DESIGN STUDY ON THE INTEGRATED UTILIZATION SYSTEM OF MEDIUM TEMPERATURE WASTE HEAT AND LNG VAPORIZATION COLD ENERGY FOR 200000 DWT LNG-POWERED VESSELS

by

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Original scientific paper
https://doi.org/10.2298/TSCI220326146Y

The study object for this work is a 215000 ton very large crude carrier – liquefied natural gas – powered vessel, intending to integrate the use of medium temperature flue gas waste heat from the exhaust turbine and cold energy from liquefied natural gas vaporization. It proposes a Rankine cycle power generating system with a two stage booster and three stage lateral nesting following the principle of "temperature matching, stepped utilization", taking into account real demands and circumstances of the vessel. The study shows that in this tonnage vessel, through the design and optimization of the stepped utilization scheme, the cold energy released during the vaporization of liquefied natural gas fuel from the ME-GI host machine and the medium temperature waste heat from the exhaust turbine can be fully utilized, and the system structure is tight and simple. After the non-azeotropic mixed working media was optimized and the operational parameters were optimized using genetic algorithm, the system designed in this paper can reach 54.61% exergy efficiency and 187.83 kW net output of power generation. The annual income of the final designed system can reach CNY 1, 133, 240. The capital recovery cycle is expected to be 5.06 years if the system is put into operation.

Key words*: liquefied natural gas, vaporization cold energy, exergy efficiency, waste heat utilization, parameter optimization*

Introduction

As the economy and society have progressed, the globe put forward increasingly stringent requirements for environmental quality, energy conservation, and emission reduction. liquefied natural gas (LNG) has shown great advantages in replacing the use of traditional energy sources such as coal and oil [1]. When LNG vaporizes, its high quality low temperature cold energy is very precious. During the complete conversion of 1 ton of LNG into gaseous natural gas, the cold energy released is as high as 830 MJ. If all this energy is converted into electricity, it is about 231 kWh [2]. Therefore, if this cold energy is not used, not only will there be a massive waste of energy, but it will also pollute nearby nature.

Up to now, the utilization of LNG vaporized cold energy is mostly aimed at land-based LNG vaporization stations, and the utilization of vaporized cold energy includes seawater

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desalination [3-6], cold storage [7-9], air conditioning [10, 11], and power generation [12-15]. In shipping LNG-powered vessels, the use of LNG vaporized cold energy is still in the early stages of research. Li *et al.* [16] took a very large crude carrier (VLCC) as the research object, and the cold storage and air conditioning on board were powered by LNG cryogenic energy. The simulation analysis shows that the LNG cryogenic energy utilization can significantly save shipping costs. Subsequently, Li *et al.* [17] further discussed several LNG cryogenic energy power production methods of LNG carrier for LNG carrier with steam power propulsion device and simulated and analyzed the power and exergy efficiency that can be generated under different schemes. The results show that using a cold energy power generation system can save a lot of operating costs and obtain very good economic benefits. Ma *et al.* [18] proposed an air conditioning system for a 3000 DWT LNG-diesel dual-fuel power ship using LNG cryogenic energy, and the simulation analysis revealed that the system can significantly cut down on operating energy consumption.

The aforementioned studies mostly targeted at the LNG cryogenic energy utilization of single-purpose, and the energy efficiency of LNG is low. After that, Li *et al.* [19] devised a comprehensive cold energy utilization scheme including vessel air conditioning, seawater desalination, Rankine cycle power generation, and vessel cold storage, taking VLCC as a representative vessel type under design and combining it with vessel flue gas waste heat. Yao *et al.* [20, 21] proposed a variety of energy systems stepped utilization schemes for 25000 DWT and 16300 DWT LNG fuel-powered chemical ships, combining seawater desalination, cold storage, power production, and air conditioner, and taking into account the existing residual heat resources of the ship. Through simulation analysis, the schemes are evaluated from two aspects of economical efficiency and exergy efficiency, and the optimization system scheme suitable for the integrated utilization of LNG cryogenic energy of corresponding ships is established. Han *et al.* [22] put forward a power generation process using LNG vaporization cold energy for a three-stage Rankine cycle for LNG power vessel with 190 °C flue gas waste heat, 90 °C cylinder liner cooling water, and seawater as heat sources.

