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Performance optimization of organic Rankine cycles for waste heat recovery for a large diesel engine

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Abstract In order to recover the low grade waste heat and increase system fuel economy for main engine 10S90ME-C9.2-TII(part load, exhaust gas bypass) installed on a 10000 TEU container ship, a non-cogeneration and single-pressure type of waste heat recovery system based on organic Rankine cycle is proposed. Organic compound candidates appropriate to the system are analyzed and selected. Thermodynamic model of the whole system and thermoeconomic optimization are performed. The saturated organic compound vapor mass flow rate, net electric power output, pinch point, thermal efficiency and exergy efficiency varied with different evaporating temperature are thermodynamically analyzed. The results of thermodynamic and thermoeconomic optimization indicate that the most appropriate organic compound candidate is R141b due to its highest exergy efficiency, biggest unit cost benefit and shortest payback time.

Keywords: Low grade waste heat; Organic Rankine cycle; Thermodynamic optimization; Thermoeconomic optimization; Intelligent marine diesel engine

Nomenclature

a	–	coefficient related to ship type
ANI	–	value of electricity recovered by WHRS annually, USD

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ANC_I	–	exergy loss cost, USD
ANC_{INV}	–	original equipment cost, USD
ANC_o	–	annual operating cost, USD
ANC_T	–	annual total cost, USD
A	–	heat exchange areas outside of tubes, m^2
A_B	–	heat exchange areas inside of tubes, m^2
C_{Tg1}	–	specific heat of exhaust gas at temperature T_{g1}, J/kg K
C_{Tg2}	–	specific heat of exhaust gas at temperature T_{g2}, J/kg K
C_z	–	compensation factor for heat transfer piping
d	–	diameter of tube
E_{in}	–	exergy of exhaust gas at the inlet of exhaust gas boiler, W
E_{out}	–	exergy of exhaust gas at the outlet of exhaust gas boiler, W
g	–	gravitational acceleration, m/s^2
K	–	overall heat transfer coefficient, W/m^2¡K
h_2	–	specific enthalpy of organic compound at the outlet of condenser, kJ/kg
h_3	–	specific enthalpy of organic compound liquid at preheater inlet, kJ/kg
h_{3s}	–	specific enthalpy of organic compound at the end of isentropic compression, kJ/kg
h_4	–	specific enthalpy of organic compound liquid at preheater outlet, kJ/kg
h_5	–	specific enthalpy of saturated organic compound vapor at specific exhaust gas boiler working pressure, kJ/kg
h_6	–	specific enthalpy of organic compound vapor at expander outlet, kJ/kg
h_{6s}	–	specific enthalpy of organic compound vapor at the end of isentropic expansion, kJ/kg
h_{p5}	–	fin height under staggered arrangement, m
h_{Tg1}	–	specific enthalpy of exhaust gas at boiler inlet, kJ/kg
h_{Tg2}	–	specific enthalpy of exhaust gas at pre-heater inlet
h_{Tg3}	–	specific enthalpy of exhaust gas at boiler outlet, kJ/kg
I	–	exergy loss of system, W
$\dot{m}_g$	–	mass flow rate of exhaust gas, kg/h
$\dot{m}_{cl}$	–	mass flow rate of cooling water(sea water), kg/h
$\dot{m}_R$	–	mass flow rate of organic compound, kg/h
Δp_{cf}	–	total flow resistance of the circulating pipelines in condenser, Pa
Δp_{Rf}	–	total flow resistance of the circulating pipelines in boiler, Pa
Nu	–	Nusselt number
Q	–	mass flow rate of organic compound, kg/h
Pr	–	Prandl number
r_{in}	–	fouling resistance inside of tubes, m^2K/W
r_{out}	–	fouling resistance outside of tubes, m^2K/W
s	–	entropy, kg/(kg °C)
$S_{p\delta}$	–	fin spacing under staggered arrangement, m
T	–	temperature
W	–	system total power output, W
W_P	–	exhausted power output, W
W_T	–	total power output of expander-generator, W
v	–	kinematic coefficient of viscosity, m^2/s
v_2	–	specific volume of organic compound at the outlet of condenser, m^3/kg
v_3	–	specific volume of organic compound at the outlet of pump, m^3/kg
v_{clin}	–	specific volume of cooling water at the inlet of condenser, m^3/kg
v_{clout}	–	specific volume of cooling water at the outlet of condenser, m^3/kg

Greek symbols

α	–	convection heat transfer coefficient, W/m^2K
$\alpha_{out,tp}$	–	two-phase convection heat transfer coefficient of organic compound which happens in condenser outside of tubes, $W/(m^2K)$
$\alpha_{out,sp}$	–	single-phase convection heat transfer coefficient of organic compound outside of tubes, $W/(m^2K)$
Γ	–	condensation heat transfer, W
η_B	–	exhaust gas boiler efficiency considering the radiation loss, %
η_g	–	efficiency of expander-generator, %
η_m	–	mechanical efficiency, %
η_{Ps}	–	isentropic compression efficiency of pump, %
η_{Ts}	–	isentropic expansion efficiency of expander, %
η_{pump}	–	compression efficiency of pump, %
$\eta_{cl,pump}$	–	efficiency of cooling water pump, %
μ	–	coefficient of kinetic viscosity, Pa s
λ	–	coefficient of heat conductivity, W/m K
ν	–	vapour
Σ	–	total sum

