

Article

Thermodynamic Analysis and Optimization of Shipborne Cascade ORC/TCO₂ Power System Utilizing LNG Cold Energy

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Abstract

Liquefied natural gas (LNG)-powered ships possess available utilizable energy resources, including LNG vaporization cold energy and main engine flue gas waste heat. To improve the ship energy utilization efficiency, this paper takes a 215,000-ton Very Large Crude Carrier (VLCC) as the research object, and couples the organic Rankine cycle (ORC) and transcritical carbon dioxide (TCO₂) Rankine cycle to achieve the combined recovery of ship cold energy and waste heat. To systematically investigate the thermal performance of the integrated system, a simulation model is established by using Aspen HYSYS to acquire basic thermal parameters. Advanced exergy analysis is adopted as the core research method for the first time to evaluate the thermal performance of the system, and the obtained results are compared with those of the conventional exergy analysis. Conventional exergy analysis reveals that Heat Exchanger 1 is the core component restricting the performance optimization of the whole system. Further results from advanced exergy analysis show that the proportions of avoidable exergy destruction and endogenous exergy destruction of the overall system are 40.46% and 58.72%, respectively. Key optimization links are further identified based on the advanced exergy analysis results, and the genetic algorithm (GA) is applied for targeted system optimization. After optimization, the system output power reaches 1772.06 kW and the exergy efficiency is 48.32%, which are increased by 5.56% and 15.54% correspondingly. Economic evaluation indicates the annual net profit reaches 3.8336 million CNY, with a static payback period of 5.49 years.

Keywords: LNG; advanced exergy analysis; flue gas waste heat; cold energy cascade utilization; traditional exergy analysis; thermal performance optimization

Academic Editors: Andrea Baccioli,
Alberto Maria Gambelli and
Jan Taler

Received: 19 May 2026

Revised: 27 June 2026

Accepted: 8 July 2026

Published: 9 July 2026

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1. Introduction

The LNG is stored at −162 °C and turned from a vapor into a gas and used as fuel in the main engine. Vaporization took the release of cold power, totaling about 830 kilojoules of strength for every kilogram of content [1]. Additionally, the engine used in the power generation creates a substantial amount of waste heat in the power generation process. The exhaust of the waste heat contributes to increased environmental stresses, in addition to increased energy losses, which amount to around 50% of unused fuel energy [2]. Consequently, the novel strategy of coupling cold energy utilization with waste heat recovery for power generation applications has garnered substantial research interest. Specifically, of all the energy conversion systems at hand, the organic Rankine cycle (ORC) system has

been recognized as having substantial potential [3]. Zou et al. [4] took a 97,741-ton LNG-powered ship as the research object, considered four typical operating conditions in actual ship operation, and adopted the ORC to realize the combined recovery of ship LNG cold energy, main engine jacket water waste heat, and flue gas waste heat. The superiority of the proposed system was verified by comparison with the traditional two-stage parallel ORC system. Liu et al. [5] proposed a novel combined heat and power (CHP) system integrating a solid oxide fuel cell/gas turbine/CO₂ cycle and ORC, which utilizes the cold energy of LNG. The system performance was evaluated via energy and exergy analyses. The results demonstrate that the overall energy utilization efficiency, net power generation efficiency, and total exergy efficiency of the system reach 79.48%, 79.81% and 62.29%, respectively. This system has the potential to supply electricity, heating and cooling for central business districts, large-scale process industries and other scenarios simultaneously. Aiming at the high fuel consumption and large CO₂ emissions of the regasification system in floating storage and regasification units (FSRU), Naveiro et al. [6] put forward an innovative closed-loop regasification system. In this system, LNG cold energy is recovered for ORC power generation, as well as for CO₂ capture, compression, drying and liquefaction processes. The results reveal that the proposed system can fully meet the power demand of FSRU without adopting dual-fuel engines, while the CO₂ capture efficiency of boiler flue gas exceeds 90%. Zonfrilli et al. [7] investigated various schemes for cold energy recovery during LNG regasification, including power generation, refrigeration, cryogenic separation, as well as CO₂ capture and liquefaction. Among these schemes, Cases B1–B4 conducted a detailed analysis of ORC power generation cycles using LNG as the cold source under different heat sources. Compared with other LNG cold energy recovery technologies, Cases B1–B4 feature lower power generation costs, yet they also present inherent limitations such as lower thermal efficiency and higher capital investment. Taking the MAN Energy Solutions Marine Engine-Gas Injection (ME-GI) series LNG dual-fuel diesel engine as the research object, Wang et al. [8] established an ORC-based low-temperature cold energy power generation system, and analyzed and investigated the effects of operating parameters and regenerative cycles on system performance. At the industrial application level, two major LNG receiving terminals in Japan, Senboku [9] and Negishi [10], commissioned the world's first batches of LNG cold power generation units based on cryogenic Rankine cycles successively in the 1970s and 1980s. A 1450 kW Rankine generator set was put into operation at the Senboku site in 1979, followed by a 6000 kW combined cycle unit for capacity expansion in 1982. The Negishi terminal was equipped with ORC cold energy power stations adopting the same technical route. These facilities realized the commercial application of organic Rankine power generation using LNG as the cold source and ambient heat as the heat source, and serve as classic reference cases for the engineering implementation of LNG-ORC systems at present.

For such thermodynamic systems integrating LNG cold energy and waste heat recovery, domestic and international researchers have conducted feasibility studies from multiple dimensions, including thermodynamic performance, economic costs, and technical adaptability. In thermodynamics, it has been shown that exergy analysis is an effective technique in pinpointing the location, extent, and underlying causes of thermodynamic irreversibility within such systems [11]. The traditional exergy analysis method is a basic framework for identifying the magnitude and position of the destruction of exergy in thermodynamic systems. However, the above-mentioned technique has limitations in addressing the inter-relationships of the components and the potentials of the critical components. Identifying the inter-relationships and the potentials of the critical components is apparently crucial for achieving the system's optimal functionality. The current modern energy analysis techniques allow a further and deeper analysis when contrasted with traditional exergy analysis approaches. They not only examine the interactions between sys-

tem elements by decomposing them into endogenous and exogenous sources, but also further distinguish the actual potential of system components, subdividing them into two levels: unavoidable and avoidable parts. The combination of these two decomposition methods enables advanced exergy analysis to delve deeper into identifying the specific sources of exergy dissipation and determining which components are subject to avoidable exergy loss. Advanced thermodynamic analysis can more accurately explain the interaction relationships between components and effectively capture the irreversible characteristics caused by economic and technological limitations [12].

The advanced exergy analysis method originated in the late 20th century. In 1996, Feng et al. [13] first proposed the splitting method of avoidable and unavoidable exergy destructions for the optimization of thermal systems. In 1997, Feng and Zhu [14] extended the application of this method to complex systems by combining pinch technology. On the basis of previous studies, Tsatsaronis [15] introduced the definitions of endogenous and exogenous exergy destructions in 1999. He clarified that the exergy destruction of a component is restricted by both its own performance and the operating conditions of other components in the system. Eventually, he integrated two classification systems, i.e., endogenous/exogenous exergy destructions and avoidable/unavoidable exergy destructions, and established a complete four-dimensional framework for advanced exergy analysis. At present, in refrigeration, heat pump and other energy exchange systems, as well as condensing power plants, cogeneration power plants, and other electrical energy production engineering systems, in the petrochemical, petroleum, power, metallurgy, refrigeration and other technical fields and energy management departments, the advanced exergy method evaluation and guidance of energy use practice has been widely used. Based on the thermodynamic model of the solid oxide fuel cell (SOFC), Song et al. [16] adopted a plug flow reactor to simulate the external methane reforming reaction and incorporated the reaction heat into the exergy balance formula. The advanced exergy analysis results revealed that the endogenous exergy destruction of the system accounts for 87.5% of the total exergy destruction, indicating that the overall system exergy loss is mainly caused by the inherent deficiencies of the equipment itself. The avoidable exergy destruction of the stack accounts for 38.43%, which constitutes the primary source of avoidable exergy destruction in the system. Huang [17] conducted an advanced exergy analysis on a novel flexibly coupled ejector-compression refrigeration system to improve system performance. The total avoidable exergy destruction related to each component indicates that the compressor and evaporator possess the greatest improvement potential in the new system, followed by the condenser and ejector. Wang et al. [18] established a thermodynamic model and performed an advanced exergy analysis for a waste heat-driven composite energy recovery device for green ships. The results show that endogenous exergy destruction accounts for 89.73% of the total effective energy loss of the entire system. The avoidable endogenous exergy destruction of the gas turbine reaches 92.95%, demonstrating a significant potential for performance improvement. Jiang et al. [19] developed a supercritical Carbon Dioxide power cycle system with waste heat recirculation and designed an integrated system suitable for large-scale ship propulsion. Through conventional and advanced exergy analyses of the system, it is concluded that both heat regeneration and flow splitting can significantly improve the waste heat recovery performance of the CO₂ power cycle. Liao et al. [20] presented an assessment study of the combination of flue gas low- and medium-temperature waste heat harvesting in an ORC system by employing energy and exergy analytical approaches. Their study demonstrates that in every component of the system, the endogenous exergy always exceeds the exogenous one. The research particularly identified turbine 1 and condenser 1 as critical components for potential technical improvements, owing to their significant contributions to avoidable exergy destruction derived from internal sources.

The aforementioned studies mainly focused on the combination of the cold energy generated during LNG regasification processes and the residual low-temperature heat from engine exhaust systems to establish a unified framework for energy recovery and optimization, while lacking in the deep integrated exploitation of medium-grade waste heat derived from marine vessels. Obviously, this leads to the separation between the steam power cycle dedicated to medium-grade residual heat recovery and the power cycle configurations tailored for low-grade residual heat and LNG cryogenic energy exploitation. In the space-constrained environment of marine engine rooms, the coexistence of these two sets of systems is difficult to achieve [21]. In addition, the above studies indicate that the current ORC technology is widely applied in the comprehensive recuperation of low-temperature waste heat and LNG cryogenic energy. However, to achieve high-efficiency exploitation of medium-grade waste heat from ship exhaust streams, it is imperative to explore a power cycle configuration better suited to this temperature range. The transcritical carbon dioxide (TCO₂) Rankine cycle is an innovative power cycle where the working fluid is supercritical carbon dioxide and involves a non-isothermal transcritical phase change in the cycle. Because of this special feature, there is a marked reduction in thermodynamic irreversibility during heat exchange processes, as well as a notable enhancement in the thermal efficiency of the system owing to the alignment with medium-temperature heat sources. Meanwhile, it also boasts advantages such as a compact system structure and high operational safety, making it a cutting-edge technical solution for power generation through the recovery of medium and low-temperature waste heat. For this reason, adopting the TCO₂ cycle to recover medium-grade marine exhaust gas residual heat, utilizing the ORC for low-grade marine flue gas waste heat recovery, and simultaneously using LNG cryogenic energy as the system cold source to develop and design a reasonable cascade coupled utilization system suitable for marine medium-temperature flue gas waste heat and LNG cold energy can not only effectively reduce LNG vaporization energy consumption and ship thermal load as well as improve energy utilization efficiency, but also fundamentally solve part of the ship emission problems and yield considerable environmental and economic benefits. Furthermore, a review of published literature reveals that no scholars have yet conducted research on the thermodynamic performance analysis and optimization of such coupled power generation systems integrating marine LNG cryogenic energy and main engine medium-temperature waste heat from the perspective of advanced exergy analysis.

Accordingly, the innovation of the present research lies in the first attempt to adopt the advanced exergy analysis method for evaluating the thermodynamic performance of an integrated system for marine medium-temperature flue gas residual heat and LNG vaporization cryogenic energy proposed by the authors. A 215,000-ton VLCC-LNG dual-fuel vessel was chosen as the target ship in this paper. Combined with the actual ship operation data and practical application scenarios of ships of this scale, an innovative full-power generation system with two-stage pressurization and three-stage longitudinal nesting, which leverages both the ORC and TCO₂ Rankine cycle, was proposed to realize the cascade integrated recovery and exploitation of marine medium-temperature flue gas waste heat and LNG vaporization cryogenic energy. Moreover, a comparative analysis of the system was conducted using the advanced exergy analysis method and the traditional exergy analysis method to explore the specific causes of the low exergy efficiency of the system and clarify the direction for system optimization. Guided by the results of advanced exergy analysis, the optimization direction of the system was clarified, and then the genetic algorithm was applied to conduct targeted optimization of the system. This work not only provides an important reference for future research on multi-stage coupled utilization systems of marine waste heat and LNG cold energy, but also holds significant importance for fully improving energy utilization efficiency, reducing fuel consumption,

and further enhancing the economic efficiency of ships. Meanwhile, it expands the application boundary of the advanced exergy analysis method in such complex thermodynamic systems and enriches the empirical research of this method in the marine field.

2. System Design

2.1. Vessel Type Parameters

The study selects a 215,000-ton VLCC with LNG propulsion capabilities as the target ship. The ship replaces fuel oil with LNG as fuel, and its main engine is an ME-GI, operating at 85% load. The detailed parameters of the main engine [22] are listed in Table 1.

