CONCEPTUAL DESIGN AND PERFORMANCE ANALYSIS OF WASTE HEAT RECOVERY SYSTEM FOR INTELLIGENT MARINE DIESEL ENGINES. PART 1: IMPRACTICAL ANALYSIS OF TRADITIONAL WHR SYSTEMS

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ABSTRACT

Intelligent Marine Diesel Engines have been state of the art main engines employed by value-added new ship including large container vessel. The distinguishing characteristics of the large two-stroke intelligent marine diesel engines refer to their very low exhaust gas temperature (about 250℃ at NCR, Normal Continuous Rating) after turbochargers and their very huge amount of exhaust gas. Due to its very low exhaust gas temperature after the turbocharger, it is more difficult to recover their waste heat. However, the mass flow rate of the exhaust gas is quantitatively huge so that to recover waste heat from the large two-stroke intelligent marine diesel engines is hot and full of challenge. In this paper, thermodynamic models for two traditional WHR systems—ordinary Rankine Cycle (RC) conceptual waste heat recovery system and Organic Rankine Cycle (ORC) conceptual waste heat recovery system matching MAN B&W 10S90ME—a typical intelligent marine diesel engine—are derived and numerically calculated. Numerical results indicate that the very low exhaust gas temperature of this intelligent marine diesel engine has led the impracticable installation of traditional WHR systems onboard.

1. INTRODUCTION

In order to meet the increasing requirement of fuel economy, reliability and stringent emission regulations, intelligent marine diesel engine comes into being and has been becoming the state of the art trend of marine diesel engine design. Intelligent diesel engine means using electronic control system mainly to do key control rather than using mechanical construction. The leading manufacturers of marine diesel engines, say MAN B&W and Wartsila-Sulzer, developed and manufactured intelligent marine diesel engines in 1 990s and successfully installed their modern intelligent marine diesel engine to propel large ships since the beginning of the 21st century. Nowadays, the representative intelligent marine diesel engines employed by large vessels include ME series of MAN B&W and RT-flex series of Wartsila-Sulzer.

The distinguishing characteristics of the large two-stroke intelligent marine diesel engines refer to their very low exhaust gas temperature (about 250℃ at NCR, Normal Continuous Rating) after turbochargers and their very huge amount of exhaust gas. Taking 9K98ME-C7.1-TII of MAN B&W for example, its exhaust gas temperature after turbochargers is 244.8℃ and flow rate of exhaust gas is 440 400kg/h at NCR (46 053kW) under ISO ambient conditions (thus total enthalpy of the exhaust gas is about 31MW) though thermal efficiency of the engine exceeds 49.22%(SFOC: 171.3g/kW h). It is no doubt that considering waste heat recovery (WHR) techniques to extract heat energy from exhaust gas and other waste heat sources (such as jacket water and scavenge air heat) could further increase system fuel economy of intelligent marine diesel engine. Exhaust gas boilers are usually installed on vessels to recover waste heat of exhaust gas to yield saturated steam for heating service, which could not recover all available waste heat effectively. For non-intelligent marine diesel engines (their exhaust gas temperature after turbochargers is more than 300℃ and mass flow rate of exhaust gas is huge at NCR), traditional WHR systems installed onboard are cogeneration systems (combined heat and power generation systems), and use a big exhaust gas boiler to yield saturated steam for heating service and superheated steam for electricity generation in steam turbine, and usually use water as working fluid in Rankine Cycles (RC) [1-4]. However, for intelligent marine diesel engines, their very low exhaust gas temperature makes traditional WHR systems(combined heat and power generation systems) less efficiency even fail.

This paper attempts to check whether or not ordinary Rankine Cycle (RC) conceptual waste heat recovery system and Organic Rankine Cycle (ORC) conceptual waste heat recovery system could match with a typical intelligent marine diesel engine, i.e. 10S90ME of MAN B&W, which may be the first choice of main engine for 10 000 TEU container vessels. Corresponding thermodynamic models are derived and system performance analysis and comparison are further carried out. Meanwhile, this paper also does feasibility analysis on WHR system for 10S90ME of MAN B&W marine diesel engine installed on large ships and some constructive suggestions are given.

