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Thermodynamic analysis and performance optimization of an Organic Rankine Cycle (ORC) waste heat recovery system for marine diesel engines

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ABSTRACT

Escalating fuel prices and imposition of carbon dioxide emission limits are creating renewed interest in methods to increase the thermal efficiency of marine diesel engines. One viable means to achieve such improved thermal efficiency is the conversion of engine waste heat to a more useful form of energy, either mechanical or electrical. Organic Rankine Cycle (ORC) has been demonstrated to be a promising technology to recover waste heat. This paper examines waste heat recovery of a marine diesel engine using ORC technology. Two separated ORC apparatuses for the waste heat from both the jacket cooling water and the engine exhaust gas are designed as the traditional recovery system. The maximum net power output is chosen as the evaluation criterion to select the suitable working fluid and define the optimal system parameters. To simplify the waste heat recovery, an optimized system using the jacket cooling water as the preheating medium and the engine exhaust gas for evaporation is presented. The influence of preheating temperature on the system performance is evaluated to define the optimal operating condition. Economic and off-design analysis of the optimized system is conducted. The simulation results reveal that the optimized system is technically feasible and economically attractive.

1. Introduction

Diesel engines occupy a large proportion of marine propulsion plants due to their robust thermal performance, wide power output range, compact system structure and high operational reliability. The primary energy consumption of marine diesel engines is increasing rapidly with the development of the navigation shipping industry. There is a strong motivation in the marine sector to increase the propulsion system energy efficiency, primarily because of increasing fuel prices and stricter upcoming regulations [1]. The International Maritime Organization (IMO) adopted Annex VI to MARPOL 73/78 in 1997 [2]; the regulation went into effect on May 19, 2005. In 2007, the Marine Environment Protection Committee (MEPC) proposed the Energy Efficiency Design Index (EEDI) of Ships, which was a stricter requirement for pollutant discharge and efficient operation of marine engine. In October 2008, the IMO revised Annex VI, and over 53 countries implemented the regulation to address the crucial issue of the depleted energy supply and environmental deterioration [2,3].

In a typical marine diesel engine, less than 45% of the fuel energy is converted into useful power output, while the remaining energy is mainly lost through the exhaust gas, the jacket cooling water and other means, such as the air cooling system and the lubrication system [3]. It is apparent that the energy recovery potential gained from the waste heat of a marine diesel engine is appreciable. Waste heat recovery technology for marine engines appeared in 1970s in the United States and Europe [4,5] as a consequence of the first oil crisis. Driven by the business opportunity, shipbuilders and marine engine manufacturers were the first to join in this research field. The MAN Group presented an exhaust gas utilization system, including a heat recovery boiler, a steam turbine and a generator, which could bring a 10% increment to the engine efficiency [2]. Wärtsilä developed a recovery system containing a dual-pressure boiler and turbine and observed that the engine efficiency could be improved by 11.4% [2]. ABB Ltd. proposed two types of recovery systems for marine diesel engines with different capacity of power outputs; these systems could efficiently utilize the waste heat from engine exhaust gas [2].

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Nomenclature

G	mass flow rate, kg/s
Q	heat load, kW
h	specific enthalpy, kJ/kg
c_p	specific heat capacity, kJ/kg K
T	temperature, K
P	pressure, kPa
I	exergy destruction rate, kW
W	power, kW

Greek symbols

ε	heat loss ratio
η	efficiency
ζ	pressure loss ratio

Subscripts

w	working fluid
tot	total
HS	heat source
preh	preheater
evap	evaporator
exp	expander
cond	condenser
c	cooling water
ave	average

Acronyms

ORC	Organic Rankine Cycle
IMO	International Maritime Organization
MARPOL	Maritime Agreement Regarding Oil Pollution
MEPC	Marine Environment Protection Committee
EEDI	Energy Efficiency Design Index
GWP	Global Warming Potential
ODP	Ozone Depletion Potential
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers

Among all of the existing technologies, Organic Rankine Cycle (ORC) has proved to be a promising energy conversion technology for the utilization of medium and low-temperature heat sources due to its high efficiency, system simplicity and reliability [6]. Derived from conventional steam Rankine cycle, ORC is now considered technically feasible. ORC has been widely applied in solar, geothermal and biomass systems, in addition to engine waste heat recovery. Larsen et al. [1] presented an applicable methodology to determine the optimum working fluid, boiler pressure and Rankine cycle process layout for marine engine heat recovery. Teng and Regner [7,8] designed an ORC-WHR system to recover heat from engine exhaust gas, charge air cooler and EGR cooler; the case study demonstrated an increase in engine power output of up to 20%. Srinivasan et al. [9] examined the exhaust waste heat recovery potential of a dual fuel low temperature combustion engine using an ORC system; the fuel conversion efficiency was improved by an average of 7%. Zhang et al. [10] analyzed the characteristics of a novel system combining a vehicle diesel engine with a dual loop ORC that recovered waste heat from the exhaust gas, intake air and coolant. Hountalas et al. [11] studied an ORC system for a heavy-duty truck diesel engine, and the simulation result revealed that when exhaust heat and EGR heat were both recovered, the improvement of brake specific fuel consumption ranged between 6% and 7.5%. Bombarda et al. [12] compared the thermal performances of ORC and Kalina cycle for waste heat recovery of diesel engines; although the obtained useful power capacities were actually equal, ORC was more suitable due to its simple plant structure. Yu et al. [13] presented an ORC system to recover the waste heat from both engine exhaust gas and jacket water using R245fa as the working fluid; the influence of evaporating pressure and engine conditions on the system performance was observed. Vaja et al. [14] proposed three different cycles to recover the engine waste heat: a simple cycle with the use of only engine exhaust gas, a simple cycle with the use of exhaust gas and engine cooling water and a regenerated cycle.

