

Article

Analysis of a Boil-Off Gas Recovery System for Liquid Hydrogen Vessel Shipping with Cryogenic Compressed Hydrogen Storage

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Abstract

During the marine transport of liquid hydrogen, heat ingress leads to the generation of boil-off gas (BOG), which increases the pressure in the liquid hydrogen storage tanks. Effective BOG management is therefore essential to ensure tank safety and minimize hydrogen loss. This study develops a cryogenic compression recovery and storage system for BOG generated during the marine transport of 160,000 m³ liquid hydrogen. The core process involves compressing a portion of the BOG and subsequently utilizing the BOG's inherent cold energy to cool the compressed hydrogen, ultimately enabling the storage of the final cryogenic compressed hydrogen product. ASPEN-PLUS software was employed to analyze the proposed system's specific energy consumption (*SEC*) and ψ (hydrogen density/*SEC*) for producing cryogenic compressed hydrogen (CCH₂) across a temperature range of 53 to 110 K and a pressure range of 40 to 100 MPa. Seven optimal sets of state parameters were identified for the cryogenic compressed hydrogen product. Based on a specified optimal parameter set of 80 K and 50 MPa, a simulation of the proposed system's performance yielded a *SEC* of 2.25 kWh/kg CCH₂ and an exergy efficiency of 87.88% with BOG feed at 53 K and 0.1 MPa, along with the exergy loss and exergy efficiency for each component. Compared to a BOG re-liquefaction system and a MRJT CCH₂ system under identical conditions, the proposed system achieves 31.81% and 64.9% reduction, respectively, in *SEC* and 17.32% and 94.6% improvement, respectively, in exergy efficiency. Furthermore, the effects of feed temperature and cryogenic compressed hydrogen product mass flow rate on the proposed system's *SEC* and exergy efficiency were investigated.

Keywords: hydrogen; boil-off gas; CCH₂; exergy efficiency; specific energy consumption



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1. Introduction

The long-term consumption of fossil fuels has led to severe environmental issues and energy crises, drawing significant global attention [1]. Countries worldwide now face the formidable challenge of achieving large-scale decarbonization. Consequently, the development of new energy sources and transition from traditional to new energy systems are crucial for the establishment of a low-carbon and efficient modern society. Hydrogen energy is widely regarded as one of the most promising high-quality renewable and clean energy carriers, often termed the “ideal energy source of the 21st century” [2]. This recognition stems from its carbon-free and pollution-free utilization, i.e., producing only water as a byproduct. In addition, hydrogen, with a high gravimetric energy density of 142 kJ/g [3], can be produced from diverse and abundant sources and has broad applica-

bility across multiple critical sectors such as transportation, industry, energy storage, and power generation [4].

Hydrogen storage represents a critical link in the hydrogen energy application chain. The inherently small size of hydrogen molecules, which facilitates permeation leakage and can induce hydrogen embrittlement in materials, poses significant technical challenges for achieving safe, economical, and sustainable storage [5]. These challenges are further compounded by hydrogen's wide flammability range (4% to 75%) and very low ignition energy (0.02 mJ). Currently, large-scale commercial hydrogen storage methods primarily include high-pressure gaseous hydrogen storage, solid-state hydrogen storage, underground hydrogen storage, and liquid organic hydrogen carriers. Compressed gaseous hydrogen at 70 MPa achieves a density of approximately 39.3 kg/m³ [6] at ambient temperature, with a compression energy consumption of 4.4 kWh per kilogram [7]. Liquid hydrogen offers a higher density of 70.8 kg/m³ at 20 K, but incurs a high liquefaction energy penalty of 10–15 kWh per kilogram [8]. Solid-state storage using metal hydrides achieves high volumetric density but suffers from low gravimetric capacity (magnesium hydride offers about 7.6 wt%) and requires high release temperatures of 350–400 °C [9]. Liquid organic hydrogen carriers (LOHCs) store hydrogen through chemical bonding, achieving gravimetric capacities above 5.8 wt% [10], but their round-trip efficiency is not high due to energy-intensive hydrogenation and dehydrogenation processes. Underground storage in salt caverns or depleted reservoirs is the most cost-effective option for large-scale storage. For salt caverns, the levelized storage cost is approximately \$0.38–0.92 per kilogram [11]. Depleted and lined rock caverns cost slightly more than salt caverns, but all are geographically limited.

Choosing the right hydrogen storage method improves energy efficiency and reduces costs. A well-matched storage and transport scheme offers greater cost advantages. Hydrogen transport includes on-land tube trailers, liquid hydrogen tank trucks, pipelines, and maritime shipping. For large-scale interregional transport, maritime shipping is better [12]. Maritime transport faces strict constraints: limited vessel space demands high storage density; energy use during transport must be low; and the system must fit large-scale logistics [13]. Among storage methods, compressed hydrogen has low density and bulky tanks, reducing efficiency. Solid-state and LOHC methods suffer weight penalties or energy losses during loading/unloading. In contrast, liquid hydrogen is most suitable for maritime transport. Its density of 70.8 kg/m³ is the highest among physical methods, allowing more cargo per voyage [8]. Studies show that for large-scale, long-distance transport, liquid hydrogen offers significantly better cost-effectiveness than other methods [14]. However, even minimal heat ingress can cause liquid hydrogen to evaporate, which generates lots of boil-off gas (BOG) to raise the internal pressure of the storage vessel. This phenomenon is further intensified during maritime transport owing to the unavoidable sloshing of the liquid hydrogen within tanks [15,16]. Large-scale storage tanks typically for maritime transport of liquid hydrogen have relatively low maximum allowable working pressures. To maintain pressure within safety limits, the BOG must be vented, resulting in a loss of both valuable cold energy and the hydrogen itself [17,18]. For example, the WENET project [19] reported the hydrogen boil-off rate was around 0.2–0.4% per day for two 200,000 m³ tanks, leading to a hydrogen BOG venting rate of 28,340 kg to 56,680 kg per day. Obviously, it is worth recovering hydrogen BOG from the perspectives of economic cost and resource conservation. The primary recovery methods for hydrogen BOG include compression recovery, re-liquefaction, and direct use as a fuel. The three main BOG recovery methods all have inherent drawbacks. These drawbacks become especially problematic in the context of maritime liquid hydrogen transport. Compression recovery is technically simple. However, it fails to recover the cryogenic exergy in BOG. This leads to a significant waste of cold energy. In addition, compressed hydrogen at 70 MPa has a density of only 39.3 kg/m³ [20],

which is just 55% of liquid hydrogen's. As a result, it requires more storage space—a critical issue on space-limited vessels. Re-liquefaction, on the other hand, preserves hydrogen mass, but it consumes a large amount of energy. Even for optimized re-liquefaction processes, the specific power consumption ranges from 3.12 to 3.39 kWh per kilogram of liquid hydrogen [21], while conventional re-liquefaction processes would have higher energy consumption. On long-haul voyages, this puts a heavy burden on the ship's power system. It also reduces the environmental benefits of hydrogen as a clean fuel. Direct use of BOG as fuel is often a last resort. It helps relieve pressure buildup, but it consumes the very cargo meant for delivery. For a vessel designed to carry hydrogen, using the payload as fuel makes little sense—both economically and operationally. These limitations highlight the need for a better BOG management strategy. Such a strategy must balance energy efficiency, cargo preservation, and the practical constraints of maritime operations.

