

SIMULATION STUDY ON UTILISING COLD ENERGY FUEL TO LIQUEFY CAPTURED CO₂ ON LNG-FUELLED VESSELS

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ABSTRACT

This study addresses the issues of abundant cold energy in fuel and the high energy consumption of CO₂ liquefaction capture by the shipboard carbon capture system in LNG-fuelled vessels. Two liquefied CO₂ schemes are proposed: an LNG cold energy and refrigeration cycle integrated CO₂ liquefaction system (Scheme 1) and an LNG cold energy and seawater diversion liquefied CO₂ system (Scheme 2). The two systems are simulated in Aspen HYSYS software and, based on the simulation data, multiple thermodynamic parameters of the system, including exergy efficiency, cold energy utilisation rate, and energy consumption, are calculated under different vessel operating conditions, thereby verifying the feasibility of the system. On this basis, the systems are optimised, enhancing their overall performance. Through a comparative analysis of the two schemes, Scheme 1 was selected to conduct an economic analysis of typical vessel routes and calculations were conducted to determine the reduction in energy consumption and the decrease in carbon emissions achieved by utilising LNG cold energy for CO₂ liquefaction and capture. The results prove that the system should have good practical applications.

Keywords: LNG-fuelled vessel; LNG cold energy; CO₂ liquefaction; carbon emission reduction

INTRODUCTION

With the increasingly strict requirements of the International Maritime Organization (IMO) on pollutant emissions from vessels, many current vessels power systems are transitioning to cleaner fuels [1-3]. Among these, liquefied natural gas (LNG) has emerged as the preferred choice for retrofitting older vessels and fuelling new builds, owing to its advantages of being clean, pollution-free, and low price [4]. Compared to Marine Fuel Oil (MFO), LNG combustion produces no SOX emissions, reduces NOX emissions by up to

85%, and also lowers CO₂ emissions by 25% [5,6]. However, the combustion of LNG still generates CO₂, which cannot meet the strict requirements of the IMO for vessel exhaust emissions in the future.

To better achieve the goal of decarbonising the shipping industry and facilitate the green transformation of vessels, Carbon Capture and Storage (CCS) technology has become a critical transitional technology for vessels moving towards zero-carbon fuels [7,8]. The application of CCS technology on LNG-fuelled vessels aims to separate CO₂ from the vessel's exhaust and store it in liquid form in CO₂ storage tanks.

LNG fuel is stored at a temperature of $-163\text{ }^{\circ}\text{C}$ and needs to be vaporised and preheated to the required supply temperature for the vessel's main engine before being fed to the main engine for fuel. During the heating process, due to the phase change of LNG and the significant temperature difference between the LNG storage temperature and the supply temperature, substantial cold exergy is dissipated before LNG is supplied to the main engine [9-11]. If this cold energy is immediately released, it will not only result in significant energy waste but, also, cause damage to the surrounding ecological environment of the vessel. Therefore, a method for effectively harnessing the cold energy inherent in LNG is a critical issue that LNG-fuelled vessels need to address.

The utilisation of cold exergy from the LNG on board vessels has yet to become widespread, with primary applications including refrigeration for cold storage and air conditioning [12], low temperature power generation [13,14], and seawater desalination [15]. Yao et al. [16] proposed a highly efficient energy utilisation system by replacing the waste heat recovery system of LNG-fuelled vessels with a novel integrated transcritical CO_2 -ORC hybrid system: a synergistic recovery of marine engine waste heat and LNG cold energy was achieved. He et al. [17] developed a cascaded LNG cold energy recovery system through gradient-matching optimisation, demonstrating enhanced thermodynamic efficiency via phased energy extraction across low to medium temperatures and near-ambient thermal zones. Karlsson [18] proposed a practical solution for recovering LNG cold energy for HVAC (Heating, Ventilation, and Air Conditioning) systems, utilising the recovered LNG cold energy from the vessel's air conditioning system, thereby reducing the system's fuel consumption. Xiang et al. [19] employed a cascade utilisation approach in a semi-closed CO_2 cycle system to utilise LNG cold energy, reducing system energy consumption whilst substantially elevating the utilisation rate of LNG cold energy. Liu et al. [15] selected LNG as the cooling medium for condensing water vapour in humid air on LNG-fuelled vessels, comparing its efficiency with that of the traditional cooling medium systems, significantly improving the efficiency of seawater desalination. Mazzoni et al. [20] improved annual gas turbine efficiency by 2% through inlet air pre-cooling utilising LNG's cryogenic potential.

Currently, the cooling load of a vessel's cold storage, air conditioning system and seawater water purification processes is relatively small, only requiring a limited amount of cooling energy [21]. However, as LNG-fuelled vessels are increasingly trending toward larger sizes, the amount of LNG fuel required by these vessels is growing, resulting in a greater amount of cold energy being made available. Utilising LNG cold energy solely for vessel cold storage, air conditioning, and seawater water purification processes still results in a significant waste of cold energy. Meanwhile, the liquefaction of CO_2 in the carbon capture system of LNG-fuelled vessels requires pressurising and cooling the captured CO_2 , a process that demands the precise conditioning of CO_2 from cold energy to cryogenic temperatures, between -30 and $-50\text{ }^{\circ}\text{C}$ [22,23]. As vessels increase in size, their carbon emissions are also

increased. If the cold energy of LNG is utilised to liquefy the captured CO_2 , it can enhance the energy utilisation efficiency of vessels, reduce energy consumption, and offer substantial practical value. Therefore, the application of cold fuel energy to liquefy captured CO_2 on LNG-fuelled vessels is studied in this paper.

SELECTION OF PARENT VESSEL AND SIMULATION OF CARBON CAPTURE SYSTEM

SELECTION OF PARENT VESSEL AND CALCULATION OF RELATED PARAMETERS

The 250,000 DWT LNG-fuelled vessel "Shandong Harmony" was selected as the parent vessel, as shown in Fig. 1. The main engine power of this vessel can reach 18,600 kW. The principal parameters of the parent vessel are listed in Table 1.



Fig. 1. The 250,000 DWT "Shandong Harmony" vessel

Tab 1. Principal parameters of the parent vessel

Parameter	Numerical value	Parameter	Numerical value
Length, overall (m)	325.00	Designed draft (m)	18.00
Breadth, moulded (m)	57.00	Main engine power (kW)	18600
Depth (m)	25.00	Design speed (knot)	14.20

When a vessel is navigating at sea, the operating condition of its main engine is dynamically influenced by environmental factors such as wind, waves, and weather conditions [24]. When the main engine uses MFO, the consumption rate of MFO is approximately 180 g/kWh. Given that the calorific value of MFO is 42 MJ/kg and the calorific value of LNG is 52 MJ/kg, the required LNG supply for the main engine when using LNG can be calculated by comparing the calorific values of LNG and MFO [13]. Five typical vessel operating conditions were selected and the LNG supply rates of the main engine under these five conditions are shown in Table 2.

Tab. 2. Power and LNG supply rates under different vessel operating conditions

Operating condition (%)	85	75	70	65	60
Power (kW)	15810	13950	13020	12090	11160
MFO supply (kg/h)	2845.8	2511.0	2343.6	2176.2	2000.8
LNG supply rate (kg/h)	2298.5	2028.1	1892.9	1757.7	1622.5

According to the calculation results, when the vessel operating conditions are at 60%, the LNG consumption can reach 1.6 tons per hour. Referring to the relevant literature, it is known that consuming 1.6 tons of LNG produces 4.5 tons of CO₂ [25,26]. Therefore, the consumption of LNG fuel and CO₂ emissions during ship navigation are both considerable.

ESTABLISHMENT AND SIMULATION OF THE CO₂ CAPTURE SYSTEM

A CO₂ capture system model was established in Aspen HYSYS software. The simulation process is illustrated in Fig. 2.