Subscripts

c	–	convective heat transfer
cl	–	cooling water
in	–	inlet
nb	–	nucleate boiling
out	–	outlet
tp	–	convective boiling

1 Introduction

With the increasingly highlighted requirement of fuel economy, reliability and stringement on emission regulations, a large two-stroke intelligent marine diesel engine has been the best choice as a 10000 TEU container ship's main engine and waste heat recovery (WHR) has been one of the most promising methods to improve ship's performance of energy-saving and emission-reduction. However, traditional waste heat recovery equipment onboard, which mainly adopting exhaust gas boiler to yield saturated steam for heating service, could not extract heat energy effectively from exhaust gas and other waste heat sources (such as jacket water and scavenge air) for their low temperatures. Distinguishing characteristics of large two-stroke intelligent marine diesel engine is its very low exhaust gas temperature after turbochargers (about 250 °C at normal continuous rating, NCR) which has brought a dilemma to install traditional steam turbine cogeneration systems onboard even though the total enthalpy of exhaust gas is tremendous [1]. How to recover waste heat effectively onboard has been one of the state of the art issues in shipbuilding industry.

One effective method is to adopt thermo efficiency system (TES) developed by MAN B&W [2]. After main engine's specially redesigned for waste heat recovery, TES could redistribute the exhaust gas heat from high amount/low temperature to low amount/high temperature so that it could recover waste heat more easily. However, the cost specific fuel oil consumption (SFOC) increases from 0.0% to 1.8% compared with the standard main engine version before redesigned. Ma and Yang [3] designed three conceptual waste heat recovery systems, i.e., TES (which uses water as working fluid), TES-ORC (which uses organic fluid as working fluid) and TES-SEG (which uses a screw expander-generator instead of the steam turbine-generator), and numerical results indicated that the above three systems were more feasible than traditional steam turbine cogeneration systems. Though the total power yield of TES-ORC was the highest, TES and TES-SEG were more advantageous for their system simplicity and safety.

Another promising method is to employ organic Rankine cycle (ORC). ORC can give a better performance to recover low grade waste heat onboard, and there are several advantages for economical utilization of energy resources, small system applications and environmental impacts [4]. Durmusoglu *et al.* [5] theoretically designed an energy saving and power solution using ORC for a container ship and proposed three performance analysis criteria. However, they did not discuss appropriate organic compounds for marine ORC use and did not do thermodynamic optimization and thermoeconomic analysis. Yue *et al.* [6] designed an ORC based waste heat recovery system for a marine diesel engine which employed isopentane as working fluid, presented first law and second law results and designed the most important components for ORC system-steam turbine. However, they did not identify how to choose the optimum evaporating temperature. Yang and Ma [7] designed two conceptual waste heat recovery steam turbine cogeneration systems which used Rankine cycle (RC) and ORC, respectively. They found that the very low exhaust gas temperature brought a dilemma to install traditional steam turbine cogeneration systems onboard for intelligent marine diesel engine without adopting TES.

In the published literatures, not much is reported on the thermodynamic and thermoeconomic optimization of ORC WHR systems onboard. Also, the designed WHR systems employing RC and ORC are traditional cogeneration systems and use forced circulation exhaust gas boiler. In this paper, a non-cogeneration ORC WHR system which uses natural circulation exhaust gas boiler is proposed to recover waste heat of main engine 10S90ME-C9.2-TII (part load, exhaust gas bypass) on a 10000 TEU container ship to increase fuel economy. Thermodynamic and thermoeconomic optimization models based on first law and second law are formulated. Effect of evaporating temperature on mass flow rate of the saturated organic compound vapor, net electric power output, pinch point, thermal efficiency and exergy efficiency are thermodynamically analyzed. In addition, payback time analysis has also analysed.

2 Main diesel engine and organic Rankine cycle waste heat recovery system

Effects of engine load on mass flow rate and temperature of the exhaust gas after turbochargers of 10S90ME-C9.2-TII (part load, exhaust gas bypass) type marine diesel engine which is forseen to be equipped to a 10000 TEU container ship are shown in Figs. 1 and 2, respectively. MAN B&W pointed out that it is probably more realistic to use the ISO ambient temperatures

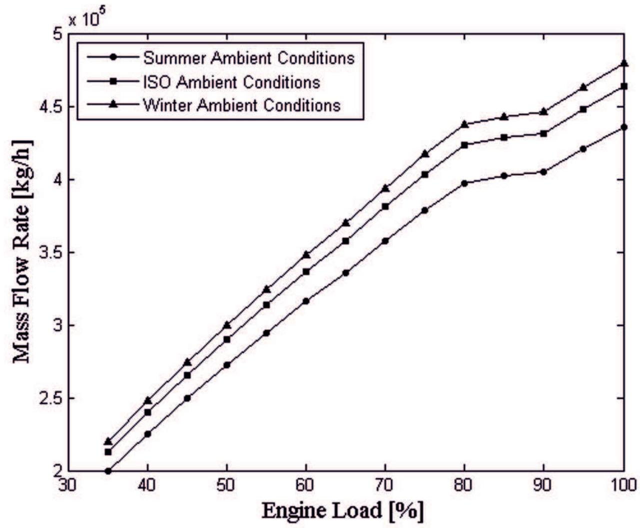

Figure 1: Effect of engine load on mass flow rate of the exhaust gas after turbochargers.

as the average ambient temperatures in worldwide operation [2]. Therefore, design point of ORC WHR system can choose main engine's normal continuous rating point (90%SMCR) under the International Organization for Standardization (ISO) ambient conditions, i.e., the mass flow rate and temperature of the exhaust gas after turbochargers are 431 500 kg/h and 266.8 °C, respectively. In addition, the ISO ambient conditions refer to a standard state where the ambient air suction temperature and cooling water temperature are both 25 °C.

