Journal of the Korea Institute of Applied Superconductivity and Cryogenics
Vol.8, No.4, (2006), pp.34~38

Cycle Analysis on LNG Boil-off Gas Re-Liquefaction Plant

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***Abstract*-- Cycle analysis was performed in order to find the optimum design point of the LNG Boil-off gas re-liquefaction system. Thermodynamic analysis revealed the system could be defined by three state variables. Thus the system performance could be described by the three cold endpoint temperatures of the three-pass heat exchanger. This enabled us to investigate the cycle performance in terms of the heat exchanger parameters. To get access to the cycle states of higher system performances, larger heat exchangers were found necessary. Also the thermal pinch in cryogenic heat exchangers was found to act as a limiting factor to the system performance.**

1. INTRODUCTION

Lately, there has been a significant increase in the level of interest on environmentally friendly and economically viable solutions for the transport of Liquid Natural Gas (LNG) [1-2]. Actually, LNG carriers have been driven by steam turbines and boil-off gas (BOG) from the LNG cargo has been used as fuel or vented. The high fuel consumption of steam turbines compared to the last-generation diesel engines will eventually motivate their replacement. This will also be encouraged by environmental concerns and future regulations. Alternative propulsion systems such as diesel engines are being equipped on the LNG carriers for better fuel efficiency. Therefore the liquefaction of boil-off gases on LNG carriers results in increased cargo deliveries and allows owners and operators to choose the most optimal propulsion system. Instead of the common application by using the boil-off gas as fuel, the LNG BOG re-liquefaction system establishes a solution to liquefy the boil-off gas and the LNG back to the cargo tanks. The LNG re-liquefaction system has some merit such as large savings in total fuel consumption and improved propulsion redundancy.

The reverse Brayton cycle is widely used as the LNG re-liquefaction plant [3-4]. In the reverse Brayton cycle, the expander is used as the main component to remove energy from the nitrogen gas stream. To increase the efficiency of the re-liquefaction system, a lower temperature at the expander outlet is required. In that case, the liquid droplet may be formed at the expander outlet. The liquid has a much lower compressibility than the gas; therefore, if liquid were formed in the expander, high momentary stresses would result. However, the Claude cycle, which we have adopted for the re-liquefaction of LNG BOG, has two expansion units such as expander and expansion valve.

We can obtain higher system efficiency by using the expander and achieve lower temperature by using the expansion valve. In this study, the focus will be on the thermodynamic analysis of the system, which will form a base for the future optimization of the system components such as heat exchangers and turbo-expanders, etc.

2. LNG BOG RE-LIQUEFACTION PLANT

The LNG BOG re-liquefaction plant, as shown in Fig. 1, is basically made of two cycles, that is, the BOG cycle and the nitrogen cycle, and the nitrogen cycle is based on the Claude cycle [5].

In a BOG cycle, BOG at $-120°C$ is released from the LNG tank when the tank pressure reaches a preset level slightly above the atmospheric pressure. BOG is withdrawn by means of a two-stage compressor and compressed to $P_{BOG} = 3bar$. The BOG temperature at the compressor discharge side is estimated $T_{BOG,1} = -57.8°C$ with a compressor efficiency of 75% and pure methane for the BOG composition was assumed.

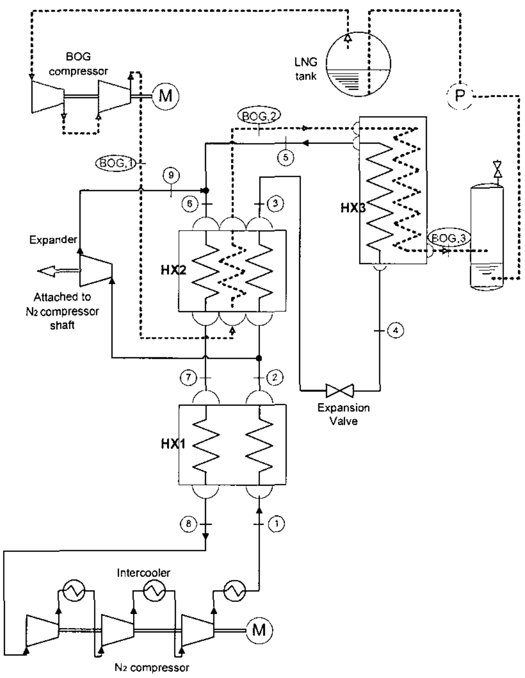


Fig. 1. Schematic of LNG BOG re-liquefaction plant.