The existing literature research has shown that the combination of medium temperature waste heat from LNG-powered vessels and the cold energy released during LNG vaporization, and the proposed energy utilization scheme from the *temperature matching, stepped utilization* point of view, can bring considerable economic benefits and is following the green shipping development trend. It is worth pointing out that although the existing literature has designed LNG cold energy utilization schemes for some specific ships. However, combined with the medium temperature flue gas residual heat discharged from the exhaust turbine, the research on process design, working fluid optimization and operating parameter optimization of LNG energy comprehensive utilization system is still blank.

Based on this, this study takes a 200000 DWT large VLCC-LNG-powered vessel as the research object. It combines the medium temperature flue gas waste heat from exhaust turbine, takes the phased usage of LNG vaporization cryogenic energy as a target, and carries out the design research of the efficient integrated utilization scheme of the whole vessel energy from the energy gradient utilization point of view. By designing a set of integrated utilization system of the whole power generation, and then optimizing the working fluid by using non-azeotropic mixed working fluid and optimizing the operating parameters by using Genetic Algorithm, to provide an optimal design solution for the integrated utilization of cryogenic energy of 200000 DWT LNG-powered vessels.

Vessel technical parameters and proposed stepped utilization system

Vessel technical parameters

In this essay, a 215000 DWT VLCC-LNG-powered vessel is selected as the research object, which uses LNG fuel instead of fuel oil, and the host machine of the vessel is ME-GI, and the system design parameters are taken at 85% load. The main specific engine parameters [19, 23] are listed in tab. 1.

Table 1. Host machine-related parameters

Parameters	Values
Host machine power	25480 kW
Air intake volume	3000 kg/h
Inlet pressure of host machine	30 MPa
Intake air temperature of host machine	20 °C
Waste heat temperature of flue gas	350 °C

System scheme design

Since the Rankine cycle is the most economical and efficient way to utilize high grade LNG cold energy, from the viewpoint of *temperature matching, stepped utilization*, according to the distribution of LNG cryogenic exergy across a wide range of temperatures, this paper adopts the following design mentality to build the system. According to LNG vaporization curves under different pressures [24], Rankine cycle working medium at different stages shall be matched with corresponding LNG vaporization curves as far as possible, so as to minimize exergy destruction caused by heat transfer temperature difference. This paper adopts the method of level-by-level segmentation. Firstly, a two-stage low temperature Rankine cycle power generation system is designed for temperatures ranging from –162 °C to –40 °C to fully use the cold energy of the LNG liquid phase and wet steam section. Comprehensively considering the thermodynamic properties, environmental properties, at the initial stage, physicochemical performance, and safety features of the circulating working fluid, the first and second Rankine cycle working fluids are identified to be ethane and propane, both of which have a high latent heat of vaporization [25]. Then, a compact CO_2 trans critical Rankine cycle is implemented between the temperature of –40 °C and the exhaust gas discharged from the exhaust turbine of vessel to improve usage efficiency of the low grade thermal energy of the exhaust gas and the LNG cryogenic energy in the shallow cold zone. In addition, considering that the intake pressure of the host machine is 30 MPa, while the pressure of LNG in storage tank is only 110 kPa. The single-stage pressurization will cause the LNG high pressure pump to consume a lot of electrical power and bring huge exergy destruction the system, so secondary pressurization between the Rankine cycle power production system is considered to meet the inlet requirements of the host machine. Finally, due to the limited space in the cabin of the vessel, when designing the LNG-based cryogenic energy power production system, the number and size of equipment should be reduced as much as possible to form a compact system.

