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Thermodynamic and Economic Analysis of Cargo Boil-Off Gas Re-Liquefaction Systems for Ammonia-Fueled LCO₂ Carriers

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Abstract: In this study, cargo boil-off gas (BOG) re-liquefaction systems for ammonia-fueled liquefied carbon dioxide (LCO₂) carriers were analyzed. These systems use cold energy from ammonia to reliquefy the CO₂ BOG. In this study, a system that can completely reliquefy the CO₂ BOG at all engine loads using only one heat exchanger is proposed, instead of the existing cascade system that requires multiple components. R744, which has a low global warming potential, was used as the working fluid for the refrigeration cycle in the CO₂ BOG re-liquefaction system. The organic Rankine cycle (ORC) was used to reduce the net power consumption of the system. The existing and proposed systems were classified into Case 1 (existing system), Case 2 (our proposed system), and Case 3 (Case 2 combined with an ORC). Thermodynamic and economic analyses were conducted. Case 2 is a system with a simpler configuration than Case 1, but it has a similar thermodynamic performance. Case 3 has a higher exergy destruction rate than Cases 1 and 2, owing to the ORC, but it can significantly reduce the net power consumption. The economic analysis shows that Cases 2 and 3 reduce the total annual costs by 17.4% and 20.1%, respectively, compared to Case 1. The proposed systems are significantly more advantageous for long-term operation than existing systems.

Keywords: boil-off gas; re-liquefaction system; liquefied carbon dioxide carrier; ammonia-fueled ship; organic Rankine cycle



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1. Introduction

The International Maritime Organization adopted a new greenhouse gas reduction strategy at the 80th Marine Environmental Protection Committee. Specifically, it aims to achieve net-zero greenhouse gas emissions from ships by 2050 [1,2]. As global warming causes serious changes to ecosystems, regulations on carbon dioxide (CO₂) emissions are being strengthened [3,4].

A representative method of reducing CO₂ emissions uses alternative fuels such as hydrogen (H₂) and ammonia (NH₃) [5,6]. Carbon capture, utilization, and storage (CCUS) has recently attracted attention as a strategy for achieving net-zero emissions. CCUS is a technology that captures CO₂ generated from fossil fuels and stores it in the ground through a compression and transportation process, or uses the captured carbon where it is needed [7]. CCUS has received worldwide attention and there have been several related studies [8–15].

With the development of the CCUS market, liquefied CO₂ (LCO₂) carriers are required to transport CO₂. The carrier stores LCO₂ in a cargo tank so that it can be transported to long-distance destinations [16]. Cargo tanks require an insulation layer to maintain the CO₂ in a liquefied state [17]. However, the generation of boil-off gas (BOG) is inevitable because of the heat transferred from the surroundings to the cargo tanks [18]. The BOG increases the pressure in the cargo tanks; therefore, LCO₂ carriers must be able to handle the BOG properly and safely. The most common method for treating BOG in LCO₂ carriers is re-liquefaction [19].

Currently, research on BOG re-liquefaction systems for LCO₂ carriers is in the early stages of development [20]. Representative systems that can be considered as CO₂ BOG re-liquefaction systems use an open-loop refrigerant cycle [16,18,19,21–23]. Such systems reliquefy CO₂ using the Joule–Thomson (J–T) effect without a separate vapor-compression refrigeration cycle (RC). A BOG that cannot be reliquefied must be recirculated into the cargo tanks. The power consumption of the compressor increases as the BOG increases. There is also a closed-loop refrigerant cycle that uses the RC [17,21,24]. This RC system uses a separate refrigerant and can reliquefy a larger quantity of BOG than an open-loop refrigerant cycle. This enables the amount of recirculated BOG and power consumption of the compressor to be reduced. Recently, Lee et al. [20] proposed a subcooling system. This system pumps LCO₂ from the cargo tank, exchanges heat with the RC to lower its temperature, and then returns it to the tank. In this system, the creation of BOG itself is suppressed through the circulation of LCO₂, and the system is more economical than existing systems because it does not require a BOG compressor. However, in this study, the efficiency of the subcooling system was low when the cargo tank pressure was < 1200 kPa. Thus, a subcooling system is advantageous when high-pressure cargo tanks are used. However, these tanks are expensive and unsuitable for high-capacity carriers. Therefore, when using an economical low-pressure cargo tank with a pressure of approximately 700 kPa, a closed-loop RC is advantageous.

Meanwhile, there have been studies conducted on CO₂ BOG re-liquefaction using the cold energy of the fuel. Yoo [22] studied liquefied natural gas (LNG)-fueled ships transporting LCO₂ cargo. This study proposed a CO₂ BOG re-liquefaction system with an LNG fuel supply system. To supply LNG fuel to an engine, it must be vaporized. Yoo proposed obtaining part of the latent heat of vaporization required for LNG fuel from the CO₂ BOG. At this time, the CO₂ BOG can be reliquefied using the LNG cold energy. However, because LNG is a carbon-containing fuel, achieving complete net-zero carbon emissions is difficult. Therefore, ships using alternative fuels, such as NH₃, have recently received attention [25].

Lu et al. [17] proposed a BOG re-liquefaction system using the cold energy of NH₃ fuel. The RC absorbed the thermal energy of the BOG and transferred it to the NH₃ fuel, which needed to be heated. The system reliquefied the BOG using the cold energy of NH₃ fuel. This is advantageous in terms of thermodynamic performance and economics, compared with classic cascade systems [17]. However, in this system, when the engine load was low, the flow rate of the supplied NH₃ fuel decreased, making it difficult to reliquefy the CO₂ using only the cold energy of the fuel. Therefore, a separate system was required for the complete re-liquefaction of the CO₂ BOG, especially at low engine loads. Lu et al. [17] also indicated that a non-environmentally friendly refrigerant was used as the working fluid in the RC. Specifically, the global warming potentials (GWPs) of R23 and R22, used in the RC of this system reach 14,600 and 1900, respectively [26,27]. In addition, R22 is a refrigerant in the hydrochlorofluorocarbon series that destroys the ozone layer [27].

In this study, a BOG re-liquefaction system for NH₃-fueled carriers transporting LCO₂ was analyzed. NH₃ cold energy and a vapor-compression RC were used; however, an improved system was proposed compared to that of Lu et al. [17]. This system allows the CO₂ BOG to be completely reliquefied, even at low engine loads, without relying on a separate system. Additionally, the RC of this system uses a refrigerant with an ozone depletion potential of zero and an extremely low GWP. In addition, this study considered the organic Rankine cycle (ORC), which can be added as an option to the above system. The ORC is being studied extensively because it can produce electricity after recovering the energy of low- and medium-temperature heat sources using organic fluids [28,29].

The remainder of this paper is organized as follows. The Background section explains the CO₂ properties, cargo tanks, BOG re-liquefaction system, and NH₃ fuel supply system. In the System Simulation section, the existing system is explained, simulated, and validated. The proposed system is then described. The case definitions, independent variables, and analysis models are presented in the Analysis Method section. In Results and Discussion,

energy, exergy, and an economic analysis of each system are performed and discussed. In the Conclusion, the thermodynamic characteristics and economic feasibility of the proposed system are summarized and compared with those of existing systems.

In this way, novel and high-performance systems are presented and analyzed in this study. Therefore, the results of this study can provide valuable knowledge in the field of ammonia-fueled LCO₂ carriers.

2. Background

2.1. CO₂ Properties and Cargo Tank

Figure 1 shows the pressure–temperature diagram of CO₂. The triple point of CO₂ is 518 kPa, −56.6 °C, and the critical point is 7380 kPa, 31.1 °C [30]. When the pressure of CO₂ is reduced to a pressure below the triple point, it changes directly from a vapor to a solid. In other words, dry ice is formed when the pressure of CO₂ is reduced to atmospheric pressure (101.325 kPa). Dry ice disrupts fluid flow and damages fluid machinery, heat exchangers, valves, and pipes; thus, its formation must be suppressed.

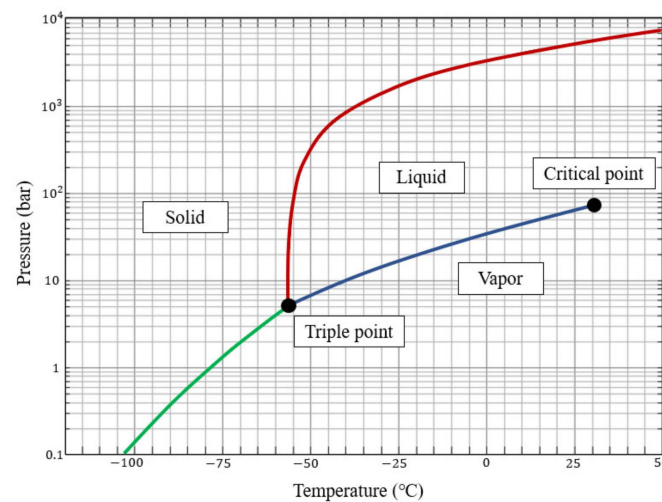


Figure 1. CO₂ pressure–temperature diagram.

Therefore, to prevent dry ice formation, the pressure in the CO₂ cargo tank must be maintained above the triple point. However, because high-pressure cargo tanks increase the capital expenditure (CAPEX), an appropriate pressure selection is necessary [20]. The temperature of CO₂ cargo tanks is generally between −20 °C and −50 °C. As the temperature decreases, the density of LCO₂ increases, allowing more cargo to be stored in the tanks [17,21,31]. However, low-temperature cargo tanks require a high operating expenditure (OPEX) to maintain this temperature [21]. In this study, considering the commercial potential of high-capacity LCO₂ carriers, the ideal pressure and temperature of the cargo tank were determined to be 659 kPa and −50.0 °C, respectively [17,31,32]. The systems shown in Figures 2 and 3 were studied based on the conditions in the CO₂ cargo tank.

2.2. BOG Re-Liquefaction System

Figure 2 shows a schematic of the CO₂ BOG re-liquefaction system analyzed in this study [17]. The BOG generated in the LCO₂ cargo tank flowed into the BOG compressor. The BOG, which was raised to a high pressure by the compressor, was condensed through a vapor-compression RC. It was then throttled at the J–T valve. The reliquefied LCO₂ was then returned to the cargo tank. Meanwhile, the working fluid in the RC vaporized by the BOG flowed into the refrigerant compressor. It was then condensed in a heat exchanger coupled to an NH₃ fuel supply system. The NH₃ fuel was heated by the working fluid of the RC. The working fluid that passed through the heat exchanger expanded in the J–T valve and exchanged heat with the BOG again.