Once the BOG is compressed, it is led inside the cryogenic plate-fin heat exchanger and goes through a two-stage cooling process. At the second heat exchanger (HX2), the BOG is pre-cooled down to a certain temperature $T_{BOG,2}$ and then condensed in the third heat exchanger (HX3) from $T_{BOG,2}$ to the design temperature of $T_{BOG,3} = -160°C$, after which the LNG is pumped back to the LNG tank by means of a cryogenic return pump.

The nitrogen cycle is composed of a three-stage centrifugal compressor, inter/after-coolers and three plate-fin heat exchangers. The nitrogen gas enters the three-stage compression and cooling process at $T_8 = 35°C$, $P_L = 14bar$ flows out at $T_1 = 40°C$, $P_H = 58bar$. The compressor work for the process is estimated $w_c = 177.7kJ/kg_{N2}$ for a compressor efficiency of 75%. The high-pressure nitrogen flow, after a pre-cooling in the first heat exchanger (HX1), divides at cycle location 2 and only a fraction $(1-r)$ proceeds to the expansion valve. The remainder bypasses HX2 and is taken instead to a turbo-expander. The gas drops in pressure when passing through the expander in consequence of the work performed therein. This cold fluid from the expander passes back to the compressor through the cold side of HX2, thereby providing a means of pre-cooling the high-pressure fluid before it enters HX3 at cycle location 3. A fraction $(1-r)$ of the high-pressure nitrogen, as it crosses the expansion valve, turns into a mixture of saturated liquid and vapor at $-164°C$, $14bar$. This provides a cold heat, the temperature of which is low enough to liquefy the BOG in HX3. The nitrogen vapor flows out of HX3 at the BOG condensation temperature of $T_5 = -146.4°C$ and returns to the compressor through the cold sides of HX2 and HX1.

3. CYCLE ANALYSIS

3.1. Thermodynamic Analysis

Application of the steady-flow energy balance successively to HX1, HX2, HX3 and the mixer yields the following equations:

$$h_1 - h_2 + h_7 - h_8 = 0 \tag{1}$$

$$(1-r)(h_2 - h_3) + h_6 - h_7 + m_{BOG}(h_{BOG,1} - h_{BOG,2}) = 0 \tag{2}$$

$$(1-r)(h_3 - h_5) + m_{BOG}(h_{BOG,2} - h_{BOG,3}) = 0 \tag{3}$$

$$(1-r)h_5 + rh_9 - h_6 = 0, \tag{4}$$

where m_{BOG} is the ratio of the BOG mass to the total high-pressure nitrogen mass and r is the ratio of the expanded mass through the expander to the total high-pressure nitrogen mass. For the expander, the adiabatic efficiency is defined by

$$\eta_e = \frac{h_2 - h_9}{h_2 - h_{9s}}, \tag{5}$$

where $\eta_e = 85\%$ was assumed. The cooling of the hot streams by the cold within HX2 can be ensured if the following constraints were imposed on the terminal temperatures of HX2:

$$T_6 \le T_3 \tag{6}$$

$$T_6 \le T_{BOG,2} \tag{7}$$

$$T_7 \le T_{BOG,1}. \tag{8}$$

Since T_8 is fixed to a temperature $5°C$ lower than T_1 as shown in Table 1, the inequality $T_7 \le T_2$ is satisfied by itself and thus need not be implemented. Also note that there is no inequality in the relationships between T_3 and $T_{BOG,2}$, and between T_2 and $T_{BOG,1}$. Since the saturation temperature of the nitrogen at $14bar$ is $-164°C$, the system operation avoiding the liquid droplet formation at the expander outlet can be reflected by

$$T_9 \ge -160°C \tag{9}$$

where a safety margin of $-164°C$ was considered.