This paper investigates waste heat recovery of a marine diesel engine manufactured by Hudong Heavy Machinery Co., Ltd. ORC technology is used to utilize waste heat from both the jacket cooling water and the engine exhaust gas. Two separated ORC systems are designed as the traditional recovery system, with the maximum net power output being used as the evaluation criterion to select the suitable working fluid and define the optimal system parameters. To simplify the waste heat recovery, an optimized ORC system using the jacket cooling water as preheating medium and the engine exhaust gas for evaporation is presented. The influence of preheating temperature on the system performance is evaluated to define the optimal operating condition. Economic and off-design analysis of the optimized system is conducted. The simulation results reveal that the optimized system is technically feasible and economically attractive.

2. System description

2.1. Marine diesel engine

The selected diesel engine is an inline six-cylinder turbocharged engine for marine propulsion plants, manufactured by Hudong Heavy Machinery Co., Ltd. The main parameters of the engine under the design condition are presented in Table 1. The measured composition of the exhaust gas is listed in Table 2. According to the calculation result by REFROP 9.0, the heat capacity of the exhaust gas is approximately 1.1 kJ/kg K. The total heat load capacity of the exhaust gas will be nearly 600 kW if it is cooled to the ambient temperature; while that of the jacket cooling water is 199.9 kW. The total heat load of the two waste heat sources reaches 800 kW, which further confirms that efficient recovery of the waste heat will significantly improve engine efficiency and reduce emission.

2.2. ORC system

An ORC system consists of a working fluid pump, an evaporator, an organic expander and a condenser. Fig. 1 shows the schematic

Table 1
Main parameters of the marine diesel engine.

Property	Unit	Value
Power output	kW	996
Rotation speed	r/min	1500
Torque	N m	6340
Temperature of the exhaust gas	K	573.15
Mass flow rate of the exhaust gas	kg/h	7139
Inlet temperature of the jacket cooling water	K	338.15
Outlet temperature of the jacket cooling water	K	363.15
Mass flow rate of the jacket cooling water	kg/h	6876

Table 2
Composition of the engine exhaust gas.

Composition	Content (g/kW h)	Molecular weight (g/mol)	Fraction (%)
O_2	1366	32.00	14.83
CO_2	552	44.00	4.36
H_2O	321	18.01	6.20
CO	0.28	28.01	–
SO_2	0.46	64.06	–
NO_x (NO_2)	11.17	46.01	–
HC (CH_4)	0.19	16.04	–
N_2	6872	28.01	74.61

diagram of a basic ORC system for waste heat recovery. The liquid organic working fluid from the condenser is pumped into the evaporator, where it is converted into saturated or superheated vapor. Next, the organic vapor expands in the expander to produce power, which is later converted into electricity by a generator. Afterwards, the exhaust organic gas from the expander is condensed to liquid in the condenser via cooling water.

The process of ORC is shown in Fig. 2, which can also be described as follows:

The mass flow rate of the organic working fluid can be calculated using the following equation:

$$G_w = \frac{Q_{tot} \cdot (1 - \varepsilon)}{h_4 - h_2} \tag{1}$$

where Q_{tot} is the total heat load of the waste heat source, and ε is the heat loss ratio of the heat exchanger. Q_{tot} can be calculated as

$$Q_{tot} = G_{HS} \cdot c_{p,HS} \cdot (T_{HS,in} - T_{HS,out}) \tag{2}$$

where $c_{p,HS}$ is the average specific heat capacity of the waste heat source, and $T_{HS,in}$ and $T_{HS,out}$ are defined as its inlet and outlet temperatures, respectively.

Process 1 to 2 in the working fluid pump is given by

$$W_{pump} = \frac{G_w \cdot (h_{2s} - h_1)}{\eta_{pump}} \tag{3}$$

where h_{2s} is the isentropic enthalpy of the working fluid after being compressed in the working fluid pump, and η_{pump} is the efficiency of the pump.

Process 2 to 4 in the evaporator is given by

$$Q_{evap} = G_w \cdot (h_4 - h_2) \tag{4}$$

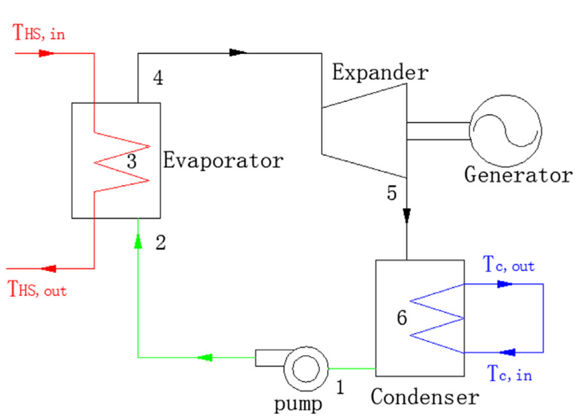

Fig. 1. Schematic diagram of an ORC system for waste heat recovery.

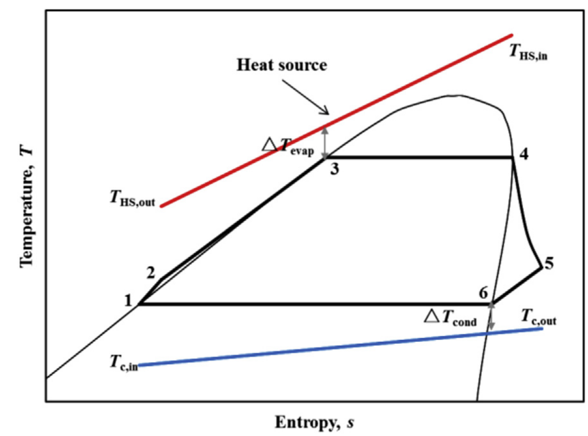

Fig. 2. T–s diagram of ORC.