Table 1. Main engine parameters.

Parameter	Unit	Value
Mainframe Power	kW	25,480
Air inflow	kg/h	3000
Exhaust gas flow	kg/h	29,399
Mainframe air intake pressure	MPa	30
Mainframe air intake temperature	K	293
Residual heat temperature of flue gas	K	623

2.2. System Description

The significant disparity between LNG vaporization temperatures and flue gas emission levels presents a considerable challenge in identifying an appropriate working fluid capable of optimizing the thermal gradient between cryogenic LNG energy and waste heat from exhaust gases. With the objective of enhancing the efficiency of energy utilization, the current study employs the method “temperature matching, step utilization” to analyze the distribution of cold energy and the exergy conversion of liquefied natural gas based on the difference in temperatures. On the basis of the analysis mentioned above, the current study has organized the system into three levels. The major ideas and concepts include the following. The current system optimizes the latent cooling ability developed in the evaporation of the liquefied natural gas in order to enhance the heat transfer between the liquefied natural gas and the working fluid of the primary power generation cycle. The major function of this heat transfer system is to conduct effective condensation in the initial stage of the cycle. After ensuring the completion of the condensation process in the first-stage power generation cycle, the second-stage and third-stage power generation cycles will reutilize the residual LNG cryogenic energy and the underutilized cryogenic energy from the preceding-stage cycle for heat exchange, thereby fulfilling the operation of the low-temperature cycle. By adopting the afore-mentioned method, the current study optimizes the low-temperature potential available in the different levels of the liquefied natural gas systems to achieve the highest degree of optimal optimization of energy utilization efficiency [23]. Meanwhile, in the power generation system, the medium-temperature waste gas serves as a heat source, and the thermal energy it generates is sequentially used to drive the third-stage, second-stage, and first-stage power generation cycles. Among them, the three-stage power generation cycle adopts the reheating mode, the first and second-stage cycle processes extract a certain proportion of circulating working medium and LNG for heat transfer and condensation in advance, so as to enable the cascade utilization of LNG cold energy and enhance overall system efficiency. Moreover, the significant pressure difference between the main engine’s intake requirement (30 MPa) and the LNG storage tank pressure (110 kPa) poses a substantial challenge. Therefore, adopting a two-stage pressurization scheme is crucial to ensure that the main engine’s suction requirements are met.

Based on the previously outlined design principles, the present study develops a two-stage pressurized, three-stage longitudinal nested full power generation system between medium-temperature flue gas and low-temperature LNG, which can satisfy the need for effective utilization of waste heat recovery in a medium-temperature environment, along with the cryogenic potential available during the phase transition of liquefied natural gas from its liquid to gaseous state, for a 200,000-ton LNG-powered VLCC. The system process flow of the final design is shown in Figure 1.

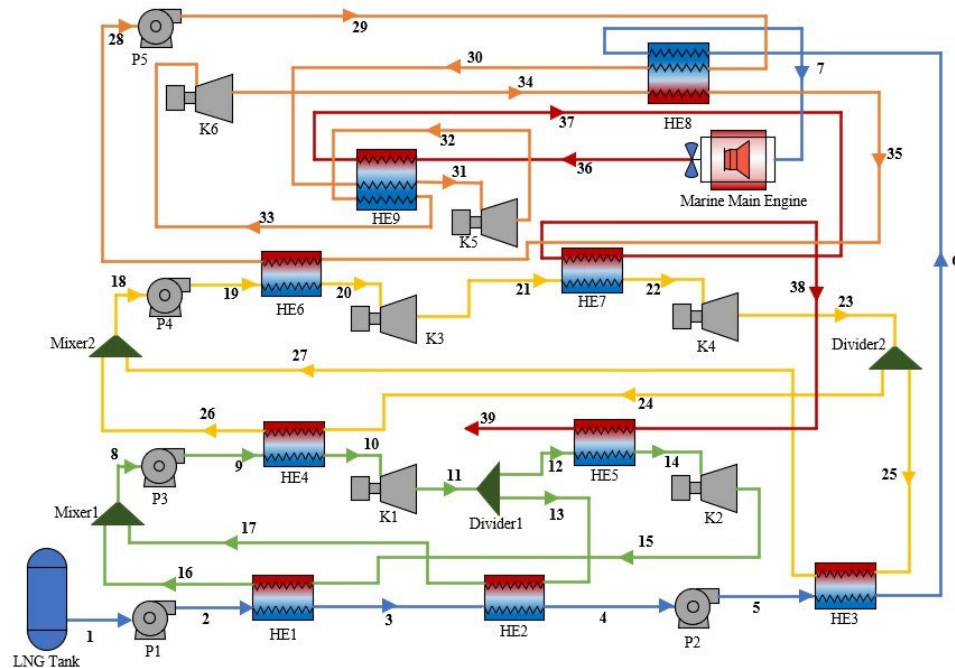


Figure 1. Simplified process flow diagram of the integrated system.

The system's detailed operational process is shown:

LNG process: The LNG released from the low-pressure storage vessel undergoes sequential pressurization and heating processes. Pump 1 (P1) elevates the LNG's pressure to the target value of 15 MPa at a time. Subsequently, the pressurized LNG flows successively through two thermal exchange units, designated as Heat Exchanger 1 (HE1) and Heat Exchanger 2 (HE2), respectively. During this process, high-grade low-temperature cold energy is initially supplied to the first-stage power generation system to meet its cooling demand. Then, the high-pressure Pump 2 (P2) is pressurized twice to 30 MPa, and then the Heat Exchanger 3 (HE3) and Heat Exchanger 8 (HE8) are used to provide cooling capacity for the second and third stage electricity generation cycles, and ultimately introduced into the ship's main engine for combustion.

First-stage generation cycle: The working fluid at the condenser outlet (Stream 8) is pressurized by Pump 3 (P3) and then fed into Heat Exchanger 4 (HE4) to absorb heat released by the working fluid of the second-stage cycle. Afterwards, it flows into Expander 1 (K1) for expansion power generation and subsequent flow splitting. One stream (Stream 13) exchanges heat directly with LNG for cooling before returning to the condenser. The other stream (Stream 12) absorbs waste heat from flue gas for reheating in Heat Exchanger 5 (HE5), and then enters Expander 2 (K2) to generate power. Finally, it mixes with the former stream (Stream 16), is cooled via heat exchange with LNG, and flows back to the condenser to complete the first-stage cycle.

Second-stage generation cycle: The working fluid exiting the condenser (Stream 18) is compressed by the Pump 4 (P4) and then directed into the Heat Exchanger 6 (HE6) from

the third stage of the power generation medium, and then enters the Expansion Machine 3 (K3) after power generation through the Heat Exchanger 7 (HE7) is reheated by the exhaust gas, and then enters the Expansion Machine 4 (K4) after the expansion of power generation shunt, a working medium and the first electrical mass heat exchange cooling into the condenser, The other working medium and LNG heat exchange and cooling into the condenser to complete the cycle.

Third-stage transcritical CO₂ Rankine cycle: The CO₂ from Heat Exchanger 6 (Stream 28) is supercharged by Pump 5 (P5) and enters Heat Exchanger 8 for preheating, then absorbs heat from Heat Exchanger 9 (HE9) and enters Expansion Machine 5 (K5) to generate electricity, and then enters Heat Exchanger 9 again for reheating and generates electricity through Expansion Machine 6 (K6). Finally, Heat Exchangers 8 and 6 are cycled through one after the other to capture cold capacity.

3. System Modeling and Validation

3.1. Initial Data and Assumptions

3.1.1. Initial Data

The LNG components used for system simulation are listed in Table 2, and the flue gas compositions are presented in Table 3.

Table 2. LNG fractions in the simulation [24].

Component	Methane	Ethane	Propane	Nitrogen
Volume Fraction	92.28%	5.94%	1.18%	0.60%

Table 3. Components of flue gas [25].

Component	N ₂	O ₂	CO ₂	H ₂ O
Volume Fraction	74.00%	9.90%	5.30%	10.80%

3.1.2. Assumptions

In the simulation of the proposed system configuration, several foundational premises have been established to ensure modeling accuracy and computational efficiency.

- (1) The system remains in a stable configuration, with all its variables unchanged during the analysis timeframe.
- (2) The ambient temperature is 298 K.
- (3) The Rankine cycle mass subcooling is 2 K.
- (4) Ignore heat loss from heat exchanger leakage.

The core components of the system comprise pumps, expanders, and heat exchangers. The adiabatic efficiency of pumps, isentropic efficiency of expanders, minimum temperature approach, and pressure drop in heat exchangers collectively constitute fundamental metrics characterizing the energy conversion behavior of the thermodynamic system, revealing intrinsic deviations between actual processes and idealized states. Table 4 shows the operating conditions under the assumptions of actual cycles, unavoidable cycles, and ideal cycles, respectively. Ideal operating conditions correspond to theoretical efficiency boundaries, which are based on the precept that all system elements function reversibly. Unavoidable operating conditions represent the maximum attainable efficiency in real-world engineering, constrained by current technological limits in materials science and manufacturing capabilities. Practical operating conditions mean the actual state of the system when it is put to work in engineering, every part has irreversible behavior.

Pumps change mechanical energy from the input to fluid pressure energy. The adiabatic efficiency is the ratio of the least ideal work to compress the fluid from the low-pressure state to the desired high pressure to the work actually consumed. Under ideal operating conditions, where no energy losses occur, the theoretical adiabatic efficiency of pumps is 1. But in fact, we also have to bear other irreversible losses such as flow, internal leakage, and fluid vortices, which consume more actual work, resulting in an adiabatic efficiency consistently less than 1. Dissipated energy shows up as waste heat, which raises the temperature of the fluid and brings about internal entropy generation. In engineering applications, pump adiabatic efficiency typically ranges from 0.7 to 0.9 [26]. Aligning with Ma Xingmin's [27] parameter selection for liquid pumps in compressed air energy storage systems, this study assigns a realistic adiabatic efficiency of 0.75 to the pumps. For now, accounting for those technological barriers that cannot be overcome, we see that the maximum adiabatic efficiency under unavoidable external operation will be 0.98 [28].

Contrary to pump operations, expanders allow high-pressure fluid to be expanded and decreased in pressure in order to create shaft work. Isentropic efficiency is equal to the actual work output divided by the largest possible theoretical power that the gas will deliver as it grows less pressurized. Under the perfect operation without any form of energy loss, the isentropic efficiency is 1. But in reality, because of irreversible factors like kinetic energy loss, blade friction, and unstable flow, the actual work output is less than the ideal, so the efficiency on an isentropic basis is always less than 1. Based on parametric alignment with pump specification, the study employs an actual isentropic efficiency of 0.75 for the expander and sets the highest achievable efficiency of the expander under unavoidable operating circumstances to 0.98.

As core components facilitating energy transfer in thermodynamic systems, heat exchangers fundamentally operate by driving heat transfer through temperature differentials between hot and cold fluid streams. The minimum temperature approach is referred to as the temperature closest to the outlet terminal of the heat exchanger, which serves as a demarcation for the reversibility boundary of the heat transfer process. The pressure drop comes from the friction caused by the fluid passing through the tube or the shell side; this is some kind of hydrodynamic limitation. Under ideal operating circumstances with zero irreversible loss from heat exchange as well as fluid flow, the theoretical values for minimum approach temperatures and pressure drops would be zero. The unavoidable operating conditions refer to the current technological limits achieved by materials science and manufacturing technologies. This is the research by Morosuk and colleagues [29] about three additional heat exchange devices; in this case, it is set that the minimum temperature approach is 0.5 K, and the pressure drop is 0.5% under unavoidable circumstances. For practical operating conditions, considering real engineering usage, and taking into account the parameters chosen by Zhang Shaobo in [30] for heat exchangers in Kalina cycle waste heat recovery systems, the actual lowest temperature approach is 5 K, and the pressure drop is 1%. Note that heat exchange equipment is very large when heat exchange equipment is at a very high temperature, such as HE4, HE5, and HE8; if the excessively small value temperature approach is used, then the required heat exchange area is very large, and the pressure drop loss is greater. Consequently, practical designs for such heat exchangers must strike an engineered trade-off between thermal efficiency and pressure constraints, justifying appropriately elevated temperature approach assumptions.

Table 4. Assumptions for cycles under real, ideal, and unavoidable conditions.