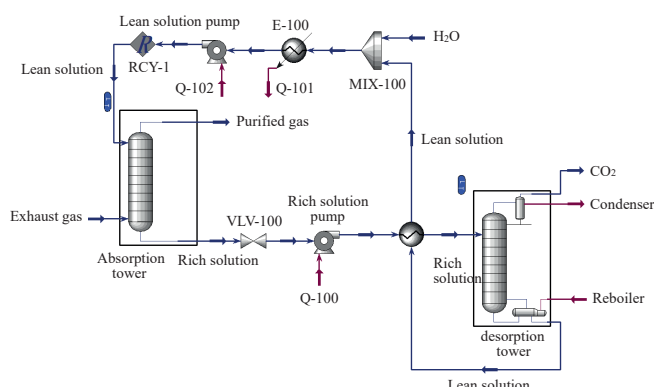


Fig. 2. Simulation diagram of the flow process of the CO₂ capture system

The established CO₂ capture system model was used to simulate 75% of the vessel's operating conditions, with the mass fraction of the selected absorbent MEA solution set at 30%. The other primary parameters of the system are configured in Table 3.

Tab. 3. The primary reaction parameters initially configured for the CO₂ capture system of the vessel

Simulation name	Parameter	Numerical value
Absorption tower	Exhaust gas temperature °C	50.0
	Exhaust gas pressure (bar)	1.0
	Poor liquid temperature °C	40.0
	Poor liquid pressure (bar)	2.0
Desorption tower	Inlet tower temperature °C	90.0
	Inlet tower pressure (bar)	1.2

DESIGN OF LNG COLD ENERGY UTILISATION FOR CO₂ LIQUEFACTION SYSTEM

LNG COLD ENERGY AND REFRIGERATION CYCLE-INTEGRATED CO₂ LIQUEFACTION SYSTEM DESIGN

Process flow description

To achieve the complete liquefaction and storage of the captured CO₂, the LNG cold energy and refrigeration cycle integrated CO₂ liquefaction system (Scheme 1) was proposed, with a CO₂ liquefaction and storage temperature set at -50 °C. The process flow diagram is illustrated in Fig. 3.

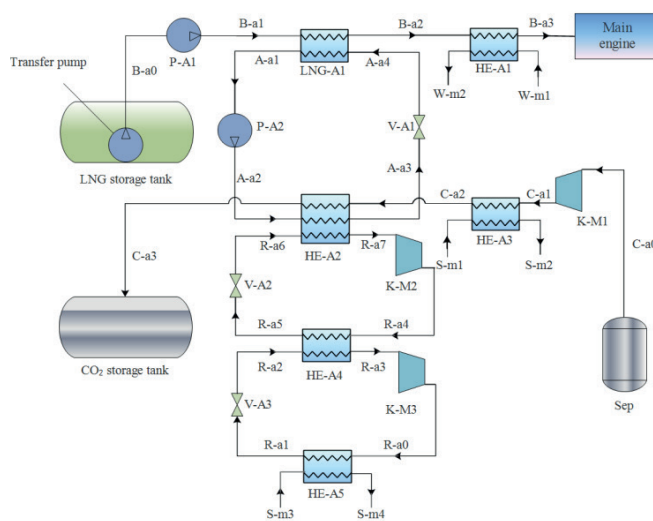


Fig. 3. Process flow diagram of the LNG cold energy and refrigeration cycle integrated CO₂ liquefaction system

As shown in Fig. 3, the fuel in the LNG storage tank is conveyed to the LNG heat exchanger (LNG-A1) through the transfer pump and pump P-A1, which exchanges heat with refrigerant 1 from the refrigerant circulation system and transfers its cold energy to refrigerant 1. The LNG, after heat exchange, then enters the heat exchanger HE-A1, where it is heated to the required temperature for main engine combustion using jacket water, subsequently supplied for combustion. In the seawater refrigeration cycle system, seawater enters the heat exchanger HE-A5 and exchanges heat with refrigerant 2 from the primary refrigeration cycle. Refrigerant 2, having absorbed the cold energy from the seawater, then enters the heat exchanger HE-A4 to exchange heat with refrigerant 3 from the secondary refrigeration cycle. Refrigerant 3 then enters the heat exchanger HE-A2, where it works together with refrigerant 1 from the refrigerant circulation system to cool the CO₂. The CO₂ captured by the vessel is compressed by the compressor K-M1 and sent to the heat exchanger HE-A3, where it is preliminarily cooled using seawater. It subsequently enters the heat exchanger HE-A2, where the cold energy from refrigerants 1 and 3 ensures the complete liquefaction of the CO₂.

System establishment and simulation

Scheme 1 achieved the complete liquefaction and storage of the captured CO₂ by leveraging the cold energy released from LNG and the refrigeration cycle. In order to validate the system's feasibility and ensure its stable and continuous operation, the system was analysed under the vessel's 75% operating conditions. Aspen HYSYS was employed to simulate the system; the simulation flow diagram is illustrated in Fig. 4.

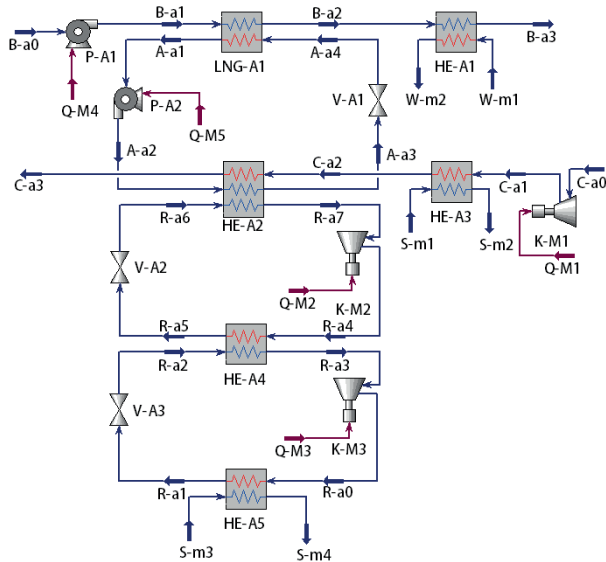


Fig. 4. Simulation diagram of the LNG cold energy and refrigeration cycle integrated CO₂ liquefaction system

In this simulation system, refrigerant 1 (in the refrigerant circulation system) was selected as R170, while refrigerants 2 and 3 in the seawater refrigeration cycle system were chosen as R717 and R23, respectively. The properties of the media in the system, including LNG, R170, R717, R23 and CO₂, were all based on the Peng-Robinson (P-R) equation of state. For the jacket water heating system and seawater heat exchange system, the NBS Steam equation of state was adopted. The pump and compressor efficiency was configured to 75% and the pressure drop in the heat exchangers was set to 20 kPa. The CO₂ liquefaction temperature was set to -50 °C, with an inlet gas temperature of 40 °C. The LNG storage conditions were -163 °C and 2 bar, while the LNG temperature for the vessel's main engine was maintained at 40 °C. At node A-a1, the temperature of R170 was set to -147 °C, with a flow rate of 4000 kg/h, ensuring the stable and continuous operation

Tab 4. Settings of main node parameters in the system

Node	Pressure (kPa)	Temperature °C	Flow rate (kg/h)	Node	Pressure (kPa)	Temperature °C	Flow rate (kg/h)
A-a1	-	-147	4000	C-a0	200	40	5165.52
A-a2	500	-	-	C-a1	699	-	-
A-a3	-	-55	-	C-a2	-	20	-
B-a0	200	-163	2030	C-a3	-	-50	-
B-a1	1600	-	-	W-m1	200	80	-
B-a3	-	40	-	W-m2	-	68	-
S-m1	200	15	-	S-m3	200	15	-

of the entire system. The parameter settings for the main nodes of the system are shown in Table 4.

Summary of simulation results

Based on the parameter settings for the vessel's 75% operating conditions in Table 4, we modelled and simulated the system in Aspen HYSYS. The established simulation system demonstrated a stable and continuous operation. The key parameters at the major nodes under steady-state conditions are summarised in Table 5.