Based on the aforementioned design ideas, this paper constructs a Rankine cycle power generating system with a two-stage booster and three-stage lateral nesting, which is able to meet the requirements of multiple utilization of medium temperature residual heat and LNG vaporization cryogenic energy of 200000 DWT LNG-powered VLCC. Figure 1 depicts the final proposed system process flow.

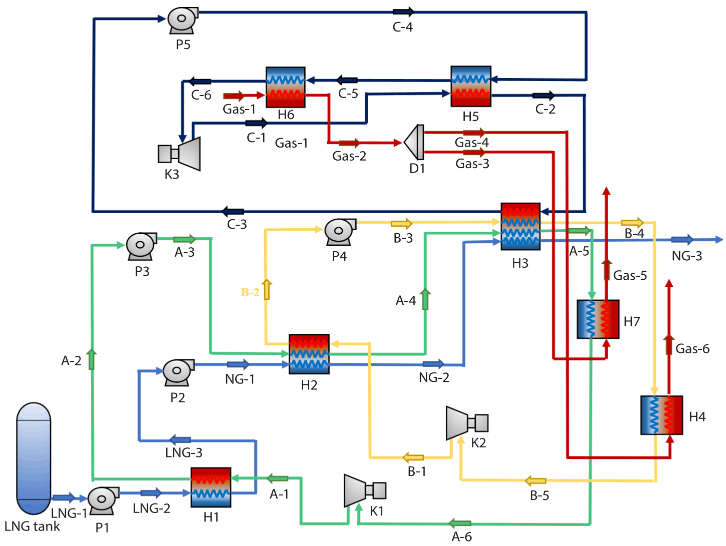

Figure 1. Integrated system design flow chart

The specific workflow of the system is as follows.

The LNG process: LNG-1 discharged from the low pressure storage tank flows into heat Exchanger 1 (H1) for heat exchange after being pressurized to 15 MPa for the first time. The cooling capacity needs of the first stage power generation are met firstly with high quality low temperature cooling energy. Following that, the flow passes via a high pressure pump for secondary pressurization 30 MPa, heat Exchangers 2 (H2) and 3 (H3) for heat exchange, and ultimately, flows into the vessel host machine for combustion.

First stage Rankine cycle: heat Exchanger 1 receives ethane from Expander 1 (K1) for heat exchange with LNG, after pump pressurization, it passes via heat Exchangers 2 and 3 for heat exchange with the second stage Rankine cycle, the third stage trans-critical CO_2 Rankine cycle, then heat Exchanger 7 (H7) for heat exchange with flue gas, and finally enters expander 1 for expansion complete the cycle.

Second stage Rankine cycle: propane from Expander 2 (K2) is condensed in heat Exchanger 2 and pumped by the pump. After that, it passes via heat Exchanger 3 for heat exchange, heat Exchanger 4 (H4) for heat exchange with flue gas, and Expander 2 for expansion finish the cycle.

Trans-critical CO_2 Rankine cycle: CO_2 enters heat Exchanger 5 (H5) from Expander 3 (K3) for precooling before being cooled by heat Exchanger 3 and pumped by the pump. The CO_2 then absorbs heat through heat Exchangers 5 and 6 (H6), before entering Expander 3 to expand and finish the cycle.

System simulation and analysis

Simulation parameters

Table 2 shows the LNG components which are used in system simulation.

The input assumptions of the simulation are:

– The ambient temperature is 25 °C.
– The undercooling of Rankine cycle working fluid is 2 °C.
– Peng-Robinson-Stryjek-Vera equation is selected as the state equation during simulation.

– The adiabatic efficiency of the pumps and the isentropic efficiency of the expanders are both 0.75.
– The minimum end difference of heat exchangers is 4 °C, ignoring the heat leakage losses and pressure drops in the heat exchangers.