Component	Parameter	Real	Ideal	Unavoidable
P1	Adiabatic efficiency	0.75	1	0.98
HE1	Minimum end difference	5 K	0 K	0.5 K
	Pressure drop	1%	0%	0.5%

HE2	Minimum end difference	5 K	0 K	0.5 K
	Pressure drop	1%	0%	0.5%
P2	Adiabatic efficiency	0.75	1	0.98
HE3	Minimum end difference	5 K	0 K	0.5 K
	Pressure drop	1%	0%	0.5%
HE8	Minimum end difference	50 K	0 K	10 K
	Pressure drop	1%	0%	0.5%
P3	Adiabatic efficiency	0.75	1	0.98
HE4	Minimum end difference	50 K	0 K	10 K
	Pressure drop	1%	0%	0.5%
K1	Isentropic efficiency	0.75	1	0.98
HE5	Minimum end difference	50 K	0 K	10 K
	Pressure drop	1%	0%	0.5%
K2	Isentropic efficiency	0.75	1	0.98
P4	Adiabatic efficiency	0.75	1	0.98
HE6	Minimum end difference	5 K	0 K	0.5 K
	Pressure drop	1%	0%	0.5%
K3	Isentropic efficiency	0.75	1	0.98
HE7	Minimum end difference	5 K	0 K	0.5 K
	Pressure drop	1%	0%	0.5%
K4	Isentropic efficiency	0.75	1	0.98
P5	Adiabatic efficiency	0.75	1	0.98
HE9	Minimum end difference	5 K	0 K	0.5 K
	Pressure drop	1%	0%	0.5%
K5	Isentropic efficiency	0.75	1	0.98
K6	Isentropic efficiency	0.75	1	0.98

3.2. Thermodynamic Analysis Methods

3.2.1. Traditional Exergy Analysis

Each component is regarded as a control volume. The mass and energy conservation equations are applied to describe the system components:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out} \quad (2)$$

where $\dot{Q}$ is the heat transfer rate of the control volume, in kW; $\dot{W}$ is the output power of the control volume, in kW; $\dot{m}_{in}$ and $\sum \dot{m}_{out}$ are the mass flow rates of working fluid at the inlet and outlet, respectively, in kg/s; h_{in} and h_{out} denote the specific enthalpy of the working fluid at the inlet and outlet, respectively, in kJ/kg. The subscripts in and out represent inlet and outlet conditions.

The exergy destruction of each component is calculated based on the Gouy–Stodola theorem:

$$\dot{E}_{D,K} = T_0 \cdot \dot{S}_{gen,K} \quad (3)$$

where $\dot{E}_{D,K}$ is the exergy destruction of the K-th component; T_0 is the temperature at the environmental reference state, in K; $\dot{S}_{gen,K}$ is the entropy generation of the K-th component.

According to Tsatsaronis [15], the exergy balance equations are expressed as

$$\dot{E}_{F,K} = \dot{E}_{P,K} + \dot{E}_{D,K} \quad (4)$$

$$\dot{E}_{F,total} = \dot{E}_{P,total} + \sum \dot{E}_{D,K} + \dot{E}_{Loss,total} \quad (5)$$

where $\dot{E}_{F,K}$ and $\dot{E}_{P,K}$ are the fuel exergy and product exergy of the K-th component, respectively; $\dot{E}_{Loss,total}$ stands for the total exergy released to the environment. The fuel exergy refers to the total exergy input required for component operation, including the exergy difference in working fluid between inlet and outlet, input shaft work, and exergy from external heat supply. The product exergy is the effective available exergy produced during the thermal process, manifested as output mechanical work or exergy increment of working fluid. The difference between fuel exergy and product exergy is the irreversible exergy destruction, which arises from irreversible phenomena such as heat transfer under finite temperature difference and flow friction, representing permanent loss of available energy in the system.

Exergy flow rate is defined as the product of specific exergy and mass flow rate:

$$\dot{E} = \dot{m}e \quad (6)$$

Neglecting the changes in kinetic and potential exergy of working fluid flow, the specific exergy consists of physical specific exergy and chemical specific exergy:

$$e = e_{ph} + e_{ch} \quad (7)$$

Formulation of the exergy values by theoretical methods generates specific mathematical expressions for the physical and chemical aspects, as follows:

$$e_{ph} = h - h_0 - T_0 (S - S_0) \quad (8)$$

$$e_{ch} = \sum x_K \bar{e}_K^{CH} + \bar{R}T_0 \sum x_K \ln x_K \quad (9)$$

where x_K represents the mole fraction of each component in the mixture; the subscript “ph” signifies the physical properties sensitive to the temperature and pressure differential, while “ch” denotes the chemical properties in terms of molecular interactions. $\bar{R}$ is the universal gas constant; $\bar{e}_K^{CH}$ is the standard chemical exergy of each component.

For the overall system, the exergy efficiency is formulated as

$$\eta = \frac{\sum W_{Turbine}}{\Delta E + \sum W_{Pump}} \quad (10)$$

where η is the system exergy efficiency; ΔE is the total exergy difference between system inlet and outlet; $W_{Turbine}$ and W_{Pump} represent the power generated by turbines and the power consumed by pumps, respectively.

The overall net output power is

$$W_{net} = \sum W_{Turbine} - \sum W_{Pump} \quad (11)$$

Under the traditional exergy analysis framework, the basic calculation formula for the Kth component in the system is shown in the following Table 5 [31].

Table 5. Common parameters in traditional exergy analysis.

Parameter	Expression
Exergy destruction due to irreversibility in the Kth component	$\dot{E}_{D,K} = \dot{E}_{F,K} - \dot{E}_{P,K}$
Exergy efficiency	$\varepsilon_K = \frac{\dot{E}_{P,K}}{\dot{E}_{F,K}}$
Exergy destruction	$y_{D,K} = \frac{\dot{E}_{D,K}}{\dot{E}_{F,K}}$

Relative exergy destruction

$$y_{D,K}^* = \frac{\dot{E}_{D,K}}{\dot{E}_{D,tot}}$$

$y_{D,K}^*$ and $y_{D,K}$ serve as crucial indicators for assessing exergy destruction. The former serves as a metric for determining the fraction of exergy destruction attributed to the Kth individual unit in relation to the aggregate exergy destruction of the entire system, while the latter characterizes the correlation between the energy destruction in the Kth component and the total fuel-related exergy destruction.

The net power generation per unit mass of LNG is defined as

$$E_{kg} = \frac{W_{net}}{m_{LNG}} \quad (12)$$

where E_{kg} denotes the net power generation per kilogram of LNG, in kWh/kg-LNG; m_{LNG} is the hourly feed mass flow rate of LNG, in kg/h.

The LNG cold energy recovery rate quantifies the recovery degree of LNG vaporization cold energy by the cycle system from the perspective of energy quantity. It is defined as the ratio of the total actual cold energy absorbed from LNG by the system to the theoretical total vaporization cold energy of LNG:

$$\eta_{LNG} = \frac{Q_{LNG,use}}{Q_{LNG,total}} \times 100\% \quad (13)$$

$$Q_{LNG,total} = m_{LNG} \cdot q_{LNG} \quad (14)$$

where η_{LNG} is the LNG cold energy recovery rate; $Q_{LNG,use}$ is the total heat absorbed from the LNG side by the system; $Q_{LNG,total}$ is the theoretical total cold energy of LNG; q_{LNG} is the vaporization cold energy per unit mass of LNG, which is set to 830 kJ/kg [1].

3.2.2. Advanced Exergy Analysis

The advanced exergy analysis method intends to address two major tasks: (1) to provide relationship building among the different components of a system, and (2) to explore improvement possibilities of system operational effectiveness. The method uses a structured approach in which the exergy destruction in the system components is distributed into two categories, namely, the endogenous and exogenous parts, as well as avoidable and unavoidable portions, respectively, to improve system performance [32]. The relationship of the split and compound exergy destruction of the Kth component is evident in Figure 2.

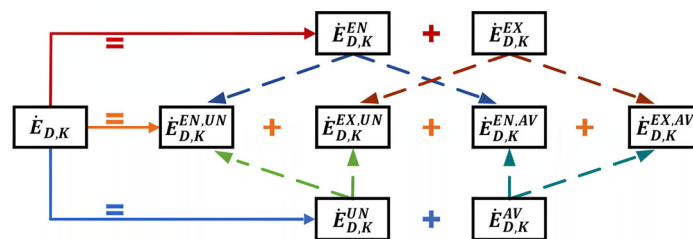


Figure 2. Split-combination relationship in advanced exergy use analysis.

- Avoidable and unavoidable exergy destruction;

Regarding the analysis of thermal systems, the amount of lost energy from the Kth component appears in two forms: unavoidable and avoidable loss of energy, respectively [33].

$$\dot{E}_{D,K} = \dot{E}_{D,K}^{AV} + \dot{E}_{D,K}^{UN} \quad (15)$$

$$\dot{E}_{D,K}^{UN} = \dot{E}_{P,K}^{real} \left[\frac{\dot{E}_{D,K}}{\dot{E}_{P,K}} \right]^{UN} \quad (16)$$

$\dot{E}_{D,K}^{UN}$, unavoidable exergy destruction: This refers to the fundamental exergy loss that cannot be eliminated or reduced by optimizing operating parameters, improving equipment structures, or adjusting system configurations under the constraints of current technologies, materials, and economic conditions. As the limit loss is determined by the inherent characteristics of equipment and practical engineering boundaries, it is unaffected by artificial optimization and regulation. Specifically, restricted by the high-temperature resistance, high-pressure resistance, and thermal conductivity of existing engineering materials, the machining and assembly precision of equipment, as well as the cost of equipment retrofitting and technological upgrading, such irreversible energy loss inevitably remains even when the component operating conditions and system matching are optimized to the current technical optimum. It represents the minimum exergy loss during the operation of thermal equipment. During the simulation, component K is operated under unavoidable conditions. When the inlet streams of the component retain the same thermodynamic parameters as those under actual working conditions, the expression of $(\dot{E}_{D,K}/\dot{E}_{P,K})^{UN}$ is obtained by isolating component K from the entire system [34]. Subsequently, the unavoidable exergy destruction of component K can be calculated via Equation (16), and the avoidable exergy destruction of component K is determined as the difference between total exergy destruction and unavoidable exergy destruction.

$\dot{E}_{D,K}^{AV}$, avoidable exergy destruction: This is the residual part of total component exergy destruction after deducting unavoidable exergy destruction. This redundant exergy loss can be effectively reduced or eliminated via manual regulation and has great potential for optimization and retrofitting. It is mainly caused by adjustable irreversible factors, including off-design operating conditions, unreasonable system matching, local flow disturbance, excessive heat transfer temperature difference, throttling loss, and insufficient utilization of waste heat. In engineering practice, avoidable exergy destruction can be continuously reduced, and the energy efficiency of components and the overall system can be improved by optimizing operating parameters, modifying internal equipment structures, enhancing system coupling, and cutting local energy losses.

The ratio $\dot{E}_{D,K}^{AV}/\dot{E}_{D,K}$ denotes the proportion of avoidable exergy destruction of the K-th component, which serves as a core indicator for evaluating the optimization potential. A higher ratio means larger redundant exergy loss and greater room for energy-saving retrofitting and condition optimization. On the contrary, a lower ratio indicates that the component operates close to the current technical optimum with limited optimization potential.

- Endogenous and exogenous exergy destruction.

In order to study interdependencies between systems, it is necessary to classify different components of exergy destruction in the Kth element into two basic forms: endogenous and its exogenous counterpart [33].

$$\dot{E}_{D,K} = \dot{E}_{D,K}^{EN} + \dot{E}_{D,K}^{EX} \quad (17)$$

$$\dot{E}_{D,K}^{EN} = \dot{E}_{P,K}^{real} \left[\frac{\dot{E}_{D,K}}{\dot{E}_{P,K}} \right]^{EN} \quad (18)$$

$\dot{E}_{D,K}^{EN}$, endogenous exergy destruction: This specifically refers to the exergy loss caused solely by internal irreversible processes within the K-th component, representing its inherent loss characteristics. When component K operates under real conditions, and all

other components of the system run under ideal conditions, the expression of $(\dot{E}_{D,K}/\dot{E}_{P,K})^{EN}$ can be obtained. Afterwards, the endogenous exergy destruction of component K can be calculated via Equation (18), and the exogenous exergy destruction of component K is determined by subtracting the endogenous exergy destruction from the total exergy destruction.

$\dot{E}_{D,K}^{EX}$, exogenous exergy destruction: This is the residual part of the total exergy destruction after subtracting the endogenous portion. Physically, it stands for the additional exergy loss of the K-th component indirectly induced by irreversible processes of other components in the thermal system. Components in a thermal system are mutually coupled and interact with one another. Operating deviations and irreversible losses of other components propagate to the K-th component via fluctuations of fluid temperature, pressure, flow rate and state parameters, disturbing its ideal operation and triggering extra exergy losses. Such losses are not inherent to the component itself but derived from mismatched operating conditions and coupling interference among components, reflecting the correlation and integrity of the overall system.

The ratio $\dot{E}_{D,K}^{EN}/\dot{E}_{D,K}$, namely the proportion of endogenous exergy destruction in total exergy destruction, is a key parameter to identify the root cause of component performance degradation and clarify the main source of losses. A higher ratio indicates intensified internal irreversible processes and deteriorated performance in fluid flow and heat transfer, meaning the component itself is the primary contributor to the increased system exergy destruction. A lower ratio implies that internal irreversible losses of the component are effectively mitigated and its operational performance is improved. This will further enhance the energy utilization efficiency of the entire thermal cycle and optimize the overall operating conditions of the system.