Tab. 5. Summary of simulation results for key nodes in the system

Node	Pressure (kPa)	Temperature °C	Flow rate (kg/h)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·°C)
A-a1	440.0	-147.00	4000.00	-3610.0	3.005
A-a2	500.0	-147.00	4000.00	-3610.0	3.005
A-a3	480.0	-55.00	4000.00	-3397.0	4.260
A-a4	460.0	-55.02	4000.00	-3397.0	4.260
B-a0	200.0	-163.00	2030.000	-5400.0	4.609
B-a1	1600.0	-162.20	2030.00	-5396.0	4.623
B-a2	1580.0	-112.20	2030.00	-4976.0	7.457
B-a3	1560.0	40.00	2030.00	-4483.0	9.867
C-a0	200.0	40.00	5165.52	-8936.0	3.830
C-a1	699.0	163.00	5165.52	-8825.0	3.896
C-a2	679.0	20.00	5165.52	-8959.0	3.530
C-a3	659.0	-50.00	5165.52	-9366.0	1.746
R-a0	901.4	143.80	1381.00	813.7	8.787
R-a2	169.9	-22.28	1381.00	-691.0	3.909
R-a3	149.9	-25.00	1381.00	462.4	8.566
R-a4	1427.0	24.52	1381.00	-9406.0	2.284
R-a5	1407.0	-20.00	7412.00	-9621.0	1.448
R-a6	408.7	-53.83	7412.00	-9621.0	1.468
R-a7	388.7	-55.00	7412.00	-9452.0	2.245

On this basis, simulations were conducted for the system under other typical operating conditions of the vessel and the system was able to operate normally in all cases. This demonstrates that the proposed scheme in Fig. 3 has strong practical feasibility.

LNG COLD ENERGY AND SEAWATER-DIVERSION LIQUEFIED CO₂ SYSTEM DESIGN

Process flow description

Scheme 1 was proposed in Section 3.1, achieving complete liquefaction of the captured CO₂. This section presents the design of the LNG cold energy seawater diversion liquefaction CO₂ system (Scheme 2). This scheme adopts a split-flow parallel configuration to fully liquefy the captured CO₂. The liquefaction process flow is illustrated in Fig. 5.

As shown in Fig. 5, LNG fuel is transported from the LNG storage tank and heat is exchanged with the refrigerant in the heat exchanger LNG-B1,

transferring the cold energy of the LNG to the refrigerant. CO₂ is compressed by the compressor K-N1 and then delivered to the seawater heat exchanger HE-B3, where it undergoes preliminary cooling by exchanging heat with seawater. Subsequently, the CO₂ undergoes splitting into two parts for liquefaction. One part enters the heat exchanger HE-B2 in the refrigerant circulation system, where it exchanges heat with the refrigerant, and the cold energy carried by the refrigerant is transferred to the CO₂, causing it to liquefy. The other part enters a multi-stage compression refrigeration system, where it is first compressed by the compressor K-N2 and then sent to the heat exchanger HE-B4 for cooling by seawater. After, it is compressed again by the compressor K-N3 and sent to the heat exchanger HE-B5 for further cooling by seawater. The cooled CO₂ expands through the expansion valve to form a gas-liquid mixture, which enters the separation tank SEP. The liquid CO₂ (LCO₂) in the tank is directly combined with the CO₂ which was liquefied using LNG cold energy and sent to the storage tank for storage. The gaseous CO₂ is returned to the compressor K-N2, where it undergoes compression and seawater cooling again, until all of the gaseous CO₂ is liquefied and sent to the storage tank for storage.

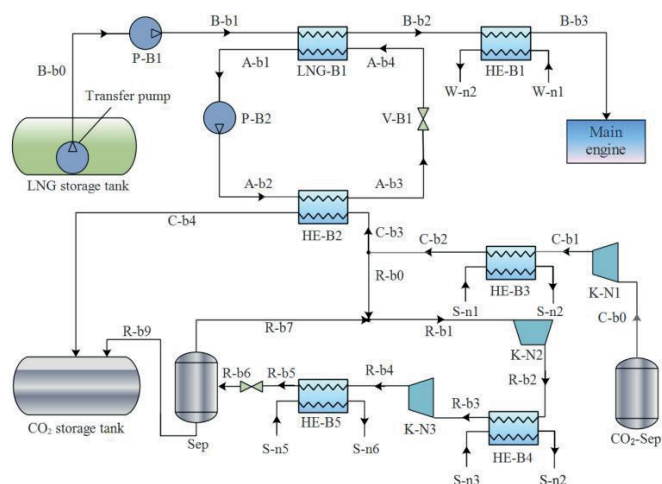


Fig. 5. Process flow diagram of the LNG cold energy and seawater diversion liquefied CO₂ system

System establishment and simulation

To validate the feasibility of the system, an analysis was conducted under the vessel's 75% operating conditions. Aspen HYSYS software was employed to simulate the system and the simulation flow diagram is illustrated in Fig. 6.

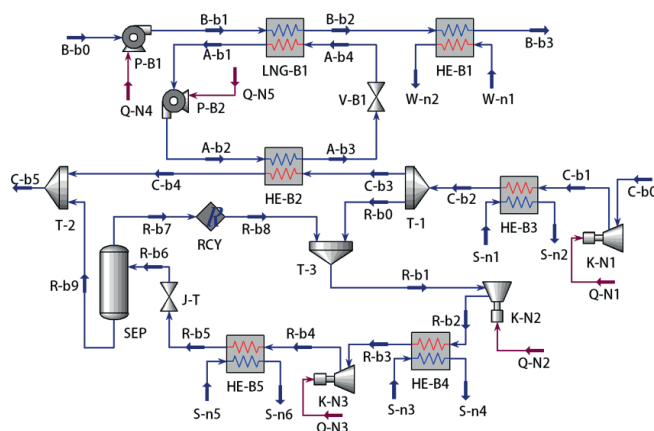


Fig. 6. Simulation diagram of the LNG cold energy and seawater diversion liquefied CO₂ system

In this simulation system, R170 was selected as the refrigerant for the refrigerant circulation system. In the multi-stage compression refrigeration system, the initial seawater inlet temperature of 15 °C and inter-stage cooling temperature of 20 °C were configured, the compression ratio being uniformly fixed at 3.5. The P-R equation of state was adopted for the properties of LNG and R170 in this system. The parameters of the pumps and compressors were set at the same values as those in Scheme 1. The parameter settings for the main nodes of the system are summarised in Table 6.

Summary of simulation results

Based on the parameter settings for the vessel's 75% operating conditions in Table 6, we modelled and simulated the system in Aspen HYSYS software. The established simulation system demonstrated stable and continuous operations, with the key parameters at major nodes under steady-state conditions, as summarised in Table 7.

Tab. 6. Settings of main node parameters in the system

Node	Pressure (kPa)	Temperature °C	Flow rate (kg/h)	Node	Pressure (kPa)	Temperature °C	Flow rate (kg/h)
A-b1	-	-147	4000.00	W-n1	200	80	-
A-b2	500	-	-	W-n2	-	68	-
B-b0	200	-163	2030.00	S-n1	200	15	-
B-b1	1600	-	-	S-n3	200	15	-
B-b3	-	40	-	S-n5	200	15	-
C-b0	200	40	5165.52	R-b2	-	-	2305
C-b1	699	-	-	R-b3	-	20	-
C-b2	-	20	-	R-b4	-	-	8068
C-b3	-	-	3050.00	R-b5	-	20	-
C-b4	-	-50	-	R-b6	-	-50	-

Tab. 7. Summary of simulation results for key nodes in the system

Node	Pressure (kPa)	Temperature °C	Flow rate (kg/h)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·°C)
A-b1	440	-147.00	4000.00	-3610	3.005
A-b2	500	-147.00	4000.00	-3610	3.005
A-b3	480	-53.88	4000.00	-3300	4.707
A-b4	460	-55.02	4000.00	-3300	4.709
B-b0	200	-163.00	2030.00	-5400	4.609
B-b1	1600	-162.20	2030.00	-5396	4.623
B-b2	1580	-89.62	2030.00	-4783	8.617
B-b3	1560	40.00	2030.00	-4483	9.867
C-b0	200	40.00	5165.52	-8936	3.830
C-b1	699	163.00	5165.52	-8825	3.896
C-b2	679	20.00	5165.52	-8959	3.530
C-b3	679	20.00	3050.00	-8959	3.530
C-b4	659	-50.00	3050.00	-9366	1.746
R-b0	679	20.00	2116.00	-8959	3.530
R-b1	658.6	-15.68	3996.93	-8991	3.419
R-b2	2305	96.56	3996.93	-8899	3.491
R-b3	2285	20.00	3996.93	-8977	3.258
R-b4	8068	143.00	3996.93	-8885	3.314
R-b6	658.6	-50.00	3996.93	-9204	2.473
R-b7	658.6	-50.00	1881.00	-9021	3.292
R-b9	658.6	-50.00	2115.00	-9366	1.746

Building on this, simulations were conducted for the system under other typical vessel operating conditions and the system demonstrated normal operations in all cases. This confirms that the proposed scheme in Fig. 5 possesses strong practical feasibility.

