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Development of partial liquefaction system for liquefied natural gas carrier application using exergy analysis

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A R T I C L E I N F O

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A B S T R A C T

The cargo handling system, which is composed of a fuel gas supply unit and cargo tank pressure control unit, is the second largest power consumer in a Liquefied Natural Gas (LNG) carrier. Because of recent enhancements in ship efficiency, the surplus boil-off gas that remains after supplying fuel gas for ship propulsion must be reliquefied or burned to regulate the cargo tank pressure. A full or partial liquefaction process can be applied to return the surplus gas to the cargo tank. The purpose of this study is to review the current partial liquefaction process for LNG carriers and develop new processes for reducing power consumption using exergy analysis. The developed partial liquefaction process was also compared with the full liquefaction process applicable to a LNG carrier with a varying boil-off gas composition and varying liquefaction amounts. An exergy analysis showed that the Joule–Thomson valve is the key component needed for improvements to the system, and that the proposed system showed an 8% enhancement relative to the current prevailing system. A comparison of the study results with a partial/full liquefaction process showed that power consumption is strongly affected by the returned liquefied amount.

1. Introduction

Boil-Off Gas (BOG) generated from a cargo tank during a voyage in a Liquefied Natural Gas (LNG) carrier (LNGC) should be treated for cargo tank protection. Environmental regulations have caused BOG to be used as a main propulsion fuel, and the Fuel Supply Unit (FSU) has been developed accordingly. Because of reduced fuel consumption from both the enhancement of ship efficiency and low speed operations, fuel gas consumption is not sufficient to treat all generated BOG. A liquefaction facility is therefore required to return surplus BOG to the cargo tank. The cargo handling system composed of an FSU and liquefaction unit is the second largest power consumer during a voyage.

The liquefaction process is well known, because it entails significant power consumption. Many methods have been studied for increasing liquefaction efficiency, with various processes using various heat exchanger types and refrigerants. Remeljej and Hoadley, 2006 compared the various processes for an offshore LNG production facility, using a Single Mixed Refrigerant (SMR), N_2, and CH_4. The results showed that SMR exhibited higher performance than other refrigerants, but an N_2 process and open loop cycle using CH_4 were recommended because of the compactness of the system. Shin and Leeb (2009) emphasized the value of non-flammable refrigerant offshore, and studied the liquefaction process using an N_2 refrigerant, which was applied to the reliquefaction of a large LNGC. Recently, adopting multiple refrigerants has been investigated as a method for increasing liquefaction efficiency in an LNG plant. Morosuk et al. (2015) suggested optimizing a PRICO liquefaction process where a pre-cooler (using propane as additional refrigerant) is added to the Mixed Refrigerant (MR) process. Ding et al. (2016) developed a pre-cooled propane N_2–CH_4 expansion process, and their results indicated that system performance fell between an MR process and a N_2 expansion process. Chang (2015) reviewed cryogenic refrigeration cycles for the liquefaction of natural gas, such as the Joule–Thomson and Brayton cycles, with pure and mixed refrigerant and thermodynamic irreversibility. Lee and Sanggyu et al. (2012) proposed that the cycle consists of pre-cooled carbon dioxide and a nitrogen expander liquefaction cycle for LNG FPSO. Most research into liquefaction has focused on natural gas applied in an on/off shore LNG plant. However, the composition of BOG in LNGC would be slightly different from that of natural gas.

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Nomenclature

BOG	Boil-off gas
C3MR	C3 (propane) mixed refrigerant
E_x	Exergy [kJ/kg]
FG	Fuel Gas
FSU	Fuel Supply Unit
h	Specific Enthalpy [kJ/kg]
I	Irreversibility, Exergy destruction [kW]
JTV	Joule–Thomson Valve
LNGC	Liquefied Natural Gas Carrier
$\dot{m}$	Mass flow rate [kg/h]
SMR	Single mixed refrigerant
NG	Natural Gas
NOx	Nitrogen oxide or dioxide
PRS	Partial Reliquefaction System
s	Specific Entropy [kJ/kg k]
SMR	Single mixed refrigerant
SPC	Specific Power Consumption [kW/kg]
$\dot{W}$	Shaft Power [kW]
Subscripts	
ex	exergy
o	environmental state (25 °C, 1 bar)

From a ship operations point of view, the optimal cargo handling system should have both good liquefaction performance and good fuel gas supply performance. In particular, the required amount of liquefaction is relatively small relative to the fuel gas, so that it may be appropriate in an LNGC to apply different processes than those used in a conventional LNG plant. D.K. Choi et al. (2014) announced a Partial Reliquefaction System (PRS) with an open loop cycle using BOG as a refrigerant, which is optimized for supplying fuel gas to an engine that requires a gas supply pressure of 300 bar and can liquefy a small amount of surplus BOG.

The purpose of this study is to develop a new PRS for a medium pressure gas engine met Tier III requirement. An exergy analysis of the system applying the PRS for medium pressure gas engine was performed. To improve system performance based on specific power consumption through an exergy analysis, a modified PRS was proposed and compared. It was also carried out as a case study, with varying BOG compositions and liquefaction flow amounts.