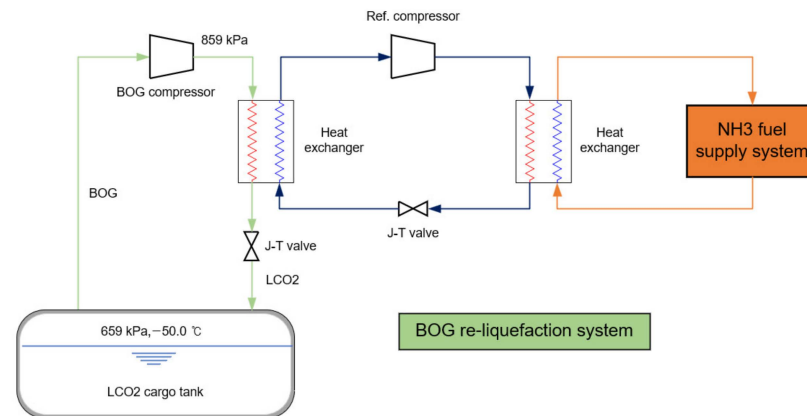


Figure 2. CO₂ BOG re-liquefaction system with vapor-compression RC and NH₃ fuel supply system.

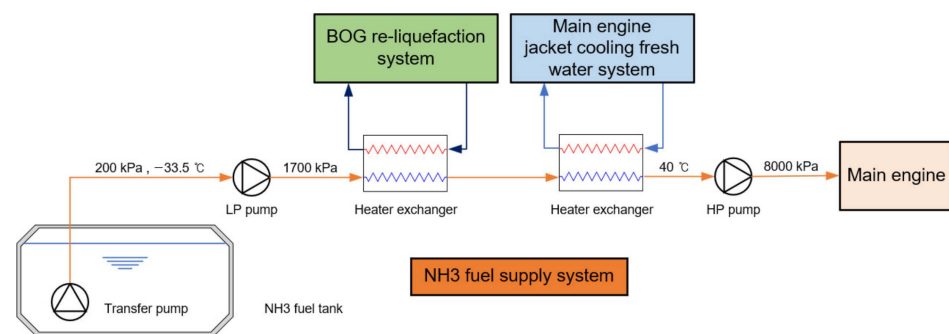


Figure 3. NH₃ fuel supply system with BOG re-liquefaction system and main engine jacket cooling fresh water system.

2.3. NH₃ Fuel Supply System

Figure 3 shows a schematic of the NH₃ fuel supply system. The fuel stored in the NH₃ fuel tank was supplied to the engine by flowing into a low-pressure (LP) pump through a submerged transfer pump. The LP pump increases the fuel pressure from 200 kPa to 1700 kPa. The NH₃ fuel was then heated using a heat exchanger combined with the BOG re-liquefaction system, as shown in Figure 2. In other words, the latent heat of the CO₂ BOG was used to heat the NH₃ fuel. It was then heated to 40 °C by fresh water in a heat exchanger linked to the main engine jacket cooling fresh water system, pressurized to 8000 kPa by the high-pressure (HP) pump, and supplied to the main engine [17,25].

3. System Simulation

In this study, the existing and proposed systems for CO₂ BOG re-liquefaction were simulated. Aspen HYSYS V12.1 was used for the system simulation, and the Peng–Robinson equation was used as the equation of state [17,25].

3.1. Simulation and Validation of Existing Systems

Figure 4 shows a CO₂ BOG re-liquefaction system that used the cold energy of NH₃, which is the fuel for the main engine [17]. This system used NH₃ cold energy to significantly improve the thermodynamic efficiency and economic feasibility of the CO₂ BOG re-liquefaction system. However, R23, which has a high GWP, was used in the RC. In addition, when the engine load was low, the flow rate of the NH₃ fuel flowing into N-1 decreased, so that the system was unable to completely reliquefy the CO₂ BOG. A separate additional system was therefore required.

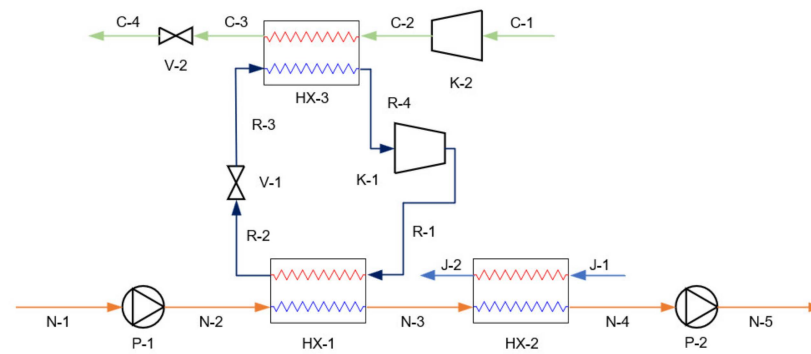


Figure 4. CO₂ BOG re-liquefaction system described by Lu et al. [17].

Table 1 lists the input variables used to simulate Figure 4. The simulation conditions were as follows: Lu et al. [17] analyzed the system by changing the temperature of R-2 by 1 °C and they selected the lowest interpretable value. This was used to obtain the lowest power consumption for the refrigerant compressor K-1. Meanwhile, the pinch point temperature difference (PPTD) of HX-1 was <1 °C. The temperature of R-2 was adjusted so that the PPTD of HX-1 was 0.5 °C, and its vapor fraction (VF) was set to zero. The efficiencies of the NH₃ fuel pumps (P-1 and P-2) were set to 75%. In addition, the efficiencies of the refrigerant compressor (K-1) and BOG compressor (K-2) were assumed to be 80% and 75%, respectively. To ensure that the pressure in C-4 was the same as that in the cargo tank, the pressure drop in V-2 was set to 200 kPa. The pressure drop in each stream in the heat exchanger (HX) was assumed to be 20 kPa; however, this was ignored for fluid flows below 1000 kPa [17]. Table 2 compares the results simulated in this study with those reported by Lu et al. [17]. Consequently, it was confirmed that the system was well simulated with a low error rate.

Table 1. Input parameters for simulation of Figure 4.

Stream	Fluid	t (°C)	p (kPa)	$\dot{m}$ (kg/h)	VF
N-1	NH ₃	−33.5	200	5018	-
N-2	NH ₃	-	1700	-	-
N-4	NH ₃	40	-	-	-
N-5	NH ₃	-	8000	-	-
J-1	H ₂ O	90	200	6.3×10^4	-
R-3	R23	−55	-	-	-
R-4	R23	-	-	-	1
C-1	CO ₂	−50	-	2470	1
C-2	CO ₂	-	859	-	-
C-4	CO ₂	−50	-	-	-

Table 2. Validation results for simulation of Figure 4.

Component	This Study	Lu et al. [17]	Error Rate (%)
K-1 (kW)	150.3	149.0	0.87
K-2 (kW)	9.6	9.4	2.13
Total (kW)	159.9	158.4	0.95

Figure 5 shows a CO₂ BOG re-liquefaction system using a cascade system [17]. This system uses seawater to reliquefy the BOG. Therefore, it has the advantage of being completely independent of the fuel supply system. However, when used alone, it is less efficient than the system shown in Figure 4 under normal engine load conditions. Because there is a large temperature difference between the seawater and CO₂, this system configures two RCs in a cascade to reliquefy the BOG. The disadvantage of this system is that it requires the installation of two expensive refrigerant compressors. Additionally, this

system uses the non-environmentally friendly refrigerants R22 and R23. The simulation conditions were determined based on those of the system in Figure 4. The parameters that must be input to simulate the cascade system are listed in Table 3. Table 4 lists the validation results of this system, which shows that the system was simulated well.

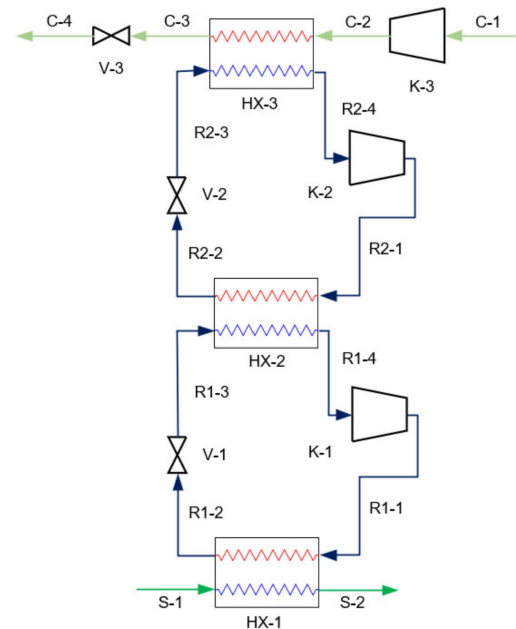


Figure 5. CO₂ BOG re-liquefaction system using a cascade system [17].

Table 3. Input parameters for simulation of Figure 5.

Stream	Fluid	t (°C)	p (kPa)	$\dot{m}$ (kg/h)	VF
S-1	H ₂ O	30	200	9.0×10^4	-
R1-2	R22	40	-	-	0
R1-4	R22	-30	-	-	1
R2-2	R23	-25	-	-	0
R2-4	R23	-55	-	-	1

Table 4. Validation results for simulation of Figure 5.

Component	This Study	Lu et al. [17]	Error Rate (%)
K-1 (kW)	152.0	152.2	0.13
K-2 (kW)	51.7	51.2	0.98
K-3 (kW)	9.6	9.2	4.35
Total (kW)	213.3	212.6	0.33

Figure 6 shows the CO₂ BOG re-liquefaction system proposed by Lu et al. [17] that can be used for all of the engine loads. The system shown in Figure 4 is advantageous for improving the efficiency of CO₂ BOG re-liquefaction because it uses NH₃ cold energy. However, it is impossible to maintain a constant pressure in the cargo tank of a high-capacity LCO₂ carrier using this system alone. Therefore, complete CO₂ BOG re-liquefaction requires the system shown in Figure 6, which is a combination of the systems shown in Figures 4 and 5. This system can reliquefy the CO₂ BOG generated in a cargo tank by dividing it into two systems.