2. BASIC PERFORMANCE DATA OF THE TARGET INTELLIGENT MARINE DIESEL ENGINE—10S90ME OF MAN B&W

Exhaust gas temperature and amount after turbocharger of intelligent marine diesel engine—10S90ME of MAN B&W is shown in Figure 1. The temperature is 223.8℃ and the amount of the exhaust gas at NCR is 450 742kg/h. Specific Fuel Oil Consumption (SFOC) is shown in Figure 2. It is obvious that the distinguishing characteristics of this intelligent marine diesel are low exhaust gas temperature、low exhaust gas amount and low SFOC.

To design WHR systems for this intelligent main engine and make performance analysis, two traditional WHR systems are taken into account: ordinary Rankine Cycle (RC) conceptual waste heat recovery system and Organic Rankine Cycle (ORC) conceptual waste heat recovery system. For better compare and contrast, detailed thermodynamic models have been built for the above two traditional WHR systems, respectively. Following assumptions are made in the analysis [5]:

•Steam is at steady state;

•No pressure drops on steam side;

•Pressure drop on exhaust gas side does not affect its temperature;

•Approach point is negligible;

•The expansion process in the steam turbine is an isentropic expansion process;

•The specific enthalpy of the superheated steam entering the steam turbine is equal to the specific enthalpy of the superheated steam left the superheater.

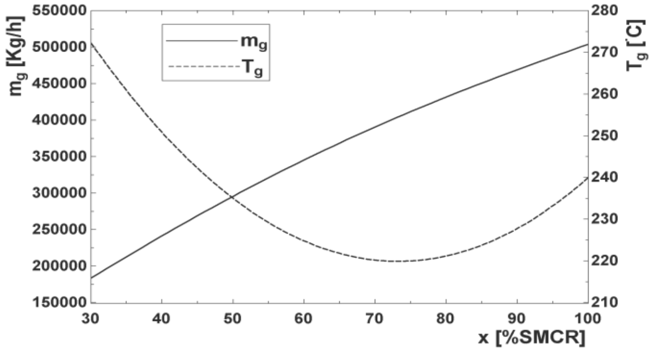

Figure 1 Exhaust gas temperature and amount after turbocharger of intelligent marine diesel engine—10S90ME of MAN B&W

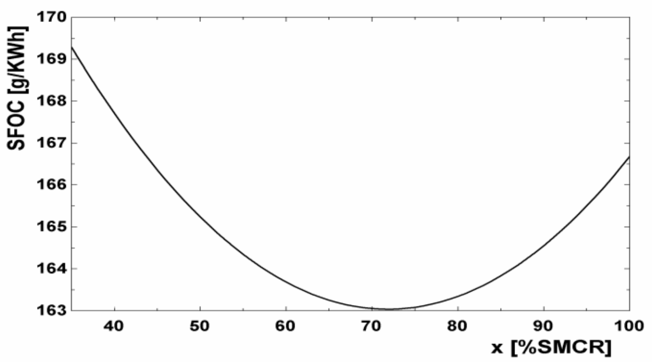

Figure 2 Specific Fuel Oil Consumption(SFOC) of intelligent marine diesel engine—10S90ME of MAN B&W

3. THERMODYNAMIC MODELS OF ORDINARY RANKINE CYCLE(RC) CONCEPTUAL WASTE HEAT RECOVERY SYSTEM

Ordinary Rankine Cycle(RC) conceptual waste heat recovery system diagram is shown in Figure 3.Temperature profiles of the exhaust gas and steam/water in the exhaust gas boiler are shown in Figure 4 and T-S diagram is shown in Figure 5, respectively.

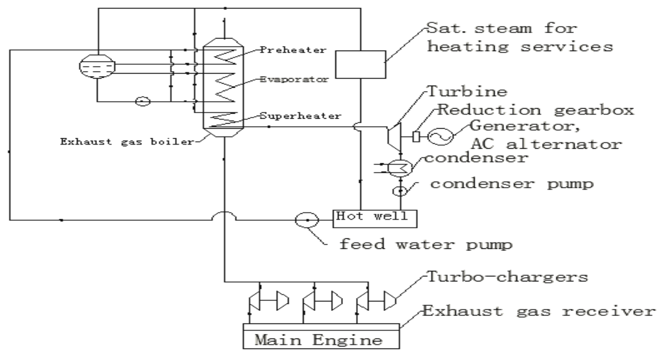

Figure 3 Ordinary Rankine Cycle(RC) conceptual waste heat recovery system diagram