Table 1 summarizes the given conditions for cycle analysis. Since the five equations, e.g.,(1)-(5), involve the eight cycle variables, e.g., T_2, T_3, T_6, T_7, T_9, $T_{BOG,2}$, m_{BOG}, and r, the cycle state is defined by three cycle variables. Their domains are bounded by the inequality relations of (6)-(9). In this study, the cycle state will be mainly described by the three terminal temperatures of HX2, e.g., T_3, T_6 and $T_{BOG,2}$.

TABLE I
GIVEN CONDITIONS FOR CYCLE ANALYSIS.

	T(℃)	P(bar)
BOG press (P_{BOG})		3
Cycle location BOG,1 ($T_{BOG,1}$)	-57.8	
Cycle location BOG,3 ($T_{BOG,3}$)	-160	
N_2 high press (P_H)		58
N_2 low press (P_L)		14
Cycle location 1 (T_1)	40	
Cycle location 5 (T_5)	-146.4	
Cycle location 8 (T_8)	35	
N_2 –compressor work, $w_c = 177.7kJ/kg_{N2}$		
Expander efficiency, $\eta_e = 85\%$		

Fig. 2 exemplifies the domain of the cycle states permitted by (1)-(9). For a given value of $T_{BOG,2} = -140°C$, each of (6)-(9) is represented by a line in the T_3-T_6 plane and the points within the closed contour are the states which the system can take. It is noted that (8) and (9) are reflected through the governing equations and take part in marking the boundary of the state domain. T-s diagram of the cycle state **S** is given in Fig. 3, which illustrates other cycle variables of the state **S**.

The performance of a re-liquefaction cycle is commonly evaluated by the amount of the expander work and the liquefied BOG mass achieved by a unit mass of refrigerant circulating the cycle. The performance indicator is the specific liquefaction work defined by:

$$w_s = \frac{w_c - w_e}{m_{BOG}}, \tag{10}$$

where w_e is the expander work by unit mass of compressed nitrogen, $w_e = r(h_2 - h_9)$. Application of energy conservation to the nitrogen stream from 1 to 8 yields

$$w_e - m_{BOG}(h_{BOG,1} - h_{BOG,3}) = h_1 - h_8. \tag{11}$$

Substituting (11) into (10), it follows that

$$w_s = \frac{w_c - (h_1 - h_8)}{m_{BOG}} - (h_{BOG,1} - h_{BOG,3}) \tag{12}$$

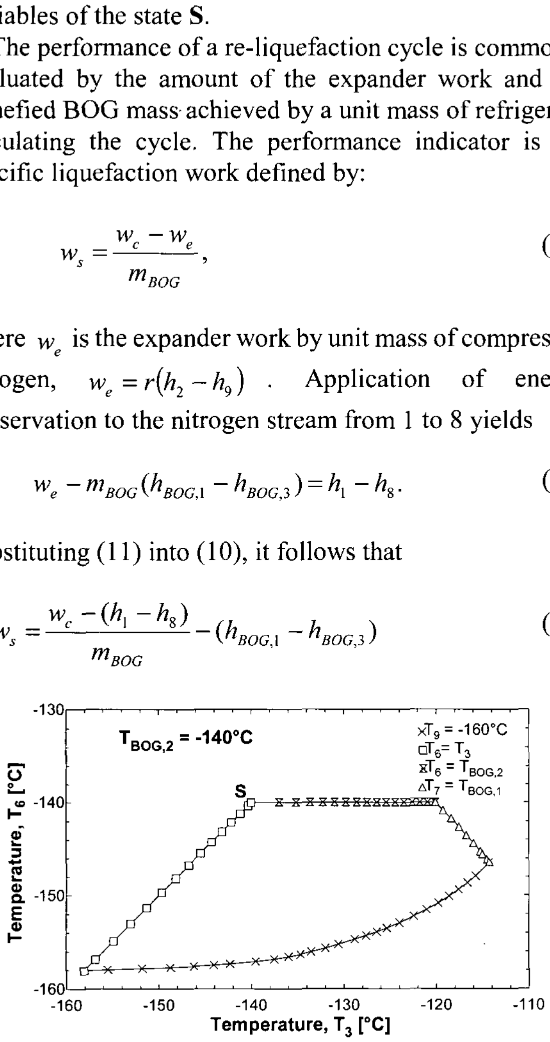


Fig. 2. Domain of cycle states.