2. System description

2.1. System condition

An FSU that meets the fuel gas conditions required by the main propulsion engine and a reliquefaction unit that liquefies the remaining BOG in the fuel supply are located in the cargo compressor room, as shown in Fig. 1. Consequently, system compactness is more important here than in other cases. An open loop cycle using process fluid as a refrigerant could be a promising solution for reducing space in the compressor room. System boundary conditions were considered as follows for analyzing and comparing the system.

The pressure of the cargo hold is maintained at 1.06 bar and the generated BOG is saturated vapor. However, the BOG entering the cargo room is heated by heat penetration through the pipe during transport. As opposed to LNG, BOG is primarily composed of methane and nitrogen. Shin and Leeb (2009) assumed the typical conditions of BOG to be a temperature of −120 °C, comprising approximately 8.5% nitrogen and 91.5% methane during the LNGC liquefaction process development. Those BOG composition ratios change over time. Initially, the nitrogen composition of BOG is larger than in LNG and decreases over time. Additionally, main propulsion gas consumption is not constant, and depends on operating conditions such as voyage mode, engine fuel mode, etc. This means that the load to be liquefied varies with fuel gas consumption and BOG rate.

Table 1 presents the system boundary conditions for the process analysis. In this research, it was assumed that 4000 kg/h of pure methane BOG was generated from the cargo tank. Of that BOG, 62.5% was supplied for fuel gas, and the rest was returned to the cargo tank through the liquefaction unit at the design stage. Performance was also reviewed by changing the nitrogen composition of BOG from 0% to 20%, and the flow amount of liquefied gas was reviewed by changing from 1000 to 2000 kg/h for a given BOG flow amount.

The process analysis was carried out in Aspen HYSYS with Peng–Robinson equation used as the equation of state. When compared with other systems, the assumptions and conditions for each component should be kept constant. Pressure loss in system components was not considered. A thermodynamic system analysis was performed based on the following general assumptions:

- Adiabatic efficiency of compressor: 75%;
- Pressure ratio in multi stage compressor: below 4 (according to GPSA guidance);
- Minimum approach temperature in heat exchanger: 5 °C; and
- Outlet process temperature of the inter coolers: 40 °C.

3. System analysis

3.1. Overall system efficiency

In process optimization studies, specific objective functions are optimized for specific variables. Cao et al. (2006) have compared SMR and N_2 expander liquefaction processes for small-scale natural gas liquefaction. They focused on the fact that temperature differences in the heat exchanger are a key parameter for optimizing power consumption. Nogal et al. (2008) compared the capital cost of a cascade mixed cycle with the required power for that cycle by changing the approach temperature of the heat exchanger. Alabdulkarem et al. (2011) optimized the required power consumption for a MR process using genetic algorithm optimization technique. Although the objective functions and process optimization variables vary among studies, ultimately many of these objective functions have aimed to minimize power consumption. The Specific Power Consumption (SPC) associated with fuel supply and liquefaction for a given BOG amount are defined as below. In this study, SPC is the main objective function used to optimize the system and it is key parameter, in which the developed system was compared with other systems.

$$\mathrm{SPC} = \left(\frac{\sum \dot{W}_{comp}}{\dot{m}} \right)_{re-liquefaction} \tag{1}$$

3.2. Exergy analysis

Exergy analysis is a useful method for measuring the qualitative/quantitative use of energy in components and the overall system.

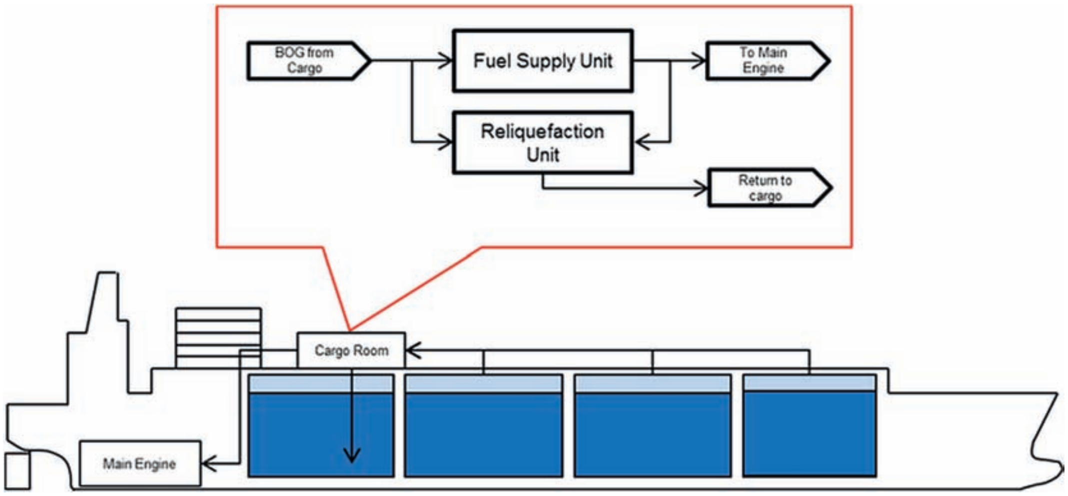

Fig. 1. Conceptual diagram of cargo handling system in LNGC compressor room.

Table 1
System Boundary Conditions.