- Combination of two splitting options.

To get an in-depth understanding of how different components interact, the exergy destruction caused by the Kth component can be conveniently categorized into four types: the endogenous avoidable part, the exogenous avoidable part, the endogenous unavoidable part, and the exogenous unavoidable part [35].

$$\dot{E}_{D,K} = \dot{E}_{D,K}^{EN,AV} + \dot{E}_{D,K}^{EN,UN} + \dot{E}_{D,K}^{EX,AV} + \dot{E}_{D,K}^{EX,UN} \quad (19)$$

$\dot{E}_{D,K}^{EN,AV}$ and $\dot{E}_{D,K}^{EX,AV}$ represent the internally and externally generated avoidable exergy destructions, respectively. These can be optimized by system optimization techniques. $\dot{E}_{D,K}^{EN,UN}$ and $\dot{E}_{D,K}^{EX,UN}$ represent two groups of irreducible energy destructions. The former indicates the unavoidable exergy destruction inherently related to system component "K", whereas the latter reflects the unavoidable destructions induced by the system as a whole due to the interaction between the system and other elements. The intersection of the aforementioned two frameworks of exergy destruction provides significant optimal system insights. Here, the avoidable sector provides the real scope of optimization. Therefore, it is necessary for system designers to focus more on the minimization of avoidable destructions not only at the system component level but also at the interaction level between system components.

For the comprehensive assessment of the endogenous/exogenous and unavoidable/avoidable exergy destruction concepts, $\dot{E}_{D,K}^{EN,UN}$ is calculated first using the following equation. The values for $\dot{E}_{D,K}^{EX,UN}$, $\dot{E}_{D,K}^{EN,AV}$ and $\dot{E}_{D,K}^{EX,AV}$ can then be assessed using the following equation and the relation shown in Figure 3.

$$\dot{E}_{D,K}^{EN,UN} = \dot{E}_{P,K}^{EN} \left[\frac{\dot{E}_{D,K}}{\dot{E}_{P,K}} \right]^{UN} \quad (20)$$

$$\dot{E}_{D,K}^{UN,EX} = \dot{E}_{D,K}^{UN} - \dot{E}_{D,K}^{UN,EN} \quad (21)$$

$$\dot{E}_{D,K}^{EN,AV} = \dot{E}_{D,K}^{EN} - \dot{E}_{D,K}^{EN,UN} \quad (22)$$

$$\dot{E}_{D,K}^{AV,EX} = \dot{E}_{D,K}^{EN} - \dot{E}_{D,K}^{EN,EX} \quad (23)$$

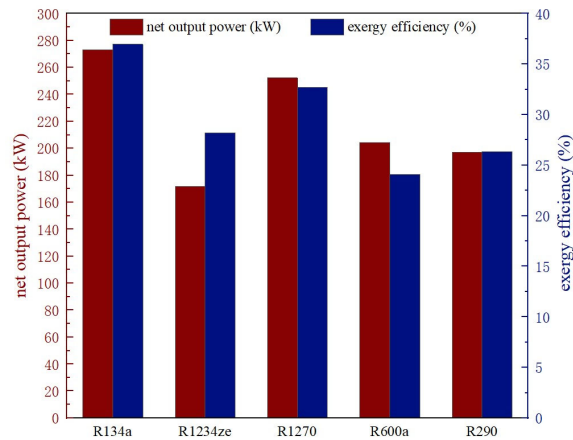


Figure 3. ORC system performance of various working fluids.

The advanced exergy analysis offers important findings through the analysis of endogenous avoidable exergy destructions, which can serve as key indicators for optimization in thermal systems. The avoidable endogenous exergy efficiency $\varepsilon_K^{AV,EN}$ and the relative avoidable endogenous exergy destruction $\gamma_K^{*AV,EN}$ are used as indicators for making key decisions about improving the thermal system [33,36].

$$\varepsilon_K^{AV,EN} = \frac{\dot{E}_{P,K}}{\dot{E}_{F,K} - \dot{E}_{D,K}^{UN} - \dot{E}_{D,K}^{AV,EX}} \quad (24)$$

$$\gamma_K^{*AV,EN} = \frac{\dot{E}_{D,K}^{AV,EN}}{\dot{E}_{D,tot}} \quad (25)$$

3.3. Economic Analysis

According to the capital cost function of the system's input equipment given in Reference [37], see Table 6 below.

Table 6. Capital investment cost functions of individual components.

Component	Capital Cost Function
Pump	$1120 \cdot W_p^{0.8}$
Heat exchanger	$2143 \cdot A_H^{0.514}$
Turbine	$4405 \cdot W_{Tur}^{0.89}$

The total system investment cost is defined as [38]

$$C_{total} = \sum_k (Z_{CI} + Z_{OM})_k \quad (26)$$

where $Z_{CI,k}$ is the investment cost of the system components, and $Z_{OM,k}$ is the operation and maintenance cost of the system.

$$Z_{CI,k} + Z_{OM,k} = \frac{Z_k \cdot \phi}{N \cdot 3600} CRF \quad (27)$$

Here, Z_k represents the investment cost of each component of the system, ϕ denotes the system maintenance coefficient with a value of 1.06 [39], N stands for the annual operating hours of the system, and the capital recovery factor (CRF) is defined as follows:

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (28)$$

Here, n represents the service life of the system with a value of 20 years [40], i denotes the annual interest rate with a value of 14% [39]. The power production cost (EPC) and the total annual net income (ATNI) of the system are expressed as follows, respectively [38]:

$$EPC = \frac{3600C_{total}}{W_{net}} \quad (29)$$

$$ATNI = 7200(EP - EPC) \cdot W_{net} \quad (30)$$

Here, W_{net} represents the net output power, EP is the current electricity price with a value of 1.5 CNY/(kW·h). EPC refers to the unit power generation cost of the system (with a system service life of 20 years), and the difference between EP and EPC is the system net profit per unit power generation.

3.4. Model Validation

3.4.1. Validation of ORC and TCO₂ Sub-Cycles

The simple ORC and TCO₂ cycles presented in Amirhassan's study [41] are adopted as the validation benchmark. Mathematical calculations have been carried out under the same input parameters, which have been rigorously evaluated by drawing comparisons with existing results produced by ASPEN HYSYS V14 software, as described in Tables 7 and 8. This clearly confirms the accuracy of the calculation procedure adopted in performing this research. Although both power cycles are constructed based on ORC and TCO₂ systems, the conventional series configuration in Amirhassan's work differs from the novel integrated scheme proposed herein, which features two-stage pressurization and three-stage longitudinal nesting. This research targets the actual operating conditions of large VLCCs and adopts advanced exergy analysis for refined performance evaluation. Accordingly, the research framework and analytical methods are more comprehensive.

Table 7. Comparison of calculation results between the ORC and the literature [41].

	Calculation Results in This Paper	Results of the Literature [41]	Error (%)
Net output power [MW]	7.8967	8.1026	2.54
Mass flow rate of ORC working medium [kg/s]	29.58	29.19	1.34
Mass flow of cooling water [kg/s]	506.39	507.44	0.21

Table 8. Comparison of calculation results between the TCO₂ cycle and the literature [41].

	Calculation Results in This Paper	Results of Liter- ature [41]	Error (%)
Net output power [MW]	6.6275	6.8559	3.33
Mass flow of CO ₂ [kg/s]	32.69	32.75	0.18
Mass flow of cooling water [kg/s]	359.72	358.43	0.36

Since model validation cannot be fully accomplished merely by comparing simulation results, the experimental data reported by Shi et al. [42] and Liu et al. [43] are further adopted as benchmarks. Corresponding models consistent with the experimental rig lay-

out are established in Aspen HYSYS, and all input parameters are set to the values under the specified operating conditions given in the literature. The calculated results are compared with experimental data, as presented in Tables 9 and 10. It is found that the key performance parameters of the ORC and TCO₂ cycles from the simulation agree well with the measured experimental data in the two references. According to Table 9, the maximum relative error of core parameters for the TCO₂ cycle, including net output power, mass flow rate of CO₂, system exergy efficiency and thermal efficiency, is only 4.09%. As shown in Table 10, the maximum relative error of all ORC performance parameters is 4.15%. All errors of the validated parameters are within 5%.

The above results fully demonstrate the engineering accuracy and practicability of the ORC and TCO₂ sub-cycle models. The established models are reliable and effective, and can support the subsequent performance analysis and optimization of the integrated system.

Table 9. Comparison between calculated results of the TCO₂ cycle and experimental data in Ref. [42].

	Calculation Results in This Paper	Experimental Data [42]	Error (%)
Net output power [kW]	1.78	1.71	4.09
Mass flow rate of CO ₂ [kg/s]	0.122	0.12	1.67
System exergy efficiency [°C]	8.53	8.80	3.07
System thermal efficiency [°C]	4.65	4.53	2.65

Table 10. Comparison between calculated results of ORC and experimental data in Ref. [43].

	Calculation Results in This Paper	Experimental Data [43]	Error (%)
Net output power [kW]	5.08	5.30	4.15
Mass flow rate of ORC working medium [kg/min]	11.72	11.53	1.65
System exergy efficiency [°C]	25.13	24.34	3.25
System thermal efficiency [°C]	9.84	10.14	2.96

3.4.2. Validation of the Integrated System Model

Independent validation of only the ORC and TCO₂ Rankine cycle subsystems cannot verify the coupling characteristics of cold and heat sources, as well as the overall mass and energy matching of the entire system, which fails to guarantee the simulation accuracy of the full-process integrated system. Therefore, the overall model validation is further carried out for the integrated system of cascade utilization of LNG cold energy and ship waste heat. The data from the reference [44] is adopted as the benchmark.

In the validation, the Aspen HYSYS model established with the Peng–Robinson equation of state and the defined LNG composition parameters is used. All other parameters and boundary conditions are consistent with the design conditions of Case 3 in the reference [44]. The simulation results are compared with the benchmark data, as shown in Table 11. It can be seen that the relative errors of all key performance parameters fall within a reasonable range for engineering simulation with controllable deviation. This verifies the reliability of the selected property method, LNG composition settings and modeling strategy in Aspen HYSYS, proving that the proposed model is scientifically sound for the simulation of the integrated system.

Both this work and the reference focus on the collaborative utilization of marine engine waste heat and LNG cold energy with similar application scenarios, facilitating

model comparison and analysis. Different from the reference that merely adopts ORC to recover LNG cold energy together with low-temperature flue gas and jacket water waste heat of ships, this study targets the integration of medium-temperature ship flue gas and LNG cold energy. Following the principle of cascade energy utilization, medium-temperature waste heat is recovered by the TCO₂ cycle while low-temperature waste heat is harvested by the ORC. The established TCO₂-ORC cascade coupling structure further improves the comprehensive utilization rate of waste heat at all temperature grades and LNG cold energy.

Table 11. Model validation results of the integrated system with LNG cold energy and ship waste heat.

	Calculation Results in This Paper	Results of the Literature [44]	Error (%)
Net output power [kW]	210.83	218.51	3.51
System exergy efficiency [°C]	26.54	27.30	2.78
System thermal efficiency [°C]	16.07	15.62	2.88

3.5. Selection of Working Fluids

To scientifically select working fluids for the ORC, multiple factors were comprehensively considered, including thermodynamic performance, environmental performance (mainly global warming potential (GWP) and ozone depletion potential (ODP)), physico-chemical properties and safety characteristics [45]. Six candidate working fluids were chosen: R1150, R134a, R1234ze, R1270, R600a and R290. The properties of the selected working fluids are presented in Table 12.

Table 12. Properties of candidate working fluids.

Working Fluid	Critical Temperature (°C)	Critical Pressure (kPa)	Boiling Point at 1 atm (°C)	Freezing Point at 1 atm (°C)	Safety Performance	GWP	ODP
R1150	9.2	5040	−103.8	−169.4	Low toxicity, highly flammable	0.2	0
R134a	101.1	4067	−26.1	−101	Low toxicity, non-flammable	14.3	0
R1234ze	109.4	3636	−18.95	−104.5	Low toxicity, slightly flammable	0.01	0
R1270	91.9	4550	−47.7	−185.2	Low toxicity, highly flammable	0.2	0
R600a	134.8	3655	−11.79	−159.6	Low toxicity, highly flammable	0.2	0
R290	96.75	4257	−42.19	−187.7	Low toxicity, highly flammable	0.2	0

Taking the net output power and total exergy destruction of the ORC system as evaluation indices, the working fluids for the ORC are screened from the perspectives of thermodynamics, economy and environmental protection. The first-stage ORC utilizes high-grade cold energy from LNG, which features a relatively low temperature. Therefore, working fluids with low boiling points should be adopted to avoid considerable irreversible losses caused by excessive temperature difference between hot and cold streams inside the heat exchangers. As shown in Table 12, R1150 has the lowest boiling point among all candidate fluids. Its freezing point is −169.4 °C, which prevents solidification during operation. Accordingly, R1150 is selected as the working fluid for the first-stage ORC. R134a, R1234ze, R1270, R600a and R290 are taken as candidate working fluids for the second-stage ORC. Under identical operating conditions, their effects on the net output power and total exergy destruction of the ORC system are compared, as presented in Figure 3. The system achieves the best performance when R134a is used for the second-stage ORC. Nevertheless, Table 12 indicates that R134a has a relatively high GWP. Although the ORC system using R1270 delivers slightly inferior performance, this fluid exerts less environ-

mental impact. After a comprehensive assessment, R1270 is determined as the working fluid for the second-stage ORC. Consequently, R1150, R1270 and CO₂ are employed as the working fluids for the first, second and third stage power generation cycles, respectively.