Nowadays, an emerging storage technology, cryogenic compressed hydrogen (CcH₂) storage [22], has gradually been paid attention to. Cryogenic compressed hydrogen storage offers distinct advantages, especially including its lack of need to cool hydrogen to 20 K and almost total avoidance of ortho-para conversion in the absence of a catalyst. This results in significantly lower energy consumption compared to hydrogen liquefaction [23], while achieving a storage density that can exceed that of liquid hydrogen. There are two primary methods for producing cryogenic high-pressure hydrogen. One is liquid hydrogen vaporization and pressurization. Researchers at Lawrence Livermore National Laboratory in the United States were pioneers in the study of this technology [24]. They used a high-pressure cryogenic pump at refueling stations to draw liquid hydrogen from a low-pressure Dewar. During this process, the liquid hydrogen was pressurized and vaporized to form CcH₂, which is directly delivered at the parametric range of 30 K to 60 K and 20.0 MPa to 87.5 MPa into the storage vessel. While this method utilizes liquid hydrogen as the feedstock and employs its high-grade cold energy to cool the storage tank, the overall energy consumption for hydrogen storage remains high due to the large amount of energy consumed in the process of hydrogen liquefaction. Consequently, this approach fails to leverage the intrinsic low energy consumption advantage of cryogenic compression storage. The other method involves a multi-stage compression of hydrogen at ambient temperature, followed by cooling to the storage temperature using an external cold source to produce CcH₂ [25]. Compared to the first mentioned method, this “compress-then-cool” approach eliminates the process to cool gaseous hydrogen to its liquefaction temperatures, thereby bypassing the energy intensive liquefaction step so as to significantly lower overall energy consumption. Furthermore, since the ortho-para conversion of gaseous hydrogen is negligible without a catalyst [23], the converter is eliminated to further simplify the process and reduce energy requirements. Evidently, the “compress-then-cool” method holds distinct advantages over liquid hydrogen vaporization in terms of both energy consumption and process complexity. If cryogenic compressed hydrogen storage technology is applied to the recovery of boil-off gas (BOG) from marine liquid hydrogen transport, it can overcome the major limitation of the high energy consumption associated with traditional BOG re-liquefaction methods. Furthermore, by adjusting the temperature and pressure conditions, it can achieve higher storage density while effectively utilizing the BOG. This technology can also flexibly adjust the storage conditions in response to fluctuations in the BOG generation rate, providing superior regulation capability to meet various voyage requirements.

Large-scale maritime transport employing cryogenic compressed hydrogen storage seems promising because more transportation volume can be obtained, owing to the density of cryogenic compressed hydrogen being higher than that of liquid hydrogen. But it may require a wholly new design for large containers capable of withstanding both high pressure and low temperatures, which face more severe challenges of safety and costs than liquid

hydrogen tanks. A feasible approach is to integrate cryogenic compression storage with conventional liquid hydrogen shipping on large ocean-faring vessels. This strategy can control costs well and ensure maritime safety since the studies on special small-sized containers for cryogenic compressed hydrogen storage are more mature [26–28]. Specifically, given that substantial volumes of liquid hydrogen carried in large-scale liquid hydrogen ships vaporizes inevitably, converting the hydrogen BOG into cryogenic compressed hydrogen for storage presents a prospective scheme to reduce hydrogen loss in marine transportation. This scheme offers a novel solution to the management of massive hydrogen BOG. In this study, a cryogenic compression recovery system for BOG on liquid hydrogen carriers is proposed and analyzed from the perspectives of energy consumption and exergy efficiency.

2. Model Development

2.1. Design of Novel Cryogenic Compressed Boil-Off Gas Recovery System

According to Zhao et al. [29], as the number of compressor stages increases, the total compression energy consumption decreases. The difference in total power consumption between five-stage and six-stage compression is less than 3%, and the power consumption of five-stage compression is only 47% of that of single-stage compression. Considering both system complexity and energy consumption, we use five-stage compression to construct a cryogenic compressed boil-off gas recovery system (see Figure 1). Low-temperature hydrogen starts at 0.1 MPa and is pressurized step by step to the target pressure. During compression, three seawater coolers (CR-1, CR-2, and CR-3) cool the compressed hydrogen. This helps control energy consumption and ensures material safety. The cooling module also uses the hydrogen fuel stream as an auxiliary cold source. Through heat exchangers HEX-3, HEX-4, and HEX-5, part of the compressed hydrogen is pre-cooled, reducing the load on the seawater coolers. Then, the ambient-temperature compressed hydrogen stream (H13) passes through HEX-2 and HEX-1. It is sequentially cooled by the low-temperature hydrogen fuel (HF1) and the incoming BOG (H1), finally becoming product H16. Meanwhile, the hydrogen fuel stream, after multiple heat exchanges, warms up to near-ambient temperature. It is then sent to a fuel cell system for power generation, providing auxiliary propulsion and electricity for the ship. The entire system recovers, pressurizes, cools, and cascades the energy of BOG without any external refrigeration.

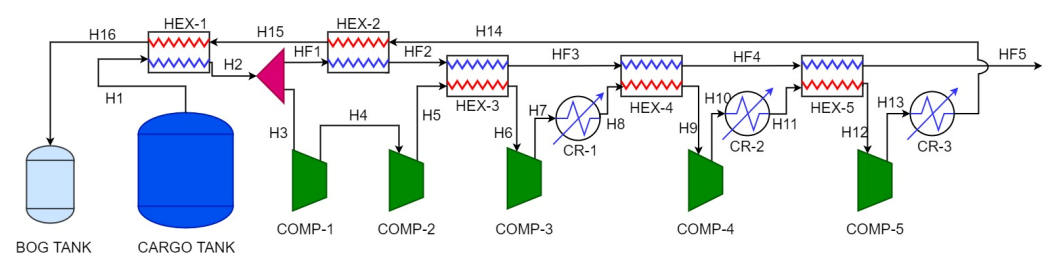


Figure 1. Novel cryogenic compressed boil-off gas recovery system.