	BOG	Fuel Gas	Return to Tank
Mass flow [kg/h]	4000	2500	1500
pressure [bar]	1.06	16	1.06
Temperature [°C]	−100	40	−161
Component (mole %)	Min	Max	Design
Nitrogen (N_2)	0	20	0
Methane (C_4)	80	100	100

Table 2
Definition of exergy destruction and efficiency in components.

	Exergy destruction, I [kW]	Exergy efficiency, η_{ex}
Compressor	$\sum \dot{m}_i e_i + W - \sum \dot{m}_o e_o$	$\frac{\sum \dot{m}_i e_i - \sum \dot{m}_o e_o}{W}$
Cooler	$\sum \dot{m}_i e_i - Q - \sum \dot{m}_o e_o$	$\frac{\sum \dot{m}_i e_i - \sum \dot{m}_o e_o}{Q}$
Heat exchanger, Mixer, Separator, JTV	$\sum \dot{m}_i e_i - \sum \dot{m}_o e_o$	$\frac{\sum \dot{m}_o e_o}{\sum \dot{m}_i e_i}$

Exergy (the maximum work available from the system relative to the environmental conditions as a reference in the stream) is defined as below.

$$E_x = \dot{m}e_x = \dot{m}[(h - h_o) - T_o(s - s_o)] \tag{2}$$

where h and s are the specific enthalpy and entropy, respectively, and T_o is the reference environmental temperature. If the process in a component or system from state 1 to state 2 is reversible, then the change in exergy is simplified as a function of enthalpy, entropy, and ambient temperature in each state, as follows.

$$\Delta E_x = \dot{m}(e_{x_2} - e_{x_1}) = \dot{m}[(h_2 - h_1) - T_o(s_2 - s_1)] \tag{3}$$

In actual processes, exergy changes because of the irreversibility of the processes that occur, referred to as exergy destruction (I). The secondary thermal efficiency is defined as the change in exit exergy relative to incoming exergy. The exergy destruction and exergy efficiency for each component are summarized in the Table 2 below.

Even with equipment with the same thermal efficiency, the amount of energy we can actually use depends on the surrounding environment of the given component. In contrast with thermal efficiency, exergy efficiency indicates how efficiently each component operates in a reversible process in a given environment, and exergy destruction indicates the amount of energy loss through a process. Because the thermal efficiency of the equipment is characteristic of the equipment itself, it does not indicate how efficiently the system operates. Therefore, the optimum operating conditions or configuration of a system shall be derived by calculating the efficiency of the entire system while changing the process variables. On the other hand, because exergy efficiency quantitatively indicates how efficiently equipment operates under given process conditions, it is very useful information at the initial stage of determining system configurations or optimizations. Exergy analysis is a promising method for system optimization by focusing on exergy efficiency and its destruction.

Yumrutas et al. (2002) investigated the effect of condensing temperature in refrigeration system on exergy loss, the second law of efficiency, and COP. Tirandazi et al. (2011) analyzed equipment exergy in the liquefaction process, and compared the exergy changes of total and individual equipment according to design variable changes. In this study, an exergy analysis of an existing open cycle was used to determine the component causing major exergy destruction. The developed system was also compared with an exergy analysis.

4. Results and discussion

4.1. Partial reliquefaction system

The PRS proposed from the D. K. Choi was optimized for engines requiring fuel gas at approximately 300 bar of pressure. It was modified for supplying 16 bar pressurized fuel gas, as shown in Fig. 2. The system was optimized by changing the inlet gas pressure of LNG-100 after the compressors. Increasing the gas pressure after the final compressor is done to reduce the recycled flow, which reduces capacity but enhances discharge pressure. Below 150 bar, the recycled flow amount is dominant in compressor power consumption, because the cooling capacity of BOG is not sufficient to create a fully saturated liquid at a given pressure. However, recycled flow does not significantly change with pressure above 150 bar, in which pressurized gas is fully liquefied through the LNG-100. In the design condition, the results showed that the SPC of the system is 0.325 [kW/kg s] for gas pressurized to up to 150 bar.

An open system such as PRS is well known as a simple system with a relatively small number of equipment components that must be considered in liquefaction. However, system efficiency is not sufficient compared to other full liquefaction systems. However, the results indicated that the open cycle system provides good performance when both the fuel supply and liquefaction system are considered simultaneously.

An exergy analysis was performed for the design condition and results. The exergy efficiency and exergy destruction for each component are shown in Fig. 3. The LNG-100 exhibited an exergy efficiency of approximately 0.967. The exergy efficiency of the compressors was approximately 0.8. In this study, the modification of compressor configurations was not considered as a method for increasing system efficiency, because the exergy efficiency of a compressor is strongly related to adiabatic efficiency. Therefore, the exergy efficiencies of each compressor were similar, even though their process conditions were different. However, the amounts of exergy destruction for the front three compressors were different from the others, because the flow amounts for the last compressors were reduced because of fuel gas supply.