4. Analysis of Simulation Results

The target system was simulated using HYSYS V14 software, the Peng–Robison equation of state was used in the simulation, and the system flowchart is shown in Figure 4, while the thermodynamic properties of each stream parameter are listed in Table 13.

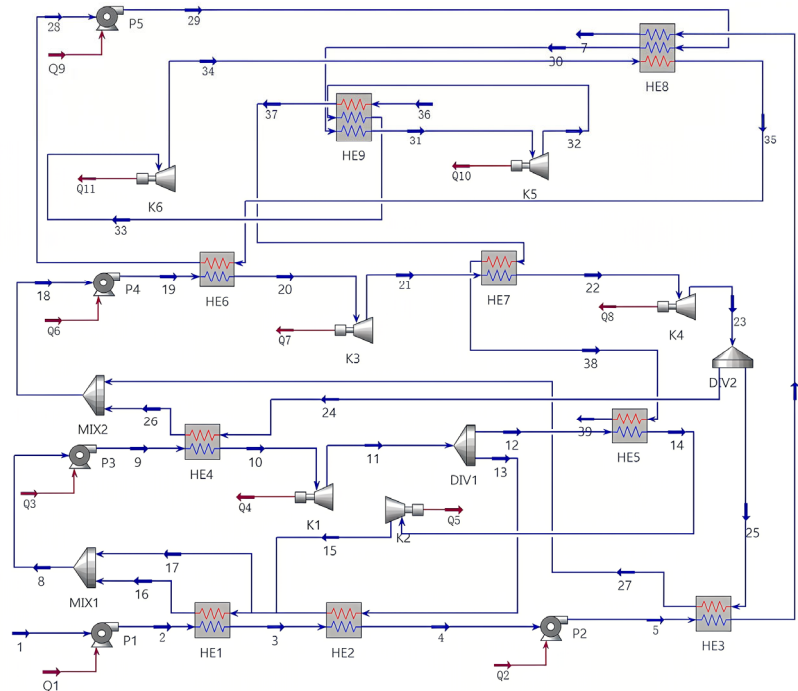


Figure 4. Aspen HYSYS software simulation system flow chart.

Table 13. Thermodynamic properties of each stream parameter.

Stream	Temperature (K)	Pressure (kPa)	Vapor Fraction	Mass Flow Rate (kg·h ⁻¹)	Mass Enthalpy (kJ·kg ⁻¹)	Mass Entropy (kJ·kg ⁻¹ ·°C)
1	111	110	0	3000	−5405	4.60
2	119	15,000	0	3000	−5359	4.74
3	153	15,000	0	3000	−5247	5.56
4	174	15,000	0	3000	−5171	6.03
5	188	30,000	0	3000	−5118	6.11
6	268	30,000	1	3000	−4839	7.35
7	298	30,000	1	3000	−4734	7.72
8	171	120	0	23,461	317	−0.83
9	172	1200	0	23,461	319	−0.82
10	263	1200	1	23,461	884	1.73
11	223	500	1	23,461	843	1.79
12	223	500	1	21,115	843	1.79
13	223	500	1	2346	843	1.79
14	345	500	1	21,115	1033	2.47
15	284	120	1	21,115	939	2.58
16	147	120	0	21,115	269	−1.13
17	202	500	1	2346	746	1.32

18	229	120	0	23,622	−507	2.66
19	234	8000	0	23,622	−489	2.68
20	433	8000	1	23,622	130	4.52
21	354	2000	1	23,622	76	4.57
22	433	2000	1	23,622	237	4.98
23	340	120	1	23,622	90	5.13
24	340	120	1	20,787	90	5.13
25	340	120	1	2835	90	5.13
26	209	120	0	20,787	−548	2.47
27	229	120	1	2835	−206	3.97
28	269	3100	0	6113	−9265	2.15
29	286	20,000	0	6113	−9241	2.17
30	393	20,000	1	6113	−8987	2.93
31	618	20,000	1	6113	−8675	3.56
32	281	12,000	0	6113	−9247	2.18
33	473	12,000	1	6113	−8833	3.36
34	371	3100	1	6113	−8903	3.43
35	329	3100	1	6113	−8948	3.3
36	623	110	1	29,399	−566	6.06
37	483	110	1	29,399	−717	5.79
38	389	110	1	29,399	−814	5.56
39	328	110	1	29,399	−887	5.34

4.1. Results of Traditional Exergy Analysis

The exergy balance equations of each component in the target system are presented in Table 14, and Table 15 lists the results obtained from the detailed conventional exergy analysis of the target system.

The relative exergy destruction values $y_{D,K}^*$, acting as the primary criterion for the evaluation of the contribution of each device towards the overall exergy destruction of the system, represent a critical parameter in the improvement of the overall performance of the system. A total of 20 equipment components are evaluated in this study. Among them, Heat Exchanger 1, Expander 5 and Expander 4 exhibit remarkably high exergy destruction ratios, accounting for 48.40%, 13.81% and 7.46% of the total system exergy destruction, respectively, and they are the primary sources of system exergy destruction.

The main cause of exergy destruction inside Heat Exchanger 1 lies in the poor matching between the temperature and flow rate of the cold and hot fluids. The inlet temperature difference between the cold and hot fluids reaches 165 K, leading to a rapid degradation of energy quality during the heat transfer process, which constitutes the core source of exergy destruction. Meanwhile, the flow rate on the heat source side (21,115 kg/h) is much higher than that on the cold source side (3000 kg/h), which further exacerbates the imbalance of the heat transfer temperature difference and amplifies the exergy destruction caused by heat transfer. Under actual operating conditions, large mass flow rates, substantial temperature drops, and pressure drops of the working fluid will aggravate the internal irreversible losses inside expanders. For Expander 4, the extremely high working fluid mass flow rate of 23,622 kg/h brings severe flow losses and markedly increases its irreversible exergy destruction. As for Expander 5, its inlet and outlet pressures are 20,000 kPa and 12,000 kPa, respectively. The large pressure difference, together with the relatively high working fluid flow rate of 6113 kg/h, jointly result in its high exergy destruction.

Table 14. Exergy balance equation for components in the system.

Component	Exergy Balance Equation
P1	$E_1 + W_{P1} = E_2 + E_{D,P1}$
P2	$E_4 + W_{P2} = E_5 + E_{D,P2}$
P3	$E_8 + W_{P3} = E_9 + E_{D,P3}$
P4	$E_{18} + W_{P4} = E_{19} + E_{D,P4}$
P5	$E_{28} + W_{P5} = E_{29} + E_{D,P5}$
HE1	$E_2 + E_{15} = E_3 + E_{16} + E_{D,HE1}$
HE2	$E_3 + E_{13} = E_4 + E_{17} + E_{D,HE2}$
HE3	$E_5 + E_{25} = E_6 + E_{27} + E_{D,HE3}$
HE4	$E_9 + E_{24} = E_{10} + E_{26} + E_{D,HE4}$
HE5	$E_{12} + E_{38} = E_{14} + E_{39} + E_{D,HE5}$
HE6	$E_{19} + E_{35} = E_{20} + E_{28} + E_{D,HE6}$
HE7	$E_{21} + E_{37} = E_{22} + E_{38} + E_{D,HE7}$
HE8	$E_{34} + E_6 + E_{29} = E_{35} + E_7 + E_{30} + E_{D,HE8}$
HE9	$E_{36} + E_{32} + E_{30} = E_{37} + E_{33} + E_{31} + E_{D,HE9}$
K1	$E_{10} = E_{11} + W_{K1} + E_{D,K1}$
K2	$E_{14} = E_{15} + W_{K2} + E_{D,K2}$
K3	$E_{20} = E_{21} + W_{K3} + E_{D,K3}$
K4	$E_{22} = E_{23} + W_{K4} + E_{D,K4}$
K5	$E_{31} = E_{32} + W_{K5} + E_{D,K5}$
K6	$E_{33} = E_{34} + W_{K6} + E_{D,K6}$

Table 15. Results of traditional exergy analysis.

Equipment	$\dot{E}_{F,K}$ (kW)	$\dot{E}_{P,K}$ (kW)	$\dot{E}_{D,K}$ (kW)	$\dot{S}_{gen,K}$ (kW/K)	ϵ_K (%)	$\gamma_{D,K}$ (%)	$\gamma_{D,K}^*$ (%)
P1	38.33	3.55	34.78	0.12	9.26	90.74	0.69
P2	44.17	24.29	19.88	0.07	54.99	45.01	0.39
P3	32.46	13.03	19.43	0.07	40.14	59.86	0.38
P4	118.11	78.98	39.13	0.13	66.87	33.13	0.77
P5	40.75	30.63	10.12	0.03	75.17	24.83	0.20
HE1	2558.06	110.40	2447.66	8.21	4.32	95.68	48.40
HE2	53.44	28.11	25.33	0.08	52.60	47.40	0.50
HE3	75.59	39.26	36.33	0.12	51.94	48.06	0.72
HE4	1272.65	895.45	377.20	1.27	70.36	29.64	7.46
HE5	74.74	1.22	73.52	0.25	1.63	98.37	1.45
HE6	82.64	63.28	19.36	0.06	76.57	23.43	0.38
HE7	298.67	254.32	44.35	0.15	85.15	14.85	0.88
HE8	162.51	4.43	158.08	0.53	2.73	97.27	3.13
HE9	575.73	316.43	259.30	0.87	54.96	45.04	5.13
K1	383.78	267.19	116.59	0.39	69.62	30.38	2.31
K2	743.70	551.34	192.36	0.65	74.13	25.87	3.80
K3	209.97	53.46	156.51	0.52	25.46	74.54	3.09
K4	1258.02	964.57	293.45	0.98	76.67	23.33	5.80
K5	971.29	272.63	698.66	2.34	28.07	71.93	13.81
K6	154.30	118.86	35.44	0.12	77.03	22.97	0.70

4.2. Results of Advanced Exergy Analysis

The outcomes obtained from the exergy evaluation are presented in Table 16. Analysis of Table 16 reveals that avoidable exergy destruction ($\dot{E}_{D,K}^{AV}$) constitutes the dominant exergy loss component in pump systems, followed by exogenous exergy destruction ($\dot{E}_{D,K}^{EX}$). This indicates substantial optimization potential for pumps and their strong coupling with other system components. Instead of upgrading the pump's structure, reduc-

ing avoidable exogenous exergy destruction by changing related linked components seems like the most effective strategy for increasing the pump's exergy efficiency.

For heat exchangers, exergy losses fundamentally stem from energy quality degradation caused by temperature differences during heat transfer, resulting in irreversible thermal dissipation. Table 16 shows that Heat Exchanger 1 exhibits the lowest avoidable endogenous exergy efficiency ($\epsilon_K^{AV,EN}$) at 64%, with the highest relative avoidable endogenous exergy destruction ($y_K^{AV,EN}$) at 4.11%. This indicates that intrinsic performance restriction seriously restricts the whole system's exergy efficiency. So optimization efforts ought to focus mainly on structure and operations to reduce inherent losses and make the whole energy use better.

As core equipment for energy recovery and pressure conversion, expanders primarily incur exergy losses through irreversible expansion processes. Data in Table 16 indicate that all expanders achieve avoidable endogenous exergy efficiency ($\epsilon_K^{AV,EN}$) exceeding 90%, demonstrating near-optimal structural design with limited potential for further improvement via component-level modifications.

Table 16. Results of advanced exergy analysis.