Specifically, we apply the proposed system as shown in Figure 1 to a case where liquid hydrogen is stored in the 160,000 m³ cargo tank at 20 K and 0.1 MPa, and the boil-off gas (BOG) is generated at the same pressure. BOG is discharged from the ship's liquid hydrogen storage tank. After passing through pipelines and associated equipment, the BOG utilized for cryogenic compressed hydrogen production is supplied at 53 K and 0.1 MPa [30]. The BOG stream (H1), influenced by heat leakage in the collection piping, enters heat exchanger HEX-1. Subsequently, the BOG stream (H2) is split into two streams: a hydrogen fuel stream (HF1) and a cryogenic hydrogen stream (H3). The cryogenic hydrogen stream (H3) intended for storage is pressurized in compressors COMP-1 and COMP-2, resulting in a temperature increase. It then enters heat exchanger HEX-3, where it is cooled by

transferring heat to the hydrogen fuel stream (HF2). The cooled hydrogen stream (H6) subsequently undergoes a third stage of compression in compressor COMP-3. Coolers CR-1, CR-2, and CR-3 utilize seawater as the cooling medium to lower the temperature of each compressor's outlet stream to 313 K. The seawater discharged from each cooler is returned to the ocean. The hydrogen stream (H7), after being cooled in cooler CR-1, enters heat exchanger HEX-4 where it is further cooled by transferring heat to the hydrogen fuel stream (HF3). The resulting stream (H8) then enters the fourth-stage compressor (COMP-4). The discharged high-pressure hydrogen is cooled in cooler CR-2 and subsequently enters HEX-5 for additional cooling. The cooled, high-pressure hydrogen stream (H11) is compressed again in the fifth-stage compressor (COMP-5), followed by cooling in cooler CR-3. The resulting ambient-temperature, compressed hydrogen stream (H13) passes through HEX-2, where its temperature decreases through heat exchange with the hydrogen fuel stream (HF1). It then flows through HEX-1, where it is cooled by the incoming BOG, ultimately producing the final C_2H_2 product (H16) for storage in the BOG tank. Meanwhile, the hydrogen fuel stream, after four stages of heat exchange, reaches near-ambient temperature and is supplied to the fuel cell system for power generation, providing auxiliary propulsion power and supplying the onboard electrical system for the liquid hydrogen vessel shipping.

2.2. Energy Analysis

In this study, the system's specific energy consumption (*SEC*) is employed to evaluate the overall system performance. A high *SEC* indicates low energy efficiency of the process. It is defined as the total energy consumption divided by the mass flow rate of the produced C_2H_2 , as expressed below [31]:

$$SEC = \frac{\dot{W}_{\text{net}}}{\dot{m}_{\text{C}_2\text{H}_2}} \quad (1)$$

where *SEC* is the system's specific energy consumption, kWh/kg C_2H_2 , $\dot{W}_{\text{net}}$ is the net power input to the BOG recovery system, kW, and $\dot{m}_{\text{C}_2\text{H}_2}$ is the mass flow rate of cryogenic compressed hydrogen, kg/s.

2.3. Exergy Analysis

The energy is characterized not only by its quantity but also by its quality. Exergy is commonly used to evaluate the utilization efficiency of energy within a system [32]. It primarily consists of physical exergy, kinetic exergy, potential exergy, and chemical exergy. Kinetic and potential exergy are typically negligible [33]. Furthermore, since no chemical substances are transferred from the system to the environment, the chemical exergy is considered zero [33]. Therefore, only physical exergy is considered in the following analysis, which is divided into two parts: the flow exergy of the hydrogen streams and the thermal exergy associated with heat transfer across system boundaries. These are expressed by the following formulas according to literature [34], respectively:

$$\dot{E}x = \dot{m}_{\text{C}_2\text{H}_2} \cdot ex = \dot{m}_{\text{C}_2\text{H}_2} \cdot [(h - h_0) - T_0(s - s_0)] \quad (2)$$

$$\dot{E}x_Q = \int_{T_1}^{T_2} \left(1 - \frac{T_0}{T}\right) d\dot{Q} \quad (3)$$

where $\dot{E}x$ is the exergy transfer rate associated with a flow stream, kW, *ex* is the specific flow exergy, kJ/kg, *h* is the specific enthalpy, kJ/kg, *s* is the specific entropy, kJ/(kg·K), T_0 is the reference temperature, K, h_0 is the specific enthalpy at the reference state, kJ/kg, s_0 is the specific entropy at the reference state, kJ/(kg·K), $\dot{E}x_Q$ is the exergy transfer rate caused by heat exchange, kW, and $\dot{Q}$ is the heat transfer rate, kW.

The exergy balance for each individual component can be expressed as:

$$\sum \dot{E}x_{in} + \sum \dot{E}x_{Q,in} + \dot{W}_{net} = \sum \dot{E}x_{out} + \sum \dot{E}x_{Q,out} + \dot{E}x_{dest} \quad (4)$$

where $\dot{E}x_{in}$ and $\dot{E}x_{out}$ are the exergy transfer rates caused by flow streams entering and leaving the system, respectively, kW, $\dot{E}x_{Q,in}$ and $\dot{E}x_{Q,out}$ are the exergy transfer rates associated with heat transferring into and out of the system, respectively, kW, and $\dot{E}x_{dest}$ represents exergy destruction rate, kW.

Referring to literature [35], the exergy destruction rate and exergy efficiency for those compressors, coolers and heat exchangers of the novel system can further be derived, as listed in Table 1.

Table 1. Exergy destruction rate and exergy efficiency equations for the system components.

Components	Exergy Destruction Rate	Exergy Efficiency
Compressors	$\dot{E}x_{dest} = (\dot{m} \cdot ex)_{in} + \dot{W}_{net} - (\dot{m} \cdot ex)_{out} - \dot{Q}_{loss}(1 - T_0/T)$	$\eta_{ex,COMP} = 1 - \frac{\dot{E}x_{dest} + \dot{Q}_{loss}(1 - T_0/T)}{\dot{W}_{net}}$
Heat exchangers and Coolers	$\dot{E}x_{dest} = \sum (\dot{m} \cdot ex)_{in} - \sum (\dot{m} \cdot ex)_{out} - \dot{Q}_{loss}(1 - T_0/T)$	$\eta_{ex,HEX/CR} = 1 - \frac{\dot{E}x_{dest} + \dot{Q}_{loss}(1 - T_0/T)}{\sum (\dot{m} \cdot ex)_{in}}$

Notes: $\dot{m}$ represents the mass flow rate of the hydrogen stream entering or leaving the components, kg/s, $\dot{Q}_{loss}$ stands for the heat exchange rate between components and surroundings, kW, $\eta_{ex,COMP}$ denotes the exergy efficiency of the compressors, $\eta_{ex,HEX/CR}$ denotes the energy efficiency of the heat exchangers or coolers.