Table 2. The LNG components

Unit	Methane	Ethane	Propane	Nitrogen
Volume proportion	0.9228	0.0594	0.0118	0.006

Exergy analysis

Under the condition of meeting the aforementioned parameters and constraints, the exergy analysis of the vessel LNG cryogenic energy and flue gas waste heat utilization system designed in this paper is carried out. Table 3 shows the calculation findings of the exergy analysis of the design system.

Table 3. System exergy analysis and calculation results

Equipment	Pay exergy [kW]	Exergy received [kW]	Exergy destruction [kW]	Exergy efficiency
H1	186.42	114.76	71.66	61.56%
H2	87.21	76.11	11.09	87.28%
H3	60.59	45.54	15.06	75.15%
H4	3.96	1.33	2.63	33.58%
H5	9.21	1.96	7.25	21.29%
H6	223.98	199.55	24.43	89.09%
H7	2.36	0.78	1.58	33.11 %
P1	36.82	4.87	31.95	13.22%
P2	47.92	25.57	22.35	53.35%
P3	0.59	0.30	0.29	50.47%
P4	0.41	0.26	0.15	64.24%
P5	42.41	30.10	12.31	70.97%
K1	50.57	34.38	16.19	67.99%
K2	39.14	27.94	11.20	71.39%
K3	252.69	203.60	49.09	80.57%
Total turbine output power [kW]		265.92		
System exergy efficiency		49.32%		

At this phase, the total exergy efficiency of the system is 49.32%, while the sections with the larger exergy destruction are focused on the heat exchangers. It is worth mentioning that the LNG heat absorption exergy destruction at low temperatures in heat Exchanger 1 is 71.66 kW, with a heat absorption exergy efficiency of 61.56%, which is similarly low. Therefore, it is necessary to optimize the working medium to decrease the exergy destruction of the whole system and further improve the exergy efficiency of the whole system through the matching and optimization of system operating parameters.

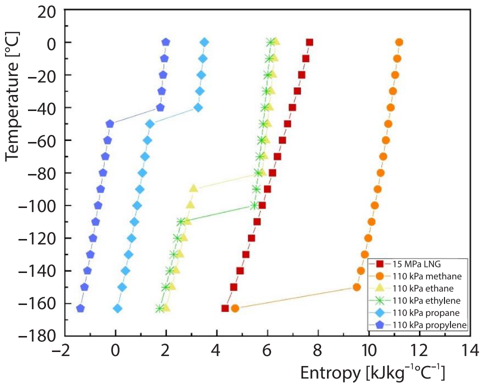

Figure 2. Tephigram of LNG (15 MPa) and common single working fluid (110 kPa)

Optimization of parameters and analysis of the results

Working fluid optimization

The non-azeotropic mixed working fluid is utilized to replace the single working fluid in the first and second Rankine cycles to further minimize exergy destruction and maximize the exergy efficiency in heat exchangers. The vaporization diagram of various typical single working fluids and LNG are shown in fig. 2. Obviously, the vaporization curves of ethane, methane, and LNG can be seen to be closer, and the vaporization curve of methane and LNG are the closest in the low temperature section (–160 °C to –150 °C). In the medium temperature section (–150 °C to –80 °C), the vaporization curve of LNG is close to ethane and deviates greatly from methane, so propane is added for adjustment. Finally, methane, ethane, and propane are selected as the mixed working fluid.

The ratio of non-azeotropic mixtures (methane: ethane: propane) in the first stage circulating is changed by Aspen HYSYS software. Judging from the simulation results, when the methane ratio is set to 40%, the mixed working fluid in the lower temperature zone can better fit the vaporization curve of LNG. Thus, the vaporization diagram of mixed working fluid and LNG is drawn, as shown in fig. 3, when the ratio of mixed working fluid is 4:5.5:0.5, 4:5:1, 4:4:2, and 4:3:3. After simulation and analysis, the best non-azeotropic mixtures ratio of the first stage circulating is methane: ethane: propane = 4:5.5:0.5. Meanwhile, in H1, the temperature-heat flow curve of cold and hot fluids is shown in fig. 4.