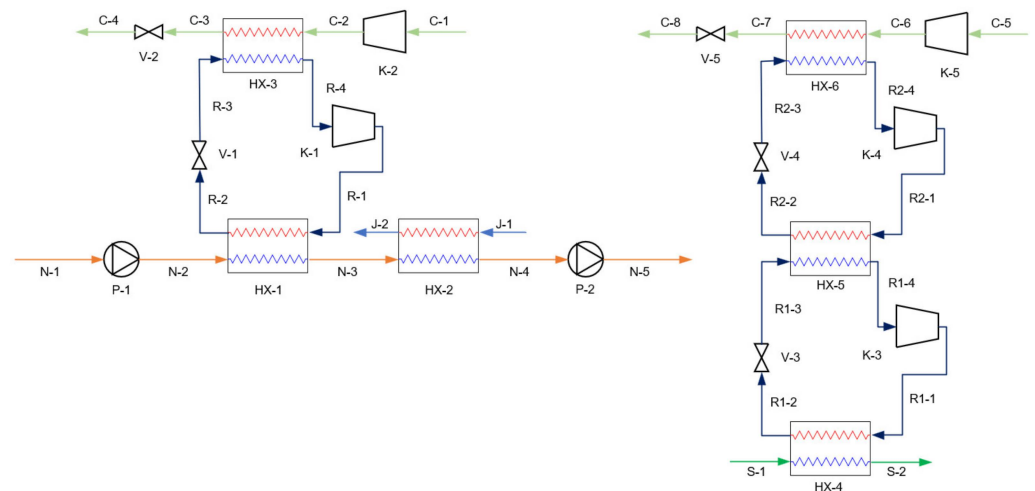


Figure 6. CO₂ BOG re-liquefaction system usable at all engine loads described by Lu et al. [17].

3.2. Simulation of Proposed Systems

Figure 7 shows the CO₂ BOG re-liquefaction system proposed in this study. This system uses a vapor-compression RC and NH₃ cold energy to reliquefy the CO₂ BOG. The heat exchanger HX-3 was added between the refrigerant compressor K-1 and the heat exchanger HX-1 in the RC. HX-3 served to cool the RC refrigerant using seawater. Therefore, HX-3 can compensate for the lack of cooling in HX-1 owing to its low engine load. Compared with the system proposed by Lu et al. [17], there is a significant advantage in that CO₂ can be completely reliquefied regardless of the engine load in a simple configuration.

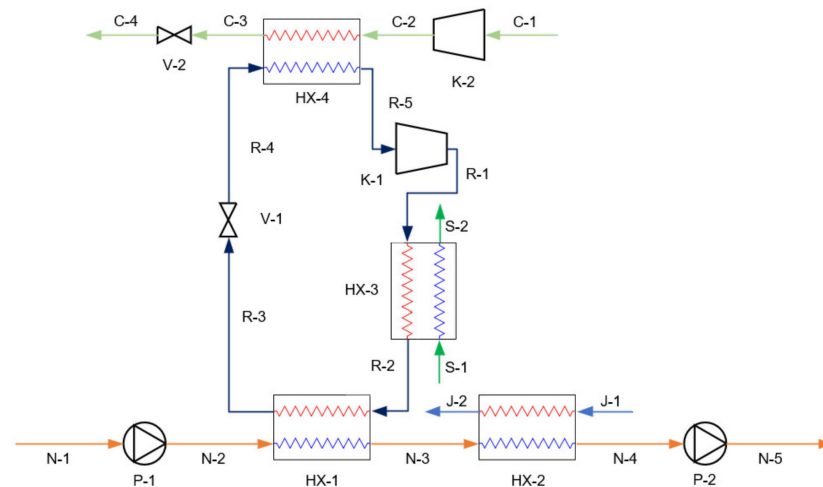


Figure 7. CO₂ BOG re-liquefaction system proposed in this study.

The refrigerant used in the RC of this system was selected as R744 (CO₂); the information regarding this refrigerant is shown in Table 5. R744 has an ozone depletion potential of zero, an extremely low GWP, no toxicity, and is very safe. Therefore, it is a suitable refrigerant for use in RCs. The system in Figure 7 refers to the simulation conditions in Figures 4 and 5 for comparison with existing systems under the same conditions. The main input parameters are listed in Table 6.

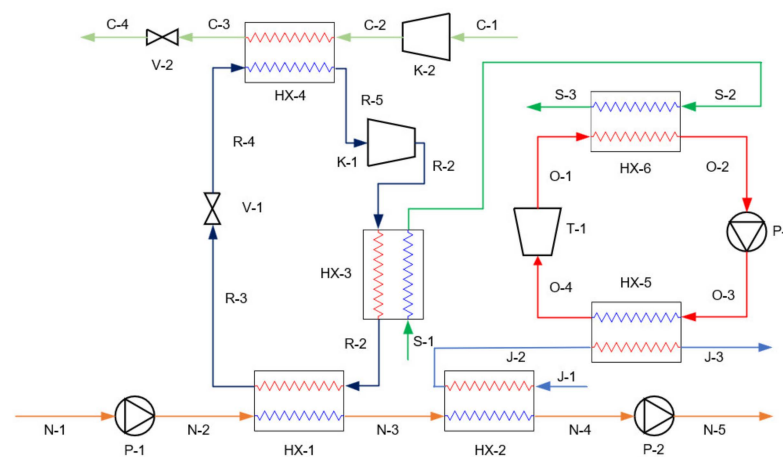
Table 5. Property information of R744 [33].

Working Fluids	ODP	GWP	t_c (°C)	p_c (kPa)	ρ_c (kg/m ³)	ASHRAE Safety Group	ASHRAE Toxicity
R744	0	1	304.19	7.38	40.8	A1	No

Table 6. Input parameters for simulation of Figure 7.

Stream	Fluid	t (°C)	p (kPa)	$\dot{m}$ (kg/h)	VF
N-1	NH ₃	−33.5	200	5018	-
N-2	NH ₃	-	1700	-	-
N-4	NH ₃	40	-	-	-
N-5	NH ₃	-	8000	-	-
J-1	H ₂ O	90	200	6.3×10^4	-
R-2	R744	35	-	-	-
R-4	R744	−55	-	-	-
R-5	R744	-	-	-	1
S-1	H ₂ O	30	200	9.0×10^4	-
C-1	CO ₂	−50	-	2470	1
C-2	CO ₂	-	859	-	-
C-4	CO ₂	−50	-	-	-

The system in Figure 7 can completely reliquefy the CO₂ BOG regardless of the load on the main engine. Adding an ORC can be considered as an option, as shown in Figure 8. Specifically, fresh water (J-2), which exchanged heat with the NH₃ fuel in HX-2, flowed to HX-5. Seawater (S-2), which exchanged heat with the refrigerant in HX-3, was allowed to flow into HX-6. HX-5 and HX-6 became the evaporator and condenser of the ORC, respectively. The turbine T-1 and pump P-3 were then added to form the ORC. NH₃ was used as the working fluid for the ORC, as indicated in the study by Kim et al. [6]. To simulate Figure 8, an additional condition for the ORC is required based on the design conditions shown in Figure 7. Table 7 lists the input parameters for the ORC analysis shown in Figure 8. Here, the pressure of O-4 was adjusted so that the PPTD of HX-5 was 5 °C. Additionally, the PPTD of HX-6 was assumed to be 5 °C. The efficiencies of turbine T-1 and pump P-3 were set to 80% and 75%, respectively.

**Figure 8.** The CO₂ BOG re-liquefaction system with an ORC proposed in this study.**Table 7.** Input parameters for simulation of Figure 8.

Stream	Fluid	t (°C)	p (kPa)	$\dot{m}$ (kg/h)	VF
O-2	NH ₃	40	-	-	0
O-4	NH ₃	-	-	-	1

4. Analysis Method

4.1. Case Classification

In this study, the system was analyzed by dividing it into three cases. Case 1 was the system proposed by Lu et al. [17]. If the system shown in Figure 4 was installed on a ship

alone, complete BOG re-liquefaction would not be possible under certain circumstances. Therefore, the optimal operation of the situation shown in Figure 6 needs to be analyzed, which includes the systems in Figures 4 and 5. Case 1 was defined as the state with the best thermodynamic performance when simultaneously using both systems shown in Figure 6, regardless of the engine load. Cases 2 and 3 correspond to Figures 7 and 8, respectively, and represent the systems proposed in this study.

4.2. Independent Variables

In this study, two major independent variables were selected and analyzed. First, the system was analyzed by varying the flow rate of the NH₃ fuel according to the engine load. The NH₃ fuel flow rate supplied according to the engine load is given in Equation (1) [17]. Here, the normal (85%) and low (65%) loads of the engine were considered. The normal and low load flow rates were 5018 and 3838 kg/h, respectively.

$$\dot{m} = \eta \times \frac{q \times P}{1000} \times \frac{q_{MDO}}{q_{NH_3}} \quad (1)$$

where,

$\dot{m}$: NH₃ fuel mass flow rate, kg/h

η : Ship load, %;

q : Fuel consumption rate, 180 g/(kW·h);

P : Ship main engine power, 14,500.5 kW;

q_{MDO} : Calorific value of marine diesel oil, 42 MJ/kg;

q_{NH_3} : Calorific value of NH₃, 18.568 MJ/kg.

Next, the amount of CO₂ generated by the BOG was varied and analyzed depending on the ambient temperature. The CO₂ BOG flow rate generated based on the ambient temperature assumed in this study is shown in Figure 9 [17].

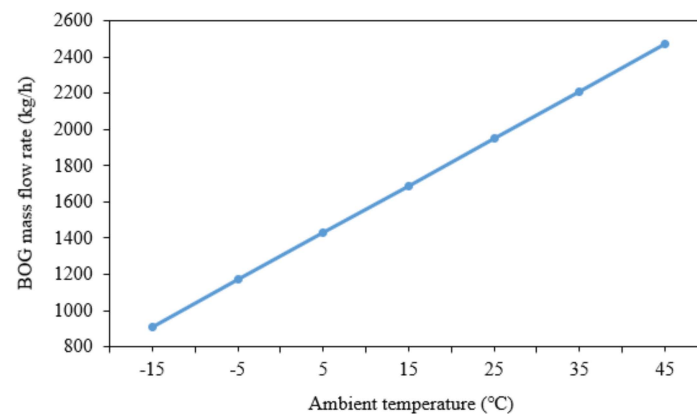


Figure 9. CO₂ BOG mass flow rate depending on ambient temperature.