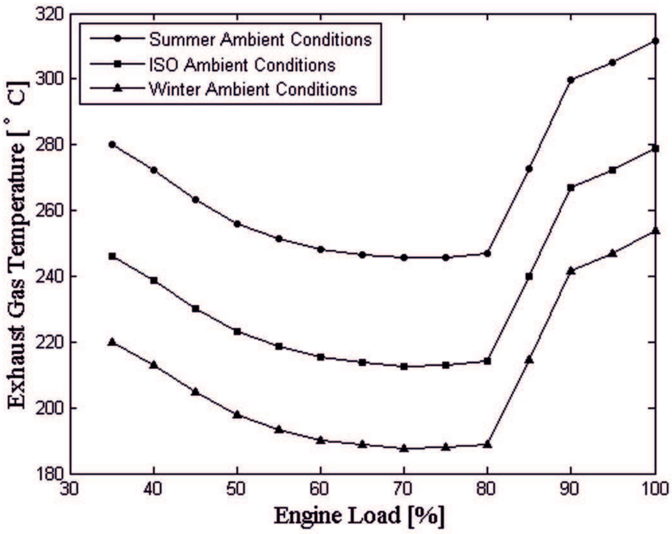

Figure 2: Effect of engine load on temperature of the exhaust gas after turbochargers.

For operational safety and cost-savings, natural circulation exhaust gas boiler has been chosen. What's more, in order to design a relatively more simple WHR system and take into account the correspondingly involved higher risks for soot deposits and fires of the exhaust gas boiler [8], a non-cogeneration and single-pressure type of ORC WHR system has been adopted. Non-cogeneration means that the exhaust gas boiler only yield saturated or superheated organic steam for power output while the needed saturaed water vapor for daily heating service should be produced by the auxiliary boiler. Mago *et al.* [9] found that ORC WHR system which makes saturated steam expand in steam turbine could achieve a satisfying performance so that there was no need to superheat organic compound. Thus, saturated organic compound vapor is adopted. Non-cogeneration and single-pressure type of ORC WHR system diagram for main engine

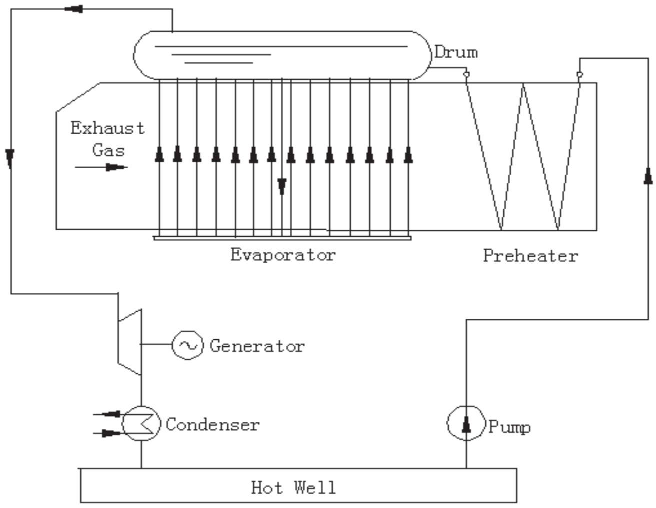

Figure 3: Non-cogeneration and single-pressure type of ORC WHR system.

10S90ME-C9.2-TII (Part load, Exhaust Gas Bypass) on a 10000 TEU container ship is shown in Fig. 3. Non-cogeneration and single-pressure type of ORC WHR system consists of an exhaust gas boiler system, a vapour turbine-generator system, a condenser system, organic compound pump, hot well and control system.

3 Thermodynamic modeling and optimization

Temperature vs. entropy (T-s) diagram of the exhaust gas and organic compound in the non-cogeneration and single-pressure type of ORC WHR system for main engine 10S90ME-C9.2-TII (part load, exhaust gas bypass) on a 10000 TEU container ship is shown in Fig. 4. The exhaust gas enters the evaporator at T_{g1}, then enters the preheater at T_{g2} and leaves the boiler at T_{g3}.