The worst exergy efficiency was shown by JTV-1, which had an exergy efficiency of 0.787. Large exergy destruction occurs for JTV-1, and it is greater than twice the exergy loss observed for the unit compressor. It was seen that the overall system efficiency was influenced by the irreversible operation of peripheral equipment such as JTV-1, although the increase in the efficiency of the compressor by the compressor directly affects increases in thermal efficiency, such as SPC. JTV-1 generates a two-phase state as depressurization occurs, with the saturated or subcooled liquid used as the inlet condition. Irreversibility increases during expansion. This irreversible process eventually appears as a loss of exergy, and the loss increases as the pressure difference through the valve increases. To improve the system, its configuration should be changed to reduce exergy loss in JTV-1.

4.2. Cascade PRS

To reduce exergy loss in the JTV, a cascade JTV was adopted for the system, as shown in Fig. 4. The compression process for the fuel supply and further compression for the liquefaction was the same as in the PRS. However, the compressed gas was divided into two streams. The branched high-pressure gas stream was subcooled through the LNG-1 during heat exchange with the BOG. It became a low-temperature two-phase state through JTV-1, and was separated into a gas and liquid in the SEP-1 separator. The liquid again expanded through the JTV-2 expansion into a pressure of tank, and the liquid in SEP-2 was returned to the tank. The other branched gas was subcooled in LNG-2 from the flashed gas from SEP-1 and SEP-2. The subcooled liquid was directly expanded in JTV-3 to the cargo tank pressure. The warm flashed gases from LNG-2 were returned to a dedicated compressor corresponding to pressure. The optimization was carried out for a case study by changing the split ratio of the pressurized gas and setting the pressure of SEP-1.

The results for exergy efficiency and destruction in a cascade PRS were compared with each component in a PRS, as shown in Fig. 5. Because the cascade PRS consists of multiple JTVs and heat exchangers, it was difficult to directly compare each components directly with its corresponding PRS component. For convenient comparison, the exergy destructions of JTVs and heat exchangers were represented by a summation of all exergy destruction for the corresponding components. The exergy efficiency of JTV and LNG in the cascade PRS indicates the ratio of total exit exergy to total inlet exergy for the respective components.

It is clear that the exergy destruction for the JTV was reduced, which was the purpose of the modifications. Exergy destruction for the JTV was reduced by 64% from PRS one, and exergy efficiency was enhanced by 15%. Additionally, exergy efficiency in the mixer was enhanced. During system optimization, the split ratio of the pressurized gas was adjusted to reduce exergy losses in the LNG-200 heat exchanger, which required that the outlet temperature for LNG-200 be warmed to near that of the incoming stream in the mixer. Generally, most researchers would focus on changing the system configuration to enhance the performance of the heat exchanger and compressor. However, the results indicated that the conditions of the stream entering the mixer would be one of key variable to affect overall system efficiency. The enhanced system efficiency would reduce the recycled flow amount relating to the

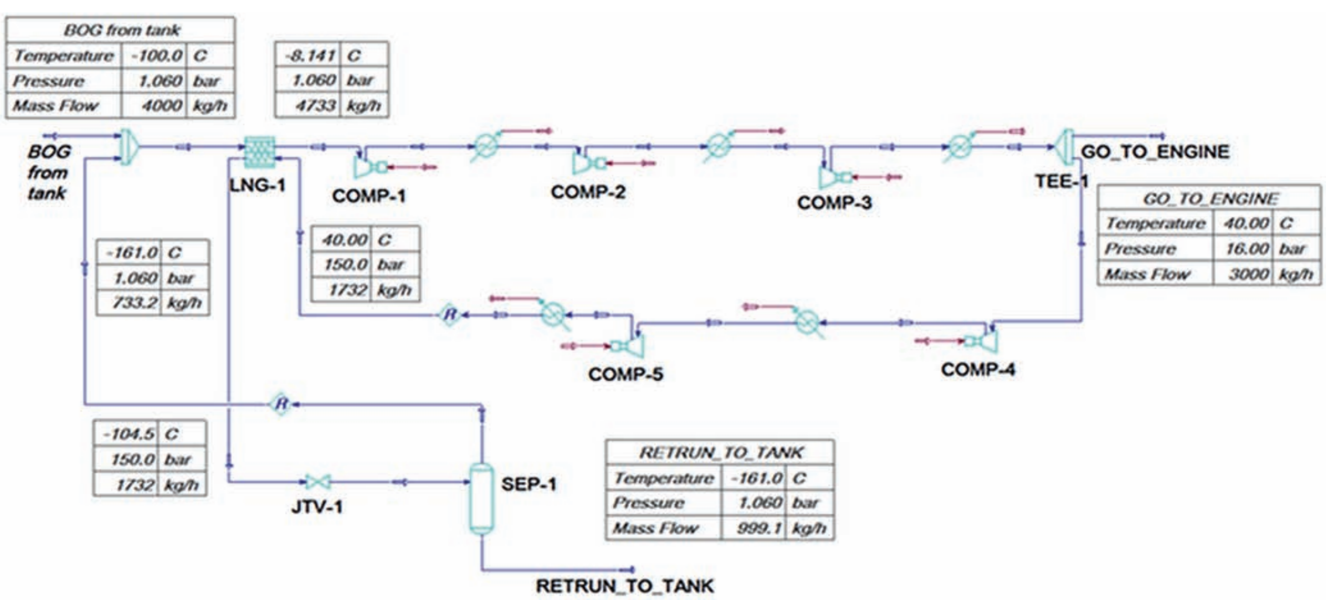

Fig. 2. Process flow diagram of partial reliquefaction system.