COMPARATIVE ANALYSIS AND OPTIMISATION OF SYSTEM SCHEMES

THERMODYNAMIC MODEL OF THE SYSTEM

Calculation model for energy consumption of major system equipment

The power equipment in this system primarily comprises heat exchangers, pumps, and CO₂ compressors, among others. The mathematical model for the gross power draw of the power equipment is described below.

Heat exchangers are mainly used for heat exchange between different working fluids. The total heat released (or absorbed) during the process, Q_{HX} , is expressed as:

$$Q_{HX} = m_{hot} \times (h_{in,hot} - h_{out,hot}) = m_{cold} \times (h_{cold,out} - h_{cold,in}) \quad (1)$$

where, m_{hot} : The mass flow rate of the hot source working fluid, kg/s;

m_{cold} : The mass flow rate of the cold source working fluid, kg/s;

h : Specific enthalpy, kJ/kg, the subscripts “in” and “out” denote the inlet and outlet working fluids, respectively.

In the system, the pump functions to pressurise the liquid working fluid to a high-pressure state and its input power W_p is given by:

$$W_p = m(h_{p,out} - h_{p,in}) \quad (2)$$

$$h_{p,out} = h_{p,in} + \frac{h_{p,out} - h_{p,in}}{\eta_p} \quad (3)$$

where, m : The mass flow rate of the working fluid in the pump, kg/s;

$h_{p,in}$: The specific enthalpy of the working fluid entering the pump, kJ/kg;

$h_{p,out}$: The specific enthalpy of the working fluid leaving the pump, kJ/kg;

η_p : The efficiency of the pump, %.

The compression work W_{comp} of the CO₂ compressor during the compression process is given by:

$$W_{comp} = \frac{m_{CO_2}(h_{out} - h_{in})}{\eta_{comp}} \quad (4)$$

where, m_{CO_2} : The mass flow rate of CO₂, kg/h;

h : Specific enthalpy, the subscripts “out” and “in” indicate the state of the working fluid at the outlet and inlet of the equipment, respectively;

η_{comp} : The efficiency of the compressor (taken as 75%).

Calculation model for system cold exergy

The exergy value E_x of the system's material flow is:

$$E_x = (H - H_0) - T_0(S - S_0) \quad (5)$$

where, H : The enthalpy of the system, kJ/kg;

H_0 : The enthalpy of the system under environmental conditions, kJ/kg;

T_0 : Ambient temperature, °C;

S : The entropy of the system, kJ/kg·°C;

S_0 : The entropy of the system under environmental conditions, kJ/kg·°C.

The ratio of the output exergy E_{gain} to the input exergy E_{input} is referred to as the exergy efficiency η_e . The exergy efficiencies of the compressor and heat exchange can be calculated as follows:

$$\eta_{e,comp} = \frac{E_{comp,gain}}{E_{comp,input}} \quad (6)$$

$$\eta_{e,HX} = \frac{E_{HX,gain}}{E_{HX,input}} \quad (7)$$

In this system, the exergy destructions of the compressor $e_{s,comp}$ and the heat exchanger $e_{s,HEX}$ are as follows:

$$e_{s,comp} = e_{comp,1} + w_{comp} - e_{comp,2} \quad (8)$$

$$e_{s,HEX} = e_{HEX,out} - e_{HEX,in} \quad (9)$$

Calculation Model for Cold Energy Utilisation Efficiency of the System

The cold energy utilisation efficiency is the ratio of effectively utilised cold energy to the total input of cold energy within the system. In this system, the utilisation rate of LNG cold energy, denoted as η_{cold} , is defined as follows:

$$\eta_{cold} = \frac{E_{out}}{E_{in}} \quad (10)$$

SYSTEM OPTIMISATION ANALYSIS AND ENERGY CONSUMPTION CALCULATION FOR SCHEME 1

Exergy efficiency analysis of the system

By implementing the parameters defined in Scheme 1 and incorporating Eqs. (1) through (10), under various vessel operating conditions, the exergy input, exergy destruction, and exergy efficiency of each system component are calculated. The results are presented in Table 8.

It can be seen from Table 8 that, as the vessel's operating conditions increase, the enhanced heat transfer between the refrigerant and LNG in heat exchanger LNG-A1 leads to greater exergy destruction, consequently resulting in reduced

exergy efficiency. In the case of heat exchanger HE-A2, as the vessel's operating conditions gradually increase, the exergy destruction also increases, while its exergy efficiency continues to improve. For other key pieces of equipment, the exergy efficiency remains constant across the five typical vessel operating conditions.

The primary objective of establishing this system is to achieve full utilisation of LNG's cold exergy. In this study, LNG fuel was stored in the tank in a low-pressure form at -163 °C and the liquefaction temperature of CO₂ was set at -50 °C. Based on these settings, in order to enhance the exergy efficiency of heat exchanger LNG-A1 (whilst reducing that of HE-A2 in an appropriate range), the exergy efficiency of node A-a1 is reasonably optimised for the system and key components in the scheme. By adjusting the thermodynamic parameters of key nodes, the simulation results show that the temperature range at the selected node A-a1 is -147 to -57 °C and the flow rate range is 4,000-15,000 kg/h. Using the Case Study function in Aspen HYSYS, a temperature step size of 10 and a flow rate step size of 1,000 were applied. Incorporating relevant exergy efficiency calculation equations, a typical vessel's 75% operating condition was selected, to analyse the exergy efficiency of the entire system and its key equipment. The simulation results are illustrated in Fig. 7.

From Fig. 7 (a), it can be observed that, as the temperature at node A-a1 (the hot stream outlet) increases, the exergy efficiency of heat exchanger LNG-A1 decreases. When the temperature of node A-a1 remains constant, the exergy efficiency of the heat exchanger LNG-A1 increases with the flow rate at node A-a1. However, with the continuous increase of the node A-a1 flow rate, the heat exchanger LNG-A1 fails to operate properly at lower corresponding node temperatures, making the exergy efficiency incalculable. At higher temperatures of node A-a1, the exergy efficiency of heat exchanger LNG-A1 increases with higher flow rates.

Tab. 8. Summary of exergy-related parameters for key equipment under different typical vessel operating conditions

Equipment name		LNG-A1	HE-A2	HE-A4	HE-A5	P-A2	K-M2	K-M3
Exergy input (kW)	60%	231.58	264.66	59.91	0.15	10.28	62.51	89.50
	65%	234.70	279.04	70.00	0.15	12.01	73.04	104.58
	70%	237.51	293.42	80.10	0.15	13.74	83.57	119.66
	75%	240.00	307.81	90.19	0.15	15.47	94.10	134.74
	85%	245.00	336.57	110.38	0.15	18.94	115.16	164.90
Exergy destruction (kW)	60%	52.26	121.82	12.88	0.11	2.66	16.03	16.76
	65%	55.38	124.29	15.05	0.11	3.11	18.73	19.59
	70%	58.19	126.78	17.23	0.11	3.55	21.43	22.41
	75%	60.69	129.25	19.40	0.11	4.00	24.13	25.24
	85%	65.68	134.21	23.74	0.11	4.90	29.53	30.89
Exergy efficiency (%)	60%	77.43	53.97	78.49	26.91	74.13	74.36	81.27
	65%	76.40	55.46	78.49	26.91	74.13	74.36	81.27
	70%	75.50	56.79	78.49	26.91	74.13	74.36	81.27
	75%	74.71	58.01	78.49	26.91	74.13	74.36	81.27
	85%	73.19	60.12	78.49	26.91	74.13	74.36	81.27

Specifically, at a node A-a1 temperature of -147°C and a flow rate of 5000 kg/h, the exergy efficiency of heat exchanger LNG-A1 reaches its maximum at 77.40%. Conversely, at node A-a1, the temperature is -57°C and the flow rate is 5000 kg/h; the exergy efficiency of heat exchanger LNG-A1 reaches a minimum of 22.02%.

Fig. 7(b) reveals that, as the temperature at node A-a1 increases, the exergy efficiency of pump P-A2 also increases. From Fig. 7(c), with progressive temperature elevation at node A-a1, the exergy efficiency of heat exchanger HE-A2 increases, reaching its maximum at a node A-a1 temperature of -57°C . Under normal system operations, at the same node A-a1 temperature, the exergy efficiency of heat exchanger HE-A2 increases with higher flow rates. However, when the node temperature reaches -57°C , the change in exergy efficiency becomes less pronounced and the maximum exergy efficiency of HE-A2 reaches 82.73%.