The overall exergy efficiency of the system is defined as the ratio of the exergy difference between the BOG and the CcH₂ product to the net power input required by the system, as shown below [36]:

$$\eta_{ex} = \frac{\dot{E}x_{CcH_2} - \dot{E}x_{BOG}}{\dot{W}_{net}} = \frac{\dot{m}_{CcH_2} \cdot ex_{CcH_2} - \dot{m}_{BOG} \cdot ex_{BOG}}{\dot{W}_{net}} \quad (5)$$

where η_{ex} is the overall exergy efficiency of the entire CcH₂ production system, $\dot{E}x_{BOG}$ is the flow exergy rate of the BOG stream, kW, $\dot{E}x_{CcH_2}$ is the flow exergy rate of the final CcH₂ product, kW, ex_{BOG} is the specific exergy of the BOG, kJ/kg, and ex_{CcH_2} is the specific exergy of CcH₂, kJ/kg.

In this system, only a portion of the total BOG generated from the liquid hydrogen storage tank is used for producing the CcH₂. The remainder serves as a refrigerant within the system and as fuel for other systems. Therefore, the exergy efficiency calculation specifically uses the mass flow rate of BOG directed to hydrogen production:

$$\dot{m}_{CcH_2} = \dot{m}_{BOG} \quad (6)$$

Combining Formulas (1) and (6) allows for the derivation of Formula (7), which expresses the relationship between exergy efficiency and specific energy consumption. This enables the calculation of the theoretical energy required to convert the feed BOG into CcH₂ from perspective of the second law of thermodynamics:

$$SEC = \frac{\dot{W}_{net}}{\dot{m}_{CcH_2}} = \frac{ex_{CcH_2} - ex_{BOG}}{\eta_{ex}} \quad (7)$$

3. Assumptions and System Simulation

3.1. Assumptions

The proposed system was simulated using ASPEN PLUS software V14, which provides an extensive database and robust methods for calculating physical properties [37,38]. The thermodynamic analysis was conducted under the following general assumptions:

- Liquid hydrogen is stored in the cargo tank at 20 K and 0.1 MPa, and BOG is generated at the same pressure. After passing through pipelines and associated equipment, the BOG utilized for cryogenic compressed hydrogen production is supplied at 53 K and 0.1 MPa [30].
- The total BOG mass flow rate is set at 1400 kg/h, corresponding to a boil-off rate of 0.3% per day for a 160,000 m³ liquid hydrogen carrier.
- Heat losses throughout the system and pressure drops across all heat exchangers and piping components are neglected.
- The minimum approach temperature in all heat exchangers is 3 K [39].
- The seawater cooler consists of a seawater–hydrogen heat exchanger and an air-cooled heat exchanger. The hydrogen stream is ultimately cooled to 313 K [30], while the temperature-rising seawater during the cooling process is first cooled to ambient temperature by the air-cooled heat exchanger, ensuring that the seawater finally discharged back to the ocean does not cause thermal pollution.
- The isentropic efficiency of all compressors is assumed to be 85% [40].
- The reference state for exergy analysis is defined as 298 K and 0.1 MPa [41].
- The pressure ratio in each compressor stage does not exceed 4, and the maximum discharge temperature is maintained below 160 °C [30].
- The system operates at steady state, and heat leakage is considered negligible.

3.2. Selection of Cryogenic Compressed Hydrogen Storage Conditions

The BOG studied in this work is available at 53 K and 0.1 MPa. Consequently, the temperature range for investigation was selected as 50 K to 110 K. Conventional hydrogen storage cylinders operate at 35 MPa as a typical working pressure. However, a patent by Nishino et al. [42] described a cryogenic liquid hydrogen tank design that demonstrates a pressure resistance of 100 MPa, confirming the structural feasibility of such extreme conditions even at low temperatures. With continuous improvements in composite materials and liner technologies, the practical working pressure of cryogenic high-pressure vessels may eventually reach 100 MPa. Therefore, to explore the theoretical performance limits of the proposed system under future material advancements, the pressure range for this investigation was set from 35 MPa to 100 MPa.

The REFPROP database, which contains extensive experimental data for hydrogen [43], was used as the reference. Utilizing its standard hydrogen density data, the hydrogen densities across the studied parameter ranges are plotted in Figure 2. The green-shaded region in the figure indicates a hydrogen density range from 70.85 kg/m³ to 97.74 kg/m³, all of which exceeds the density of liquid hydrogen at 0.1 MPa. Selecting target temperatures and pressures within this region for producing cryogenic compressed hydrogen enables a storage volume comparable to, or even smaller than, that achievable with a re-liquefaction process.

Since the system operates without external refrigeration to further cool the BOG, it cannot reduce the temperature of the produced hydrogen below 53 K. To ensure high storage density, the cryogenic compression BOG recovery system is designed to produce hydrogen within a temperature range of 53 K to 110 K and a pressure range of 40 MPa to 100 MPa. The SEC required for producing hydrogen at each state within these ranges is shown in Figure 3. The lowest SEC of 1.98 kWh/kgC₂H₂ is achieved for producing

cryogenic compressed hydrogen at 110 K and 40 MPa, which has a density of 54.99 kg/m³. The SEC increases as the temperature decreases and the pressure increases. The highest SEC of 2.81 kWh/kgCCH₂ is required to produce hydrogen at 53 K and 100 MPa, which has a density of 96.84 kg/m³.

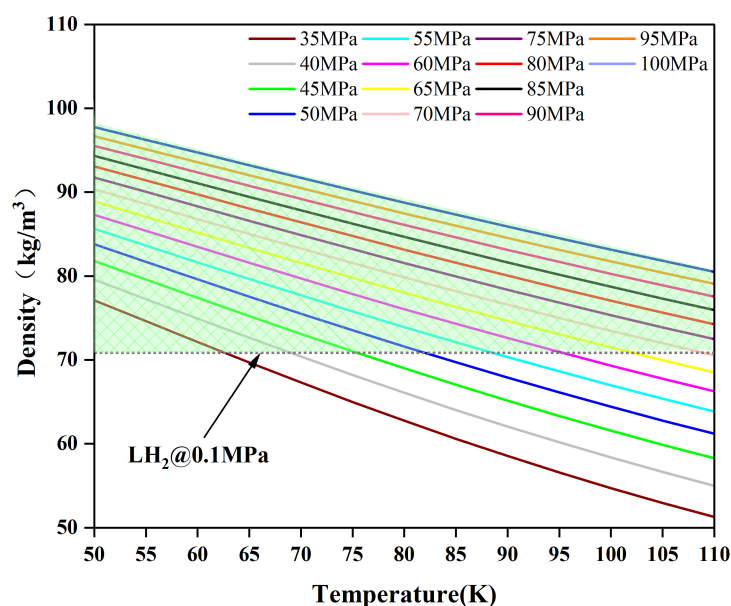


Figure 2. Hydrogen density at different temperatures and pressures.