Process 4 to 5 in the organic expander is given by

$$W_{exp} = G_w \cdot (h_4 - h_{5s}) \cdot \eta_{exp} \tag{5}$$

where h_{5s} is the isentropic enthalpy of the exhaust organic gas at the expander outlet, and η_{exp} is the efficiency of the organic expander.

Process 5 to 1 in the condenser is given by

$$G_c = \frac{G_w \cdot (h_5 - h_1)}{(1 - \varepsilon) \cdot c_{p,c} \cdot (T_{c,out} - T_{c,in})} \tag{6}$$

where ε is still the heat loss ratio of the heat exchanger; $c_{p,c}$ is the average specific heat capacity of the cooling water, $T_{c,in}$ and $T_{c,out}$ are its inlet and outlet temperatures, respectively.

The net power output of the ORC system is

$$W_{net} = W_{exp} - W_{pump} \tag{7}$$

The thermal efficiency of the ORC recovery system can be calculated as

$$\eta_{net} = \frac{W_{net}}{Q_{tot}} = \frac{W_{exp} - W_{pump}}{Q_{tot}} \tag{8}$$

The 2nd law analysis is also performed to explore the irreversible loss of the ORC system, corresponding to the exergy destruction rate of each component.

$$I_{pump} = G_w \cdot T_0 \cdot (s_2 - s_1) \tag{9}$$

$$I_{evap} = G_w \cdot T_0 \cdot \left[(s_4 - s_2) - \frac{h_4 - h_2}{T_{evap,ave}} \right] \tag{10}$$

$$I_{exp} = G_w \cdot T_0 \cdot (s_5 - s_4) \tag{11}$$

$$I_{con} = G_w \cdot T_0 \cdot \left[(s_1 - s_5) - \frac{h_1 - h_5}{T_{cond,ave}} \right] \tag{12}$$

where T_0 is the ambient temperature, which is 298.15 K in this paper; $T_{evap,ave}$ and $T_{cond,ave}$ are defined as the average temperatures during the evaporation and condensation processes, respectively.

2.3. Working fluid selection

Working fluid selection has an enormous influence on the thermal performance of the ORC recovery system under different heat source conditions. Generally, the organic working fluids can be divided into three categories: dry, isentropic and wet according to the saturation vapor curve slope, as shown in Fig. 3. A large number of research studies have been performed on the selection of suitable working fluids for different ORC systems. Mago et al. [15] chose a variety of dry and wet fluids based on different heat source temperatures and evaluated the system performance; the results indicated that system performance with dry fluids was better. Zhang et al. [16] compared the performance of an ORC system with various working fluids and optimized cycle parameters, utilizing exhaust gas from gasoline engine as the heat source. Hung et al. [17] demonstrated that the major properties affecting the system performance included latent heat, specific heat of liquid and vapor. Chen et al. [18] suggested that fluids with high latent heat and low liquid specific heat were preferable. However, Yamamoto et al. [19] suggested that low latent heat fluids were better.

In this paper, the organic working fluid candidates are selected according to the following principles:

- proper thermal properties during the cycle;
- chemical stability under the operating condition;
- slight impact on the environment – low GWP (Global Warming Potential) and low ODP (Ozone Depletion Potential);
- security in the system (non-flammable, non-explosive and non-toxic);
- availability and low cost.

3. Design and analysis of two separated ORC

Two separated ORC systems are designed to recover waste heat from both the jacket cooling water and the engine exhaust gas as the traditional system. The schematic diagram is shown in Fig. 4. On the basis of an ORC test rig constructed by Tsinghua University and Hangzhou Chinen Steam Turbine Power Co., Ltd [20] and one of the previous research studies [21] by the authors, some design parameters are given in Table 3. The condensation temperature is assumed to be 311.15 K. No superheating and no sub-cooling are considered at the exit of the evaporator and the condenser. All the properties of working fluids are acquired from REFROP 9.0, and the simulation is performed using a computer program written by the authors in the FORTRAN environment [21].

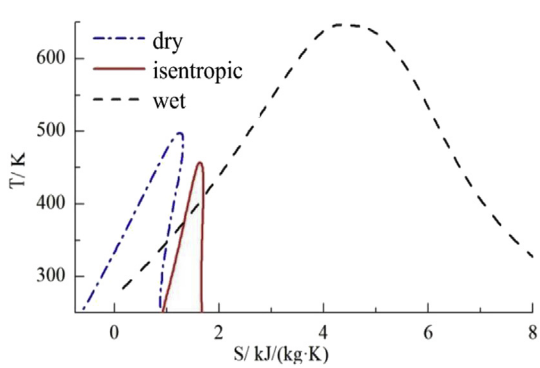

Fig. 3. *T–s* diagram of dry, isentropic and wet organic fluids.

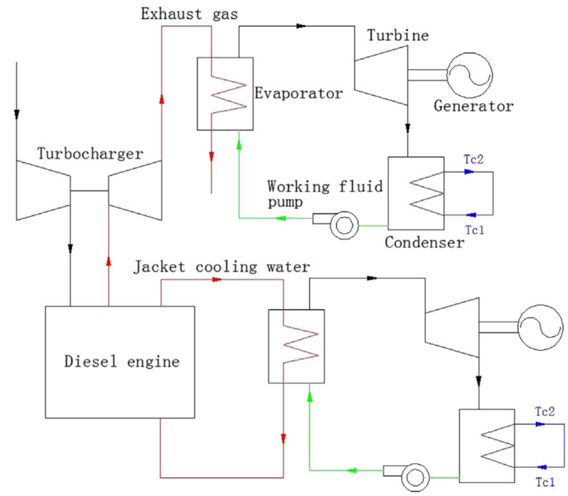

Fig. 4. Schematic diagram of the traditional ORC recovery system.