3.1 Organic compound candidates

In this paper, detailed thermodynamic and thermoeconomic optimization models have been formulated based on the first and the second law of thermodynamics. The organic compound selection has been proven to be vital to ORC plant efficiency and optimum performance [10–13]. Pre-selection of

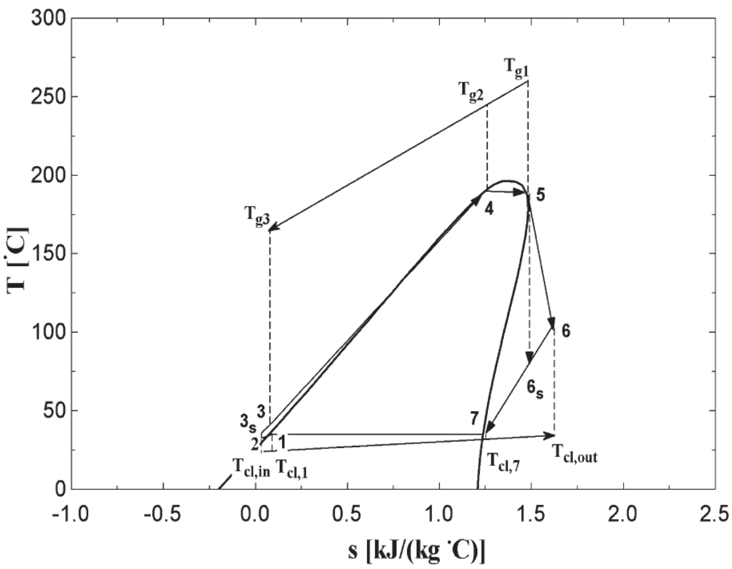

Figure 4: Temperature vs. entropy (T-s) diagram of the exhaust gas and organic compound in the non-cogeneration and single-pressure type of ORC WHR system.

organic compound is performed according to the following criteria: (1) the organic compound should have a critical temperature T_{cri} near the exhaust gas temperature; (2) the standard boiling point T_{boi} should be near the ambient air temperature; (3). It should be a well-known organic compound in the ORC field, i.e., an organic compound that has been previously studied in the scientific literature or organic compounds that are used in commercial ORC power plants, such as solkatherm, n-pentane or R134a [14]. The final selection of organic compound candidates are described in Tab. 1.

3.2 Heat exchanger model

The heat exchangers are modeled by means of the logarithmic mean temperature difference (LMTD) method for counter-flow heat exchangers. The exhaust gas boiler uses annular finned tubes with exhaust gas flowing turbulently across staggered tube bundles to obtain better heat transfer effect. Condenser is a fixed shell and tube heat exchanger which uses bare pipes.

According to the above assumption, exhaust gas boiler inlet temperature T_{g1} and outlet temperature T_{g3} have been known, so that the mass

Table 1: List of considered working fluids.

Candidate fluids	T_{cri}, °C	P_{cri}, MPa	T_{boi}, °C
R123	183.3	3.6618	27.97
R141b	204.5	4.212	32.20
R245ca	174.57	3.925	25.28
R245fa	154.16	3.651	15.29
n-pentane	196.70	3.370	36.21
isopentane	187.35	3.378	27.98

flow rate of organic compound could be calculated as follows:

$$\dot{m}_R = \frac{\dot{m}_g \eta_B \left(h_{Tg1} - h_{Tg3}\right)}{h_5 - h_3} \,. \tag{1}$$

With the same method, the specific enthalpy of exhaust gas at preheater inlet h_{Tg2} could be calculated as follows:

$$h_{Tg2} = h_{Tg3} + \frac{\dot{m}_R \eta_B \left(h_4 - h_3\right)}{\dot{m}_g \eta_B} \,. \tag{2}$$

Suppose that the specific heat of exhaust gas at temperature T_{g1} equals to the specific heat of exhaust gas at temperature T_{g2} [15], i.e., $C_{Tg1} = C_{Tg2}$, the temperature of exhaust gas at preheater inlet T_{g2} could be calculated as follows:

$$T_{g2} = \frac{h_{Tg3}}{C_{Tg1}} + \frac{\dot{m}_R \eta_B \left(h_4 - h_3\right)}{C_{Tg1} \dot{m}_g \eta_B} \,. \tag{3}$$

The pinch point temperature difference ΔT_{pp} could be calculated as follows:

$$\Delta T_{pp} = T_{g2} - T_5 \,. \tag{4}$$

For exhaust gas boiler, the overall heat transfer coefficient K_{boiler} is calculated as follows [16–18]:

$$K_{boiler} = \frac{1}{\frac{1}{\alpha_{out}} + \frac{A}{\alpha_{in} A_B}} \,. \tag{5}$$

convection heat transfer coefficient can be calculated through Eqs. (6)–(9)

$$\alpha_{out} = 0.23\, C_z\, \varphi_\sigma^{0.23} \frac{\lambda}{S_{p\delta}} \left(\frac{d}{S_{p\delta}}\right)^{-0.54} \left(\frac{h_{p\delta}}{S_{p\delta}}\right)^{-0.14} \left(\frac{w\, S_{p\delta}}{\nu}\right)^{0.65} , \tag{6}$$

$$\alpha_{in,tp} = \frac{\mathrm{Nu}\lambda}{d_{in}} , \tag{7}$$

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} , \tag{8}$$

$$\alpha_{in,tp} = \max(\alpha_c, \alpha_{nb}) . \tag{9}$$

For condenser, the overall heat transfer coefficient $K_{condenser}$ is calculated as follows [19]:

$$\frac{1}{K_{condenser}} = \frac{1}{\alpha_{out}} + r_{out} + r_{in}\left(\frac{d_{out}}{d_{in}}\right) + \frac{1}{\alpha_{in,sp}}\left(\frac{d_{out}}{d_{in}}\right) , \tag{10}$$

$$\alpha_{out,tp} = 1.51\left(\frac{4\Gamma}{\mu}\right)^{-1/3}\left(\frac{\mu^2}{\lambda^3\rho^2 g}\right)^{-1/3} , \tag{11}$$

$$\alpha_{out,sp} = j_H\frac{\lambda}{D_e}\left(\frac{c_p\,\mu}{\lambda}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} . \tag{12}$$