Equipment	$\dot{E}_{D,K}^{UN}$ (kW)	$\dot{E}_{D,K}^{AV}$ (kW)	$\dot{E}_{D,K}^{EN}$ (kW)	$\dot{E}_{D,K}^{EX}$ (kW)	$\dot{E}_{D,K}^{UN,EN}$ (kW)	$\dot{E}_{D,K}^{UN,EX}$ (kW)	$\dot{E}_{D,K}^{EN,AV}$ (kW)	$\dot{E}_{D,K}^{AV,EX}$ (kW)	$\epsilon_K^{AV,EN}$ (%)	$y_K^{AV,EN}$ (%)
P1	9.14	25.64	13.51	21.27	8.54	0.60	4.97	20.67	0.50	0.31
P2	2.74	17.14	8.09	11.79	2.19	0.56	5.90	11.24	0.78	0.42
P3	3.51	15.92	8.35	11.10	3.07	0.43	5.28	10.65	0.76	0.16
P4	11.73	27.40	18.06	21.07	9.13	2.59	8.92	18.48	0.94	0.28
P5	1.29	8.83	2.12	8.00	1.10	0.19	1.03	7.81	0.96	0.08
HE1	1429.07	1018.59	1373.95	1073.71	742.85	686.22	631.09	387.49	0.64	4.11
HE2	2.69	22.64	6.76	18.57	2.32	0.37	4.44	18.20	0.88	0.27
HE3	21.62	14.71	23.08	13.25	13.70	7.92	9.38	5.33	0.80	0.61
HE4	275.18	102.02	235.73	141.47	216.93	58.25	18.80	83.22	0.98	0.28
HE5	28.19	45.33	16.09	57.43	13.74	14.45	2.35	42.98	0.80	0.12
HE6	12.61	6.75	12.10	7.26	11.10	1.51	1.00	5.75	0.97	0.37
HE7	27.01	17.34	23.81	20.54	20.47	6.54	3.34	14.00	0.99	0.24
HE8	112.05	46.03	115.01	43.07	100.85	11.20	14.16	31.87	0.95	0.27
HE9	207.76	51.54	197.12	62.18	173.25	34.51	23.87	27.67	0.97	0.24
K1	46.85	69.74	66.28	50.31	40.00	6.84	26.27	43.47	0.91	1.82
K2	120.92	71.44	124.23	68.13	87.75	33.17	36.48	34.96	0.90	2.50
K3	108.28	48.23	128.52	27.99	107.08	21.20	21.43	6.80	0.93	1.75
K4	190.57	102.88	163.88	129.57	153.42	37.15	10.46	92.42	0.98	0.55
K5	618.98	79.68	468.33	230.20	449.27	179.71	19.05	30.62	0.98	0.25
K6	30.45	4.99	22.61	12.83	17.48	12.98	5.14	0.15	0.96	0.31

To further clarify the results of the advanced exergy analysis, Figure 5 illustrates the unavoidable and avoidable components of exergy destruction for each piece of equipment in the target system. It appears that the unavoidable exergy destruction for each component is strictly greater than zero. Firstly, this result conforms to the basic postulates of the second law of thermodynamics in the context where the inherent properties of an irreversible process dictate that any real system must have some unavoidable exergy destruction. Secondly, it draws attention to the deficiency of traditional exergy analysis, where the exclusive basis for the assessment is on the sum of the exergy destruction. In other words, there would be no differentiation in regard to inherent unavoidable losses and losses amendable by means of advancing technologies. Therefore, to determine the improvement potential of system components, focus should be placed on the avoidable ex-

ergy destruction component of each component. Its proportion directly reflects the practical potential for improving exergy efficiency of the component through design modifications or parameter optimization.

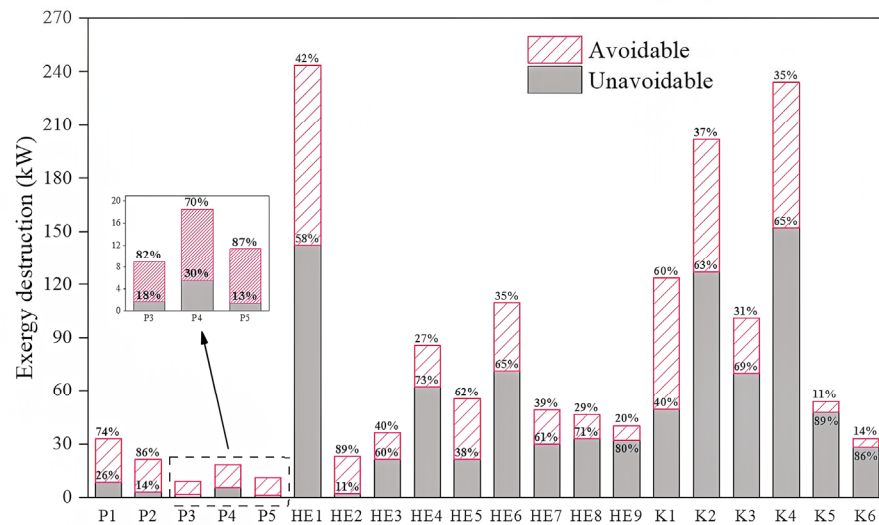


Figure 5. Avoidable/unavoidable exergy destruction based on advanced exergy analysis.

As shown in Figure 6, the data distribution on the destruction of exergy due to external as well as internal causes, within the system under consideration, has been presented comprehensively. It can be observed that the exogenous destruction of exergy in all components has a positively skewed distribution. This phenomenon reveals that reducing the associated exergy losses caused by interactions between components can effectively enhance the thermodynamic performance of each component. The components with the largest $\dot{E}_{D,K}^{EN}/\dot{E}_{D,K}$ are Expander 3 ($\dot{E}_{D,K3}^{EN}/\dot{E}_{D,K3} = 82.11\%$), Heat Exchanger 9 ($\dot{E}_{D,HE9}^{EN}/\dot{E}_{D,HE9} = 76.02\%$) and Heat Exchanger 8 ($\dot{E}_{D,HE8}^{EN}/\dot{E}_{D,HE8} = 72.75\%$), which indicates that more than 70% of the exergy destructions generated within these three components are due to their own inefficiencies.

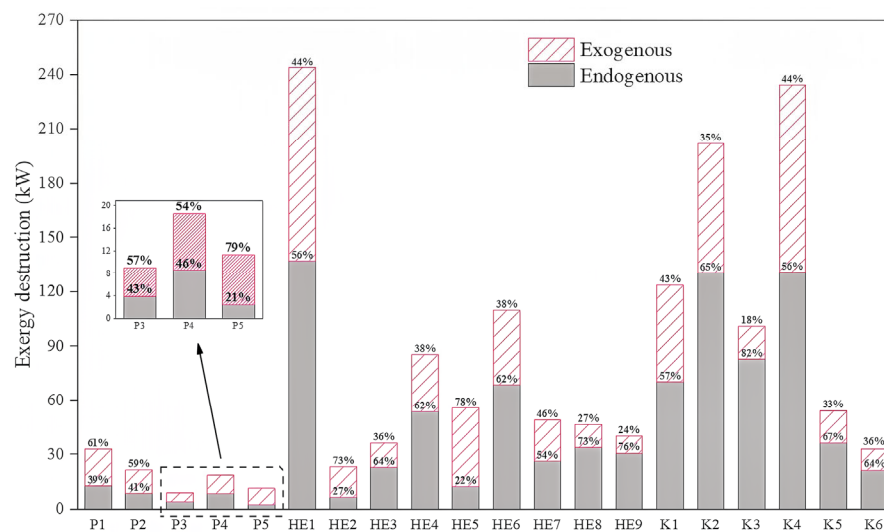


Figure 6. Exogenous/endogenous exergy destruction based on advanced exergy analysis.

According to the framework of dual-dimensional analysis incorporating exergy destruction in terms of “inevitability” and “avoidability” and being “exogenous” or “endogenous”, Figure 7 illustrates total exergy destruction in the system. The endogenous unavoidable exergy destruction accounts for the largest proportion, accounting for 43.81% of the total. The subsequent distribution consists of avoidable exogenous exergy destruction at 25.55%, followed by unavoidable exogenous and avoidable endogenous categories, representing 15.73% and 14.91%, respectively. The analysis demonstrates that endogenous exergy constitutes 58.72% of the total system exergy destruction ($\dot{E}_{D,tot}^{EN}/\dot{E}_{D,tot}$); endogenous exergy destruction accounts for over half of the total loss, so component optimization is the top priority for energy saving. Target high-loss units via exergy decomposition, and optimize structures and operating conditions of inefficient parts to reduce renovation costs. Rising endogenous exergy destruction aggravates internal irreversible losses and lowers component exergy efficiency. This impairs heat transfer and work output, and eventually reduces the system's net power, overall exergy efficiency and energy utilization. Conversely, lower endogenous exergy destruction means better component performance, which improves the thermal performance and cycle efficiency of the whole system.

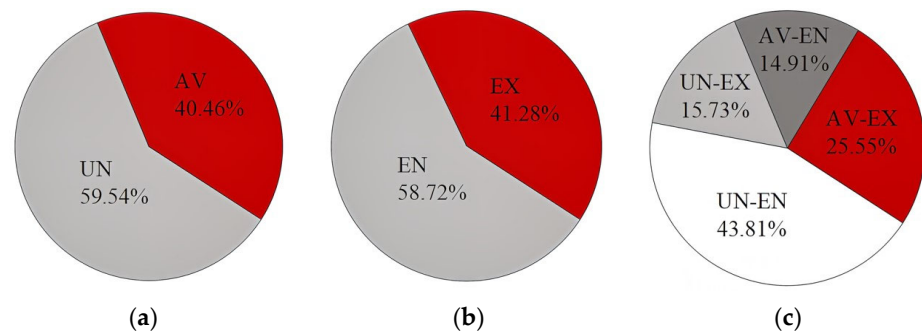


Figure 7. System exergy destruction ratio. (a) Unavoidable/avoidable; (b) Exogenous/endogenous; (c) Split-combination of unavoidable/avoidable and exogenous/endogenous.

In addition, the endogenous avoidable and exogenous avoidable components of exergy destructions should be of interest in the evaluation of thermal systems. These two components can reveal the real potential for system improvement and enable reduction through system optimization. The percentage of avoidable exergy destructions for the whole system is $\dot{E}_{D,tot}^{AV}/\dot{E}_{D,tot} = 40.46\%$, which serves as a quantitative indicator and represents the maximum achievable energy efficiency improvement for retrofitting. The avoidable endogenous exergy destruction can be reduced by optimizing individual components, while the avoidable exogenous exergy destruction can be cut via system coupling optimization, providing clear optimization routes. A decline in avoidable exergy destruction means reduced unnecessary energy consumption and irreversible losses, along with higher energy efficiency and better economic performance. In contrast, a rise indicates wasted energy-saving potential, increased exergy destruction, lower effective output under the same energy input, and higher operating energy consumption and costs.

4.3. Comparison of Traditional and Advanced Exergy Analysis Results

The comparative visualization in Figure 8 illustrates the differential assessments of relative exergy destruction across system components between conventional and advanced analytical approaches. As illustrated in the figure, a marked divergence exists between the prioritization outcomes of traditional and advanced exergy assessment methodologies. Both the traditional exergy analysis and the advanced exergy analysis identified Heat Exchanger 1 as the first component to be improved, but immediately afterwards,

the two analysis methods diverged. The traditional exergy analysis indicated that Expander 5 was the second most important component to be affected, while the advanced exergy analysis reveals that Expander 2 had the second highest improvement priority in terms of improving overall system performance due to its second highest avoidable exergy destruction.

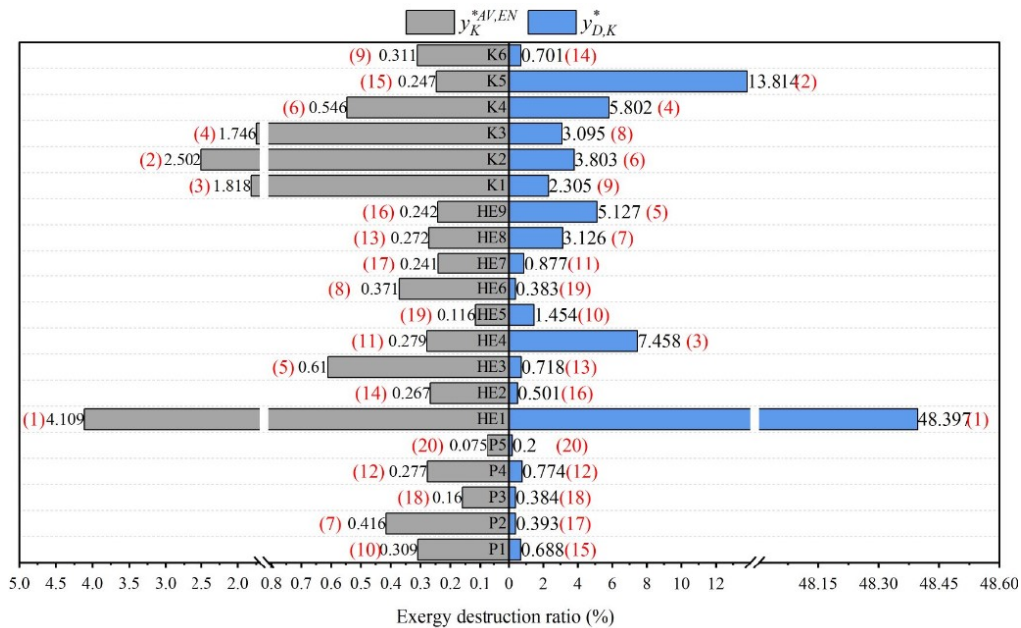


Figure 8. Comparison of the relative exergy destruction of each component in traditional exergy analysis and advanced exergy analysis.