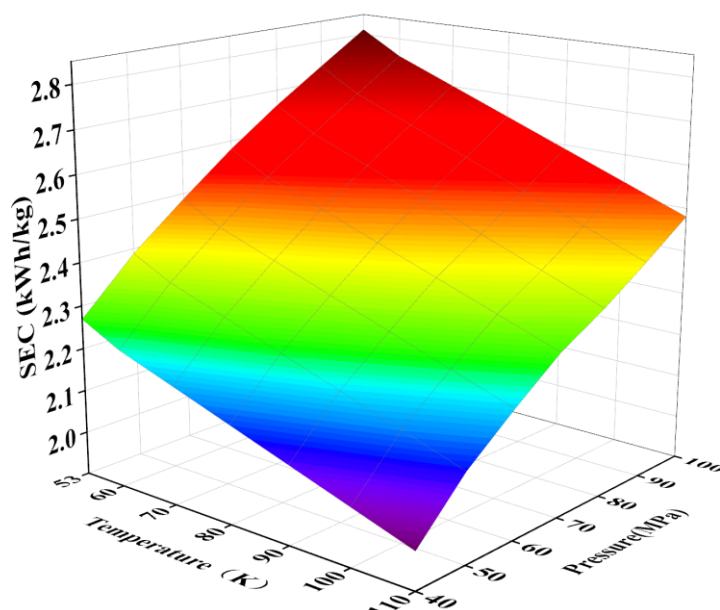


Figure 3. Specific energy consumption of hydrogen at different temperatures and pressures.

When producing cryogenic compressed hydrogen from BOG during marine transport, energy consumption alone cannot be the sole consideration. Due to the limited space on board, producing hydrogen with higher density is more advantageous for conserving valuable shipboard capacity.

Figure 4 plots the achievable hydrogen density against the SEC under different conditions. A parameter ψ is defined as hydrogen density/SEC to represent the acceptability of each storage condition. A higher ψ value indicates greater acceptability, as this evaluation metric comprehensively considers the trade-off between energy consumption and storage density. As shown in Figure 4, the highest ψ value is achieved at 53 K and 50 MPa, while the

lowest is at 110 K and 40 MPa. Furthermore, the sensitivity of the ψ value to temperature variations at a constant pressure becomes more pronounced as the pressure increases. This indicates that the temperature range of 60 K to 80 K offers more acceptable operating conditions, allowing for the selection of parameters within this range that reduce *SEC* while still ensuring a high hydrogen storage density.

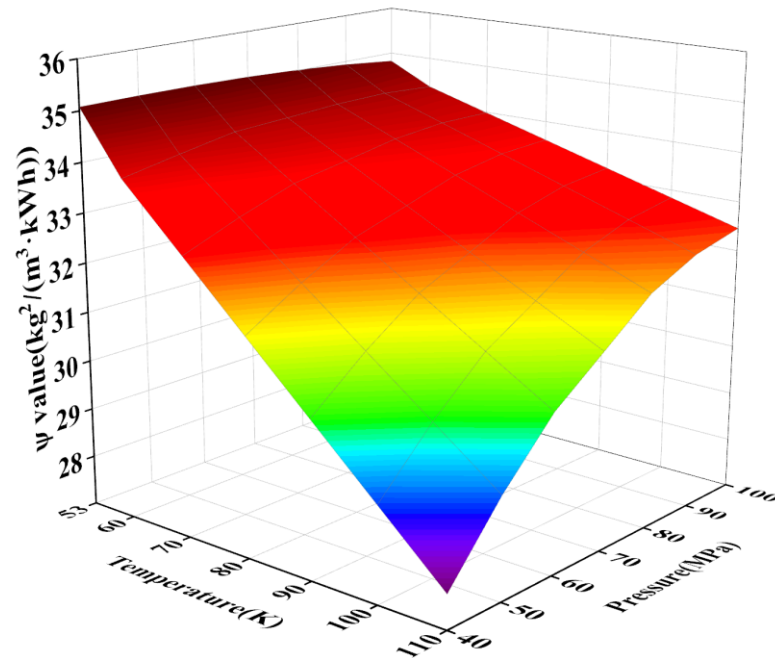


Figure 4. ψ values for cryogenic compressed hydrogen under different operating conditions.

Based on a comprehensive study of hydrogen storage density, specific energy consumption, and the ψ value, and considering the practical heat exchange constraint that prevents the final cryogenic compressed hydrogen (H15) from being cooled back to the initial boil-off gas (H1) temperature of 53 K, the specific operating conditions suitable for the cryogenic compression BOG recovery system investigated in this work are summarized in Table 2.

Table 2. Selected cryogenic high-pressure hydrogen parameters.

Parameters	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7
T (K)	60	70	70	80	90	100	110
P (MPa)	40	40	50	50	70	70	80
ρ (kg/m ³)	74.93	70.39	75.52	71.59	76.59	73.48	74.26

Once the storage capacity limit is reached, any additional hydrogen must be vented, which results in hydrogen waste. As hydrogen storage density increases, the same storage volume can hold more cryogenic compressed hydrogen. This increases the storage limit, resulting in longer sailing times and greater voyage distances. For a liquid hydrogen carrier traveling at a speed of 18 knots, the total storage volume for cryogenic compressed hydrogen is set at 5000 m³. Based on the hydrogen mass flow rate from the cryogenic BOG recovery system and the storage limit of the system, the maximum continuous sailing duration and voyage range for different hydrogen storage densities are calculated. The results are shown in Table 3.

Table 3. Sailing duration and continuous voyage range for different hydrogen storage densities in a 5000 m³ volume.

Hydrogen Storage Density (kg/m ³)	Sailing Duration (Day)	Continuous Voyage Distance (km)
70	26.92	21,540.18
73	28.08	22,463.34
76	29.23	23,386.49
79	30.38	24,309.64
82	31.54	25,232.79
85	32.69	26,155.94

At a storage density of 70 kg/m³ and above, the carrier can achieve a continuous sailing range exceeding 20,000 km. Referring to the maritime route from Comodoro Rivadavia to the port of Yokohama, Japan, which spans 21,400 km [44], the results demonstrate that the total volume of the cryogenic compressed hydrogen storage tanks designed in this study enables liquid hydrogen carriers to achieve long-distance, intercontinental transport.