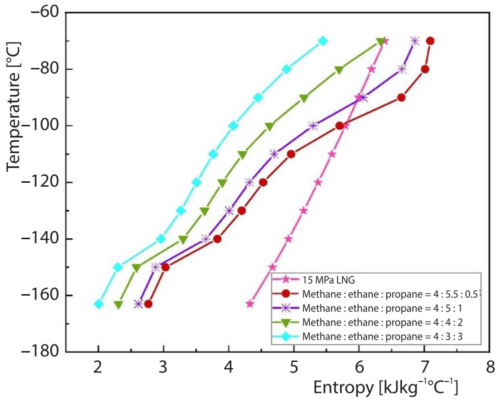

Figure 3. Tephigram of LNG (15 MPa) and mixed working fluid (110 kPa)

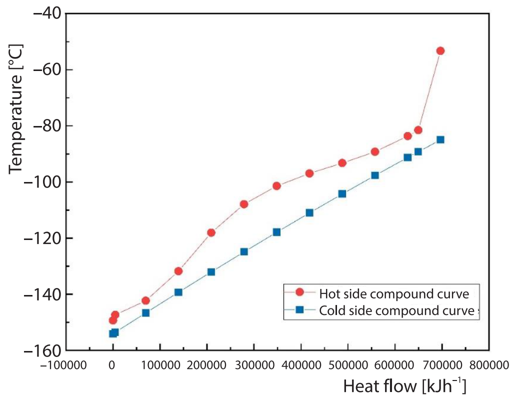

Figure 4. Temperature-heat flow diagram of cold and hot fluids in H1

The mixed working fluid composed of methane, ethane, propane, and pentane can be used in the second Rankine power generation cycle [13]. Figure 5 shows the temperature-heat flow curve of cold and hot fluids in H2 when the volume proportion of mixed working fluid of the system after simulation and adjustment is methane, ethane, propane, and pentane = 0.5:2.5:4.5:2.5. The heat transfer temperature difference between cold and hot fluids has been greatly decreased by the mixed working fluid of the first and second stage Rankine cycles, as shown in figs. 4 and 5.

Table 4 shows the simulation and comparative analysis results of the overall turbine output power and the overall system exergy efficiency before and after the working fluid optimization. The overall power output of the turbine is enhanced by 13.91 kW after employing the non-azeotropic mixtures, compared to the scheme using the single working fluid. The overall system exergy efficiency is also improved to 51.79%, a 2.47% increase over what it was before working fluid modification.

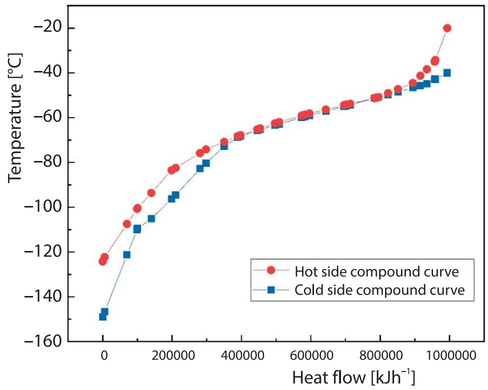

Figure 5. Temperature-heat flow diagram of cold and hot fluids in H2

Table 4. Comparison between optimized working fluid and original scheme

Scheme	Total turbine output power [kW]	Total exergy efficiency
Original working fluid combination	265.92	49.32%
Optimized working fluid combination	279.83	51.79%

Parameter optimization based on genetic algorithm

In the simulation, it is easy to find that many parameters affect the performance of the system in the comprehensive utilization scheme of system energy and are coupled with each other, such as evaporation pressure and condensation pressure in each cycle. It is difficult to realize the optimal design of the system only by adjusting and optimizing a single parameter. Facing the multi-parameter non-linear optimization problem of the system, Genetic Algorithm, as a powerful optimizationol, can optimize multiple variables at the same time [23, 26]. In the following, to achieve the global optimization of the thermal performance of the system, the genetic algorithm is utilized as the optimization technique, while exergy efficiency of the system is employed as a goals function produce superior matching value for every optimization parameter.