According to the traditional exergy analysis, the priority order of system components improvement is: Heat Exchanger 1, Expander 5, Heat Exchanger 4, Expander 4, Heat Exchanger 9, Expander 2, Heat Exchanger 8, Expander 3, Expander 1, Heat Exchanger 5, Heat Exchanger 7, Pump 4, Heat Exchanger 3, Expander 6, Pump 1, Heat Exchanger 2, Pump 2, Pump 3, Heat Exchanger 6, Pump 5.

Advanced exergy analysis shows that the priority order of system component improvement is: Heat Exchanger 1, Expander 2, Expander 1, Expander 3, Heat Exchanger 3, Expander 4, Pump 2, Heat Exchanger 6, Expander 6, Pump 1, Heat Exchanger 4, Pump 4, Heat Exchanger 8, Heat Exchanger 2, Expander 5, Heat Exchanger 9, Heat Exchanger 7, Pump 3, Heat Exchanger 5, Pump 5.

The discrepancy in results between the two analytical methods stems from the differences in their evaluation frameworks. Conventional exergy analysis adopts relative exergy destruction as the core evaluation indicator, with its central hypothesis that the higher the proportion of a component's exergy destruction to the total system exergy destruction, the greater its improvement value. However, this assessment method has two prominent flaws. First, it does not differentiate between avoidable and unavoidable exergy destruction. Based on the second law of thermodynamics, every machine, even in real-world scenarios, involves unavoidable exergy destruction, which depends on thermodynamic constraints, material characteristics, and other parameters (e.g., exergy destruction due to the minimum temperature difference in heat transfer in heat exchanger units and innate entropy creation in expander units). This exergy destruction cannot be reduced by technical upgrading. If this unavoidable exergy destruction is not taken into account, conventional exergy destruction analysis could focus excessively on systems showing high total exergy destruction but low avoidable exergy destruction. Second, this

assessment method does not differentiate between exogenous and endogenous exergy destruction. Conventional exergy destruction analysis does not account for the coupling interactions of system components with other system units; as such, this method could neglect components showing moderate total exergy destruction but high endogenous exergy destruction.

On the contrary, advanced exergy analysis offers a varied evaluation method with a double-level decomposition of the total exergy destruction, which can overcome the problem brought by the traditional exergy analysis method. Classify exergy destruction into avoidable and unavoidable parts so that the method can correctly handle the elimination of unavoidable exergy destruction from an optimization point of view. And make sure the priorities for strengthening this improvement are still technically doable. An additional breakdown of exergy destruction according to whether it is endogenous or exogenous destruction enables an assessment that not only entails the inherent performance functionality of system components, but also takes into consideration the relationship among system components. Therefore, the procedure followed in the creation of optimization priorities transcends the analysis involved in individual system component performance.

It is very obvious that the results from advanced exergy analysis are more practical. Considering at the same time both the interaction between system parts as well as the technical boundaries of each individual part, the results yielded after conducting thorough exergy analysis greatly improve the engineering significance. And it can create a good foundation for developing strong and reasonable enhancement plans.

4.4. Safety Analysis

This power generation system with cascade utilization of medium-temperature flue gas waste heat and LNG vaporization cold energy is installed in the engine room of a large VLCC. Its working fluids include cryogenic liquid LNG, high-temperature flue gas, organic working fluids R1150/R127 and carbon dioxide. The main equipment consists of high-pressure pumps, shell-and-tube heat exchangers, expanders and pressure pipelines. Considering the fluid properties and operating conditions, the system is exposed to potential hazards such as cryogenic frostbite, leakage and combustion of combustible media, high-temperature thermal damage, failure of high-pressure components and alternating thermal stress of pipe walls. In accordance with the regulations of the IMO and major classification societies, a deterministic safety analysis method is adopted for the overall safety evaluation of the system. With standardized procedures and comprehensive hazard identification, this method complies with relevant marine equipment regulations and can effectively assess all potential safety risks [46].

4.4.1. Description of Analysis Method

The deterministic safety analysis adopted in this study follows the general evaluation procedures for marine thermal equipment and consists of four main steps. Firstly, all potential hazard sources and abnormal operating conditions throughout the system are fully identified. Secondly, the causes and hazard levels of various dangerous conditions are analyzed in combination with equipment reliability, actual operating parameters and human factors. Thirdly, targeted engineering protection facilities, safety interlock systems and standard operating procedures are deployed to mitigate or eliminate identified risks. Finally, the system design and operation schemes are verified item by item to ensure compliance with domestic, international and IMO regulations as well as technical standards for the marine industry [46].

4.4.2. Hazard Identification

Based on the overall process flow, physicochemical parameters of working media and rated operating conditions of the system, Hazard and Operability Analysis (HAZOP) is conducted successively for five major units, including the LNG delivery unit, flue gas heat exchange unit, ORC unit, TCO₂ cycle unit and rotating equipment unit. The operating deviations, fault causes, direct hazards and existing protective measures of each unit are clarified. The main hazard sources of the system are summarized in Table 17.

Table 17. HAZOP results of key system units.

System Unit	Operating Deviation	Main Causes	Potential Hazards	Existing Protective Measures
Two-stage LNG pressurization pipeline	Pressure exceeding 30 MPa	Regulating valve failure, pump runaway	Pipeline deformation and rupture, LNG leakage	High-pressure sensors, emergency relief valves, pressure interlock shutdown
LNG cryogenic pipeline	Leakage of cryogenic medium	Seal aging, pipeline defects	Personnel cryogenic frostbite, pipeline brittle fracture	Low-temperature material selection, regular seal inspection, cabin temperature monitoring
Flue gas heat exchange pipeline	Excessive wall temperature/Large temperature difference between cold and hot fluids	Working fluid cut-off, abnormal flue gas parameters	Pipe wall thermal fatigue and rupture with leakage	Temperature sensors, flue gas bypass regulation, high-temperature alarm
ORC pipeline (R1150/R1270)	Working fluid leakage and overpressure	Pipeline blockage, expander stalling	Combustible gas accumulation, fire and explosion risks	Safety valves, combustible gas detectors, graded alarm system
TCO ₂ pipeline	High-pressure leakage	Pressure-bearing component failure	Medium jet, cabin oxygen deficiency	Pressure monitoring, ventilation system, inert gas protection
Pumps, expanders and other rotating equipment	Excessive vibration, shaft seal leakage	Impeller wear, lubrication failure	Medium leakage, equipment interlock shutdown	Online vibration monitoring, shaft seal leak detection, regular maintenance

4.4.3. Deterministic Simulation Under Typical Worst-Case Scenarios

Based on the established Aspen HYSYS simulation model, deterministic calculations are carried out under typical Worst-Case Scenarios (WCSs). All boundary conditions are fixed to quantify parameter variations under extreme faults.

For condensation and corrosion risks on the flue gas side: The main engine of this vessel uses LNG as fuel, and its combustion products contain no sulfides, so sulfide corrosion can be ruled out. According to the thermodynamic properties tables of water and steam, the dew point temperature of flue gas is 324.15 K. Combined with the system flow diagram (Figure 4) and stream parameter table (Table 13), the operating temperatures of flue gas streams No. 36 to No. 39 are all higher than the dew point throughout the operation. Thus, no condensation or condensation corrosion occurs on the flue gas side of the system.

For LNG pipeline leakage conditions, it is assumed that the pipeline at the outlet of the two-stage LNG pressurization system ruptures completely. The rated leakage mass flow is 3000 kg/h, equivalent to a volumetric flow rate of approximately 7.14 m³/h for liquid LNG, and the ambient temperature is set to 298 K. The simulation results show that the minimum temperature in the leakage area drops to 111 K, posing a risk of brittle fracture to pipelines and surrounding equipment under low temperatures.

For extreme operating conditions of heat exchangers, Heat Exchanger 1, which has the largest exergy destruction in the system, is selected for verification. An extreme fault where the LNG flow on the cold side is completely simulated. The calculated instantaneous

ous maximum temperature difference in the pipe wall reaches 320 K, with the corresponding thermal stress of 186 MPa. Long-term alternating temperature differences will aggravate thermal fatigue damage to the pipe wall.

For overpressure conditions of circulating working fluids: Overpressure simulations are performed on the ORC subsystem and TCO₂ subsystem, respectively. When the system pressure exceeds the design value by 15%, the pipeline pressure reaches the critical alarm threshold, bringing potential failure risks to pressure-bearing components.

4.4.4. System Safety Deficiencies and Optimization and Prevention Measures

Based on the above analysis, the existing safety deficiencies of the system are summarized as follows. The enclosed cabin environment makes leaked LNG and organic working fluids hard to disperse and prone to accumulation. Heat exchangers are subject to long-term alternating temperature differences, which may lead to thermal fatigue failure. In addition, the system adopts multi-stage pressurization and high-pressure circulation design, resulting in a higher leakage risk at high-pressure seals compared with conventional marine equipment. Corresponding optimization and prevention measures are proposed as follows:

1. Equipment structure optimization: Low-temperature and high-pressure pipelines are made of high-strength low-temperature alloy steel. Combined sealing structures are adopted for high-pressure sections. Thermal compensation components are installed on heat exchangers to relieve alternating thermal stress.
2. Safety system deployment: Temperature, pressure, and flammable gas sensors are arranged throughout the system, together with multi-stage audible and visual alarms, emergency pressure relief and interlock shutdown devices. A forced ventilation system is equipped in the engine room to accelerate the diffusion of hazardous gases.
3. Operation parameter control: The load change rate of the equipment is strictly regulated. The pressurization is optimized according to simulation results to avoid sharp fluctuations of pressure and temperature.
4. Operation and maintenance enhancement: Regular inspections, pressure tests and safety valve calibrations are implemented. Specialized safety and emergency response training is provided for crew members.

5. System Optimization and Economic Benefit Analysis

5.1. Parameter Optimization Based on Genetic Algorithm

The simulation results indicate that multiple coupled parameters, such as evaporation pressure and condensation pressure of each cycle, jointly affect system performance. Single-parameter adjustment cannot achieve the optimal design. For this multi-parameter nonlinear optimization problem, the genetic algorithm is adopted for simultaneous multi-variable optimization [47]. This study takes system exergy efficiency as the objective function to obtain the optimal combination of parameters and realize the global optimum of thermal performance.

Figure 9 presents the workflow of parameter optimization based on the genetic algorithm (GA). In this study, MATLAB R2021b is coupled with Aspen HYSYS to adjust and optimize the key decision variables via the genetic algorithm. Firstly, the independent variables and their value ranges are defined, and relevant parameters are initialized. Afterwards, a numerical simulation is performed in Aspen HYSYS. If the simulation fails to converge, the population of the genetic algorithm is updated, the parameters are reinitialized, and the above procedures are repeated. Once convergent simulation results are obtained, the data are transmitted back to MATLAB for constraint checking. If the con-

straints are not satisfied, a penalty function is applied. If all constraints are met, the results are taken as the fitness function for subsequent iterative evolution until the termination criterion is reached. Finally, it is judged whether the current solution is optimal. If not, the GA population is updated and the entire workflow is repeated. If the optimal solution is acquired, the results are decoded and exported.

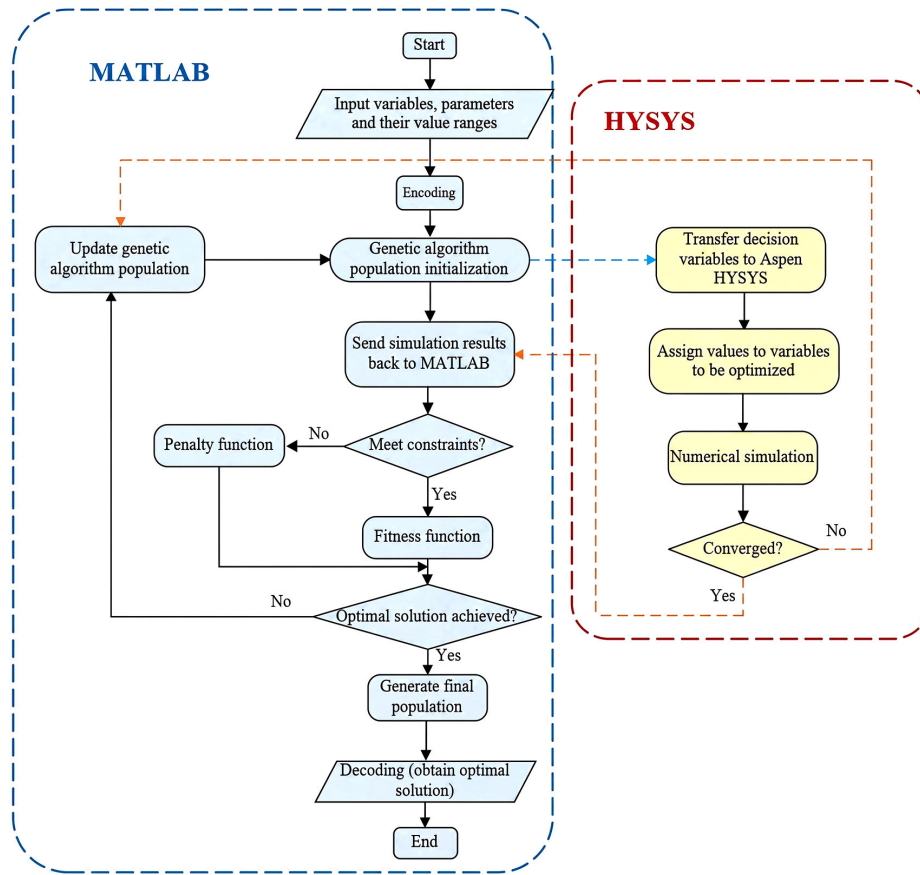


Figure 9. Parameter optimization workflow diagram based on genetic algorithm.