3.3. Simulation Analysis of Specific Operating Conditions

In Case 4, with conditions of 80 K and 50 MPa, the hydrogen density is 71.59 kg/m³. Under these operating parameters, the liquid hydrogen carrier can achieve a voyage range exceeding 21,540.18 km. This condition was selected for detailed study of the cryogenic compressed hydrogen storage system. The mass flow rate of BOG allocated for cryogenic compressed hydrogen production is 650 kg/h, while the mass flow rate used as fuel is 750 kg/h. The thermodynamic properties of each hydrogen stream obtained from the simulation were used to calculate the corresponding exergy values, as summarized in Table 4.

Table 4. Thermodynamic properties of hydrogen streams in the proposed recovery system.

Stream	Temperature K	Pressure MPa	Mass Flow kg/h	Specific Enthalpy kJ/kg	Specific Entropy kJ/(kg·K)	Exergy Rate kW
H1	53.00	0.10	1400	−3409.15	−23.86	1444.45
H2	104.37	0.10	1400	−2702.06	−14.53	638.12
H3	104.37	0.10	650	−2702.06	−14.53	296.27
H4	160.06	0.35	650	−1937.19	−13.80	394.77
H5	243.97	1.20	650	−765.17	−13.06	566.59
H6	240.28	1.20	650	−817.20	−13.27	568.75
H7	362.77	4.16	650	951.08	−12.52	847.66
H8	313.15	4.16	650	231.92	−14.65	832.53
H9	292.62	4.16	650	−61.11	−15.62	831.71
H10	440.21	14.43	650	2140.93	−14.85	1187.86
H11	313.15	14.43	650	283.31	−19.83	1120.40
H12	262.30	14.43	650	−455.24	−22.41	1125.55
H13	396.63	50.00	650	1728.12	−21.56	1474.06
H14	313.15	50.00	650	484.01	−25.08	1439.00
H15	172.00	50.00	650	−1663.57	−34.22	1543.17
H16	80.00	50.00	650	−3186.52	−46.97	1954.34
HF1	104.37	0.10	750	−2702.07	−14.53	296.27
HF2	238.79	0.10	750	−840.83	−3.09	19.31
HF3	242.00	0.10	750	−795.74	−2.90	17.05
HF4	260.00	0.10	750	−541.77	−1.89	7.12
HF5	305.00	0.10	750	98.29	0.38	−0.47

HEX-1 and HEX-2 serve as the primary cooling module, responsible for lowering the temperature of the compressed hydrogen stream. In particular, HEX-2 utilizes the hydrogen fuel stream (HF1), which is split off early in the process, as its cold sink. This stream has sufficient thermal capacity to provide substantial cooling, enabling the large temperature drop from 313 K (H14) to 172 K (H15). HEX-3, on the other hand, is designed primarily to manage the energy consumption of the compression module. Its function is to trim the temperature of the hydrogen stream entering the third-stage compressor (COMP-3). The relatively small temperature reduction (≈ 3.7 K) is the result of an optimization that balances compression work and heat recovery under the current operating conditions (product at 50 MPa and 80 K). The data show that its contribution in terms of temperature drop is indeed modest under these conditions. However, if the system were required to produce cryogenic compressed hydrogen at a higher pressure, each compressor stage would need to perform more work, leading to a higher outlet temperature from COMP-2. Under such circumstances, the role of HEX-3 would become significantly more important, as it would then provide a much larger cooling duty to protect the downstream compressor and reduce overall energy consumption. Therefore, the inclusion of HEX-3 is fully justified, especially considering the flexibility of the system to operate over a wide pressure range (up to 100 MPa in this study).

4. Analyses and Discussions

4.1. System Comparison

The energy consumption and exergy efficiency of the proposed system were compared with those of a BOG re-liquefaction system and a system producing cryogenic compressed hydrogen via the “compress-then-cool” method (e.g., MRJT CcH₂), as summarized in Table 5.

Table 5. Specific energy consumption and exergy efficiency of the three systems.

Production Process	Operating Conditions	Specific Energy Consumption (kWh/kg)	Exergy Efficiency (%)
Proposed System	From 0.1 MPa, 53 K to 50 MPa, 80 K	2.25	87.88
Re-liquefaction System	From 0.1 MPa, 53 K to 0.1 MPa, 20 K	3.30	74.90
MRJT CcH ₂ System	From 0.1 MPa, 298 K to 50 MPa, 80 K	6.42	45.15

The re-liquefaction system re-liquefies the BOG using a reversed Brayton cycle. The liquefied hydrogen is returned to the cargo tanks. The system processes BOG at 53 K and 0.1 MPa, with a total mass flow rate of 1400 kg/h. The mass flow rate directed to re-liquefaction is set at 650 kg/h (for consistency with the proposed system), while 750 kg/h is used as fuel.

The MRJT CcH₂ system compresses hydrogen to 50 MPa and cools it to 80 K. The cooling duty is provided by an external refrigeration cycle. The feed hydrogen is at ambient temperature (298 K) and 0.1 MPa. The final product is at 80 K and 50 MPa.

The cryogenic compressed BOG recovery system proposed in this study and the re-liquefaction system both process boil-off gas generated from liquid hydrogen carriers. The two systems share identical settings in terms of inlet boundary conditions, total mass flow rate distribution, and the flow ratio between recovered hydrogen and fuel hydrogen, allowing for a direct comparison from the perspective of the basic process framework. Meanwhile, the proposed system and the MRJT CcH₂ system both aim to produce cryogenic compressed hydrogen at 80 K and 50 MPa. They exhibit strong consistency in product

form, storage density, and final application, enabling a comparative analysis from the perspectives of product output and energy efficiency.

As illustrated in Figure 5, the proposed system achieves a specific energy consumption (*SEC*) of 2.25 kWh/kg and an overall exergy efficiency of 87.88% when converting boil-off gas at 0.1 MPa, 53 K to CcH₂ at 50 MPa, 80 K, while the theoretical minimum specific energy consumption corresponding to 100% overall exergy efficiency is 1.97 kWh/kgCcH₂. In terms of *SEC*, the proposed system reduces *SEC* by 31.8% compared to the re-liquefaction system and by 64.9% compared to the MRJT CcH₂ system, respectively. In terms of exergy efficiency, the proposed system improves exergy efficiency by 17.3% over the re-liquefaction system and by 94.6% over the MRJT CcH₂ system, respectively. These comparison results indicate that the proposed system outperforms both reference systems in terms of both energy consumption and energy utilization efficiency, with particularly significant gains in exergy efficiency. This fully demonstrates the performance advantages of the proposed system for onboard BOG recovery.