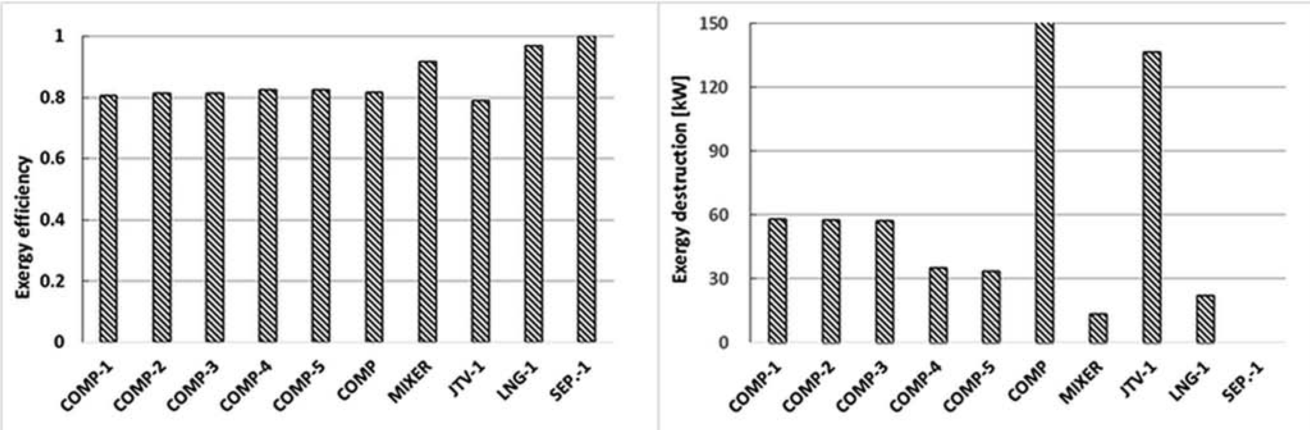

Fig. 3. Exergy analysis of PRS for design conditions.