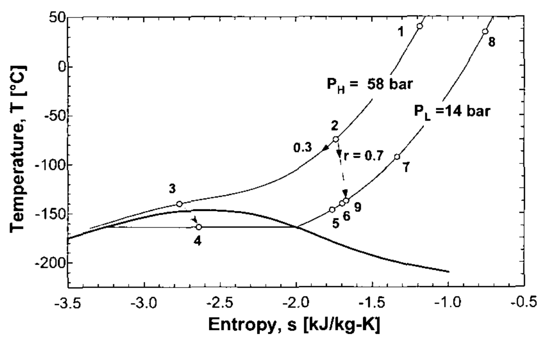


Fig. 3. T-s diagram of cycle state **S**.

Since h_1, h_8, $h_{BOG,1}$ and $h_{BOG,3}$ are constants, w_s can be treated as an another cycle variable. The iso-lines of w_s and r are shown in Fig. 4. It is seen that for a given value of T_3, a more power-saving cycle state can be found if one chooses T_6 closer to T_3. The system will be most efficient if designed to operate at cycle state **S**. However, this is not possible, which will be discussed in Sec 3.2. Further calculations with different values of $T_{BOG,2}$ were performed and the cycle state of the minimum w_s was concluded to be at $T_{BOG,2} = -140°C$.

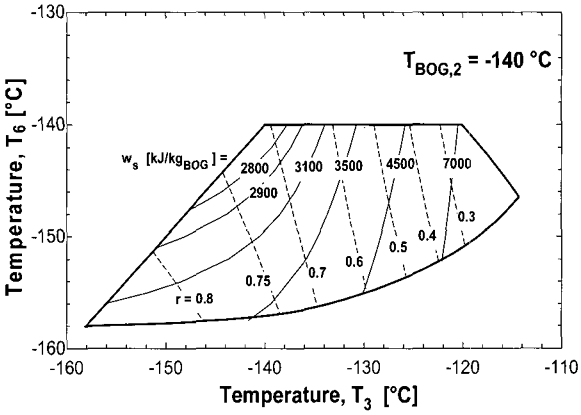


Fig. 4. Iso-lines of specific liquefaction work and ratio of expanded mass.

3.2. Heat Exchanger Analysis

The HX2 is a three-pass counter-flow heat exchanger within which two hot fluids are cooled down by a cold nitrogen stream. The relative amounts of heat removed from the two hot fluids is controlled by the ratio of the overall heat transfer coefficients

$$\kappa = U_{Bc} / U_{hc},$$

where the subscripts B, c and h refer to the BOG, cold and hot nitrogen, respectively. Application of the energy balance to the incremental length dx of the heat exchanger in the cold endpoint direction yields

$$(1-r)\dot{m}_c c_{p,h} dT_h = -Ubdx(T_h - T_c) \tag{13}$$

$$m_B \dot{m}_c c_{p,B} dT_B = -\kappa Ubdx(T_B - T_c) \tag{14}$$

$$c_{p,c} dT_c = m_B c_{p,B} dT_B + (1-r)c_{p,h} dT_h \tag{15}$$

where U is the overall heat transfer coefficient between the hot and cold nitrogen stream, b is the width of the heat exchanger, $\dot{m}_c$ is the mass flow rate of the cold nitrogen, and c_p is the constant-pressure specific heat. Equations (13)-(14), after some manipulations, can be rewritten for the local temperature differences $\Delta_h = T_h - T_c$, $\Delta_B = T_B - T_c$ and the BOG temperature T_B as