From Fig. 7(d), it can be seen that elevated temperatures at node A-a1 correspond to decreases in the exergy efficiency of the entire system. The system attains peak exergy efficiency (58.95%) under optimised operating conditions of a nodal temperature of -147°C and a mass flow rate of 5000 kg/h. At a node temperature of -57°C , the exergy efficiency of the system reaches its minimum at 54.50%.

Analysis of LNG cold energy utilisation rate

Based on the system's simulation results and using Eq. (10), the utilisation rate of LNG cold energy under five typical vessel operating conditions can be calculated. The calculation results are presented in Fig. 8.

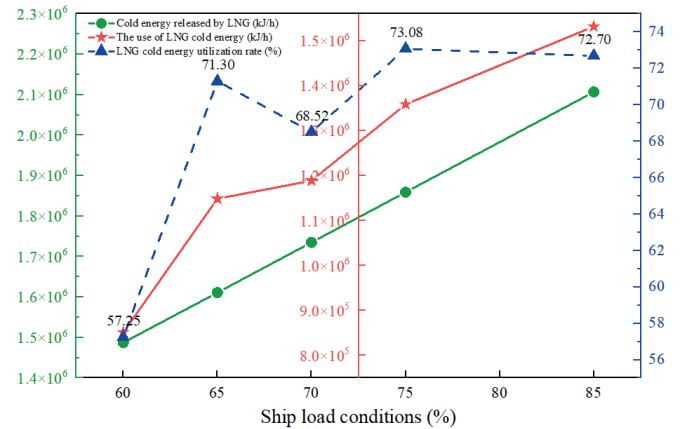


Fig. 8. LNG cold energy utilisation rate under five typical vessel operating conditions

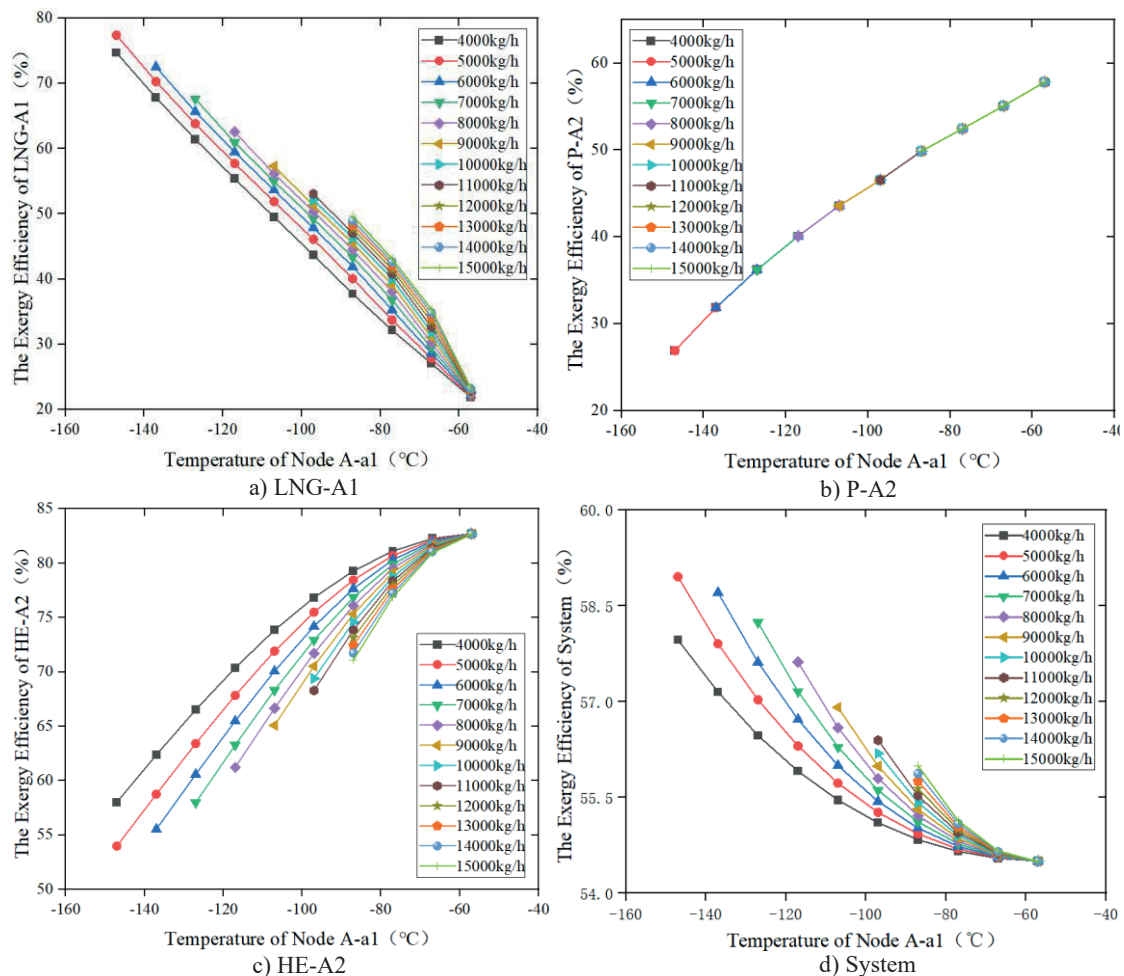


Fig. 7. Impact of different temperatures and corresponding flow rates at node A-a1 on the exergy efficiency of the entire system and key equipment at 75% vessel operating conditions

From Fig. 8, it can be observed that, as the vessel's operating load increases, the amount of cold energy released by LNG gradually rises, and the system's utilisation of LNG cold energy also increases. When the vessel's operating condition reaches 85%, the system utilises the maximum amount of LNG cold energy (1,531,683 kJ/h). When the vessel operating conditions are at 75%, the LNG cold energy utilisation rate reaches its peak at 73.08%, representing the highest efficiency of LNG cold energy utilisation in the system.

Calculation of optimised system energy consumption

In CCS technology, the CO₂ liquefaction process has a significant amount of energy. Therefore, it is essential to reduce the energy consumption of the CO₂ liquefaction process, decreasing LNG consumption and achieving cost savings and efficiency improvements. For the LNG cold energy and refrigeration cycle integrated CO₂ liquefaction system, the primary energy consumption is attributed to the compressors. The energy consumption of the compressors in the optimised system scheme was calculated and the results are summarised in Table 9.

Tab. 9. Compressor energy consumption of system

Vessel operating conditions	60%	65%	70%	75%	85%
Total energy consumption (kW)	280.31	297.28	333.59	350.35	422.95

ANALYSIS OF SYSTEM OPTIMISATION AND ENERGY CONSUMPTION CALCULATION FOR SCHEME 2

Analysis of LNG cold energy utilisation rate

The simulation results of the Scheme 2 system developed in Aspen HYSYS, combined with Eq. (10), enable the cold exergy utilisation efficiency to be calculated. The calculated results are presented in Fig. 9.

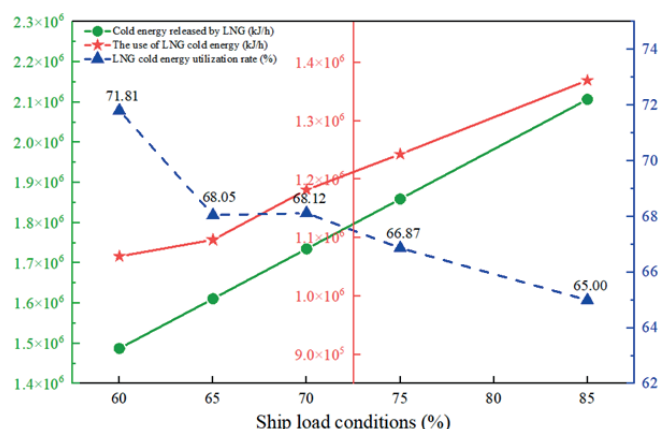


Fig. 9. LNG cold energy utilisation rate under five typical vessel operating conditions

Fig. 9 shows that, as the vessel's operating conditions increase, the cold energy released by LNG progressively increases, and the system's efficiency in using LNG cold energy also shows an increasing trend. When the vessel operating conditions reach 85%, the system utilises the maximum amount of LNG cold energy, amounting to 1,369,488 kJ/h. At a vessel operating condition of 60%, the LNG cold energy utilisation rate is 71.81%, representing the highest cold energy utilisation rate of the system under these conditions.