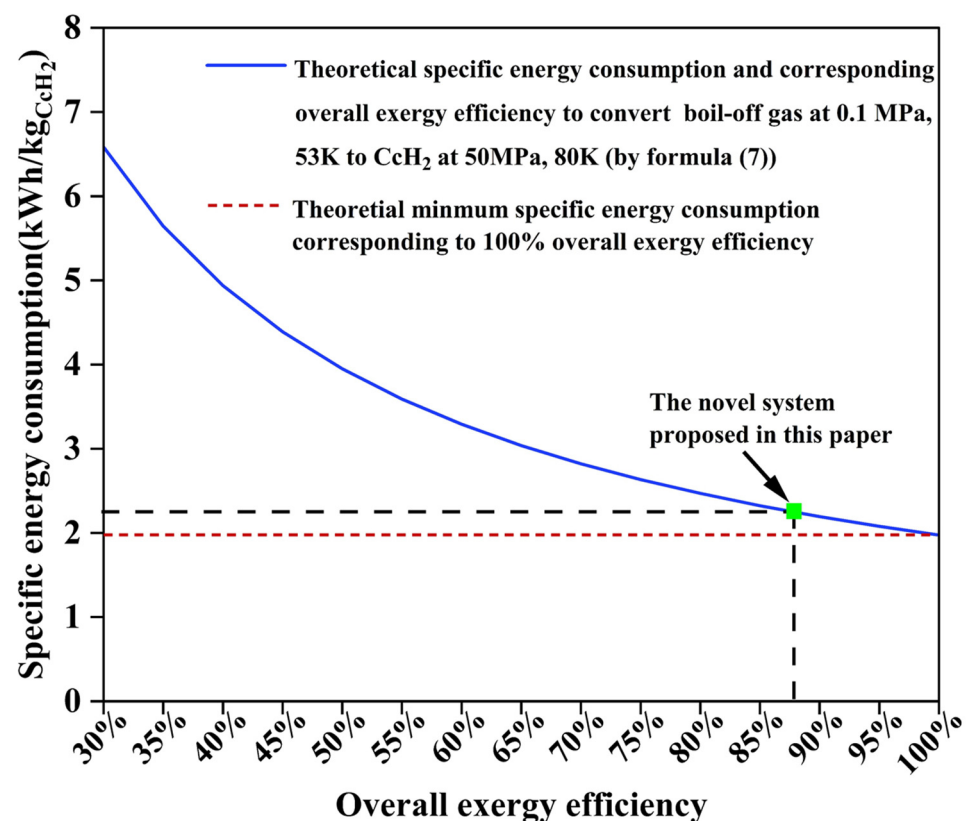


Figure 5. Exergy flow diagram of the overall system.

As shown in Table 6, compared with Table 5, compared to a BOG re-liquefaction system operating with identical BOG inlet conditions and hydrogen recovery mass flow rates, the proposed system reduces the *SEC* by 31.81% and increases the exergy efficiency by 17.32%, respectively. Furthermore, when viewed solely from the perspective of producing cryogenic compressed hydrogen, the proposed system—which generates hydrogen at 80 K and 50 MPa—demonstrates even more significant advantages over the MRJT CcH₂ system. By utilizing the inherent cold energy of the BOG, it eliminates the need for energy consumption in additional cooling modules. This results in a substantial *SEC* reduction of 64.95% and a remarkable exergy efficiency improvement of 94.64%. The exergy destruction rate, proportion of exergy rate, and exergy efficiency for each component in the system are presented in Table 6. The exergy destructions of the five compressors are relatively similar, each being approximately 40 kW.

Table 6. Exergy destruction distribution for system components.

Components	Exergy Destruction Rate (kW)	Proportion of Exergy Destruction Rate (%)	Exergy Efficiency (%)
COMP-1	39.60	4.16	71.32
COMP-2	39.80	4.18	81.19
COMP-3	40.37	4.24	87.36
COMP-4	41.45	4.36	89.58
COMP-5	45.71	4.81	88.40
HEX-1	395.16	41.55	86.77
HEX-2	218.38	22.96	87.74
HEX-3	0.08	0.01	98.00
HEX-4	10.75	1.13	98.73
HEX-5	2.44	0.26	93.72
CR-1	15.13	1.59	98.22
CR-2	67.46	7.09	94.32
CR-3	35.06	3.69	97.62

However, the exergy efficiencies of the third, fourth, and fifth compressors are significantly higher than those of the first and second compressors. Referring to Table 1, it can be observed that the inlet temperatures of the third, fourth, and fifth compressors are higher than those of the first two. This indicates that compressors operating at higher temperatures exhibit higher exergy efficiency, which can be attributed to lower entropy production under such conditions [19]. Furthermore, the exergy efficiencies of all coolers and heat exchangers exceed 86%. However, the exergy destructions in HEX-1 and HEX-2 are notably high, at 395.16 kW and 218.38 kW, respectively. These account for 41.55% and 22.96% of the total system exergy destruction. The primary cause of these significant destructions in HEX-1 and HEX-2 is the large temperature difference between the hot and cold streams. Reducing these stream-to-stream temperature differences is key to minimizing exergy destruction and improving overall system performance. In contrast, the coolers utilize seawater for cooling, and the seawater is ultimately discharged back to the ocean. This means that the greater the cooling duty, the larger the associated exergy destruction, as the thermal energy is effectively discarded to the environment.

4.2. Parametric Analyses

The temperature of the BOG generated onboard a liquid hydrogen carrier is initially close to that of the liquid hydrogen itself. However, this temperature increases due to heat ingress during its transport through pipelines before it enters the cryogenic high-pressure hydrogen system. Figure 6 illustrates the variation in the system's SEC and exergy efficiency with the BOG feed temperature. As the feed temperature increases from 42 K to 56 K, the SEC increases by 5%, while the exergy efficiency rises from 81.17% to 88.76%. This is because a lower feed temperature creates a larger temperature difference in the heat exchangers, which provides more cooling capacity from the BOG itself. Consequently, the compressors need to do less work, leading to a lower SEC. However, the larger temperature difference also increases exergy destruction during the heat transfer process, which reduces the system's exergy efficiency. It indicates that enhancing the thermal insulation of the collection pipelines to minimize heat ingress and preserve more of the BOG's inherent cold energy can reduce the energy consumption of the cryogenic compression system, albeit at the cost of a lower exergy efficiency.