3.1. Jacket cooling water waste heat recovery

The mass flow rate of the jacket cooling water is 6876 kg/h, and the inlet and outlet temperatures are 338.15 K and 363.15 K, respectively. Because the total heat load is fixed, the design of the ORC recovery system is aimed to achieve the maximum net power output, indicating the overall heat recovery efficiency as well. The computer program arrives to a proper evaporation temperature through continuous iteration, the heat load above which is utilized for evaporation and below which is used for preheating the working fluid to the bubble point temperature. Several organic fluids are selected as candidates, and their thermodynamic properties are listed in Table 4 [22–25]. Simulation of the jacket cooling water waste heat recovery is conducted, with the results listed in Table 5. The temperature of the jacket cooling water is low, which results in low net power output and low thermal efficiency of the ORC system. With R236fa or R245fa as the working fluid, the maximum net power output reaches 10.3 kW and the thermal efficiency is 5.2%. In consideration of the impact to the environment, R245fa is selected due to its low GWP value.

The 2nd law analysis is also performed for the ORC system with R245fa as the working fluid to explore the irreversible loss of each component. As shown in Fig. 5, the exergy destruction rate of the evaporator and the condenser are considerably higher due to the significant temperature differences within these two components; while that of the organic expander is only 2.5 kW, and for the working fluid pump, the irreversible loss is extremely low.

Table 3
Design parameters of the ORC system.

Property	Symbol	Value
Pump efficiency	η_{pump}	0.8
Heat loss of the heat exchanger	ε	0.01
Preheater pressure loss	ζ_{preh}	0.08
Minimum temperature difference of the evaporator	ΔT_{evap}/K	6.0
Expander inlet pipe pressure loss	ζ_1	0.02
Expander efficiency	η_{exp}	0.8
Turbine outlet pipe pressure loss	ζ_2	0.015
Cooling water initial temperature	$T_{c,in}$/K	298.15
Minimum temperature difference of the condenser	ΔT_{cond}/K	6.0

Table 4
Working fluid candidates for the jacket cooling water waste heat recovery.

Working fluid	Molecule weight (g/mol)	Normal boiling point (K)	Critical temperature (K)	Critical pressure (kPa)	GWP	ODP	ASHRAE 34[a]
R236ea	152.04	279.3	412.4	3502.0	1350	0	A1
R236fa	152.04	271.7	398.1	3200.0	9400	0	A1
R600	58.12	272.7	425.1	3796.0	~20	0	A3
R600a	58.12	261.4	407.8	3629.0	~20	0	A3
R123	152.93	301.0	456.8	3661.8	120	0.012	B1
R134a	102.03	247.1	374.2	4059.3	1300	0	A1
R245fa	134.05	288.3	427.2	3651.0	950	0	B1

[a] ASHRAE Standard 34 – Refrigerant safety group classification. 1: No flame propagation; 2: Lower flammability; 3: Higher flammability; A: Lower toxicity; B: Higher toxocity.

3.2. Exhaust gas waste heat recovery

The temperature of the exhaust gas out of the marine diesel engine is 573.15 K, and the mass flow rate is 7139 kg/h. According to the temperature condition of the exhaust gas, several organic fluids with high critical temperature are selected as candidates to keep the cycle within the subcritical range, the thermal properties of which are listed in Table 6 [26–28].

The total heat load absorbed by the ORC system varies with the outlet temperature of the exhaust gas. The simulation results are shown in Fig. 6, which demonstrates variations of the evaporation temperature, the mass flow rate of the working fluid, the net power output and the system thermal efficiency with the outlet temperature of the exhaust gas.

Fig. 6(a) shows that the evaporation temperature of each type of working fluid increases with the increment of the outlet temperature of the exhaust gas, and the variation is nearly linear. The evaporation temperature of cyclohexane is the highest, while that of toluene is the lowest. Fig. 6(b) shows that the mass flow rate of the working fluid decreases with the increment of the outlet temperature of the exhaust gas. A higher evaporation temperature corresponds to a higher power output per unit mass of working fluid and mass flow rate of working fluid directly affects the total power output of the ORC system. Variations of the evaporation temperature and the mass flow rate of the working fluid with the outlet temperature of the exhaust gas are opposite; thus it is not easy to directly determine how the net power output changes when the outlet temperature of the exhaust gas increases. Fig. 6(c) shows that the net power output decreases with increasing outlet temperature of the exhaust gas, which indicates that the effect of the decreasing mass flow rate dominates within the mentioned outlet temperature range. Fig. 6(d) demonstrates the variations of the system thermal efficiency with the outlet temperature of the exhaust gas, which is positively relevant to the evaporation temperature of the working fluid.

Note that there is sulfur element in the fuel, and the sulfur transforms into sulfur dioxide when burned in the cylinder. The sulfur dioxide is later converted into sulfur trioxide within further oxidation. If the outlet temperature of the exhaust gas is lower than the acid dew point, the sulfur trioxide will combine with the steam vapor and become sulfuric acid, which might corrode the pipe and the heat exchanger of the ORC system. It is desirable for the cooled exhaust gas to be set above 373.15 K [29]. Therefore, the outlet temperature of engine exhaust gas is specified as 378.15 K in this paper, which is 5 K higher than the safety limit. According to the simulation results in Fig. 6, cyclohexane, benzene and toluene are selected as the optimal working fluids. The operating parameters of the ORC system utilizing marine diesel engine exhaust gas are listed in Table 7. With benzene as the working fluid, the maximum net power output reaches 90.8 kW, and the thermal efficiency is 21.3%.