3.3 Expander-generator model

Isentropic expansion efficiency of expander

$$\eta_{Ts} = \frac{h_5 - h_6}{h_5 - h_{6_s}} . \tag{13}$$

Total power output of expander-generator

$$W_T{=}\dot{m}_R\eta_{Ts}\eta_g\eta_m\,(h_5 - h_6) . \tag{14}$$

3.4 Pump model

For refrigerant pump, isentropic compression efficiency is

$$\eta_{Ps} = \frac{h_{3_s} - h_2}{h_3 - h_2} . \tag{15}$$

Output power recoverd from exhausted gases is

$$W_P = \frac{\dot{m}_R \int_{p_2}^{p_3} v dp}{\eta_{pump}} , \tag{16}$$

$$\int_{p_2}^{p_3} v dp \approx \frac{v_2 + v_3}{2}(p_3 - p_2) , \tag{17}$$

$$p_3 - p_2 = (p_{evap} - p_{cond}) + \Delta p_{Rf} \ . \tag{18}$$

Driving power of the cooling water pump is

$$W_{clpump} \approx \frac{\dot{m}_{cl} \frac{v_{clin}+v_{clout}}{2} \Delta p_{cf}}{\eta_{clpump}} \ . \tag{19}$$

System thermal efficiency

$$\eta_{thermal} = \frac{W_T - W_P - W_{clpump}}{Q} \ , \tag{20}$$

$$Q = \dot{m}_g \left(h_{T_{g1}} - h_{T_{g3}} \right) \eta_B \ . \tag{21}$$

System exergy efficiency

$$\eta_{exergy} = \frac{E_{in} - \Sigma I - E_{out}}{E_{in}} \ . \tag{22}$$

3.5 Thermodynamic optimization

For cycle performance simulation, the assumptions are made as follows: (1) temperature of exhaust gas is 266.8 °C, and the mass flow rate is 431 500 kg/h; (2) all the ORC WHR systems using different organic compounds have the same exhaust gas boiler outlet temperature 165 °C; (3) ambient temperature is considered to be 25 °C, and the ambient pressure is considered to be 101.325 kPa; (4) pinch point temperature difference is 5 °C; (5) condensing temperature is 35 °C; (6) subcooling temperature at condenser outlet is 0.5 °C; (7) sea water temperature at the inlet of condenser is considered to be 25 °C, and sea water temperature at the outlet of condenser is considered to be 30 °C; (8) the whole system is assumed to reach steady state, $\Delta p_{Rf} = 0.6$ MPa, $\Delta p_{cf} = 0.3$ MPa, $\eta_{Ps} = 0.8$, $\eta_{pump} = 0.65$, $\eta_{clpump} = 0.8$; $\eta_g = 0.75$; $\eta_B = 0.98$.

In this case, the only available degree of freedom is the evaporating temperature. The thermodynamics properties of the organic compounds were calculated by standard reference data REFPROP 8.0 [20]. Effect of evaporating temperature on mass flow rate of the saturated organic compound vapor, net electric power output, pinch point, thermal efficiency and exergy efficiency are demonstrated in Figs. 5 to 9, respectively.

It is obvious that the higher evaporating temperature, the greater net power output, the bigger thermal efficiency and exergy efficiency. The maximum of net power output, thermal efficiency and exergy efficiency occurs when the evaporating temperature is close to its critical temperature. It is easy to find that organic compound R141b has the biggest net power output, thermal efficiency and exergy efficiency while organic compound R245fa has the smallest. However, the pinch point shows a reverse. Also, the needed organic compound mass flow rate differs for different fluids. The one needed for the organic fluid n-pentane is the smallest while for R123 is the biggest.

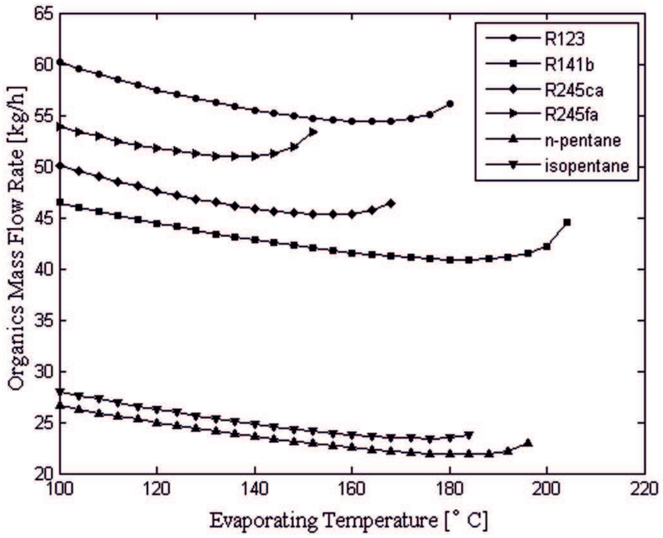

Figure 5: Effect of evaporating temperature on mass flow rate of the saturated organic compound vapor.