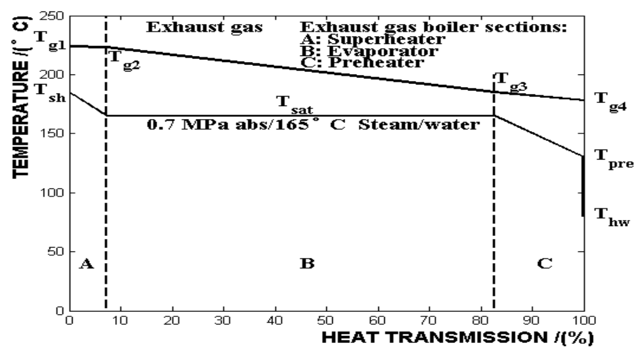

Figure 4 Temperature/heat transmission diagram of exhaust gas boiler

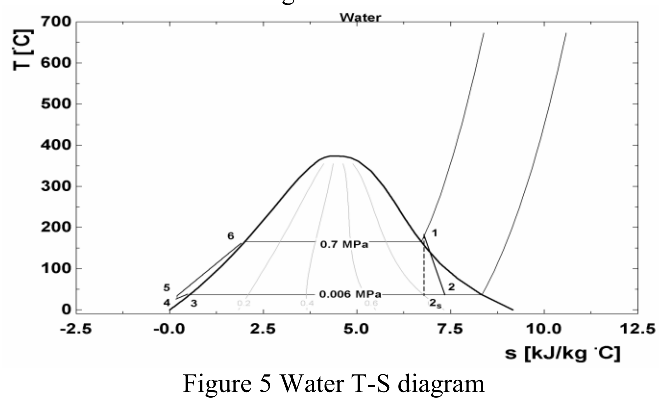

Figure 5 Water T-S diagram

3.1 Formulae of cycle

Formulae of preheater. By mixing the water from hot well ($\dot{m}_{hw}$) and a part of saturated water from the boiler ($\dot{m}_{B1}$), the water with a mass flow rate $\dot{m}$ could enter the preheater at T_{pre} (130℃). The mass flow rate $\dot{m}$ can be expressed as

$$\dot{m} = \dot{m}_{hw} + \dot{m}_{B1} \tag{1}$$

According to the law of energy conservation, there will be

$$\dot{m}\times h_{pre} = \dot{m}_{hw}\times h_{hw} + \dot{m}_{B1}\times h'_{sat} \tag{2}$$

Using the temperature profiles, one can reach

$$(\dot{m}_{hw}+\dot{m}_{B1})\times(h'_{sat}-h_{pre}) = C_g\times\dot{m}_g\times(T_{g3}-T_{g4})\times\eta_B \tag{3}$$

Formulae of Evaporator. The exhaust gas temperature entering the preheater is expressed as

$$T_{g3}=T_{sat}+PP \tag{4}$$

According to the law of energy conservation, one can obtain

$$\dot{m}_{exh,B}\times(h''_{sat}-h'_{sat}) = C_g\times\dot{m}_g\times(T_{g2}-T_{g3})\times\eta_B \tag{5}$$

Formulae of Superheater. According to the law of conservation of energy, there will be

$$\dot{m}_{sh}\times(h_{sh}-h''_{sat}) = C_g\times\dot{m}_g\times(T_{g1}-T_{g2})\times\eta_B \tag{6}$$

According to the law of mass conservation, the steam generation $\dot{m}_{exh,B}$ and the mass flow of hot well pump $\dot{m}_{hw}$ can be expressed

$$\dot{m}_{exh,B} = \dot{m}_{heating}+\dot{m}_{sh} \tag{7}$$

$$\dot{m}_{hw} = \dot{m}_{heating}+\dot{m}_{sh} \tag{8}$$

Objective Formulae. According to the above formulae, the superheated steam generation $\dot{m}_{sh}$ and temperature T_{g4} can be expressed as Eq. (9)-(10)

$$\dot{m}_{sh} = \frac{C_g\times\dot{m}_g\times(T_{g1}-T_{g3})\times\eta_B+\dot{m}_{heating}\times(h_{sh}-h''_{sat})}{h_{sh}-h'_{sat}} - \dot{m}_{heating} \tag{9}$$

$$T_{g4}=T_{g3}-\frac{(\dot{m}_{exh,B}+\dot{m}_{B1})\times(h'_{sat}-h_{pre})}{C_g\times\dot{m}_g\times\eta_B} \tag{10}$$