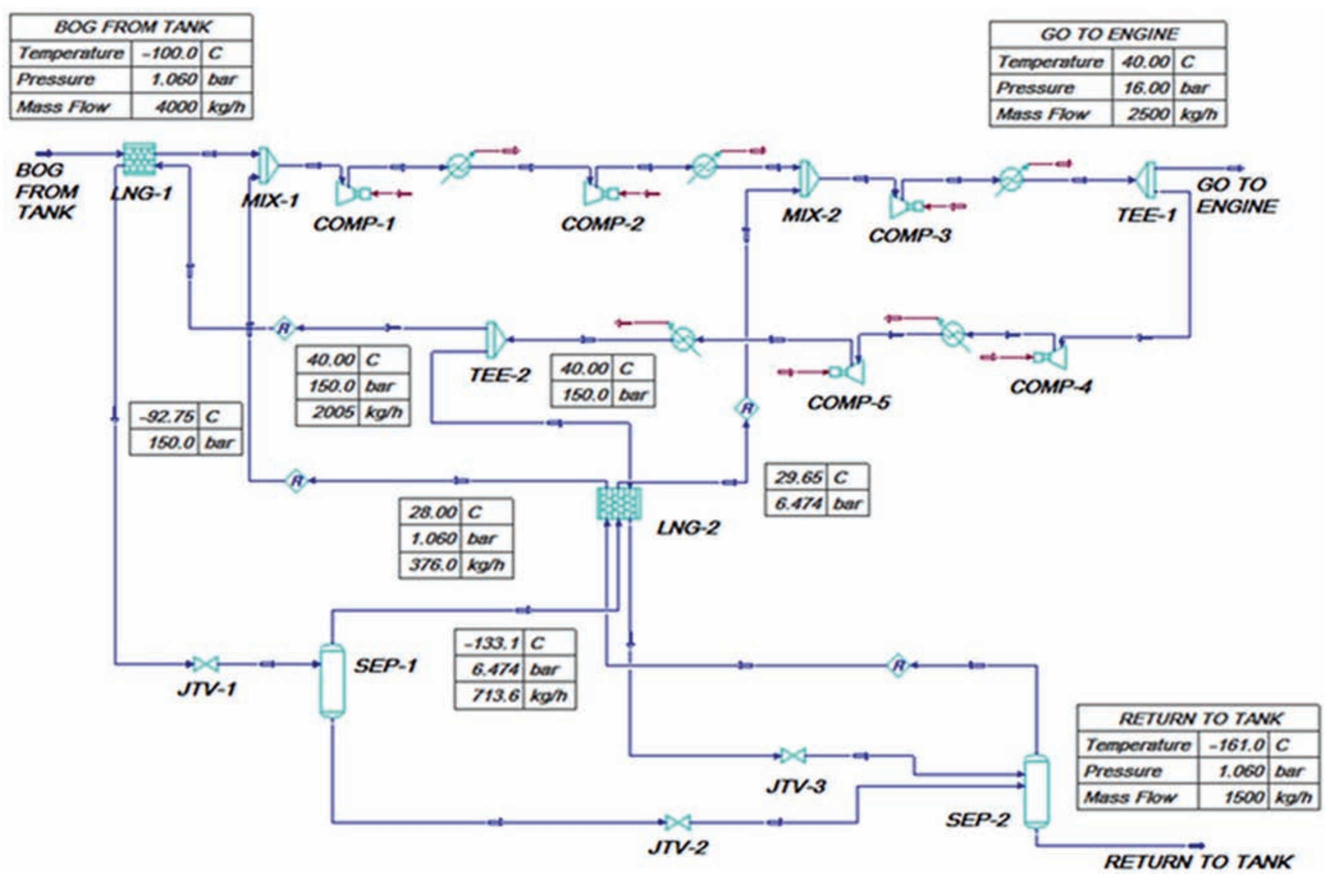

Fig. 4. Process flow diagram of cascade PRS.

mass flow of compressors. Flashed gas of the SEP-1 separator in the cascade PRS was merged after COMP-2, thereby reducing the load on the front compressors. Those phenomena would reduce quantitative exergy destruction, even for the same exergy efficiency of the compressors. The SPC of the cascade PRS was 0.299, an increase of 8% from that of the PRS, as presented in the Table 3 below.

Table 3
Specific power consumption and exergy loss in both systems.

	SPC (kWh/kg)	Exergy destruction (kW)
PRS	0.429	412
Cascade PRS	0.354	326

5. Selective study

A selective study considering changing BOG compositions and liquefaction flow ratios was conducted to analyze changes in system performance with respect to the variables that occur during operation.

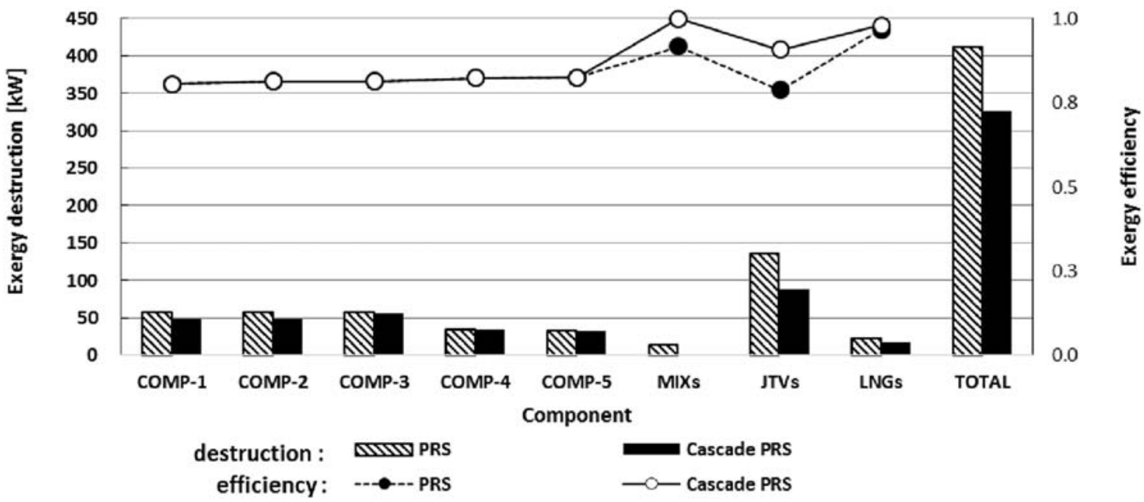

Fig. 5. System comparison using exergy analysis.

5.1. Nitrogen effect on BOG composition

The characteristics of BOG with added nitrogen, which at atmospheric pressure is saturated at temperatures below −195 °C, are

- to create a larger vapor fraction after the JTV resulting in an increase in compressor load;
- to increase the higher outlet temperature during compression, resulting in a requirement for more utility work;
- to generate reduced compression work at a given mass flow; and
- to cause the lower LHV of the fuel gas to result in the limit on output of engine

An exergy analysis was performed for the major components of the two systems: 0% and 15% nitrogen content in the BOG. The results showed that exergy destruction increases with increasing BOG nitrogen content, although there is no significant change in exergy efficiency from the addition of nitrogen (except in the JTV and LNG-100 heat exchangers), as shown in Fig. 6. Exergy loss in the JTV was due to the difference in the vapor fraction after expansion. When the entrance condition of the JTV was subcooled to the same temperature in systems, the increased nitrogen contents in BOG was less subcooled, and the vapor fraction increased for the same pressure after expansion. Those phenomena led to an increase in exergy loss in the JTV itself and an increase in the amount of gas recirculating through the system. This resulted in an increase in the exergy loss of the peripheral components. During the system optimization in the previous design stage, changes in the split ratio of the compressed gas was determined to increase the exergy efficiency of the LNG heat exchanger and mixer.

The performance of the systems was compared while varying the composition of nitrogen from 0% to 20%, as shown in Fig. 7. The results showed that as nitrogen content increases, the SPC for both systems increases. The mole fraction of nitrogen contained in the liquefied BOG returned to the tank is low relative to that of the gas entering from the BOG. This surplus nitrogen would exit by mixing with the fuel gas, or would accumulate in the system, resulting in exponential increases in the SPC. Comparing SPC changes with the nitrogen fractions of both systems, the rate of increase in the PRS is higher than that in the cascade PRS. For the case of 0% nitrogen (i.e., pure methane), the SPC of the cascade PRS was reduced by approximately 8% relative to that of the PRS, while it was reduced by approximately 15% for BOG with 20% nitrogen.