It is observed that system performance is very susceptible to the evaporating temperature on the hypothesis that the main variables, i.e., the condensing temperature and the sea water temperature at the outlet of condenser, are fixed by experience. It is necessary to determine the optimal operating parameters.

The thermal efficiency cannot reflect the ability to convert energy from exhaust gas into usable work. Therefore, the thermodynamic optimization aims at maximizing the exergy efficiency, which can evaluate the performance for waste heat recovery. The exergy efficiency varies with three variables, i.e., evaporating temperature, condensing temperature and sea water temperature at the outlet of condenser. It is assumed that the evaporating temperature varies from 100 °C to the critical temperature, condensing

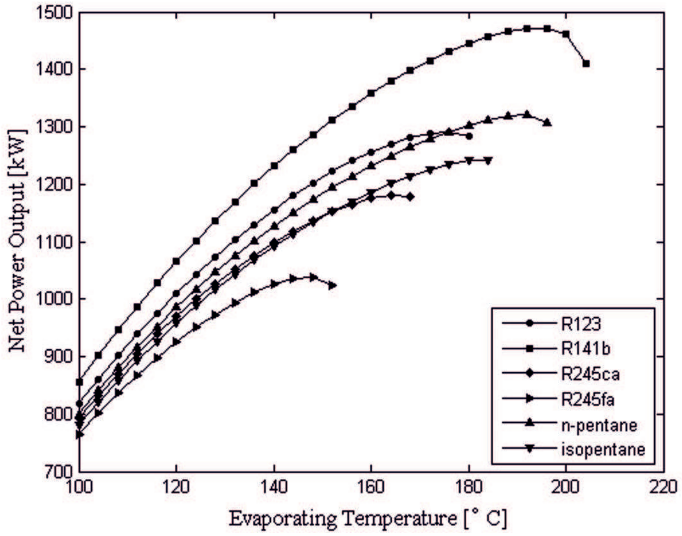

Figure 6: Effect of evaporating temperature on net electric power output.

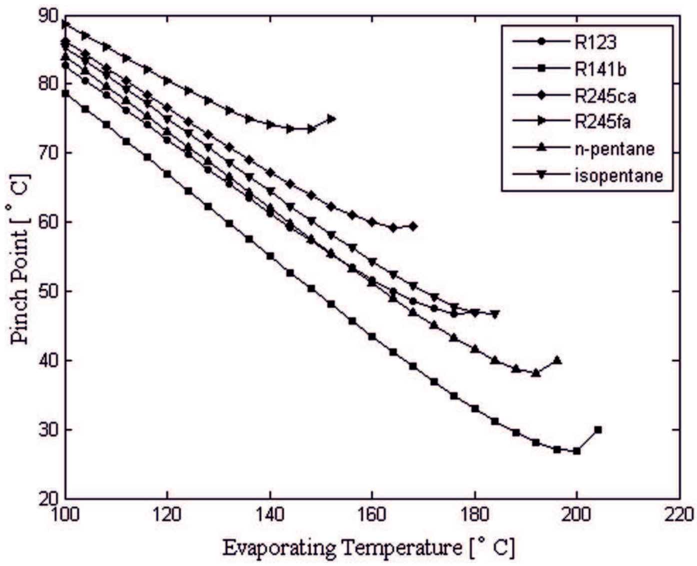

Figure 7: Effect of evaporating temperature on pinch point temperature difference.

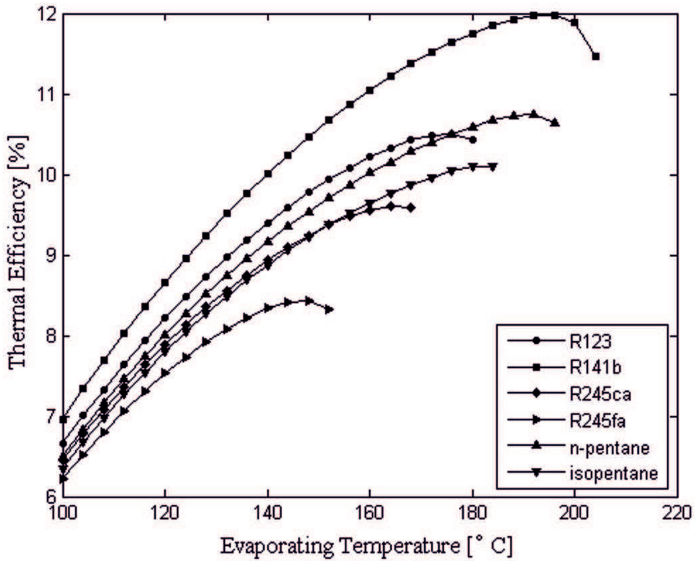

Figure 8: Effect of evaporating temperature on thermal efficiency.