Two separated ORC systems are designed to recover waste heat from both the jacket cooling water and the engine exhaust gas as the traditional system, with R245fa and benzene as the working fluids, respectively. The maximum net power output of the total system reaches 101.1 kW, which increases the marine diesel engine power by 10.2%.

Table 5
Parameters of the ORC system for the jacket cooling water.

Working fluid	Evaporation temperature (K)	Mass flow rate (kg/s)	Net power output (kW)	Thermal efficiency
R236ea	337.1	1.19	10.2	5.1%
R236fa	337.7	1.28	10.3	5.2%
R600	336.3	0.52	10.1	5.1%
R600a	336.8	0.57	10.1	5.1%
R123	335.6	1.10	10.2	5.1%
R134a	337.8	1.14	10.2	5.1%
R245fa	336.4	0.99	10.3	5.2%

4. System optimization and further analysis

4.1. System optimization

It is evident that the traditional system containing two separated ORC systems is complex in structure and requires a large amount of space, which might limit its application on ships. System optimization is implemented in this section from the view of system simplicity and technological feasibility. Because the temperature of the jacket cooling water is low, it is desirable to be used for preheating rather than evaporation [13,14]. A heat exchanger is installed in the jacket cooling water cycle as a preheater; thus only one single ORC system is required. The schematic diagram of the optimized ORC recovery system is shown in Fig. 7.

As the outlet temperature of the jacket cooling water is 363.15 K, it is assumed that the working fluid could be preheated up to 357.15 K at most, taking the pinch temperature difference of the

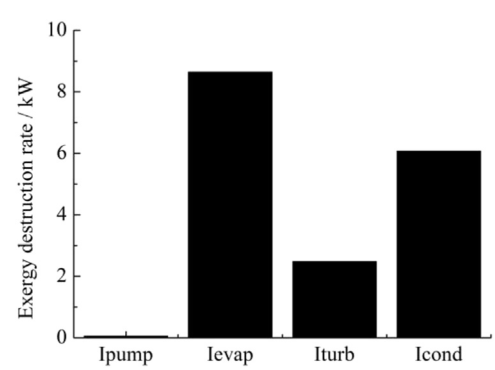

Fig. 5. Exergy destruction rate of each ORC component.

Table 6
Working fluid candidates for the engine exhaust gas waste heat recovery.

Working fluid	Molecule weight (g/mol)	Normal boiling point (K)	Critical temperature (K)	Critical pressure (kPa)	GWP	ODP	ASHRAE 34[a]
Cyclohexane	84.16	353.9	553.6	4075.0	Low	0	A3
Benzene	78.11	353.2	562.1	4894.0	Low	0	B2
Toluene	92.14	383.8	591.8	4126.3	Low	0	A3
Nonane	128.26	423.9	594.6	2281.0	Low	0	A3
Decane	142.28	447.3	617.7	2103.0	Low	0	A3

[a] ASHRAE Standard 34 — Refrigerant safety group classification. 1: No flame propagation; 2: Lower flammability; 3: Higher flammability; A: Lower toxicity; B: Higher toxocity.

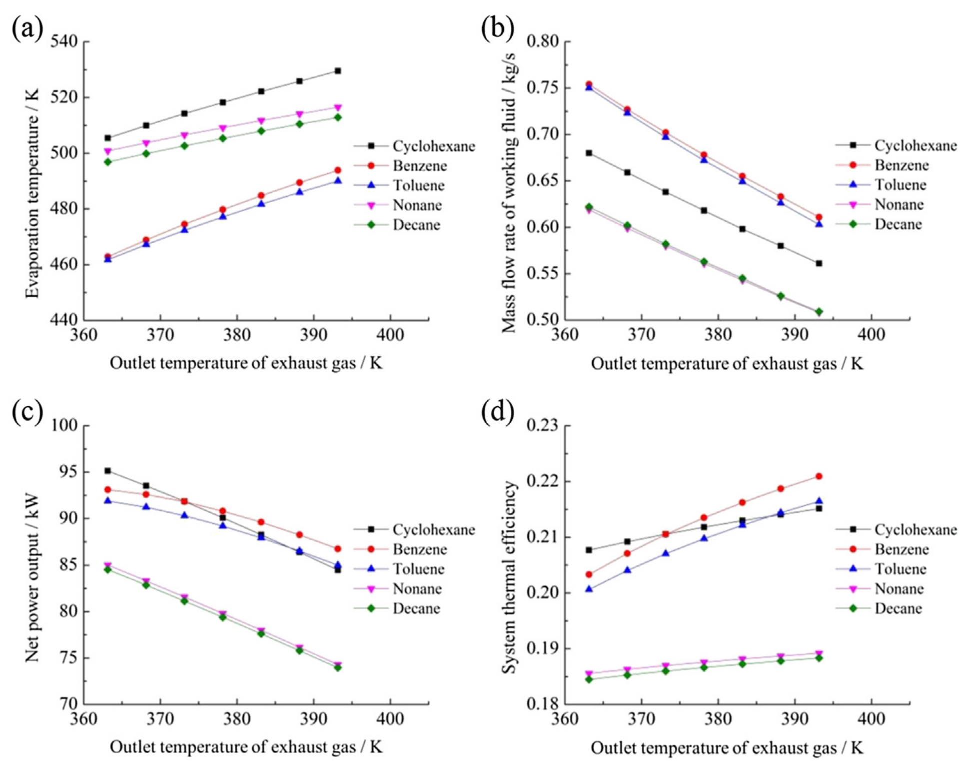

Fig. 6. Variations of the thermal parameters with the outlet temperature of the exhaust gas.

heat exchanger into consideration. Mass flow rate of the jacket cooling water used could be regulated to control the heat load in the preheater, as long as the total heat load is lower than 199.9 kW. Therefore, the preheating temperature varies in this study. Cyclohexane, benzene and toluene are still chosen as the working fluids to evaluate system performance. Fig. 8 illustrates the variations of several parameters with the preheating temperature of working fluid at the exit of the preheater, which ranges from 323.15 K to 357.15 K in this study.