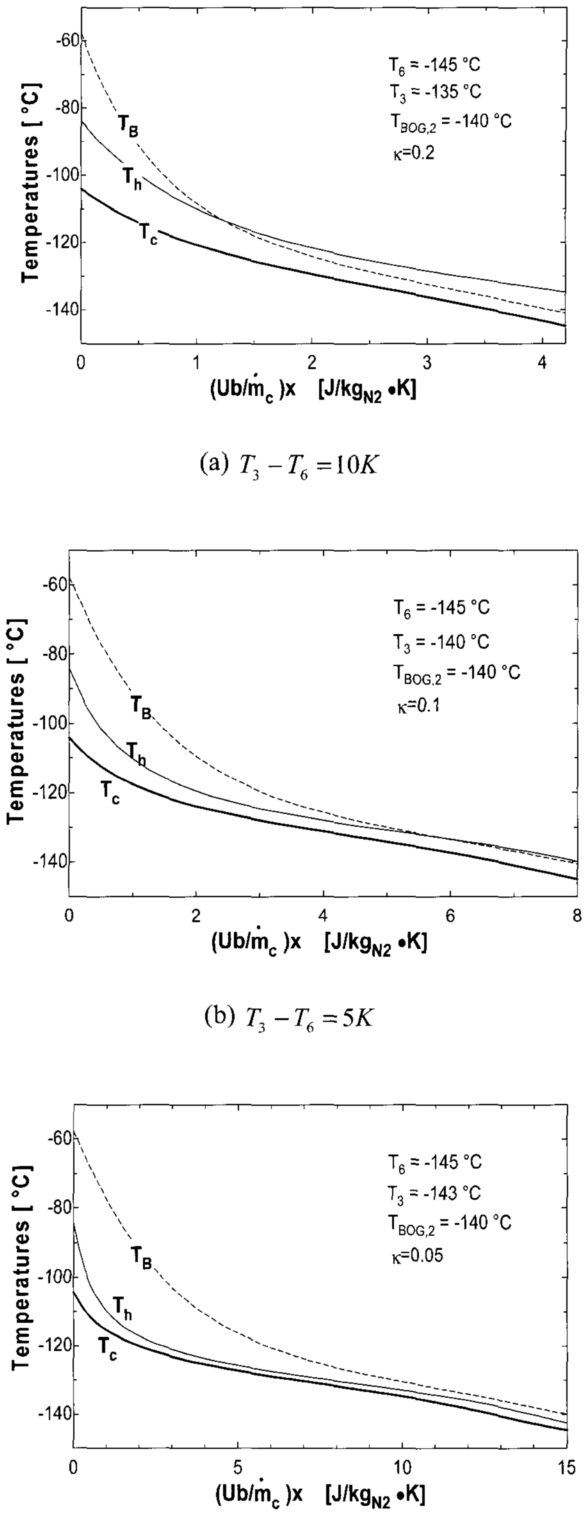
$$\frac{\dot{m}_c}{Ub}\frac{d\Delta_h}{dx} = -\left[\frac{1}{(1-r)c_{p,h}} - \frac{1}{c_{p,c}}\right]\Delta_h + \frac{\kappa}{c_{p,c}}\Delta_B \qquad (16)$$

$$\frac{\dot{m}_c}{Ub}\frac{d\Delta_B}{dx} = \frac{1}{c_{p,c}}\Delta_h - \kappa\left[\frac{1}{\dot{m}_B c_{p,B}} - \frac{1}{c_{p,c}}\right]\Delta_B \qquad (17)$$

$$\frac{\dot{m}_c}{Ub}\frac{dT_B}{dx} = -\frac{\kappa}{\dot{m}_B c_{p,B}}\Delta_B \qquad (18)$$

Once a cycle state is defined, the hot endpoint values of Δ_h, Δ_c and T_B will be given, and $\dot{m}_B$ and r as well. Thus (16)-(18) can be integrated until T_B reaches $T_{BOG,2}$. κ is adjusted to match the cold endpoint Δ_B with $(T_{BOG,2} - T_6)$. No consideration for the compatibility of the cold endpoint Δ_h with T_3 is needed since the cycle state has already satisfied the conservation of energy.

