{m}_F e_{x,F} - \dot{m}_A e_{x,A}}{W_{\text{NET}}} \]  

(6)

In this work, exergy efficiency of the system is calculated by considering only physical exergy. However, chemical exergy also becomes significant because reliquefaction of BOG is an open system, where a small part of the gas is vented out. The vent gas is mixture of methane and nitrogen. The former has a higher chemical exergy because it is a fuel gas, while the latter is a neutral gas possessing very low chemical exergy in comparison. As the methane content in the vented gas reduces, so does the loss of chemical exergy. Though the value of chemical exergy of the vented gas is much higher than the physical exergy, any reduction of loss of chemical exergy at the vent would also reflect as an increase of physical exergy output (\( \dot{m}_F \) increases in Eq. (6)) and hence would necessarily improve performance of the system even when exergy efficiency is calculated based on physical exergy alone.

The specific chemical exergy (kJ/mole):

\[ e_{x,\text{ch}} = \sum x_k e_{x,\text{ch}}^0 + \sum x_k \ln(x_k) \]  

(7)

where \( x_k \) is mole fraction of any mixture component and \( e_{x,\text{ch}}^0 \) is its standard chemical exergy, which for nitrogen = 0.72 kJ/mole, and for methane = 831.6 kJ/mole [14].

Total chemical exergy

\[ E_{x,\text{ch}} = e_{x,\text{ch}} \times MF \]  

(8)

where \( MF \) is the molar flow in moles/s.

3. Results and discussions

3.1 Effect of heat transfer area of heat exchangers on the performance of reliquefaction system

Figure 2 shows physical exergy destruction in components. Nearly half of the total exergy destruction of reliquefaction system occurs in nitrogen compressor section. Compressor efficiency may be improved by increasing the number of compression stages and improving its adiabatic efficiency. More stages mean requirement of more space and capital cost, while improvement of adiabatic efficiency requires refining the design of compressor blade profiles and also operating it at the rated condition. However, these aspects are not within the purview of the present study.

The exergy destruction of phase separator is neglected for further analysis as it is less than 1% of total exergy destruction. Exergy loss in expansion valve cannot be reduced any further because the inlet and outlet pressures and mass flow rate are fixed. Exergy efficiency of turbine is proportional to its adiabatic efficiency, which, like the compressor, depends on the efficacy of design and also on its
operation at the rated condition [4, 18]. For a given configuration of the reliquefaction system, the improvements suggested are confined to the performance of heat exchangers and the operating parameters of the BOG compressor as far as the present work is concerned.

The possible heat exchange in a heat exchanger is limited by the approach temperature between the hot and the cold stream. When the approach temperature is small, the energy savings are high but the surface area (capital investment) required is also high. As the approach temperature is increased, the investment decreases while the operating cost increases. The minimum approach temperature ($\Delta T_{\text{min}}$), which is the smallest temperature difference between the hot and the cold streams in the heat exchanger, can be used as a parameter to determine the optimal size of the heat exchanger. Sometimes in natural gas liquefaction or vaporization processes, large variation of $c_p$ in one or both the fluids results in “kinked” nature of the temperature profiles and the point of $\Delta T_{\text{min}}$ can occur anywhere in the region of internal heat exchange and not just at one end. It is called a pinch or an internal pinch. The “Pinch Point”, however, is not a point (i.e. $\Delta T_{\text{min}}$ is not zero) but two points, called hot pinch point and cold pinch point. Above the pinch point (or just pinch), the streams have $c_{p,h} \leq c_{p,c}$ while it is $c_{p,h} \geq c_{p,c}$ below the pinch (assuming mass flows remain constant). An increase in $\Delta T_{\text{min}}$ results in higher energy costs and lower capital costs. A decrease in $\Delta T_{\text{min}}$ results in lower energy costs and higher capital costs. An optimum $\Delta T_{\text{min}}$ exists where the total annual cost of energy and capital costs is minimized.