Figure 6 displays the effect of related parameters of the Rankine cycle working medium on exergy efficiency, where the gradient of the curve of related parameters reflected the sensitivity of the parameter.

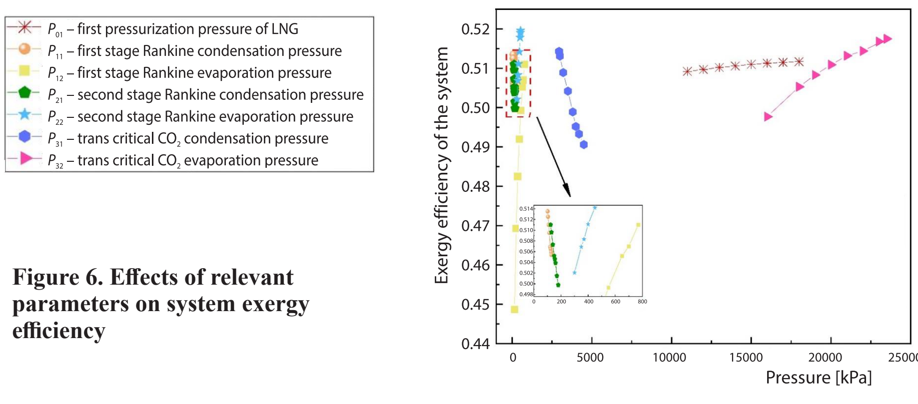

Figure 6. Effects of relevant parameters on system exergy efficiency

By observing the aforementioned curve slope, it can be seen that the exergy efficiency of the system is more sensitive to P_{11}, P_{12}, P_{21}, P_{22}, P_{31} and P_{32}. Although P_{01} has little effect on the thermal performance of the system, considering the coupling effect between them and other related parameters, the seven parameters mentioned previously are optimized together. On the premise of ensuring that there is no temperature crossover and normal operation of each heat exchanger, it is gradually adjusted through single parameter simulation find out each sensitive parameter value bound, and then find the optimal solution maximize the system exergy efficiency by jointly calling the genetic algorithm in MATLAB tool library and Aspen HYSYS software. The specific results are listed in tab. 5. It can be noted that the optimum intermediate pressure of two-stage pressurization is 16.99 MPa.

Table 5. Selection of sensitive parameters and optimization parameters

Target	Lower bound [kPa]	Upper bound [kPa]	Optimized value [kPa]
P_{01}	10000	18000	16990
P_{11}	110	130	110
P_{12}	400	770	767
P_{21}	124	180	128
P_{22}	300	520	517
P_{31}	3000	4500	3106
P_{32}	16000	23500	22500

The comparison of system performance data before and after global optimization based on genetic algorithm is shown in tab. 6. Obviously, after the genetic algorithm based global optimization, the overall turbine output power also increased from 279.83-296.95 kW. In addition, the overall system exergy efficiency improved from 51.79-54.61%, which is 2.82% higher than that of the system after working medium optimization.

Table 6. Comparison of schemes after global optimization and before

Scheme	Total turbine output power [kW]	Total exergy efficiency
After working fluid optimization	279.83	51.79%
After parameter optimization	296.95	54.61%

Economic estimate analysis

According to the method of calculating the cost of capital investment [27], tab. 7 gives the capital cost function of various equipments.

Table 7. Capital cost function of the various equipments of the system

Equipment	Capital cost function
Pump	$1120W_{\mathrm{p}}^{0.8}$
Heat exchanger	$2143A_{\mathrm{H}}^{0.514}$
Turbine	$4405W_{\mathrm{tur}}^{0.89}$

Note: W_{p} is the pump input power, A_{H} – the heat exchanger heat transfer area, and W_{tur} – the turbine output power.