4.3. Thermodynamic and Economic Analysis Model

Energy, exergy, and economic analyses were conducted for each system. In the energy analysis, the performance was analyzed using specific energy consumption (SEC). SEC refers to the net power consumption of the entire system per mass flow rate of reliquefied CO₂. The lower the SEC and net power consumption, the better the system. The SEC and net power consumption can be expressed by Equations (2) and (3) [20]. The power of each fluid machine is given by Equations (4)–(6). The power ($\dot{W}$) of each component is calculated by multiplying the mass flow rate ($\dot{m}$) and the enthalpy (h) difference between the inlet and outlet [20].

$$SEC = \frac{\dot{W}_{net}}{\dot{m}_{LCO_2}} \quad (2)$$

$$\dot{W}_{net} = \sum \dot{W}_{comp} + \sum \dot{W}_{pump} - \dot{W}_{turb} \quad (3)$$

$$\dot{W}_{comp} = \dot{m}(h_{out} - h_{in}) \quad (4)$$

$$\dot{W}_{pump} = \dot{m}(h_{out} - h_{in}) \quad (5)$$

$$\dot{W}_{turb} = \dot{m}(h_{in} - h_{out}) \quad (6)$$

In the exergy analysis, the exergy destruction rate of the system was analyzed. The exergy per unit mass is expressed using Equation (7) [34,35]. Here, the subscript 0 refers to the reference state (101.325 kPa, 25 °C). The exergy destruction rates of the compressor, pump, turbine, heat exchanger, and J-T valve that comprise the system are given in Equations (8)–(12), respectively [34].

$$e = (h - h_0) - T_0(s - s_0) \quad (7)$$

$$\dot{E}_{D,comp} = \dot{m}(e_{in} - e_{out}) + \dot{W}_{comp} \quad (8)$$

$$\dot{E}_{D,pump} = \dot{m}(e_{in} - e_{out}) + \dot{W}_{pump} \quad (9)$$

$$\dot{E}_{D,turb} = \dot{m}(e_{in} - e_{out}) - \dot{W}_{turb} \quad (10)$$

$$\dot{E}_{D,HX} = \dot{m}_{hot}(e_{hot,in} - e_{hot,out}) + \dot{m}_{cold}(e_{cold,in} - e_{cold,out}) \quad (11)$$

$$\dot{E}_{D,JT} = \dot{m}(e_{in} - e_{out}) \quad (12)$$

In the economic analysis, the annual CAPEX and OPEX were considered. To calculate CAPEX, the total capital investment (TCI) was calculated, as shown in Table 8. To obtain the TCI, the purchased equipment cost (PEC) was calculated for each component. The direct (DC) and indirect (IC) costs were calculated based on this. Subsequently, the TCI was calculated using the PEC, DC, and IC.

Table 8. Equations of total capital investment.

Category.	Equations	Reference
PEC		
Compressor	$PEC_{comp} = 8400 + 3100 \times (\dot{W}_{comp})^{0.6}$	Ref. [20]
Pump	$PEC_{pump} = 920 + 600 \times (\dot{W}_{pump})^{0.7}$	Ref. [20]
Turbine	$PEC_{turb} = 378 \times (\dot{W}_{turb}/0.746)^{0.81}$	Ref. [34]
Heat exchanger	$PEC_{HX} = 1100 + 850 \times (A_{HX})^{0.4}$	Ref. [20]
DC		
Equipment installation	47% of PEC	Ref. [36]
Piping	68% of PEC	
Electrical equipment and materials	11% of PEC	
Instrumentation and control	36% of PEC	
IC		
Engineering and supervision	8% of DC	Ref. [36]
Construction costs and contractor's profit	15% of DC	
TCI	$TCI = 1.47 \times (PEC + DC + IC)$	Ref. [37]

The annual CAPEX was calculated using Equation (13) and the calculated TCI [38,39]. The interest rate (i) and operating period (n) were assumed to be 10% and 20 years, respectively [20].

$$CAPEX = TCI \times \frac{i(1+i)^n}{(1+i)^n - 1} \quad (13)$$

The annual OPEX was calculated using net power consumption. Specifically, the electrical cost was calculated considering 16.67 USD/GJ [20]. Subsequently, the total annual costs (TAC) were obtained by adding the annual CAPEX and OPEX, as shown in Equation (14).

$$\text{TAC} = \text{CAPEX} + \text{OPEX} \quad (14)$$

Figure 10 shows the flow chart of the thermodynamic and economic analysis for each case. Here, all calculations are performed according to the given analysis conditions and models.

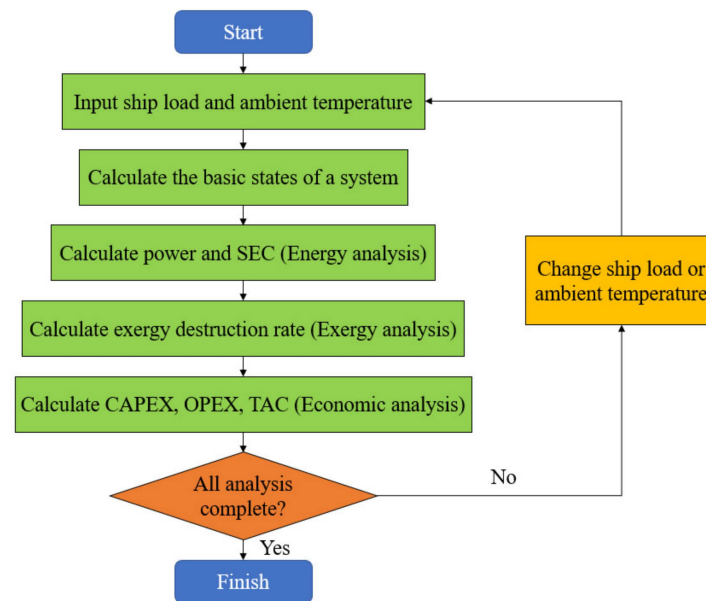


Figure 10. Flow chart of thermodynamic and economic analysis for each case.

5. Results and Discussion

5.1. Energy Analysis

The main results of the energy analysis are summarized in Table A1. Figure 11 shows the SEC and net power consumption ($\dot{W}_{net}$) with respect to the ambient temperature. Figure 11a,c show that $\dot{W}_{net}$ increases with the ambient temperature, regardless of the engine load. As the ambient temperature increases, the amount of CO₂ BOG generated increases, and the power required for the compressor to reliquefy it also increases. Figure 11b,d show that the SEC generally tends to increase proportionally to ambient temperature. However, when the ambient temperature is as low as $-15\text{ }^{\circ}\text{C}$ at a normal load, the SECs of Cases 1 and 2 increase slightly compared to that at $-5\text{ }^{\circ}\text{C}$. This is because when the ambient temperature is low, the amount of CO₂ generated by the BOG is also low.

Figure 11a,b show that $\dot{W}_{net}$ and SEC do not differ significantly between Cases 1 and 2 at normal loads. However, Case 1 required a cascade system containing multiple components to obtain a thermodynamic performance like that in Case 2. In contrast, Case 2 achieved results like those in Case 1 with only one heat exchanger. This means that Case 2 proposed in this study has significant advantages over the existing system. In Case 3, an ORC was added to Case 2. Because of the output generated by the ORC's turbine, the performance of Case 3 can be significantly improved compared to Cases 1 and 2. Specifically, the $\dot{W}_{net}$ and SEC of Case 3 can be lowered by 35.5–45.4 kW and 51.8–171.5, respectively, compared to those of Case 1.

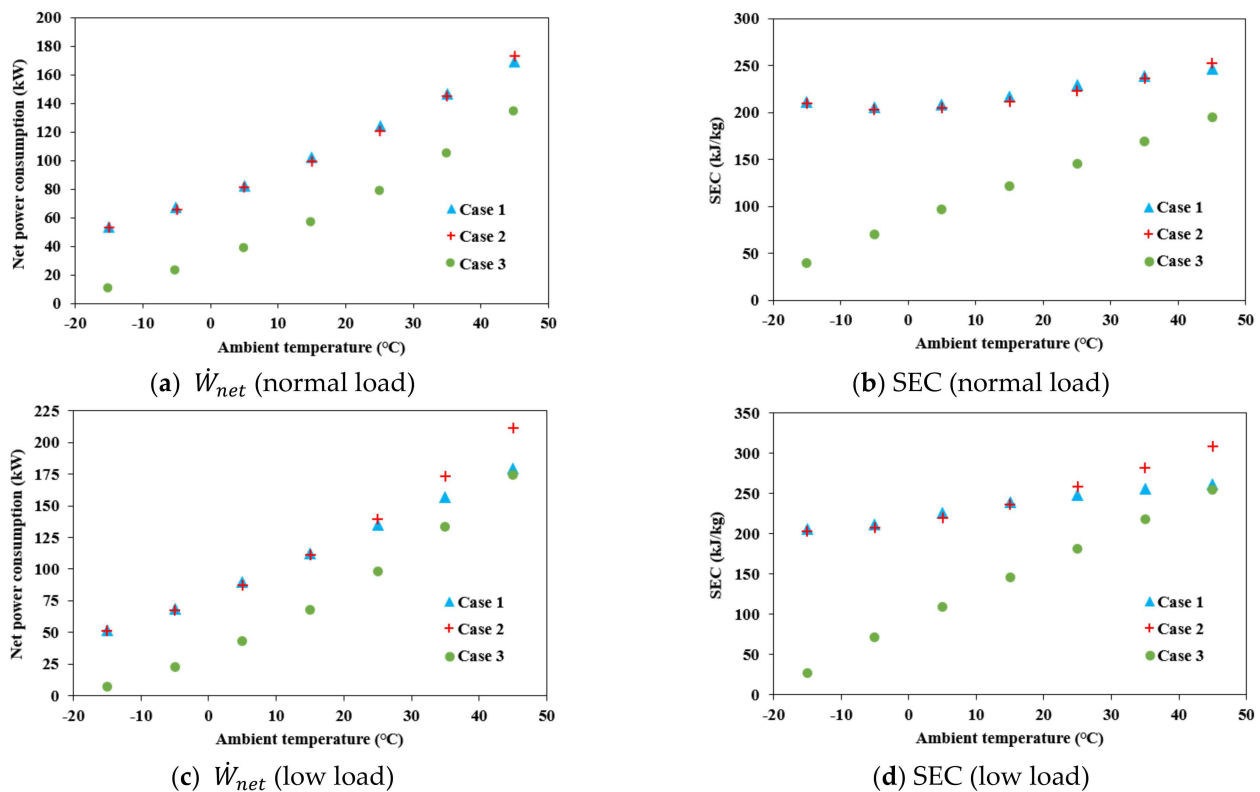


Figure 11. Net power consumption and SEC according to ambient temperature.

Figure 11c,d show the results at a low load when the cold energy of the NH_3 fuel is not fully used. The cascade system in Case 1 and HX-3 in Cases 2 and 3 then play an important role in the CO_2 BOG re-liquefaction. At 25–45 °C, the $\dot{W}_{net}$ and SEC of Case 1 are lower than those of Case 2. The cascade system requires the installation of many additional components, but it is useful in low-load and high-ambient-temperature situations. Below 15 °C, the results of Cases 1 and 2 are similar. Case 3 shows a performance difference compared to Case 1, even at a low load. Specifically, the $\dot{W}_{net}$ and SEC of Case 3 can be lowered by 5.1–46.4 kW and 7.4–178.3, respectively, compared to those of Case 1. The ORC in Case 3 can improve the performance compared with the existing system in the engine load and ambient temperature range specified in this study.