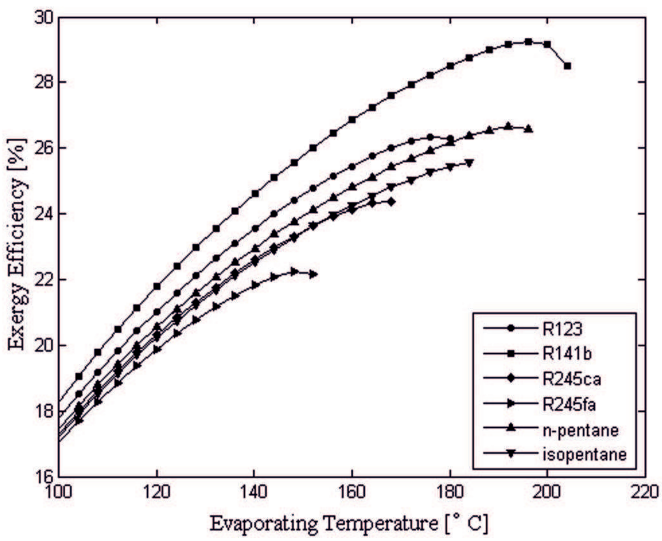

Figure 9: Effect of evaporating temperature on exergy efficiency.

temperature varies from 35 °C to 50 °C and sea water temperature at the outlet of condenser varies from 30 °C to 45 °C. The results of this optimization are presented in Tab. 2. According to Tab. 2, the maximum of exergy efficiency is found at 197.0 °C when the ORC WHR system uses organic fluid R141b while the minimum is found at 149.4 °C when the ORC WHR system uses organic fluid R245fa. In addition, all of the optimal exergy efficiencies pccur at the fixed condensing temperature 35 °C and the fixed sea water temperature at the outlet of condenser 30 °C. What's more, it is obvious that all the optimal evaporating temperatures are close to their critical temperatures.

Table 2: Performance of considered working fluids.

Candidates fluids	T_{evap}, °C	T_{cri}, °C	η_{ex}, %	$\eta_{thermal}$, %	W_{net}, kW
R123	177.8	183.68	26.34	10.490	1290
R141b	197.0	204.35	29.24	11.960	1471
R245ca	167.3	174.42	24.39	9.600	1180
R245fa	149.4	154.01	22.24	8.423	1036
n-pentane	193.7	196.55	26.70	10.730	1320
isopentane	184.7	187.20	25.54	10.090	1241

4 Thermoeconomic modeling and optimization

The goal of this section is to propose an alternative optimization for the ORC WHR working conditions: instead of the exergy efficiency, the selected objective function for this optimization is the unit cost benefit (UCB) expressed as follows:

$$UCB = \frac{ANB}{ANC_T} \,. \tag{23}$$

ANC_T is the annual total cost of ORC WHR system. ANC_T consists of three parts, i.e., the annual operating cost, ANC_o, the exergy loss cost, ANC_I, and the original equipment cost, ANC_{INV}[21]:

$$ANC_T = ANC_o + ANC_I + ANC_{INV} \,. \tag{24}$$

ANB is the annual net profit and is expressed as

$$ANB = ANI - ANC_o \,. \tag{25}$$

ANI is the value of electricity recovered by ORC WHR system annually and is expressed as

$$ANI = EP\,W_{net}\,D \,, \tag{26}$$

where: EP – price of electricity in Chinese RMB, [0.5CN¥,/(kWh)], D – annual operating time [8000 h].

ANC_o is calculated according to [22]:

$$ANC_o = 0.1ANC_T \,. \tag{27}$$

ANC_I is calculated by

$$ANC_I = k_I I_{sys} D \,, \tag{28}$$

where: k_I – per cost of exergy loss, [0.5CN¥,/(kWh)], I_{sys} – total exergy loss of ORC WHR system [kW].

ANC_{INV} is calculated according to [23]:

$$ANC_{INV} = \frac{1}{n}\sum_{1}^{6} C_{INV,i} \,, \tag{29}$$

where: n – normal service life of ORC WHR system, [15 years], $C_{INV,i}$ – original cost of different equipments, [CN¥].

$C_{INV,1}$ is the original cost of exhaust gas boiler, $C_{INV,2}$ is the original cost of expander, $C_{INV,3}$ is the original cost of generator, $C_{INV,4}$ is the original cost of condenser, $C_{INV,5}$ is the original cost of refrigerant pump and $C_{INV,6}$ is the original cost of cooling water pump. $C_{INV,i}$ are calculated respectively:

$$C_{INV,1} = 116125.3\Big[(UA)_{pre}^{0.6} + (UA)_{evap}^{0.6}\Big] \,, \tag{30}$$

$$\begin{aligned} &C_{INV,2} = 4627.6 \times (81.05 + 21.1872Y + 34.1785Y^2 - \\ &22.6463Y^3 + 5.54694Y^4 + 0.334884Y^5 + 0.165157Y^6 + \\ &0.007602Y^7) \,, \end{aligned} \tag{31}$$

$$Y = \ln(W_{net}/100) \,, \tag{32}$$

$$C_{INV,3} = 21052.8W_{net}^{0.58} \,, \tag{33}$$

$$C_{INV,4} = 1106.6A_c^{1.01} \,, \tag{34}$$

$$C_{INV,5} = 493W_P^{0.8} \,, \tag{35}$$

$$C_{INV,6} = 493W_{clpump}^{0.8} \,. \tag{36}$$

where UA_{pre} – heat transfer area of the preheater, m^2, UA_{evap} – heat transfer area of the evaporator, m^2, W_{net} – net system power output, kW, A_c – heat transfer area of the condenser, m^2, W_{clpump} – power ultilized by cooling water pump, kW.