As indicated in tab. 8, the capital investment cost of each equipment of the design scheme is calculated.

Table 8. Capital investment costs of the various equipments of the system

Equipment	Capital cost/CNY
Pump	348900
Heat exchanger	120200
Turbine	5259500
Total	5728600

According to the aforementioned table, we have estimated the equipment investment of the optimization scheme, where we define the system overall investment cost [27]:

$$C_{\text{total}} = \sum_k \left(Z_{CI} + Z_{OM}\right)_k \tag{1}$$

The initial investment cost $Z_{CI,k}$ and the running and maintaining cost $Z_{OM,}k$ for each system component are written:

$$Z_{CI,k} + Z_{OM,k} = \frac{Z_k \Phi}{N \cdot 3600} CRF \tag{2}$$

where Z_k is the initial investment cost of each equipment of the system, N – the annual operating time, and Φ – the equipment maintenance factor, taking 1.06 [28]. The capital recovery factor (CRF) can be defined:

$$CRF = \frac{i\left(1+i\right)^n}{\left(1+i\right)^n - 1} \tag{3}$$

where n is the system lifetime, taken as 20 years [29] and i – the annual interest rate, set to 14% [28]. The electricity production cost (EPC) and the annual total net income (ATNI) are written [27]:

$$W_{\text{net}} = \sum W_{\text{tur}} + \sum W_{\text{eva}} - \sum W_{\text{p}} \tag{4}$$

$$EPC = \frac{600 C_{\text{total}}}{W_{\text{net}}} \tag{5}$$

$$ATNI = 7300\left(EP - EPC\right) W_{\text{net}} \tag{6}$$

where W_{net} is the net output power, $\sum W_{\text{tur}}$ – the total turbine output power, and $\sum W_{\text{p}}$ – the total pump power. In the optimization scheme, W_{net} is 187.83 kW. The EPC and EP are, respectively defined as the unit power generation cost of the system and electricity price on board, in which EP is taken as CNY 1.50/kWh and the calculated EPC is CNY 0.67. The difference between EP and EPC is the net profit of the system per unit of power generation. The overall annual net revenue of the vessel owner under this scheme is CNY 1133240 if the annual operating days of the vessel power production system are 300 days. If the system is put into operation, the capital recovery cycle is projected to be 5.06 years.

Conclusions

- In the 200000 DWT vessel, the system combines the cold energy released during the vaporization of LNG fuel of the ME-GI host machine with the low grade thermal energy of medium temperature residual heat from the exhaust turbine flue gas. The system architecture is tight and simple, and the scheme is designed and optimized through cascade utilization, which can take full advantage of LNG high grade cold energy for power generation.

- Due to the high intake pressure of the ME-GI host machine, the two-stage pressurized fuel intake mode in this tonnage VLCC-LNG vessel can achieve the highest exergy efficiency of the system compared with other pressurized intake modes, and the best intermediate pressure is 16.99 MPa.
- Through the selection and optimization of working fluid, the best matching working fluid in the system is: the mixture ratio of methane: ethane: propane = 4:5.5:0.5 for the first stage organic working fluid Rankine cycle. While the mixture ratio of methane, ethane, propane, and pentane = 0.5:2.5:4.5:2.5 for the second stage organic working fluid Rankine cycle.
- After working fluid optimization and Genetic Algorithm based parameter optimization, the design system exergy efficiency can reach 54.61%, while the net power output is 187.83 kW. After the optimization is adopted for the whole vessel, the annual income of the design system can reach CNY 1133240. The expected capital recovery cycle is 5.06 years if the system is invested.

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Paper submitted: March 26, 2022
Paper revised: June 5, 2022
Paper accepted: June 10, 2022