Figure 12 shows the power of each item of equipment according to the ambient temperature. Each result was expressed according to the case and engine load. Considering the common results in Figure 12a,b, the power of the refrigerant compressor K-1 in the RC accounts for a significant portion of the total power consumption. As the amount of CO_2 BOG increases with ambient temperature, the power of refrigerant compressors K-3 and K-4 in the cascade system gradually increase. Conversely, as the ambient temperature decreases, the power of these compressors decrease because the role of the cascade system is reduced. Comparing Figure 12a,b, there was a difference in the flow rate of the NH_3 fuel flowing into the system owing to the engine load. At a low load, the powers of P-1 and P-2 decreased compared to the normal load. In addition, the cold energy of NH_3 fuel cannot be sufficiently used for CO_2 BOG re-liquefaction; therefore, the power required for the compressors in the cascade system at this time increases compared to the normal load.

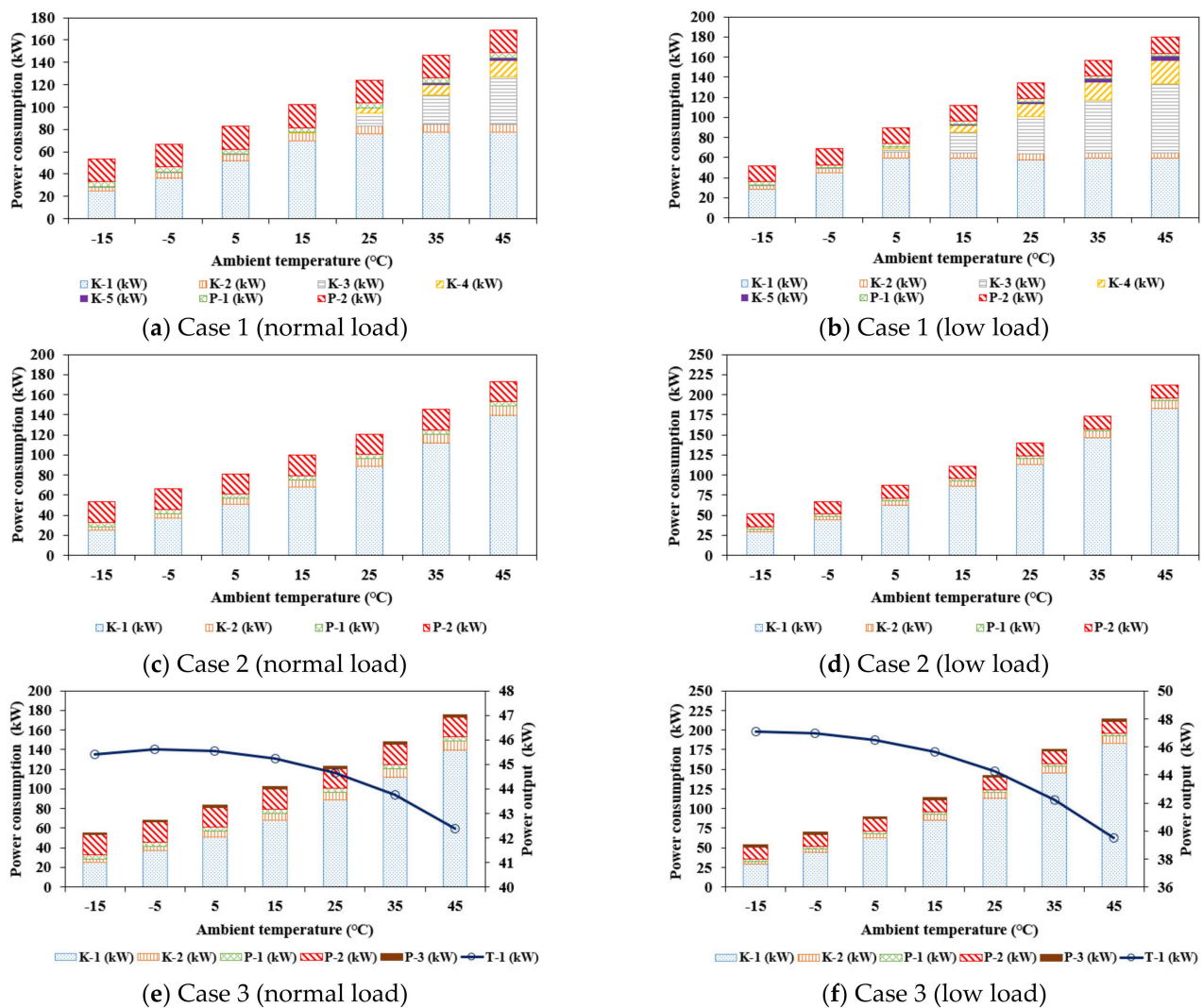


Figure 12. Power of each item of equipment according to ambient temperature.

Figure 12c–f show that the power of K-1 increases proportionally with the ambient temperature. Cases 2 and 3 did not have a separate cascade system, and the CO₂ BOG was relieved using only one RC. Therefore, the power of K-1 dominated the overall power. Like Case 1, the powers of P-1 and P-2 in Cases 2 and 3 vary depending on the engine load. Meanwhile, at the same engine load and ambient temperature, the power consumption between Figure 12c,e has only a small difference as much as the ORC pump P-3, and the same is true for Figure 12d,f. In Figure 12e,f, the power output of the ORC turbine T-1 is generated. Case 3 did not have a significant difference in power consumption compared to Case 2 under the same conditions, but the net power consumption was low owing to the output of the turbine. In Case 3, the turbine output decreased as the ambient temperature increased. As the ambient temperature increased, the amount of CO₂ BOG increased, and the temperature of the seawater passing through the heat exchanger in the RC increased. Warm seawater then flowed into the ORC condenser. Because the PPTD of the ORC condenser was assumed to be 5 °C, the outlet temperature of the seawater from the condenser was almost constant. Therefore, an increase in the temperature of the incoming seawater reduced the condenser duty and working fluid flow rate of the ORC. This caused a decrease in the output of the ORC turbine T-1.

5.2. Exergy Analysis

Detailed results of the exergy analysis are listed in Table A2. Figure 13 shows the variation in exergy destruction rate with changing ambient temperature. As shown in Figure 13a, the exergy destruction rates of Cases 1 and 2 are similar under normal loads. In both cases, NH₃ fuel cold energy can be fully used; thus, there is no significant difference in the exergy destruction rate. Figure 13b shows that the exergy destruction rate of Case 2 is higher than that of Case 1 when the ambient temperature >35 °C; however, at temperatures <25 °C, the exergy destruction rates of Cases 1 and 2 are similar. When the ambient temperature is high at a low load, Case 1, which operates optimally through the two systems, has a lower exergy destruction rate than Case 2, which does not use sufficient NH₃ cold energy. Meanwhile, the exergy destruction rate of Case 3 is 33.7–42.4 kW higher in Figure 13a and 36.1–49.7 kW higher in Figure 13b than that of Case 1. An increase in the exergy destruction rate due to the application of the ORC is inevitable.

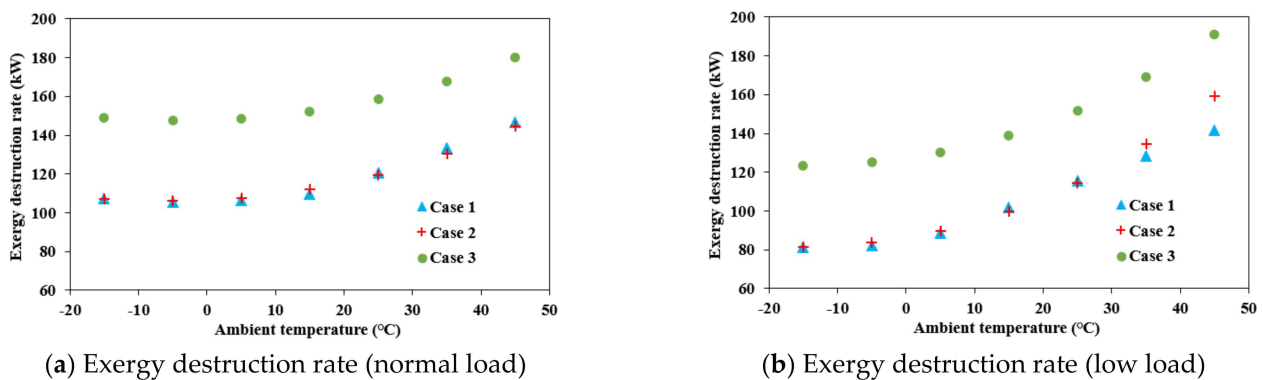


Figure 13. Exergy destruction rate according to ambient temperature.

Figure 14 shows the exergy destruction rate of each stream with respect to the ambient temperature. This was expressed by adding the exergy destruction rates of the components belonging to each stream. The definitions of each stream are listed in Table 9. The stream categories were classified as NH₃, CO₂, RC, and ORC. The heat exchangers through which the refrigerant passes were classified separately as refrigerant heat exchangers (Ref. HX).

Table 9. The stream categories of the CO₂ BOG re-liquefaction systems.

Stream	Case 1	Case 2	Case 3
NH ₃	P-1, HX-2, P-2	P-1, HX-2, P-2	P-1, HX-2, P-2
CO ₂	K-2, V-2, K-5, V-5	K-2, V-2	K-2, V-2
RC	K-1, V-1, K-3, V-3, K-4, V-4	K-1, V-1	K-1, V-1
Ref. HX	HX-1, HX-3, HX-4, HX-5, HX-6	HX-1, HX-3, HX-4	HX-1, HX-3, HX-4
ORC	-	-	P-3, T-1, HX-5, HX-6

The following common phenomena can be observed in Figure 14a–f. Considering the NH₃ stream, as the ambient temperature decreased, the temperature of the NH₃ fuel flowing into HX-2 decreased. This increased the temperature difference between the inlet and outlet of the NH₃ fuel in HX-2, causing an increase in the exergy destruction rate. In addition, owing to the differences in the flow rate of the NH₃ fuel depending on the engine load, the exergy destruction rate at a low load was less than that at a normal load. Meanwhile, the CO₂ stream is small compared to the overall exergy destruction rate. RC and Ref. HX increased proportionally with the ambient temperature. As the CO₂ BOG increased, the duty of each component to reliquefy it increased, and the exergy destruction rate increased. RC and Ref. HX have a significant impact on the exergy destruction rate; therefore, it is important to install and operate the compressors, heat exchangers, and valves that make them up under optimal conditions.

Figure 14e,f show that Case 3 requires additional consideration of the ORC exergy destruction rate. As shown in Table A2, most of the exergy destruction rate of the ORC occurs in the turbine and heat exchanger. In other words, the thermodynamic performance of the system can be improved by reducing the exergy destruction rate for the turbine and heat exchanger.