Different from conventional exergy analysis, focusing on relative exergy destruction, advanced exergy analysis reveals the actual energy-saving potential and optimization priorities via relative avoidable endogenous exergy destruction. Accordingly, components with relatively avoidable endogenous exergy destruction above 0.5% are selected for parameter optimization, including Heat Exchanger 1, Expander 2, Expander 1, Expander 3, Heat Exchanger 3, and Expander 4. The optimized parameters cover the condensation pressure of Heat Exchanger 1 (equal to the outlet pressure of Expander 2), the outlet pressures of Expander 1, Expander 3 and Expander 4 (equal to the condensation pressure of Heat Exchanger 3), and the condensation pressure of Heat Exchanger 3. In addition, the primary pressurization pressure of LNG is also optimized to determine the proper intermediate pressure of two-stage pressurization and reduce the exergy destruction of pumps.

Engineering and thermodynamic constraints are imposed to guarantee physically feasible operating conditions:

1. The outlet pressure of LNG after two-stage pressurization is set to 30 MPa, complying with the rated intake pressure of the ME-GI main engine.
2. The minimum temperature difference between hot and cold streams of all heat exchangers is no less than 5 K to prevent heat transfer pinch points.

3. The subcooling degree on the condensation side of each cycle is maintained above 2 K, consistent with the initial simulation settings.
4. The isentropic efficiency of all pumps and expanders remains constant at 0.75. Only operating conditions are optimized while equipment performance parameters are fixed.

On the premise of ensuring the normal operation of the entire system, the upper and lower limits of each sensitive parameter in the design scheme system were determined through gradual adjustment via single-parameter simulation, and the specific results are shown in Table 18.

Table 18. Range of sensitive parameters in the system.

Object	Lower Limit	Upper Limit
First-stage LNG boosting pressure (kPa)	11,000	16,500
HE1 Condensation Pressure (kPa)	110	140
K1 Outlet Pressure (kPa)	400	700
K3 Outlet Pressure (kPa)	1800	2300
HE3 Condensation Pressure (kPa)	100	130

The genetic algorithm in the MATLAB toolbox and Aspen HYSYS software are jointly used to optimize the system parameters for maximum exergy efficiency. Table 19 lists the core configuration parameters adopted for the genetic algorithm in this work. The optimized sensitive parameters are presented in Table 20.

Table 19. Main parameters of genetic algorithm.

Parameter	Value
Population size	100
Crossover rate	0.8
Mutation rate	0.1
Number of iterations	50

Table 20. Optimized values of system-sensitive parameters.

Object	Optimized Numerical Values
First-stage LNG boosting pressure (kPa)	14,000
HE1 Condensation Pressure (kPa)	132
K1 Outlet Pressure (kPa)	458
K3 Outlet Pressure (kPa)	2150
HE3 Condensation Pressure (kPa)	115

Table 21 presents a comparison of the results between the original scheme and those after parameter optimization. As can be seen from Table 14, following parameter optimization based on the genetic algorithm, the total output power of the system increased from 1678.77 kW to 1772.06 kW, the total exergy loss decreased from 2335.50 kW to 1895.28 kW, and the overall exergy efficiency rose from 41.82% to 48.32%. The entire reduction in total exergy loss stemmed from the decrease in avoidable exergy loss; this is because optimizing the key operating parameters of each component in the LNG-fueled ship energy utilization system can only reduce its avoidable exergy loss. Owing to economic and technical constraints, there is currently no available method to mitigate the unavoidable exergy loss of the system. In addition, the net power generation per unit mass of LNG increases from 0.5596 kWh/kg-LNG to 0.5907 kWh/kg-LNG, and the LNG cold energy recovery rate improves from 62.1% to 67.3%.

Table 21. Comparison of original and optimized system performance.

	Original System	Optimized System
Total net output power (kW)	1678.77	1772.06
Total exergy destruction (kW)	2335.50	1895.28
Exergy efficiency (%)	41.82	48.32
Net power generation per unit LNG (kWh/kg-LNG)	0.5596	0.5907
LNG cold energy recovery rate (%)	62.1	67.3

5.2. System Cost and Economic Benefit Analysis

For the two-stage pressurization and three-stage longitudinally nested full-power generation system proposed in this study, although the thermodynamic performance of the system has been analyzed and optimized based on the results of advanced exergy analysis, economic efficiency is the key constraint for engineering applications. While fully recovering the medium-temperature waste heat and LNG cold energy of the ship, it is necessary to balance the goal of low-cost power generation. According to the established economic model, the capital investment cost of each component of the designed system is calculated and shown in Table 22 below, and the economic benefit analysis of the system is presented in Table 23. Based on the assumption that the ship power generation system operates 300 days a year, the system achieves an annual net income of 3.8336 million CNY, and the static investment payback period is 5.49 years.

Table 22. Capital investment cost of each component of the system.

Component	Capital Costs (10 ⁴ CNY)
Pump	54.28
Heat exchanger	62.31
Turbine	1869.50
Total	1986.09

Table 23. Economic benefit analysis of the system.

Index	Value
Total output power of the system (kW)	1678.77
Exergy efficiency (%)	41.82
Capital cost (10 ⁴ CNY)	1986.09
Total annual net income (10 ⁴ CNY)	383.36
Payback Period (years)	5.49

6. Conclusions

To improve the energy utilization efficiency of ships, this paper takes a 215,000-ton VLCC as the research object, and innovatively proposes a fully power-generating system with two-stage supercharging and three-stage longitudinal nesting, which is based on the ORC and TCO₂ Rankine cycle. The system achieves cascade coupling between the medium-temperature flue gas waste heat of the main engine and the LNG vaporization cold energy. Furthermore, this study applies the advanced exergy analysis method to the thermodynamic performance analysis and optimization of such integrated systems for the first time. Through comparative analysis of thermodynamic performance, multi-parameter global optimization and economic evaluation, the following conclusions are drawn:

- (1) There are significant differences between traditional exergy analysis and advanced exergy analysis in determining the priority of system component improvement, and advanced exergy analysis is more capable of revealing the essential optimization di-

rection of the system. Taking the relative exergy loss as the index, traditional exergy analysis proposes the following improvement priority: Heat Exchanger 1, Expander 5, Heat Exchanger 4, Expander 4, Heat Exchanger 9, Expander 2, Heat Exchanger 8, Expander 3, Expander 1, Heat Exchanger 5, Heat Exchanger 7, Pump 4, Heat Exchanger 3, Expander 6, Pump 1, Heat Exchanger 2, Pump 2, Pump 3, Heat Exchanger 6, Pump 5. By dividing exergy loss into endogenous/exogenous and avoidable/unavoidable exergy loss, advanced exergy analysis clarifies the interaction among various components and their real improvement potential. Taking the relative avoidable endogenous exergy loss as the core index, it determines the improvement priority as follows: Heat Exchanger 1, Expander 2, Expander 1, Expander 3, Heat Exchanger 3, Expander 4, Pump 2, Heat Exchanger 6, Expander 6, Pump 1, Heat Exchanger 4, Pump 4, Heat Exchanger 8, Heat Exchanger 2, Expander 5, Heat Exchanger 9, Heat Exchanger 7, Pump 3, Heat Exchanger 5, Pump 5. Meanwhile, the results of advanced exergy analysis reveal that the proportion of avoidable exergy loss in the entire system is 40.46%, indicating that the system has an improvement potential of 40.46%; the proportion of endogenous exergy loss is 58.72%, suggesting that the interaction among various components of the system is relatively weak and there is a certain degree of independence between each component, which provides a theoretical basis for the targeted optimization of the system.

- (2) Based on the component improvement priority determined by advanced exergy analysis, a genetic algorithm was adopted for the global optimization of key sensitive parameters such as the initial LNG pressurization pressure, heat exchanger condensation pressure, and expander outlet pressure, resulting in a significant improvement in the system's thermodynamic performance. After optimization, the total output power of the system increased from 1678.77 kW to 1772.06 kW, with a growth rate of 5.56%; the exergy efficiency rose from 41.82% to 48.32%, an increase of 15.54%. Meanwhile, the total exergy loss decreased from 2335.50 kW to 1895.28 kW, and the entire reduction in total exergy loss was derived from the optimization of avoidable exergy loss. This verifies the precise guiding role of the advanced exergy analysis method in identifying the optimization direction of the system.
- (3) Economic evaluation verifies that the integrated system with optimized thermodynamic performance based on advanced exergy analysis results possesses favorable feasibility for engineering application. Calculated on the basis of an annual operation period of 300 days, the total initial capital investment is 19.8609 million CNY, the annual net income is 3.8336 million CNY, corresponding to a static payback period of 5.49 years, which falls within the reasonable profit range for ship energy-saving retrofit projects. This result indicates that the system not only realizes efficient energy recovery but also achieves the dual goals of satisfactory return on investment and low-cost operation.

This study introduces advanced exergy analysis into the thermodynamic performance analysis and optimization of the integrated system for marine medium-temperature waste heat and LNG cold energy for the first time. It breaks through the limitations of traditional exergy analysis, which is unable to quantify the interaction between components and technical constraints, thereby providing a novel method for the precise optimization of such coupling systems. Based on the assumptions of steady-state system operation and negligible heat leakage losses in heat exchangers, this study proposes a dual-stage pressurization system with a three-level longitudinally nested architecture. By integrating ORC with the TCO₂ Rankine cycle, it achieves cascaded utilization of medium-temperature waste heat and LNG cold energy. This design effectively addresses challenges in vessel engine rooms—especially spatial constraints and the limitations of single-cycle systems in adapting to broad temperature-range energy exchange. The research out-

comes not only offer technical references for energy recovery system designs in large LNG-powered vessels such as VLCCs but also establish a foundation for advancing the application of sophisticated exergy analysis in marine thermodynamic systems. Furthermore, they hold significant practical value for enhancing ship energy efficiency and reducing operational costs.

Author Contributions: Conceptualization, S.Y. and Y.W.; methodology, S.Y.; software, Y.H.; validation, Y.H., Y.W. and X.D.; formal analysis, Y.H.; investigation, X.D.; resources, Y.H.; data curation, Y.H.; writing—original draft preparation, Y.H.; writing—review and editing, Y.H.; visualization, Y.H.; supervision, Y.S. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Data Availability Statement: Data is contained within the article.

Acknowledgments: The study gained adequate support in terms of resources from the School of Energy and Power Engineering, Jiangsu University of Science and Technology. Specifically, the study thanks Shouguang Yao for his valuable input and assistance at various stages of the study. The inputs of Huailiang You and Daifen Chen in research methodology and data analysis are also greatly acknowledged. Finally, the study greatly appreciates the inputs of Yue Wei in the manuscript preparation. The study team finally thanks the editorial board and reviewers of *Energies* for their valuable inputs at various stages of the publication process.

Conflicts of Interest: The authors declare no conflicts of interest.

Abbreviations

The following abbreviations are used in this manuscript:

List of Abbreviations

HE	Heat exchanger
K	Expander
LNG	Liquefied natural gas
ME-GI	Main engine-gas injection
ORC	Organic Rankine cycle
P	Pump
CO ₂	Carbon dioxide
TCO ₂	Transcritical carbon dioxide Rankine cycle
TEU	Twenty-feet equivalent unit

Nomenclature

E	Exergy, kW
$E_{D,K}$	Exergy destruction of the Kth component, kW
$E_{F,K}$	Fuel exergy of the Kth component, kW
$E_{P,K}$	Product exergy of the Kth component, kW
e	Exergy of mass, kW/kg
h	Enthalpy of mass, kJ/kg
m	Mass flow, kg/h
P	Pressure, kPa
Q	Heat transfer rate, kW
s	Entropy of mass, kJ/(kg·K)
T	Temperature, K
W	Power, kW
W_{net}	Net power, kW
η	Energy efficiency, %
ε	Exergy efficiency, %
$\gamma_{D,K}$	Exergy destruction of the Kth component, %
$\gamma_{D,K}^*$	Relative exergy destruction of the Kth component, %
0	Ambient conditions

<i>ch</i>	Chemistry
<i>D</i>	Destruction
<i>in</i>	Inlet
<i>k</i>	Component k
<i>out</i>	Outlet
<i>ph</i>	Physics
<i>p</i>	Pump
<i>Q</i>	Heat
<i>t</i>	Turbine
<i>tot</i>	Total
<i>W</i>	Power generation

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