5.2. Reliquefied flow amount

The SPC of the system was compared while varying the liquefaction ratio, which is the amount to be liquefied for a given BOG quantity. The exergy efficiency of each component depends on the composition, efficiency and process condition of each stream. Since the change in liquefaction ratio does not affect the variables mentioned above, an exergy analysis was not performed for this case study. As shown in Fig. 8, the SPC and exergy destruction of the system increase as the liquefaction ratio increases, even for the same exergy efficiency in each component. Even for a JTV with the same exergy efficiency, an increase in the required amount of liquefaction leads to an increase in the amount of flashed gas generated through expansion in the JTV, which leads to more recycled flow in the compressor stream and increased compressor work.

It should be noted that the system SPC increases exponentially with an increasing liquefaction ratio. The results showed that, as the amount of liquefaction increases, the liquefaction process of an open cycle such as a PRS or cascade PRS can be less efficient than a simple liquefaction process. However, process conditions (such as the liquefaction ratio) have no effect on system efficiency when the fuel supply system and liquefaction process are operated independently. The effect of composition in independent system is less sensitive than in the open cycle. To select the proper system, it is necessary to compare system efficiency by changing process conditions, as mentioned previously.

The N_2 cycle typically adopted in large LNGCs and the C3MR cycle for the LNG plant were considered in the liquefaction process for comparison. The SPC of the N_2 cycle presented by Shin and Leeb (2009) was 0.697, and that for the C3MR presented by Vink and Nagelvoort (1998) was 0.31. Because the assumptions and compositions reflected in the SPC calculation for the mentioned processes are different, an accurate comparison is not possible. Nevertheless, the comparison of trends is reasonable, because the SPC of C3MR (which was specialized for LNG liquefaction) might increase when

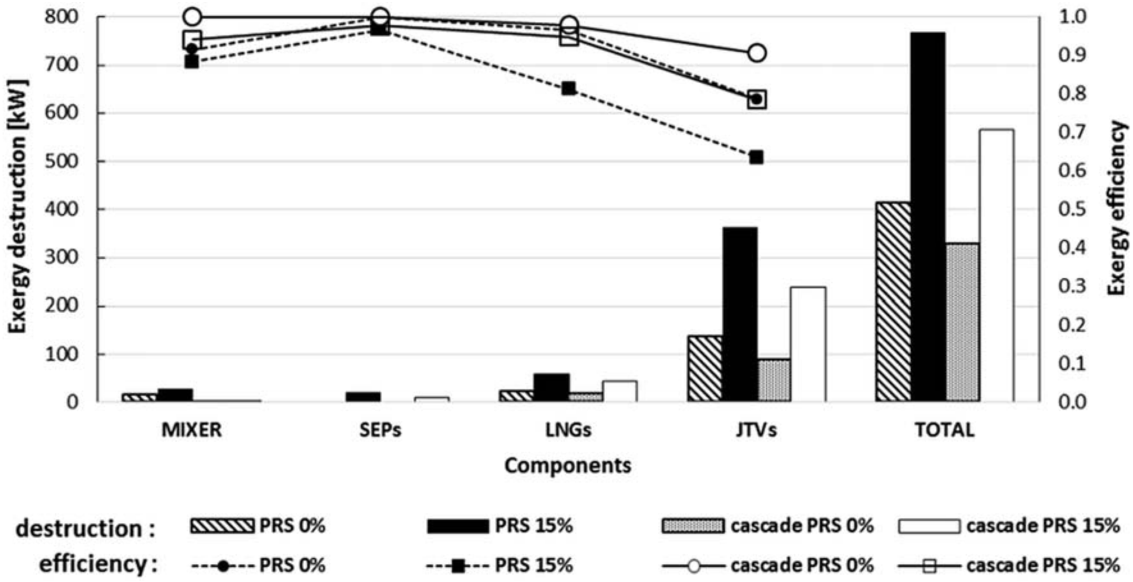

Fig. 6. Exergy efficiency and exergy destruction of each component in systems, between pure methane and mixture.

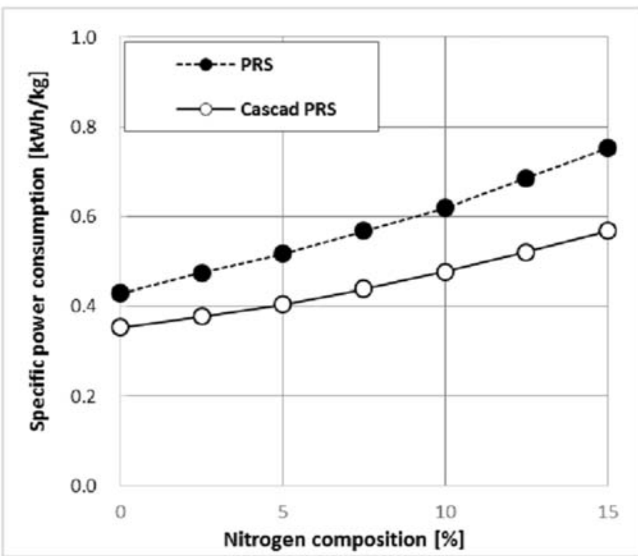

Fig. 7. Variation of specific power consumption with nitrogen content in BOG.