The total heat load absorbed by the working fluid increases with the increment of the preheating temperature, which is equivalent to the situation that the outlet temperature of the exhaust gas decreases in section 3.2. Thus the evaporation temperature decreases and the mass flow rate of the working fluid increases with the increment of the preheating temperature, as shown in Fig. 8(a) and (b), which is similar with those in Fig. 6(a) and (b). The evaporation temperature of cyclohexane is the highest, nearly 40 K higher than those of benzene and toluene under each preheating temperature condition. It is still not easy to directly determine how the net power output changes because the variations of evaporation temperature and mass flow rate of working fluid are different. In Fig. 8(c), the net power output increases with the increment of preheating temperature for cyclohexane, which indicates that the effect of the increasing mass flow rate dominates within the preheating temperature range. While for benzene and toluene, there is an optimal preheating temperature for the ORC system to reach the maximum net power output. When the preheating temperature is lower than the optimal value, the effect of increasing mass flow rate dominates; when the preheating temperature is higher than the optimal value, the effect of decreasing evaporation temperature dominates. In addition, the system with cyclohexane performs better than the other two working fluids and the maximum net power output reaches 99.7 kW when the preheating temperature is 357.15 K, which is the highest that could be reached. Fig. 8(d) demonstrates the variations of the system thermal efficiency with the preheating temperature, which is positively relevant to the evaporation temperature. The heat loads of the preheater are

Table 7
Parameters of the ORC system for the engine exhaust gas.

Working fluid	Evaporation temperature (K)	Mass flow rate (kg/s)	Net power output (kW)	Thermal efficiency
Cyclohexane	518.3	0.62	90.1	21.2%
Benzene	479.8	0.68	90.8	21.3%
Toluene	477.2	0.67	89.2	21.0%

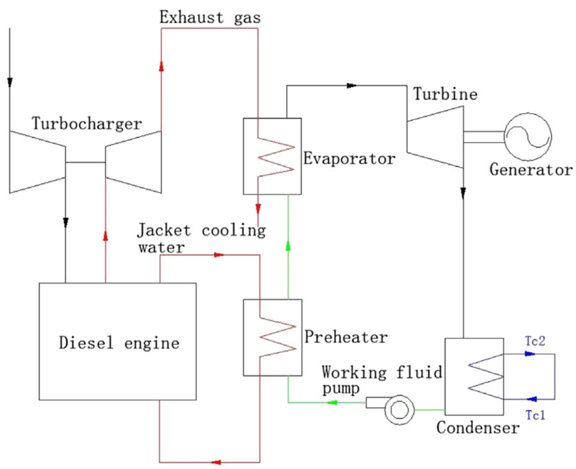

Fig. 7. Schematic diagram of the optimized ORC recovery system.

shown in Fig. 8(e) and the difference among the three working fluids are little to none. When the preheating temperature is 357.15 K, the heat load of the preheater is approximately 75 kW, only 37.5% of the total heat load of the jacket cooling water.

With cyclohexane as the working fluid, the maximum net power out of the optimized system reaches 99.7 kW, only 1.4% lower than that of the traditional system containing two separated ORCs. As a consequence of the optimization, only one ORC system and an extra heat exchanger are required instead of two subsystems. The recovery system is simplified and the space required is reduced. Furthermore, because only 37.5% of the jacket cooling water heat load is used for preheating, the remaining can be harnessed as domestic hot water on the ship.

4.2. Further analysis

The simulation results and analysis above reveal that the optimized system is more suitable due to its compact structure, although the net power output is slightly lower. Further analysis is conducted in this section from the view of economic factors and off-design conditions.