Impact of inter-stage cooling temperature on system exergy efficiency and energy consumption

When the inter-stage compression ratio remains constant, the inter-stage cooling temperature directly impacts the energy consumption of the compressor, while the seawater temperature directly influences the inter-stage cooling temperature. The exergy efficiency and energy consumption of the system are discussed at inter-stage cooling temperatures of 10, 15, 20, 25, 30 and 35 °C. The results are shown in Fig. 10 and Fig. 11.

Fig. 10 clearly indicates the variation patterns of the exergy efficiency of key equipment in the system, remaining consistent across five typical vessel operating conditions. As the inter-stage cooling temperature continues to rise, the exergy efficiency of the compressor continues at around 80%, while the exergy efficiency of the heat exchanger generally shows a declining trend.

As illustrated in Fig. 11, at five typical vessel operating conditions, the energy consumption of compressor K-N1 remains constant as the inter-stage cooling temperature increases, while the energy consumption of compressors K-N2 and K-N3 rise with the elevation of the inter-stage cooling temperature. With the progressive increase in vessel operating conditions, the total energy consumption of each compressor and the system as a whole also escalates. Across the five vessel operating conditions, the system's total energy consumption experiences a significant increase when the inter-stage cooling temperature rises from 30 °C to 35 °C. In summary, the system's energy consumption reaches its minimum at 251.69 kW, when the vessel operates at 60% load and the inter-stage cooling temperature is 10 °C. Conversely, the energy consumption peaks at 730.50 kW under 85% vessel loading at an inter-stage cooling temperature of 35 °C. This demonstrates that the system's energy consumption is notably influenced by the inter-stage cooling temperature.

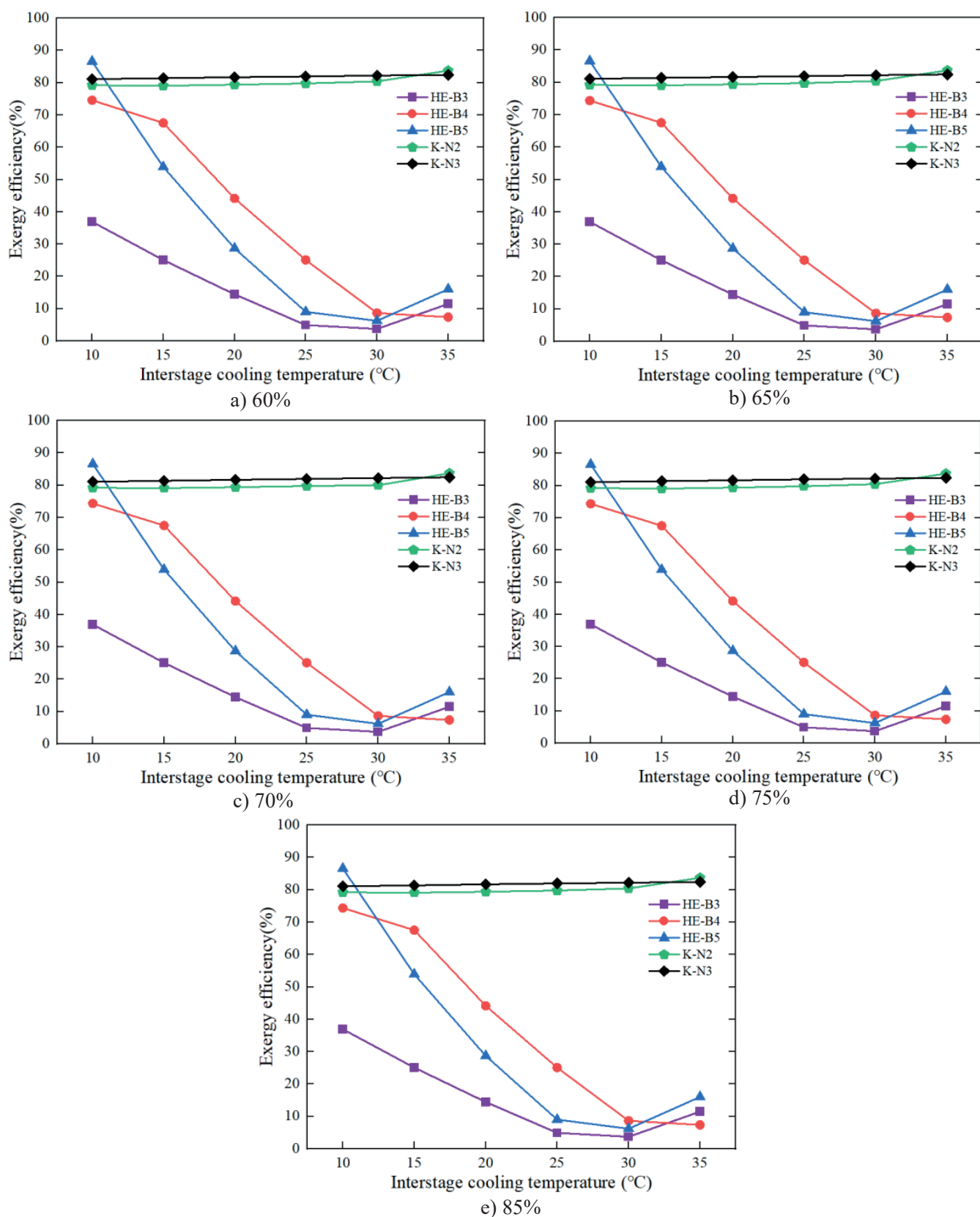


Fig. 10. Impact of different inter-stage cooling temperatures on the exergy efficiency of key system equipment at five typical vessel operating conditions