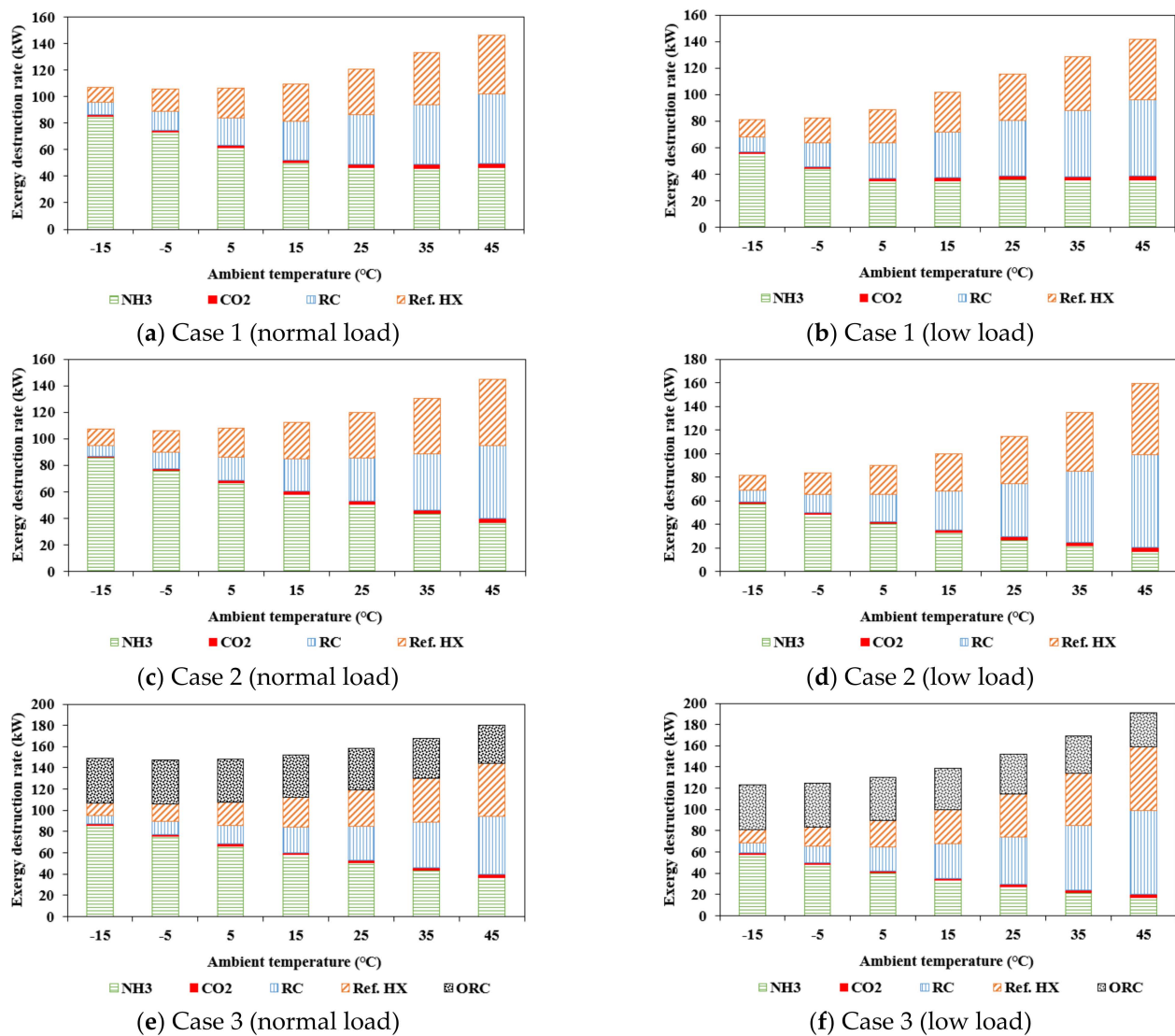


Figure 14. Exergy destruction rate of each stream according to ambient temperature.

5.3. Economic Analysis

An economic analysis was performed at a normal engine load and an ambient temperature of 45 °C, which generated large amounts of CO₂ BOG. The main results of the economic analyses are presented in Table A3. Figure 15 shows the PEC for each case after the economic analysis.

The compressors are the most expensive components. In particular, the price of the refrigerant compressor K-1, which operates in the RC between the CO₂ BOG and NH₃ fuel, was the highest in all cases. In Case 1, the prices of K-3 and K-4, which operated in a cascade system, were also high. In Cases 2 and 3, there is only one refrigerant compressor K-1; thus, the price is higher than that of Case 1.

In all cases, the cost for the BOG compressor is second only to the refrigerant compressor. Pumps, turbines, and heat exchangers are relatively inexpensive compared with

compressors. Therefore, it is economically attractive to reduce the number of compressors where possible.

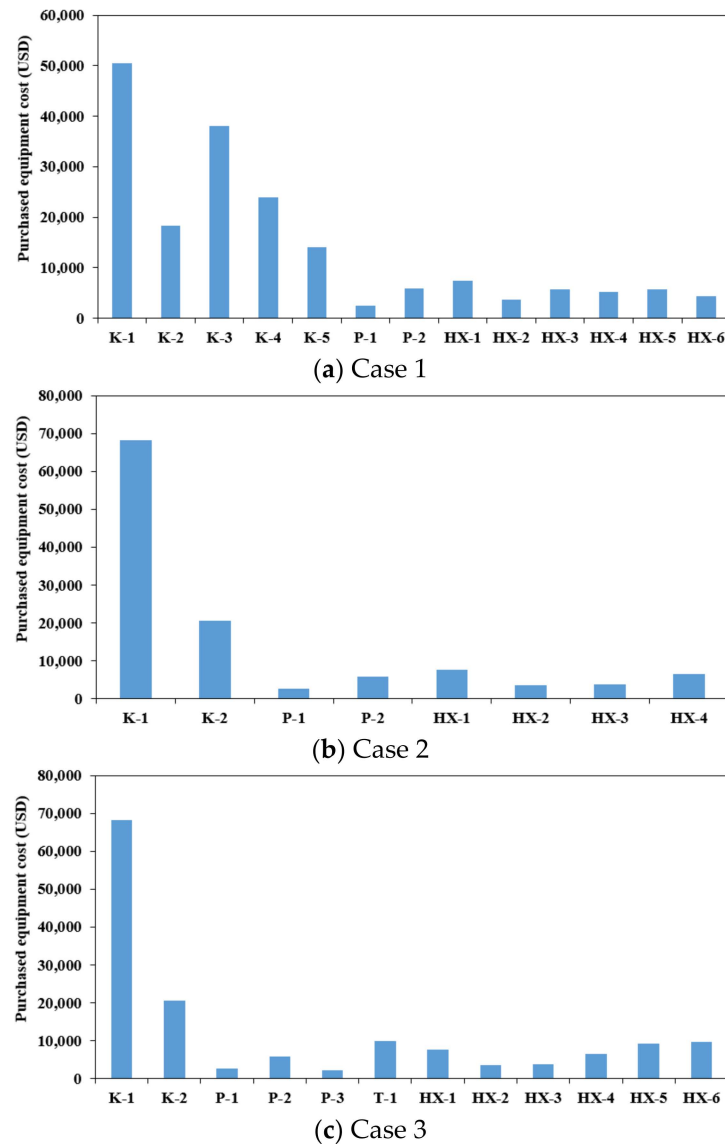


Figure 15. The purchased equipment cost of the CO₂ BOG re-liquefaction systems.

Figure 16 and Table 10 present the annual costs for each case. Specifically, they include CAPEX, OPEX, and TAC. Among the three cases, Case 1 had the highest CAPEX. This is because Case 1 required multiple compressors. Case 2 exhibits the lowest CAPEX values. This is because Case 2 has a small number of compressors and does not have a system such as an ORC, as in Case 3.

Table 10. CAPEX, OPEX, and TAC of the CO₂ BOG re-liquefaction systems.

Item	Case 1	Case 2	Case 3
CAPEX (USD/year)	95,656.253	61,255.71	77,317.00
OPEX (USD/year)	88,831.025	91,101.90	70,151.89
TAC (USD/year)	184,487.28	152,357.60	147,468.89

Table A3 shows the net power consumption ($\dot{W}_{net}$) for each case. Specifically, the values for Cases 1, 2, and 3 are 168.98 kW, 173.29 kW, and 133.44 kW, respectively. Case 1

had a lower $\dot{W}_{net}$ than Case 2, when the cascade system was optimally used. Additionally, Case 3 can significantly lower $\dot{W}_{net}$ compared with Cases 1 and 2, owing to the ORC. OPEX was calculated in proportion to $\dot{W}_{net}$. Therefore, the OPEX is the highest in Case 2 and the lowest in Case 3.

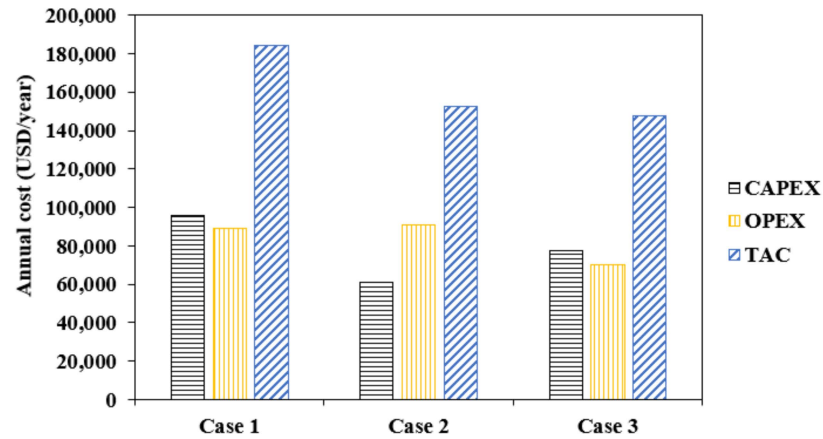


Figure 16. The annual costs of the CO₂ BOG re-liquefaction systems.

The TAC was calculated by adding CAPEX and OPEX. Case 2 had a high OPEX, but the CAPEX was lower than that of Case 1; therefore, the TAC was lower than that of Case 1, so it can be considered economical. Case 3, which adds the ORC as an option, increased the CAPEX compared to Case 2 but can significantly lower the OPEX. Therefore, Case 3 exhibits the lowest TAC among the three cases. Case 1 had the lowest economic feasibility, even though the existing system operated optimally. Cases 2 and 3 proposed in this study can reduce the total annual costs by 17.4% and 20.1%, respectively, compared to Case 1.