The temperature profiles for HX2 are shown in Fig. 5 and Fig. 6. The cold endpoint value of the x-axis will be proportional to the length of the heat exchanger if one were to construct heat exchangers with a given U and $b/\dot{m}_c$. Since $\kappa < 1$, κ is presumed not to have an influence on the length. Fig. 5 illustrates the size of the heat exchanger increases as the cycle state approaches the $T_6 = T_3$ line. This poses a problem with the performance-size trade-off since the cycle states of zero cold endpoint temperature difference (meaning lower specific liquefaction work as was seen in Fig. 4) would require too large a heat exchanger. Another problem with the cryogenic heat exchangers is the thermal pinch, which is illustrated in Fig. 6. The coefficient of the first term in (16) is responsible for this. When the heat capacity rate of the hot nitrogen becomes larger than that of the cold, that is, $(1-r)c_{p,h} > c_{p,c}$, their local temperature difference increases in the cold-end direction. In case of that, the minimum Δ_h occurs within the heat exchanger and the cycle states of zero endpoint temperature difference can never be reached even with a larger heat exchanger. This prohibits the access of the system to the cycle state of higher performances. This becomes more serious as the cycle approaches the cycle state S.

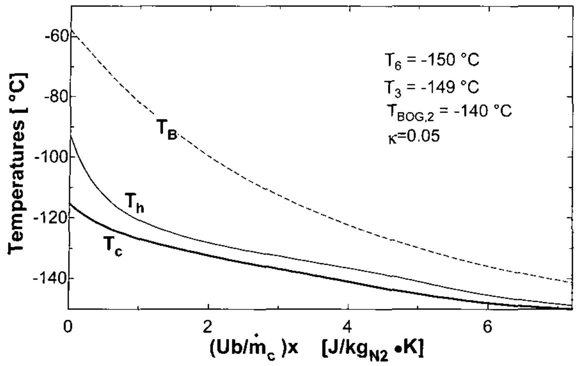


(a) $T_3 - T_6 = 10K$

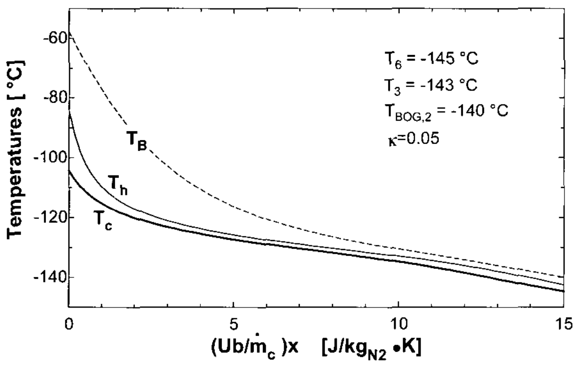
(b) $T_3 - T_6 = 5K$

(c) $T_3 - T_6 = 2K$

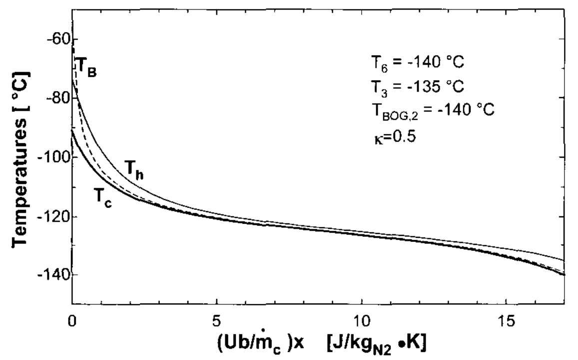
Fig. 5. Heat exchanger size and the end temperature differences.



(a) $r = 0.8$



(b) $r = 0.75$



(c) $r = 0.65$

Fig. 6. Thermal pinch.

4. CONCLUSION

Cycle analysis has been performed to find the optimum design point of the LNG BOG re-liquefaction plant. According to the thermodynamic analysis, the system state was defined by the three cycle variables. Thus the cycle performance could be expressed by the three cold endpoint temperatures of the three-pass heat exchanger. This enabled us to explain the relationship between the parameters of the heat exchanger and the cycle performance. As the cold endpoint temperature difference of the two nitrogen streams approaches a zero limit, the system performance improved. However, this was found possible only with a larger heat exchanger. The thermal pinch becomes serious as the ratio of the expanded mass through the expander to the total mass decreases which causes the increase of the heat capacity rate of the high-pressure nitrogen. This was found to limit the closeness of the two endpoint temperatures in coming together and hence act as a limiting factor in accessing the least power-consuming cycle state.

ACKNOWLEDGMENT

The author would like to express thanks to H.M. Chang at Hongik University for motivating this work and to Y.P. Lee and J.W. Moon at KIST for reviewing and permitting the release of this publication.

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