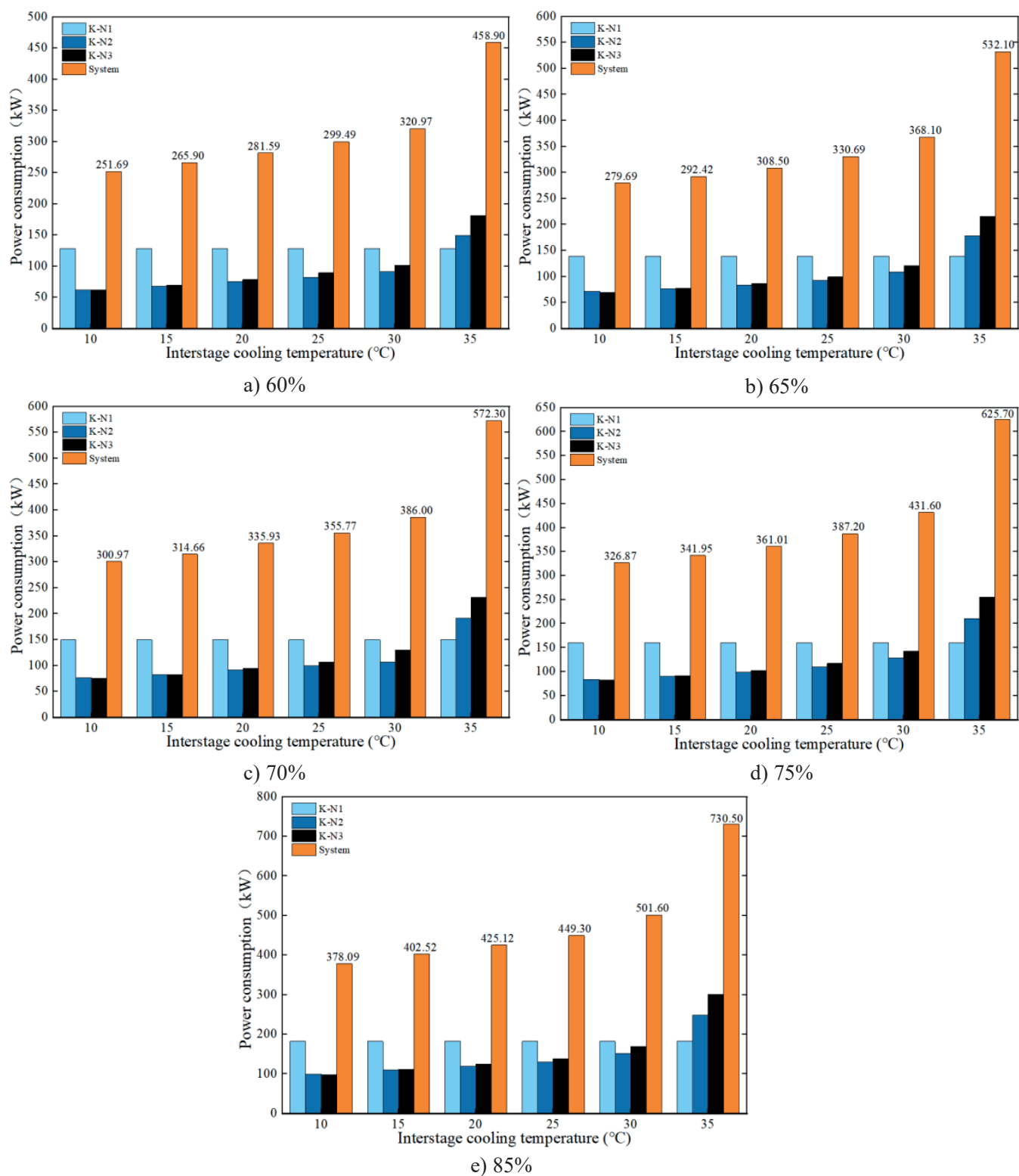


Fig. 11. Impact of different inter-stage cooling temperatures on system energy consumption at five typical vessel operating conditions

COMPARATIVE ANALYSIS OF ENERGY CONSUMPTION FOR TWO SYSTEM CONFIGURATIONS

The liquefaction and storage of CO₂ represents a critical component of vessel-based CCS technology, making the

reduction of energy consumption during the liquefaction process particularly important. A comparative analysis of the energy consumption between the two system schemes was conducted and the resulting data are graphically depicted in Fig. 12.

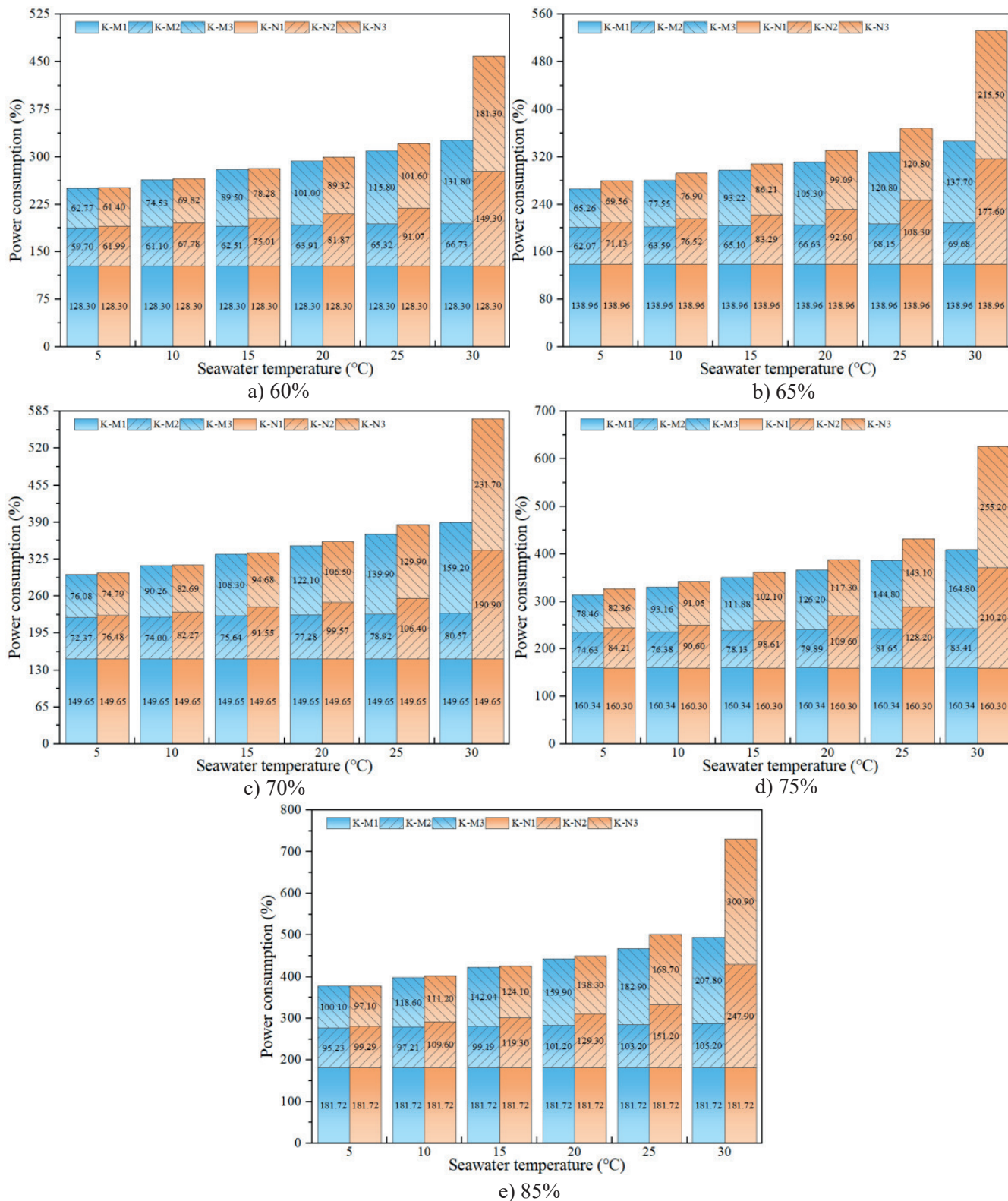


Fig. 12. Comparison of energy consumption between the two systems under typical vessel operating conditions

As illustrated in Fig. 12, at five typical vessel operating conditions, the energy consumption of both schemes increases with the rise in seawater temperature. Between 5 and 25 °C, the energy consumption of the two systems is relatively similar across all five vessel operating conditions. However, at a seawater temperature of 30 °C, the energy consumption of Scheme 2 is significantly higher compared to Scheme 1. This is because, when the seawater temperature rises to 30 °C, the CO₂ cooled by seawater in the three-stage compression of Scheme 2 remains gaseous. As a result, a larger proportion of CO₂ in the system fails to liquefy after expansion, requiring additional energy to ‘reliquefy’ this portion of CO₂. Consequently, the compressor’s power consumption also increases. In summary, both Scheme 1 and Scheme 2 exhibit relatively low overall energy consumption, enabling efficient utilisation of LNG cold energy and complete CO₂ liquefaction. Scheme 1 demonstrates comparatively lower energy consumption demand with minimal fluctuations under varying seawater temperatures, rendering it more viable for practical implementation.

ECONOMIC ANALYSIS

CALCULATION OF POWER SAVINGS

As discussed in Section 4.4, the energy consumption of Scheme 1 remains relatively stable, regardless of variations in seawater temperature, making it more economically advantageous than Scheme 2. To better demonstrate the system’s real-world applicability, its economic feasibility was assessed using the shipping route shown in Fig. 13.

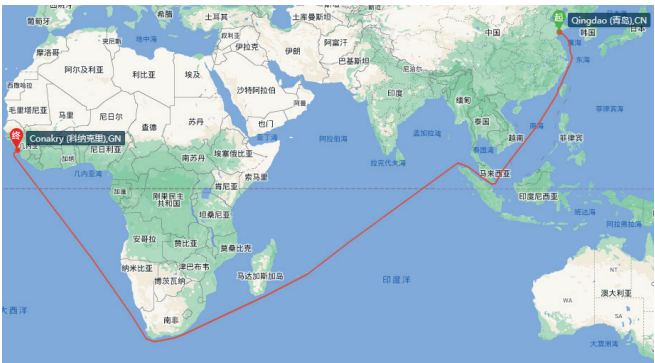


Fig. 13. Actual shipping route of the parent vessel

Fig. 13 illustrates the actual shipping route of the parent vessel transporting bauxite from Qingdao Port, China, to Conakry Port, Guinea, Africa. Given the extensive length of the route and the significant latitudinal range covered during the voyage, the seawater temperature undergoes continuous variations. To more accurately reflect practical

conditions, ten waypoints were selected along the route, designated ‘A’ to ‘J’. Based on the on-site investigation and the relevant parameters of the parent vessel, the maritime distances and corresponding sailing times between adjacent nodes were determined. For more details, please refer to Table 10.