6. Conclusions

In this study, the BOG re-liquefaction system of a high-capacity LCO₂ carrier using cold energy from NH₃ was analyzed. Instead of a cascade system that requires multiple components, a system was proposed to completely reliquefy the CO₂ BOG at all of the engine loads using only one heat exchanger. In addition, low-GWP R744 was used as the working fluid in the RC. Here, the ORC can be added as an option to reduce the net power consumption. The existing and proposed systems were classified into Cases 1, 2, and 3, and the main results of the thermodynamic and economic analyses were as follows:

- Case 2 was simpler than Case 1; however, the energy-analysis results were similar. At a low load and high ambient temperature, the performance of Case 1 was superior to that of Case 2. The $\dot{W}_{net}$ and SEC of Case 3 were lower than those of Case 1 by 35.5–45.4 kW and 51.8–171.5, respectively, at a normal load, and 5.1–46.4 kW and 7.4–178.3, respectively, at a low load. Case 3 showed an improved thermodynamic performance owing to the turbine output in the ORC.
- As a result of the exergy analysis, the results of Cases 1 and 2 were largely the same, except when the ambient temperature was high at a low load, for reasons like those expressed for the energy analysis. Meanwhile, the exergy destruction rate in Case 3 was 33.7–42.4 kW higher at a normal load and 36.1–49.7 kW higher at a low load than Case 1, due to the addition of the ORC. In Case 3, it was important to minimize the ORC exergy destruction rate.
- Case 1 exhibited the highest CAPEX because it required multiple expensive compressors. Instead, the OPEX is lower than that in Case 2, owing to the cascade system. Case 2 had the lowest CAPEX because it used a small number of components, but it used the highest OPEX. Case 3 had a higher CAPEX than Case 2 owing to the ORC, but it had an extremely low OPEX. Cases 2 and 3 proposed in this study can reduce the total annual costs by 17.4% and 20.1%, respectively, compared with Case 1.

Compared to Case 1, Case 2 proposed in this study is simpler, shows a similar thermodynamic performance, and is also superior in terms of economic feasibility. Case 3, in which an ORC is added, can greatly reduce the net power consumption of the system; therefore, it is economically advantageous when used in the long term. The results of this study can serve as a useful reference in the field of high-capacity LCO₂ carriers and BOG re-liquefaction systems that use NH₃ as fuel. In the future, environmental analyses and verification studies should be conducted.

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Nomenclature

A	Area	n	Operating period
BOG	Boil-off gas	NH ₃	Ammonia
CAPEX	Capital expenditure	ODP	Ozone depletion potential
CCUS	Carbon capture utilization and storage	OPEX	Operating expenditure
CO ₂	Carbon dioxide	ORC	Organic Rankine cycle
DC	Direct cost	P	Pump
e	Specific exergy	p	Pressure
$\dot{E}_d$	Exergy destruction rate	PEC	Purchased equipment cost
GWP	Global warming potential	PPTD	Pinch point temperature difference
h	Enthalpy	RC	Refrigerant cycle
H ₂	Hydrogen	Ref.	Reference, Refrigerant
HP	High pressure	s	Entropy
HX	Heat exchanger	SEC	Specific energy consumption
i	Interest rate	T	Turbine
IC	Indirect cost	t	Temperature
J–T	Joule–Thomson	TAC	Total annual cost
K	Compressor	TCI	Total capital investment
LCO ₂	Liquefied carbon dioxide	V	Valve
LNG	Liquefied natural gas	VF	Vapor fraction
LP	Low pressure	$\dot{W}$	Power
$\dot{m}$	Mass flow rate		
Greeks			
ρ	Density		
Subscripts			
0	Reference state	in	In
c	Critical point	JT	Joule–Thomson
$cold$	Cold	net	Net
$comp$	Compressor	out	Out
hot	Hot	$pump$	Pump
HX	Heat exchanger	$turb$	Turbine

Appendix A

Table A1. Summary of main results for energy analysis.

Ambient Temperature (°C)	−15	−5	5	15	25	35	45
Case 1 (normal load)							
K-1 (kW)	24.49	36.64	51.64	69.90	76.28	77.38	77.20
K-2 (kW)	3.50	4.51	5.52	6.53	6.85	6.90	6.89
K-3 (kW)	0.62	0.62	0.62	0.62	11.66	26.83	42.97
K-4 (kW)	0.21	0.21	0.21	0.21	3.97	9.13	14.62
K-5 (kW)	0.04	0.04	0.04	0.04	0.74	1.70	2.72
P-1 (kW)	4.14	4.14	4.14	4.14	4.14	4.14	4.14
P-2 (kW)	20.44	20.44	20.44	20.44	20.44	20.44	20.44
$\dot{W}_{net}$ (kW)	53.44	66.59	82.60	101.87	124.07	146.51	168.98
SEC (kJ/kg)	211.39	204.90	207.95	217.01	229.05	238.66	246.28
Case 1 (low load)							
K-1 (kW)	28.69	44.38	59.71	59.36	58.06	59.08	59.10
K-2 (kW)	3.50	4.51	5.30	5.28	5.22	5.27	5.27
K-3 (kW)	0.62	0.62	4.12	20.38	37.36	52.59	68.57
K-4 (kW)	0.21	0.21	1.40	6.93	12.71	17.89	23.33
K-5 (kW)	0.04	0.04	0.26	1.29	2.36	3.32	4.33
P-1 (kW)	3.16	3.16	3.16	3.16	3.16	3.16	3.16
P-2 (kW)	15.63	15.63	15.63	15.63	15.63	15.63	15.63
$\dot{W}_{net}$ (kW)	51.85	68.55	89.59	112.04	134.50	156.95	179.40
SEC (kJ/kg)	205.13	210.92	225.55	238.67	248.31	255.66	261.47
Case 2 (normal load)							
K-1 (kW)	24.84	36.72	51.13	68.31	88.54	112.07	139.11
K-2 (kW)	3.54	4.55	5.56	6.57	7.58	8.59	9.60
P-1 (kW)	4.14	4.14	4.14	4.14	4.14	4.14	4.14
P-2 (kW)	20.44	20.44	20.44	20.44	20.44	20.44	20.44
$\dot{W}_{net}$ (kW)	52.96	65.84	81.26	99.46	120.70	145.24	173.29
SEC (kJ/kg)	209.50	202.60	204.58	211.87	222.83	236.59	252.58
Case 2 (low load)							
K-1 (kW)	28.86	43.89	62.69	85.70	113.31	145.75	183.02
K-2 (kW)	3.54	4.55	5.56	6.57	7.58	8.59	9.60
P-1 (kW)	3.16	3.16	3.16	3.16	3.16	3.16	3.16
P-2 (kW)	15.63	15.63	15.63	15.63	15.63	15.63	15.63
$\dot{W}_{net}$ (kW)	51.19	67.23	87.04	111.06	139.68	173.14	211.42
SEC (kJ/kg)	202.50	206.87	219.13	236.59	257.88	282.04	308.14
Case 3 (normal load)							
K-1 (kW)	24.84	36.72	51.13	68.31	88.54	112.07	139.12
K-2 (kW)	3.54	4.55	5.56	6.57	7.58	8.59	9.60
P-1 (kW)	4.14	4.14	4.14	4.14	4.14	4.14	4.14
P-2 (kW)	20.44	20.44	20.44	20.44	20.44	20.44	20.44
P-3 (kW)	2.55	2.57	2.59	2.59	2.58	2.56	2.50
T-1 (kW)	45.41	45.59	45.53	45.24	44.66	43.75	42.36
$\dot{W}_{net}$ (kW)	10.09	22.83	38.32	56.82	78.62	104.04	133.44
SEC (kJ/kg)	39.90	70.24	96.48	121.04	145.15	169.48	194.49
Case 3 (low load)							
K-1 (kW)	28.86	43.89	62.69	85.70	113.31	145.76	183.02
K-2 (kW)	3.54	4.55	5.56	6.57	7.58	8.59	9.60
P-1 (kW)	3.16	3.16	3.16	3.16	3.16	3.16	3.16
P-2 (kW)	15.63	15.63	15.63	15.63	15.63	15.63	15.63
P-3 (kW)	2.70	2.72	2.71	2.69	2.63	2.54	2.40
T-1 (kW)	47.10	46.98	46.51	45.63	44.23	42.23	39.52
$\dot{W}_{net}$ (kW)	6.79	22.97	43.24	68.12	98.08	133.45	174.30
SEC (kJ/kg)	26.87	70.68	108.86	145.10	181.07	217.39	254.04

Table A2. Summary of main results for exergy analysis.