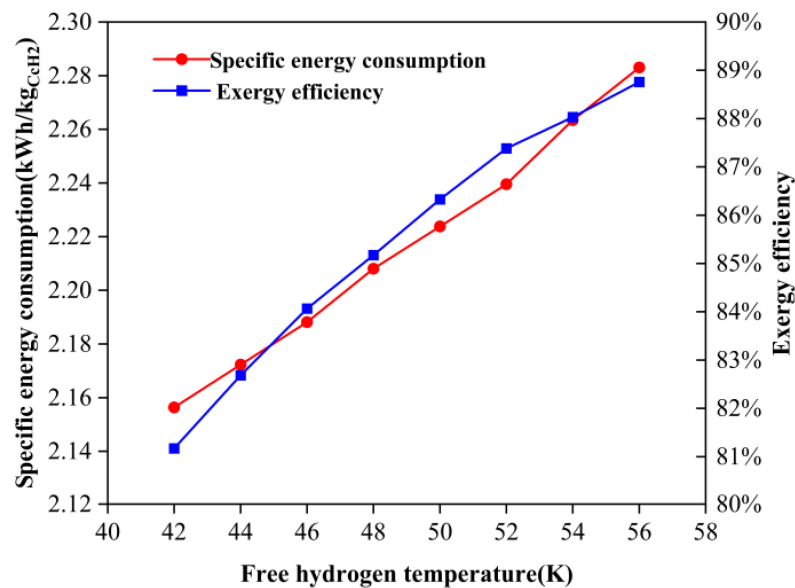


Figure 6. Effect of feed temperature on the system's specific energy consumption and exergy efficiency.

The BOG generated during the marine transport of liquid hydrogen is split into two streams: one is used to produce cryogenic compressed hydrogen for storage, while the other provides cooling for this production process before ultimately being used as fuel for power generation. Seven different flow distribution scenarios between these two streams are listed in Table 7.

Table 7. Comparison of seven mass flow rate scenarios for cryogenic compressed hydrogen production.

Case	Cryogenic Compressed Hydrogen Mass Flow (kg/h)	Hydrogen Fuel Gas Mass Flow (kg/h)	Recovery Rate (%)
1	600	800	42.86
2	625	775	44.64
3	650	750	46.43
4	675	725	48.21
5	700	700	50.00
6	725	675	51.79
7	750	650	53.57

An increase in the mass flow rate directed to cryogenic compressed hydrogen production indicates a higher BOG recovery ratio, meaning more of the evaporated hydrogen is reclaimed.

Figure 7 illustrates the impact of varying the mass flow rate allocated to cryogenic compressed hydrogen production on the system's SEC and exergy efficiency. While the total BOG flow rate remains constant at 1400 kg/h, increasing the mass flow rate for cryogenic compressed hydrogen production from 600 kg/h to 750 kg/h raises the SEC by 19.91% and decreases the exergy efficiency from 93.49% to 77.97%. This demonstrates that recovering a greater amount of BOG requires more energy input, and the system's exergy efficiency concurrently decreases. However, due to the finite total available cooling capacity within the system, increasing the production mass flow rate beyond 750 kg/h becomes infeasible, as the system can no longer meet the required cooling demands.

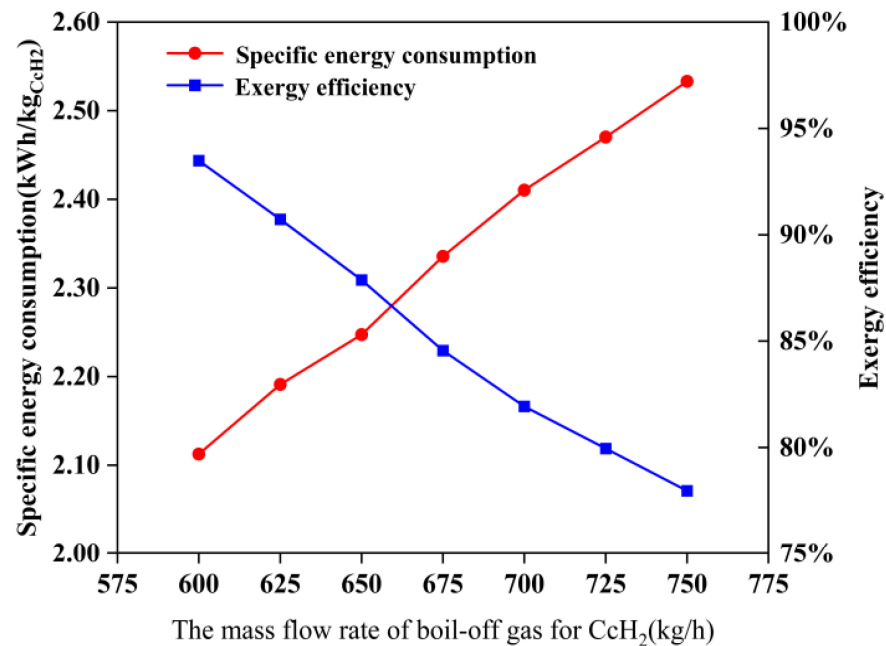


Figure 7. Effect of cryogenic compressed hydrogen mass flow rate on the system's specific energy.

5. Conclusions

This paper proposes a novel boil-off gas (BOG) management system for a 160,000 m³ liquid hydrogen carrier based on cryogenic compressed hydrogen storage. The system compresses a portion of the BOG and utilizes the cold energy of all BOG for cooling to produce high-density cryogenic compressed hydrogen for storage, while directing the remaining BOG to usage as fuel. This approach effectively reduces compressor power consumption and achieves a hydrogen storage density comparable to or even higher than that of liquid hydrogen, offering a new solution for BOG management on large-scale liquid hydrogen carriers. The key findings of this study are as follows:

- The system can produce cryogenic compressed hydrogen over a wide range of operating conditions (53–110 K, 40–100 MPa). The specific energy consumption (*SEC*) increases with decreasing temperature and increasing pressure. Through a comprehensive evaluation of *SEC* and the ψ value (density/*SEC*) under different operating conditions, seven optimal operating points that balance energy consumption and storage density are identified: 60 K & 40 MPa, 70 K & 40 MPa, 70 K & 50 MPa, 80 K & 50 MPa, 90 K & 70 MPa, 100 K & 70 MPa, and 110 K & 80 MPa.
- Under the representative operating condition (80 K, 50 MPa), the system achieves an *SEC* of 2.25 kWh/kg and an exergy efficiency of 87.88%, respectively. Compared to a BOG re-liquefaction system, the *SEC* is reduced by 31.8% and the exergy efficiency is increased by 17.3%, respectively; compared to the MRJT CcH₂ system, the *SEC* is reduced by 64.9% and the exergy efficiency is increased by 94.6%, respectively.
- Exergy analysis of individual components reveals that heat exchangers HEX-1 and HEX-2 contribute the largest exergy destructions, amounting to 395.16 kW and 218.38 kW, respectively. Meanwhile, compressors COMP-1 and COMP-2 exhibit the lowest exergy efficiencies, at 71.32% and 81.19%, respectively. Optimizing these specific components should be prioritized for enhancing the overall system performance.
- Increasing the BOG feed temperature raises both *SEC* and exergy efficiency, while increasing the BOG recovery rate (production flow rate) increases *SEC* but decreases exergy efficiency. The system is constrained by the available cooling capacity, and the maximum feasible production flow rate is 750 kg/h.

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