Figure 3 shows that pinch point occurs in BOG condenser where BOG is desuperheated, condensed and subcooled by cold nitrogen which remains in gaseous phase throughout. Two temperature profiles are shown for the BOG condenser (UA fixed at 20), while UA of PHX are different at 80 and 260. It may be observed that when the PHX UA is 80, the pinch point occurs at dew point of BOG somewhere in the middle of the heat exchanger. The corresponding heat duty is shown as $\Delta H_1$. When the PHX UA is increased to 260, more heat transfer takes place ($\Delta H_2$ is higher) and the pinch point shifts to the coldest end of the BOG condenser. Pinch temperature difference decreases for UA=260, which is the result of the higher surface area (higher capital cost). However, the heat transfer increases ($\Delta H_2 > \Delta H_1$), which results in higher performance.

Exergy efficiency of the system has been plotted in figure 4 for a range of UA of PHX at different UA of BOG condenser. It also shows variation of $\Delta T_{\text{min}}$ (in BOG condenser) as functions of UA of PHX. For UA of BOG condenser 40, when UA of PHX reaches 140, the pinch temperature difference in BOG condenser becomes zero. At this point the exergy efficiency saturates and there is no increase of UA for PHX. Any further addition of UAs will only lead to wastage of capital cost (surface area) without any improvement of performance of the system (exergy efficiency). However, for UA of BOG
condenser 10, this condition (of saturation) is reached at a much higher UA of PHX though albeit the saturation occurs at a much lower value of exergy efficiency. UA of BOG at 20 is a good compromise which gives a higher energetic efficiency and also the scope for increase of UA of PHX beyond 140. These calculations may be helpful for the designer to choose the sizes of PHX and BOG condenser for optimized capital (surface area) and operating cost (exergy efficiency) of the reliquefaction system. The $\Delta T_{\text{min}}$ can be seen to peak at a certain value of UA of PHX. However, the exergy efficiency continues to rise to the saturation value. This trend can be explained by the fact that to the left of the peak value of $\Delta T_{\text{min}}$, pinch occurs somewhere in-between the end points of the heat exchangers, while to the right of the peak, the pinch shifts to the cold end and total heat transfer ($\Delta H$) increases and the performance of the system improves.

Figure 5 shows the effects of UA of PHX on exergy destruction in components of reliquefaction system. The exergy destruction occurring in PHX reduces with increase of UA of PHX. Improved performance of PHX decreases the inlet temperature of turbine, which reduces exergy destruction in turbine. Reduction of temperature at the inlet of expansion valve also reduces exergy destruction in it. However, exergy destruction in BOG condenser shows an opposite trend. Decrease in turbine outlet temperature means that nitrogen stream enters BOG condenser colder. As a result, BOG outlet (as liquid) becomes more subcooled and condenser load increases. Exergy destruction in BOG condenser is proportional to the heat load, which increases exergy destruction in BOG condenser.

**Figure 4.** Effect of nondimensional UA of PHX on exergy destruction in components.

**Figure 5.** Effect of nondimensional UA of PHX on exergy destruction in components.

Apart from analysing performance of the system using physical exergy efficiency, it is required to quantify the amount of vent gas as well as the amount of methane in it. Figure 6 shows the effects of UA of PHX and BOG condenser on chemical exergy and mole fraction of nitrogen of vent gas. Since nitrogen is inert, the chemical exergy of BOG is almost the same as chemical exergy of methane. It may be observed that for any UA of BOG condenser, the UA of PHX has to be kept above 140 so that the methane content in vent gas is sufficiently reduced and more quantity of non-condensable nitrogen is removed. When the UA of BOG condenser is 40, occurrence of zero pinch point temperature difference limits any further addition of PHX UA. When UA of BOG condenser is 10, nitrogen vent reduces and chemical exergy of vent increases. If UA of BOG condenser is kept at 20, UA of PHX can be increased beyond 140 to save chemical exergy of BOG. Thus from figure 4 and 6, it can be concluded that the preferred UA for BOG condenser is 20 and that of PHX is 260 for better physical exergy efficiency and nitrogen removal. However, the heat exchanger sizes would finally be decided from total cost calculation which is not included in the present study.
3.2 Effect of heat transfer area with pressure drop in heat exchangers on reliquefaction system performance