Ambient Temperature (°C)	−15	−5	5	15	25	35	45
Case 1 (normal load)							
$\dot{E}_{D,K-1}$ (kW)	4.99	7.26	9.98	13.18	14.28	14.47	14.44
$\dot{E}_{D,K-2}$ (kW)	1.09	1.40	1.71	2.03	2.12	2.14	2.14
$\dot{E}_{D,K-3}$ (kW)	0.10	0.10	0.10	0.10	1.92	4.43	7.09
$\dot{E}_{D,K-4}$ (kW)	0.04	0.04	0.04	0.04	0.85	1.95	3.12
$\dot{E}_{D,K-5}$ (kW)	0.01	0.01	0.01	0.01	0.23	0.53	0.84
$\dot{E}_{D,P-1}$ (kW)	0.85	0.85	0.85	0.85	0.85	0.85	0.85
$\dot{E}_{D,P-2}$ (kW)	3.10	3.10	3.10	3.10	3.10	3.10	3.10
$\dot{E}_{D,HX-1}$ (kW)	5.33	8.41	12.11	16.35	17.75	17.99	17.95
$\dot{E}_{D,HX-2}$ (kW)	81.18	69.37	57.72	46.25	42.74	42.17	42.25
$\dot{E}_{D,HX-3}$ (kW)	6.28	8.10	9.91	11.73	12.29	12.38	12.37
$\dot{E}_{D,HX-4}$ (kW)	0.07	0.07	0.07	0.07	1.67	3.82	6.04
$\dot{E}_{D,HX-5}$ (kW)	0.05	0.05	0.05	0.05	0.86	1.98	3.18
$\dot{E}_{D,HX-6}$ (kW)	0.07	0.07	0.07	0.07	1.32	3.04	4.87
$\dot{E}_{D,V-1}$ (kW)	3.86	6.54	10.33	15.56	17.53	17.87	17.82
$\dot{E}_{D,V-2}$ (kW)	0.06	0.07	0.09	0.10	0.11	0.11	0.11
$\dot{E}_{D,V-3}$ (kW)	0.12	0.12	0.12	0.12	2.29	5.26	8.42
$\dot{E}_{D,V-4}$ (kW)	0.03	0.03	0.03	0.03	0.50	1.15	1.84
$\dot{E}_{D,V-5}$ (kW)	0.00	0.00	0.00	0.00	0.01	0.03	0.04
$\dot{E}_{D,total}$ (kW)	107.23	105.60	106.29	109.64	120.42	133.26	146.47
Case 1 (low load)							
$\dot{E}_{D,K-1}$ (kW)	5.68	8.49	11.16	11.10	10.87	11.05	11.05
$\dot{E}_{D,K-2}$ (kW)	1.09	1.40	1.64	1.64	1.62	1.63	1.64
$\dot{E}_{D,K-3}$ (kW)	0.10	0.10	0.68	3.36	6.16	8.67	11.31
$\dot{E}_{D,K-4}$ (kW)	0.04	0.04	0.30	1.48	2.71	3.81	4.97
$\dot{E}_{D,K-5}$ (kW)	0.01	0.01	0.08	0.40	0.73	1.03	1.34
$\dot{E}_{D,P-1}$ (kW)	0.65	0.65	0.65	0.65	0.65	0.65	0.65
$\dot{E}_{D,P-2}$ (kW)	2.37	2.37	2.37	2.37	2.37	2.37	2.37
$\dot{E}_{D,HX-1}$ (kW)	6.60	10.43	13.87	13.80	13.51	13.74	13.74
$\dot{E}_{D,HX-2}$ (kW)	52.77	41.05	32.10	32.29	32.98	32.43	32.43
$\dot{E}_{D,HX-3}$ (kW)	6.28	8.10	9.51	9.48	9.37	9.46	9.46
$\dot{E}_{D,HX-4}$ (kW)	0.07	0.07	0.58	2.91	5.28	7.33	9.42
$\dot{E}_{D,HX-5}$ (kW)	0.05	0.05	0.30	1.51	2.76	3.89	5.07
$\dot{E}_{D,HX-6}$ (kW)	0.07	0.07	0.47	2.31	4.24	5.97	7.78
$\dot{E}_{D,V-1}$ (kW)	5.16	9.25	13.84	13.73	13.32	13.64	13.64
$\dot{E}_{D,V-2}$ (kW)	0.06	0.07	0.08	0.08	0.08	0.08	0.08
$\dot{E}_{D,V-3}$ (kW)	0.12	0.12	0.81	4.00	7.32	10.31	13.44
$\dot{E}_{D,V-4}$ (kW)	0.03	0.03	0.18	0.87	1.60	2.26	2.94
$\dot{E}_{D,V-5}$ (kW)	0.00	0.00	0.00	0.02	0.04	0.05	0.07
$\dot{E}_{D,total}$ (kW)	81.14	82.30	88.64	102.00	115.63	128.38	141.41
Case 2 (normal load)							
$\dot{E}_{D,K-1}$ (kW)	4.75	6.80	9.19	11.94	15.07	18.61	22.59
$\dot{E}_{D,K-2}$ (kW)	1.10	1.41	1.72	2.04	2.35	2.67	2.98
$\dot{E}_{D,P-1}$ (kW)	0.85	0.85	0.85	0.85	0.85	0.85	0.85
$\dot{E}_{D,P-2}$ (kW)	3.10	3.10	3.10	3.10	3.10	3.10	3.10
$\dot{E}_{D,HX-1}$ (kW)	5.46	7.77	10.35	13.17	16.16	19.26	22.39
$\dot{E}_{D,HX-2}$ (kW)	81.92	71.99	62.78	54.32	46.56	39.52	33.16
$\dot{E}_{D,HX-3}$ (kW)	0.26	0.64	1.37	2.57	4.36	6.87	10.20
$\dot{E}_{D,HX-4}$ (kW)	6.35	8.17	9.98	11.80	13.61	15.43	17.24
$\dot{E}_{D,V-1}$ (kW)	3.25	5.39	8.32	12.24	17.34	23.84	31.94
$\dot{E}_{D,V-2}$ (kW)	0.06	0.07	0.09	0.10	0.12	0.14	0.15
$\dot{E}_{D,total}$ (kW)	107.10	106.19	107.77	112.12	119.53	130.28	144.59

Table A2. Cont.

Ambient Temperature (°C)	−15	−5	5	15	25	35	45
Case 2 (low load)							
$\dot{E}_{D,K-1}$ (kW)	5.33	7.80	10.76	14.23	18.28	22.92	28.17
$\dot{E}_{D,K-2}$ (kW)	1.10	1.41	1.72	2.04	2.35	2.67	2.98
$\dot{E}_{D,P-1}$ (kW)	0.65	0.65	0.65	0.65	0.65	0.65	0.65
$\dot{E}_{D,P-2}$ (kW)	2.37	2.37	2.37	2.37	2.37	2.37	2.37
$\dot{E}_{D,HX-1}$ (kW)	6.09	8.73	11.64	14.73	17.85	20.85	23.61
$\dot{E}_{D,HX-2}$ (kW)	54.80	45.70	37.55	30.35	24.06	18.64	14.03
$\dot{E}_{D,HX-3}$ (kW)	0.52	1.35	2.86	5.26	8.74	13.40	19.29
$\dot{E}_{D,HX-4}$ (kW)	6.35	8.17	9.98	11.80	13.61	15.43	17.24
$\dot{E}_{D,V-1}$ (kW)	4.27	7.42	11.96	18.23	26.60	37.39	50.79
$\dot{E}_{D,V-2}$ (kW)	0.06	0.07	0.09	0.10	0.12	0.14	0.15
$\dot{E}_{D,total}$ (kW)	81.53	83.67	89.58	99.76	114.63	134.46	159.28
Case 3 (normal load)							
$\dot{E}_{D,K-1}$ (kW)	4.75	6.80	9.19	11.94	15.07	18.61	22.59
$\dot{E}_{D,K-2}$ (kW)	1.10	1.41	1.72	2.04	2.35	2.67	2.98
$\dot{E}_{D,P-1}$ (kW)	0.85	0.85	0.85	0.85	0.85	0.85	0.85
$\dot{E}_{D,P-2}$ (kW)	3.10	3.10	3.10	3.10	3.10	3.10	3.10
$\dot{E}_{D,P-3}$ (kW)	0.38	0.39	0.39	0.39	0.39	0.38	0.38
$\dot{E}_{D,T-1}$ (kW)	10.79	10.84	10.82	10.75	10.62	10.40	10.07
$\dot{E}_{D,HX-1}$ (kW)	5.46	7.77	10.35	13.17	16.16	19.26	22.39
$\dot{E}_{D,HX-2}$ (kW)	81.93	71.99	62.78	54.32	46.56	39.52	33.16
$\dot{E}_{D,HX-3}$ (kW)	0.26	0.64	1.37	2.57	4.36	6.87	10.20
$\dot{E}_{D,HX-4}$ (kW)	6.35	8.17	9.98	11.80	13.61	15.43	17.25
$\dot{E}_{D,HX-5}$ (kW)	16.32	16.17	15.92	15.58	15.12	14.48	13.78
$\dot{E}_{D,HX-6}$ (kW)	14.12	13.92	13.63	13.23	12.73	12.11	11.35
$\dot{E}_{D,V-1}$ (kW)	3.25	5.39	8.32	12.24	17.34	23.84	31.94
$\dot{E}_{D,V-2}$ (kW)	0.06	0.07	0.09	0.10	0.12	0.14	0.15
$\dot{E}_{D,total}$ (kW)	148.72	147.50	148.52	152.08	158.40	167.65	180.19
Case 3 (low load)							
$\dot{E}_{D,K-1}$ (kW)	5.33	7.80	10.76	14.23	18.28	22.92	28.17
$\dot{E}_{D,K-2}$ (kW)	1.10	1.41	1.72	2.04	2.35	2.67	2.98
$\dot{E}_{D,P-1}$ (kW)	0.65	0.65	0.65	0.65	0.65	0.65	0.65
$\dot{E}_{D,P-2}$ (kW)	2.37	2.37	2.37	2.37	2.37	2.37	2.37
$\dot{E}_{D,P-3}$ (kW)	0.41	0.41	0.41	0.40	0.40	0.38	0.36
$\dot{E}_{D,T-1}$ (kW)	11.19	11.17	11.06	10.85	10.51	10.04	9.39
$\dot{E}_{D,HX-1}$ (kW)	6.09	8.73	11.64	14.73	17.85	20.85	23.61
$\dot{E}_{D,HX-2}$ (kW)	54.80	45.70	37.55	30.35	24.06	18.64	14.03
$\dot{E}_{D,HX-3}$ (kW)	0.52	1.35	2.86	5.26	8.74	13.41	19.29
$\dot{E}_{D,HX-4}$ (kW)	6.35	8.17	9.98	11.80	13.61	15.43	17.24
$\dot{E}_{D,HX-5}$ (kW)	16.37	16.12	15.71	15.14	14.37	13.41	12.32
$\dot{E}_{D,HX-6}$ (kW)	13.98	13.66	13.21	12.60	11.82	10.86	9.72
$\dot{E}_{D,V-1}$ (kW)	4.27	7.42	11.96	18.23	26.60	37.40	50.79
$\dot{E}_{D,V-2}$ (kW)	0.06	0.07	0.09	0.10	0.12	0.14	0.15
$\dot{E}_{D,total}$ (kW)	123.48	125.02	129.97	138.75	151.73	169.16	191.08

Table A3. Summary of main results for economic analysis.

	Case 1	Case 2	Case 3
PEC _{K-1} (USD)	50,467.65	68,293.81	68,296.55
PEC _{K-2} (USD)	18,268.18	20,445.53	20,445.53
PEC _{K-3} (USD)	37,998.43	-	-
PEC _{K-4} (USD)	23,898.09	-	-
PEC _{K-5} (USD)	14,044.76	-	-

Table A3. Cont.

	Case 1	Case 2	Case 3
PEC _{P-1} (USD)	2541.52	2541.52	2541.52
PEC _{P-2} (USD)	5880.05	5880.05	5880.05
PEC _{P-3} (USD)	-	-	2059.05
PEC _{T-1} (USD)	-	-	9963.49
PEC _{HX-1} (USD)	7446.02	7722.63	7723.39
PEC _{HX-2} (USD)	3715.08	3560.75	3560.77
PEC _{HX-3} (USD)	5725.92	3719.54	3719.60
PEC _{HX-4} (USD)	5193.05	6383.53	6383.54
PEC _{HX-5} (USD)	5656.06	-	9318.87
PEC _{HX-6} (USD)	4287.51	-	9738.24
PEC _{total} (USD)	185,122.31	11,8547.37	149,630.59
DC (USD)	299,898.14	192,046.74	242,401.55
IC (USD)	68,976.57	44,170.75	55,752.36
TCI (USD)	814,375.61	521,504.35	658,243.20
CAPEX (USD/year)	95,656.25	61,255.71	77,317.00
$\dot{W}_{net}$ (kW)	168.98	173.29	133.44
OPEX (USD/year)	88,831.03	91,101.90	70,151.89
TAC (USD/year)	184,487.28	152,357.60	147,468.89

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