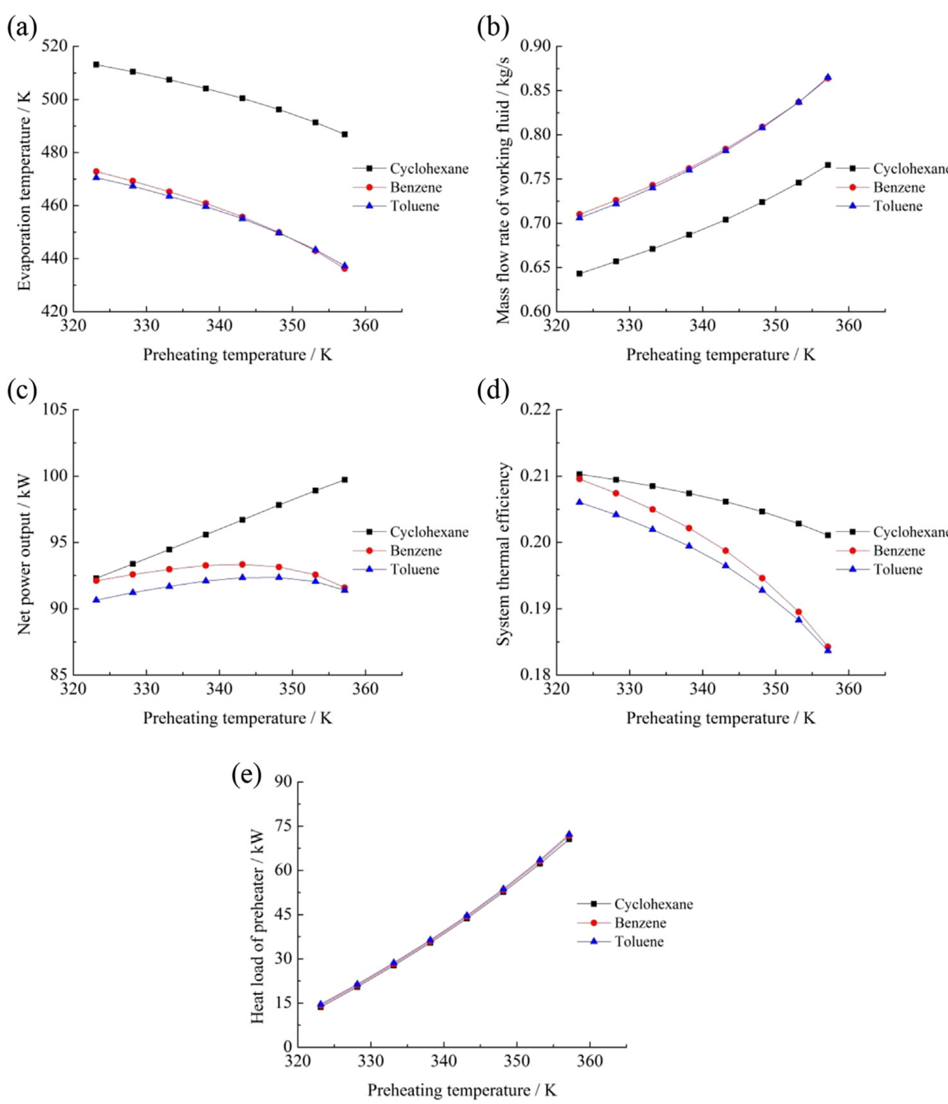

Fig. 8. Variations of the thermal parameters with the preheating temperature.

4.2.1. Economic analysis

ORC manufacturers appeared on the market at the beginning of the 1980s, and plenty of ORC systems were applied in different situations, which could be categorized according to temperature level of heat sources, system power output and target applications. In Fig. 9, the ORC module costs for waste heat recovery are plotted [30]. Note that the provided data are indicative only and partially collected from a non-exhaustive set of ORC manufacturers and from scientific publications [31–34]; thus, the scattering in the data is due to different prices of manufactures, different market strategies and different integration costs. In general, the individual cost of an ORC system should not be generalized. However, it still could be concluded from the figure that the cost tends to decrease when the power output increases.

Comparative study of the traditional and the optimized recovery system is conducted from the perspective of economics, with the results listed in Table 8.

As for the optimized system, the net power output is only 1.4% lower than that of the traditional system. However, the capital cost could be reduced by 34.5%. Thus the optimized system is recommended from the view of economics.

Generally, it is difficult to evaluate the capital cost of an ORC system due to the global economic situation, fluctuation of the component price and variability of supply and demand in market. It will be part of authors' future research to complete a relatively accurate economic estimation for the ORC system.

4.2.2. Off-design analysis

The marine diesel engine might operate under different conditions as the navigation condition changes. An off-design analysis of the ORC recovery system must be performed. Four typical operating conditions of the marine diesel engine are selected in Table 9. The temperature and the mass flow rate of the exhaust gas vary under different load condition, so does the jacket cooling water. All the data listed in the table is acquired from the engine manufacturer.

The off-design analysis is conducted for the optimized ORC recovery system with cyclohexane as the working fluid. All the other operating parameters are assumed to be the same in this section, i.e., the heat exchange efficiency, the pressure loss of the ORC system and the condensation temperature. The simulation results are shown in Fig. 10, which demonstrates the variations of the evaporation temperature, the mass flow rate of working fluid, the net power output, the system thermal efficiency and the heat load of preheater with preheating temperature. It can be concluded that a higher engine load corresponds to a higher evaporation temperature, a higher mass flow rate of working fluid and a higher net power output of the ORC system, because the temperature and the heat load of the exhaust gas are positively related to engine load capacity. Since the other conclusions are similar with those in sections 3 and 4, they are omitted in this section. Moreover, the heat load of the preheater is lower than 199.9 kW under each engine load condition, which means the optimized ORC recovery system can operate normally.

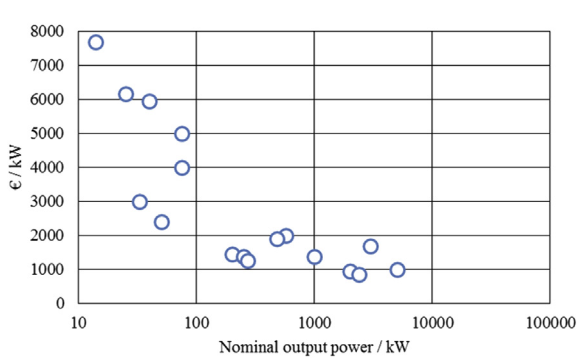

Fig. 9. Module cost of ORC systems depending on the net power output.

Table 8
Economic analysis of the ORC system.