Tab. 10. Navigation distance and travel time between different route segments

Route segment	A-B	B-C	C-D	D-E	E-F	F-G	G-H	H-I	I-J
Distance (nm)	838.9	775.1	896.4	2841.7	1531.9	1406.9	646.1	1173.2	1445.2
Time (h)	50.5	45.3	55.3	230.7	116.8	105.7	40.4	86.2	109.1

As shown in Table 10, the sailing time of the parent vessel from point A to point J was 35 days. Since the external temperature varied dynamically along the same route segment when the vessel departed from point A in different seasons, the power savings achieved by the system over the entire route also differed. Based on this, the vessel was assumed to depart from point A in different months, with the starting times sequentially denoted as T1 to T12. Taking the vessel’s 75% operating condition as an example, a comparative analysis of energy consumption between the Scheme 1 and conventional refrigeration cycles was conducted for ships departing from Point A across different months, the results obtained for reduced energy consumption are shown in Fig. 14.

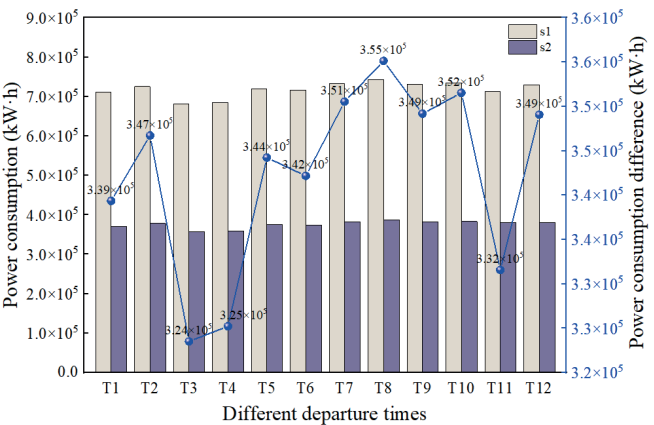


Fig. 14. Comparison of energy consumption for the CO₂ liquefaction process along the entire route when departing from Qingdao port in different months

As illustrated in Fig. 14, utilising LNG cold energy for CO₂ liquefaction significantly reduces the system’s energy consumption. The maximum energy savings for the vessel across different months reached 3.55x10⁵ kW·h. Furthermore, by calculating the power reduction under four other typical operating conditions, it can be concluded that the larger the vessel operating conditions, the greater the electricity savings achieved.

CALCULATION OF CARBON EMISSION REDUCTION

The amount of LNG fuel saved by the generator in different months can be calculated according to the electricity saved by the parent vessel applied in Scheme 1; the results are presented in Fig. 15.

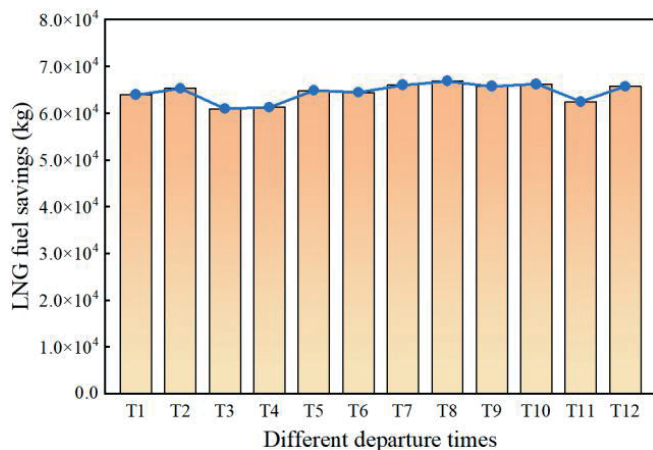


Fig. 15. The amount of LNG fuel saved by a vessel departing in different months

The carbon emission reduction mentioned in this study refers to the amount of CO₂ saved by the LNG cold energy and refrigeration cycle integrated CO₂ liquefaction system. Based on the amount of LNG saved by the parent vessel, the corresponding reduction in CO₂ emissions can be further calculated. The results are shown in Fig. 16.

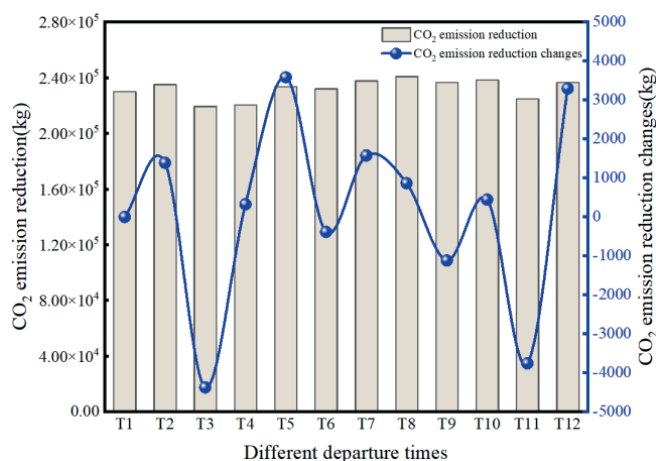


Fig. 16. CO₂ emission reduction of the parent vessel along the entire route

As shown in Fig. 16, the maximum reduction in CO₂ emissions achieved by the parent vessel along the entire route, through the utilisation of LNG cold energy, is 2.41×10^5 kg. The calculation of CO₂ emission reductions across the four remaining operating conditions demonstrates that greater vessel operating conditions yield proportionally higher CO₂ reduction quantities. This demonstrates that CO₂ liquefaction

using LNG cold energy reduces ship-based carbon emissions, aligning with the goals of global shipping energy conservation and emission reduction development trends for the future, and highlighting its significant practical application value.

CONCLUSIONS

Aiming at the problems of high energy consumption and wasted cold energy in the CO₂ liquefaction of LNG-fuelled vessels equipped with CCS systems, two system schemes for CO₂ liquefaction using the cold energy of LNG are proposed. With relevant thermodynamic parameters as optimisation objectives, the systems were simulated and analysed in HYSYS. Additionally, an economic analysis was conducted for the selected vessel route. The conclusions are as follows:

(1) A typical LNG-fuelled parent vessel was selected and an on-board CO₂ Capture System was designed and simulated. On this basis, the LNG cold energy and refrigeration cycle-integrated CO₂ liquefaction system (Scheme 1) was proposed. A simulated investigation was conducted on the system's exergy efficiency and energy consumption in Aspen HYSYS. By adjusting the temperature and flow rate at node A-a1, the system achieved the highest exergy efficiency of 58.98% at the 60% condition. Additionally, the LNG cold energy utilisation rate reached a maximum of 73.08% at the 75% condition, while the system energy consumption was minimised to 250.77 kW at the 60% condition, with a seawater temperature of 5 °C. The findings confirm the system's capability to efficiently utilise LNG cold energy and achieve complete CO₂ liquefaction, highlighting its significant practical application value.

(2) The LNG cold energy and seawater diversion liquefied CO₂ system (Scheme 2) was proposed. The feasibility of the system under different operating conditions was simulated and validated in Aspen HYSYS, and the influence of inter-stage cooling temperature on the system's exergy efficiency and energy consumption was analysed. The results indicate that the LNG cold energy utilisation rate of the system reached a maximum of 71.81% at the 60% condition, demonstrating significant energy-saving effects. The total energy consumption of the system increased significantly with higher inter-stage cooling temperatures, the increase in energy consumption being very significant when the sea temperature reaches 30 °C. Compared to Scheme 1, this system's energy consumption is more influenced by seawater temperature.

(3) By selecting a typical route, the economic performance and carbon emission reduction effect of the combined LNG cold energy and refrigeration cycle liquefied CO₂ system were analysed. The results show that system operation yields an electrical energy reduction of 3.73×10^5 kW·h and reduces CO₂ emissions by 2.53×10^5 kg under different operating conditions. This significantly improves the utilisation efficiency of energy, aligns with the development trend of energy conservation and emission reductions in vessels, and demonstrates strong practical application value.

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