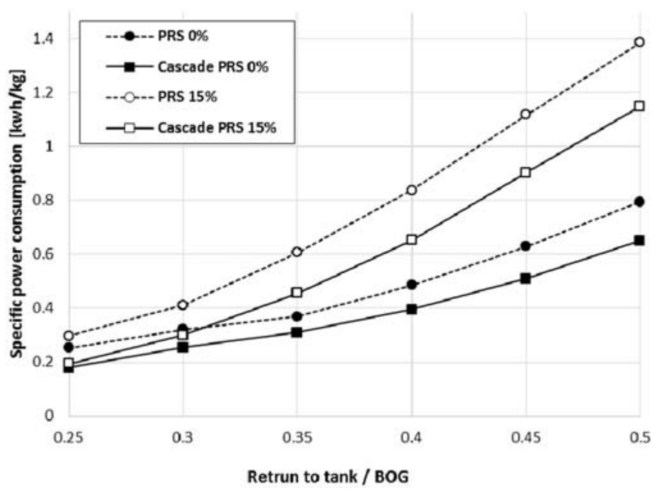

Fig. 8. Variation of specific power consumption with liquefaction ratio.

applied to BOG liquefaction. Therefore, the SPC of the N_2 and C3MR cycles referred to the abovementioned research results. The liquefaction and FGS processes were separately operated in independent systems, and the SPC, according to the liquefaction, amount was compared for a PRS and cascade PRS.

As shown in the Fig. 9, the results indicated that, as liquefaction increases, the nitrogen cycle and the C3MR cycle SPC was kept constant, whereas the PRS and cascade PRS increase exponentially. The results also showed that liquefaction ratio should be considered at the system selection stage. Open cycles, such as PRS or cascade PRS, provided less SPC when a smaller amount of liquefaction was required. When the liquefaction ratio is less than 30%, PRS is more efficient than C3MR, and it is more efficient than N_2 cycle below a 45% liquefaction ratio. Similarly, in the case of a cascade PRS, the performance is better than that of a C3MR for liquefaction ratio lower than 40%, and better than that of an N_2

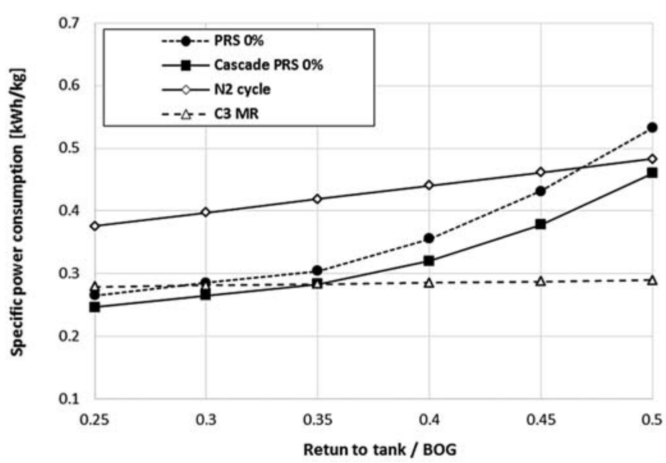

Fig. 9. SPC comparison of applicable systems with various return ratios.

cycle for liquefaction ratio lower than 50%. The results of this study could be an important index for selecting the type of gas handling system used in LNGCs.

6. Conclusion

In this study, an exergy analysis of PRS was performed to develop a new cargo handling system for the medium pressure fuel supply and partial liquefaction of BOG during LNGC operations. A cascade PRS, which is a modified system, was proposed and was compared with an existing PRS and applicable liquefaction processes. The findings of the study are as follows:

- Exergy efficiency for each PRS component was calculated under the design conditions. The exergy efficiency of the JTV was the worst of the components, and the exergy loss from the JTV was more than twice that from the compressor. To improve the overall efficiency of the system, it was necessary to increase the efficiency of the JTV.
- To increase the exergy efficiency of the JTV, a multi-stage cascade PRS was constructed. Overall system performance was improved by approximately 8%, and the exergy efficiency of the JTV was improved by approximately 15%. The multi-stage JTV reduced compressor work, because the flashed gas from the first JTV expansion bypasses the first two compressors.
- The performance of the system was compared by varying the BOG composition and liquefaction ratio. As the proportion of nitrogen in the BOG increases, exergy losses in the JTV increase. As a result, the recycled flow increases, and the SPC of the whole system increases exponentially. As the required liquefaction amount increases, the recycled flow amount increases, although the exergy efficiency of each component is the same. As a result, the SPC of the system increases exponentially.
- System performance, according to variations in the required liquefaction ratio, was compared with that of a conventional liquefaction process. When the required liquefaction amount is relatively small, PRS and cascade PRS systems show excellent performance. However, when the required liquefaction amount exceeds a certain ratio, it loses its competitiveness relative to a conventional liquefaction process. These criteria should be considered when designing the cargo handling system of a ship.

Acknowledgement

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