For the thermoeconomic optimization, assumptions are made: (1) detailed structure parameters of the ORC WHR system have been carefully determined, which means that the thermoeconomic optimization is only related to thermodynamic parameters; (2) condensing temperature is 35 °C and the sea water temperature at the outlet of condenser is 30 °C.

Therefore, the only available degree of freedom is the evaporating temperature. The results of this optimization are presented in Tab. 3. According to Tab. 3, the maximum of UCB is found at 195.3 °C when the ORC WHR system uses organic compound R141b while the minimum happens at 147.9 °C when the ORC WHR system uses organic compound R245fa. It is easy to find that the optimal evaporating temperature value for the UCB does not coincide with the optimal evaporating temperature value for the exergy efficiency.

Table 3: Results of the thermoeconomic optimization.

Candidates fluids	T_{evap}, °C	UCB	ANB 10^6, ¥/year	η_{ex},%	W_{net}, kW
R123	176.0	0.2976	3.867	26.32	1292
R141b	195.3	0.3909	4.690	29.22	1472
R245ca	165.9	0.2468	3.364	24.37	1182
R245fa	147.9	0.1885	2.712	22.22	1038
n-pentane	192.5	0.3121	4.001	26.69	1321
isopentane	183.5	0.2758	3.646	25.53	1242

The payback time of ORC WHR system is expressed as

$$t = \frac{\lg \frac{C}{C - i\,ANC_{INV}}}{\lg(1+i)} \,, \tag{37}$$

where: t – payback time of ORC WHR system [year], C – annual saved fuel cost [USD/year], i – annual interest rate [8%].

For the payback time simulation, assumptions are made as follows: (1) mass flow rate and pressure of the needed saturated steam (water vapor) which is produced by the auxiliary boiler for heating service is 3834 kg/h and 0.7 MPa; (2) thermal efficiency of the auxiliary boiler is 0.93; (3) thermal efficiency of the diesel generators is 0.9; (4) the specific fuel oil consumption (SFOC) of diesel generators is 185 g/(kW h);(5) lower calorific value (LCV) of the fuel is 42700 kJ/kg.

The variation of average price of fuel with different time is shown in Fig. 10 [24]. The optimal evaporating temperatures in different ORC WHR systems are decided by the thermoeconomic optimization. The variation of payback time with price of fuel is illustrated in Fig. 11. Suppose the price of fuel oil is still high in a long time, to install such ORC WHR system on large or ultra-large merchant ships will be a great benefit to the ship owners in the long-period service of the ship (normally 25 years).

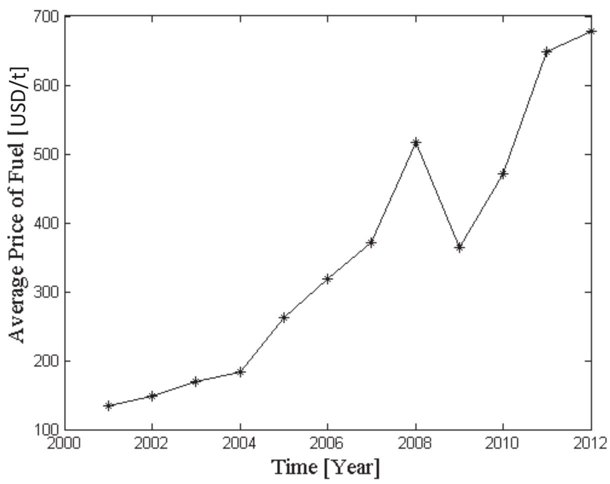

Figure 10: Variation of the average price of fuel with different years.

5 Conclusion

Thermodynamic and thermoeconomic optimization of a non-cogeneration and single-pressure type of ORC WHR system are performed in this paper. The most appropriate organic compound candidate is R141b for the main engine 10S90ME-C9.2-TII (part load, exhaust gas bypass) on a 10000 TEU

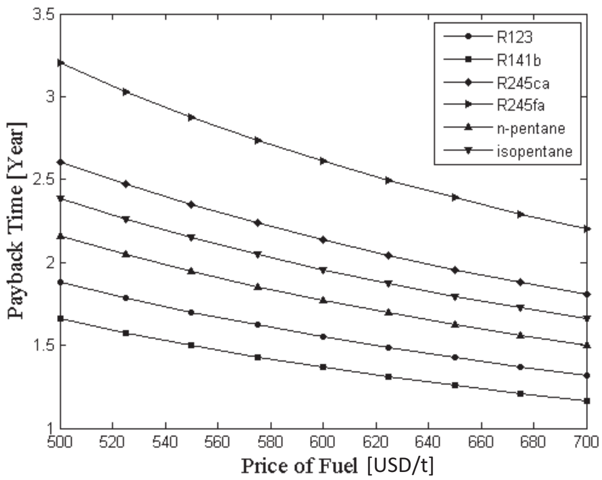

Figure 11: Variation of the payback time with the price of fuel.

container ship due to its highest exergy efficiency, biggest unit cost benefit and shortest payback time. Further, operation performance under part load conditions and system arrangement design onboard of such concept ORC WHR system will be done.

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