For a particular configuration of flow passage, increase of UA linearly increases pressure drop in both heat exchangers. Pressure drop reduces exergy output of the system and hence its effect on the performance needs to be evaluated. For UA = 50 for both heat exchangers, we have chosen pressure drops as 0.2 bar for both sides of PHX and 0.1 bar and 0.03 bar on nitrogen side and BOG side respectively in BOG condenser as shown in table 3. For evaluating the effect of pressure drop in heat exchangers, the pressure drop of each stream considered in previous simulation is increased in steps of 2: by two times and four times. Figure 7 shows that the impact of pressure drop becomes significant for UA of PHX above 100. While exergy efficiency rises continuously with increase in UA of PHX, it reaches maximum values when pressure drop is taken into consideration. Optimum UA of PHX shifts to lower (than saturated values without considering pressure drop) values as pressure drop increases.

Figure 6. Effect of nondimensional UA of PHX on vent gas exit conditions.

Figure 7. Effect of nondimensional UA of PHX and pressure drop (PD) on exergy efficiency.

3.3 Effect of BOG compressor exit pressure on reliquefaction system performance

Figure 8 shows effect of compressor exit pressure on physical exergy efficiency of the system and fraction of chemical exergy lost through the vent gas. The values remain unchanged from 3 to 6 bar(a).

Figure 8. Effect of variation of BOG compressor exit pressure on physical exergy efficiency of reliquefaction system (without pressure drop) and fraction of chemical exergy of vent gas [UA of PHX = 260, UA of BOG Condenser = 20].

Figure 9. Effect of variation of BOG compressor exit pressure on exergy destruction in components of reliquefaction system (without pressure drop) [UA of PHX = 260, UA of BOG Condenser = 20].
From figure 9 it may be observed that when the BOG compressor exit pressure is varied, the effect of decrease of exergy destruction in PHX is almost nullified by the increase in exergy destruction in BOG condenser and turbine. Total exergy destruction in these component remains almost the same. Higher pressure generates more heat of compression, which is removed in BOG condenser increasing its exergy destruction. Higher temperature of BOG stream increases exit temperature of nitrogen stream from BOG condenser which in turn increases the exit temperature of high pressure nitrogen stream of PHX. As a result the inlet temperature of turbine increases and its exergy destruction increases. Exergy destruction in PHX reduces because the temperature difference between cold and hot fluid decreases.

4. Conclusions
Exergy analysis reveals the effects of variation of UA of both the heat exchangers and BOG compressor exit pressure on performance of LNG boil-off gas reliquefaction system. Process heat exchanger has to be significantly larger than the BOG condenser to get a reasonable system performance. UA of PHX of about 140 reduces the vent gas and its methane content. System is found to become more sensitive to pressure drop when the UA of PHX is greater than 100. With increasing pressure drops in heat exchangers, overall system performance deteriorates and the optimum UA of PHX reduces. For fixed UAs of heat exchangers, optimum discharge pressure of BOG compressor that gives maximum physical exergy efficiency and minimum chemical exergy of vent gas remains flat for a significant range allowing a low discharge pressure to be selected.

There is a scope of improving the system performance further by studying the effect of other process parameters like nitrogen compressor system discharge pressure, ratio of mass flow rates of nitrogen to BOG etc. Of course, improvement can always be achieved by increasing the number of stages and adiabatic efficiency of nitrogen and BOG compressor systems. Modification of the configuration of RBC with precooling stages, multiple expander cycle, arrangement of heat exchangers etc. may further increase overall exergy efficiency of the reliquefaction system. All these remain subjects of proposed future studies.

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