System	Capital cost per unit of power (€/kW)	Net power output (kW)	Total capital cost (€)
Traditional system	7500	10.3	304,250
	2500	90.8	
Optimized system	2000	99.7	199,400

5. Conclusions

This paper focuses on waste heat recovery of a marine diesel engine manufactured by Hudong Heavy Machinery Co., Ltd. ORC technology is used to recover waste heat from both the jacket cooling water and the engine exhaust gas. The maximum net power output is chosen as the evaluation criterion to select the suitable working fluids and define the optimal set of thermodynamic parameters, and system simplicity, technological and economic feasibility are considered in the optimization. The primary conclusions are summarized as follows:

(1) Two separated ORC systems with R245fa and benzene as the working fluids are designed to utilize waste heat from both the jacket cooling water and the engine exhaust gas. The total net power output was found to reach 101.1 kW, which results in an efficiency increment of 10.2% for the marine diesel engine.
(2) An optimized ORC recovery system is presented. The low-temperature jacket cooling water is used to preheat the working fluid, while the high-temperature exhaust gas is utilized for evaporation. One single ORC system and an extra heat exchanger are required for the two waste heat sources. With cyclohexane as the working fluid, the maximum net power output of the optimized system reaches 99.7 kW, only 1.4% lower than that of the two separated systems.
(3) Further analysis is conducted from the view of economic factors and off-design conditions. The optimized system is found to be economically attractive because the capital cost is significantly reduced. Four typical operating conditions of the marine diesel engine are selected to make off-design analysis for the optimized system. A higher engine load corresponds to a higher evaporation temperature, a higher mass flow rate of working fluid and a higher net power

Table 9
Four typical conditions of the marine diesel engine.

Load	Exhaust gas		Jacket cooling water		
	Temperature (K)	Mass flow rate (kg/h)	Temperature (K)	Mass flow rate (kg/h)	Heat load (kW)
100%	573.15	7139	363.15	6876	199.9
85%	557.15	6381	363.15	5918	172.1
75%	553.15	5835	363.15	5187	150.8
50%	548.15	4189	363.15	3528	102.6

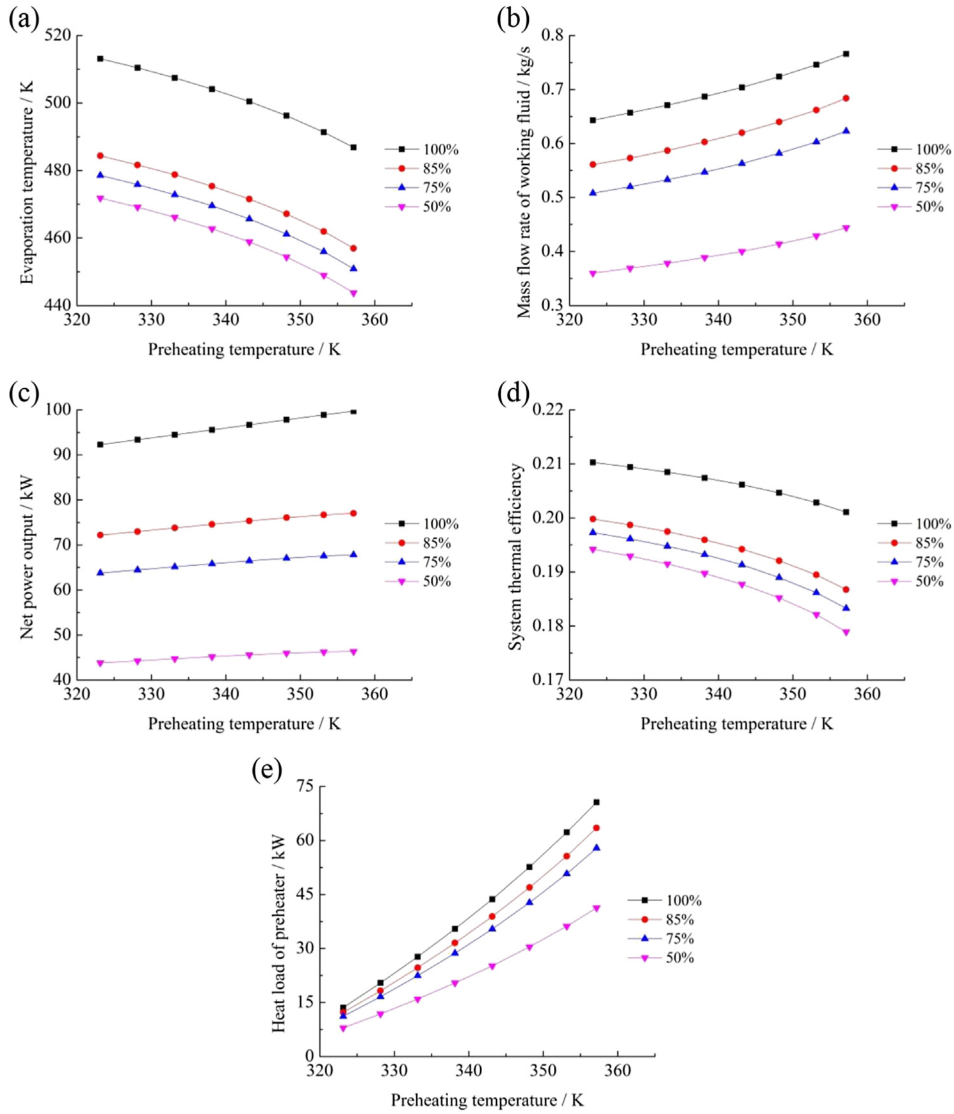

Fig. 10. Variations of the thermal parameters with the preheating temperature under different engine load conditions.

output because the temperature and the heat load of the exhaust gas are positively related to the engine load capacity.

(4) Although the net power output is slightly lower, the compact system structure and low capital cost are the advantages of the optimized ORC system. Therefore, the optimized system is recommended for practical application in marine engine waste heat recovery. The analytical method and optimization progress presented in this paper could be applied to similar waste heat recovery cases.

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