The electric power yield of the steam turbine generation W_T, the net electric power of waste heat recover system W_{net}, the heat recovery efficiency η_{hr} can be written as Eq.(11)-(14)

$$W_T=\dot{m}_{sh}\times(h_{sh}-h_{2s})\times\eta_s\div3600 \tag{11}$$

Where η_s is the isentropic efficiency of the turbine.

The exhausted electricity of feed water pump is

$$W_{pp}=\dot{m}_{hw}\times(h_5-h_4)\div\eta_{pp}\div3600 \tag{12}$$

Where η_{pp} is the efficiency of feed water pump.

$$W_{net}=W_T-W_{PP} \tag{13}$$

The heat recovery efficiency is expressed as

$$\eta_{hr}=\frac{T_{exh,in}-T_{exh,out}}{T_{exh,in}-T_{amb}} \tag{14}$$

Where $T_{exh,in}$ is the exhaust gas temperature at boiler inlet and $T_{exh,out}$ is the exhaust gas temperature at boiler outlet and T_{amb} is the ambient temperature.

3.2 Exergy analysis of cycle

The environment reference state is as follows:

The enthalpy of water at reference state: $h_0=109kJ/kg$, the entropy of water at reference state: $s_0=0.3809kJ/(kgK)$ ($T_0=299.15K$, $p_0=0.1MPa$).

The enthalpy of exhaust gas at reference state: $h_0=299.5kJ/kg$, the entropy of exhaust gas at reference state: $s_0=6.867kJ/(kgK)$ ($T_0=299.15K$, $p_0=0.1MPa$).

Available exergy of exhaust gas. The available exergy of exhaust gas $E_{exh,gas}$ is as follows:

$$E=E_{exh,gas}=(e_{exh,T_{g1}}-e_{exh,T_{g4}})\times\dot{m}_g\times\eta_B= \tag{15}$$
$$[(h_{exh,T_{g1}}-h_{exh,T_{g4}})-T_0\times(s_{exh,T_{g1}}-s_{exh,T_{g4}})]\times\dot{m}_g\times\eta_B$$

Exergy loss of exhaust gas boiler. The exergy loss of exhaust gas boiler is as follows:

$$\Delta E_B=E_{exh,gas}-E_{sat,steam}-E_{sup,steam} \tag{16}$$

Exergy loss of turbine. Exergy loss of turbine ΔE_T is as follows:

$$\Delta E_T=(e_1-e_2)\times\dot{m}_{sh}-(h_1-h_2)\times\dot{m}_{sh}=T_0\times(s_2-s_1)\times\dot{m}_{sh} \tag{17}$$

Exergy loss of condenser. Exergy loss of condenser ΔE_C is as follows:

$$\Delta E_C=(e_2-e_4)\times\dot{m}_{sh} \tag{18}$$

Exergy loss of feed water pump. The irreversible loss in compression is ignored, so the exergy loss of feed water pump is zero.

$$\Delta E_{pp}=0 \tag{19}$$

4. THERMODYNAMIC MODELS OF ORGANIC RANKINE CYCLE(ORC) CONCEPTUAL WASTE HEAT RECOVERY SYSTEM

Organic Rankine Cycle (ORC) conceptual waste heat recovery system diagram and T-S diagram are shown in Figure 6, Figure 7, respectively.

Be similar to ordinary Rankine Cycle (RC) conceptual waste heat recovery system, Organic Rankine Cycle (ORC) conceptual waste heat recovery system is not only yielding electricity, but also yielding saturated steam for heating service. The loop yielding the saturated steam for heating service and the loop yielding electricity using organic rankine cycle are independent. The loop yielding the saturated steam for heating service includes preheater and evaporator.

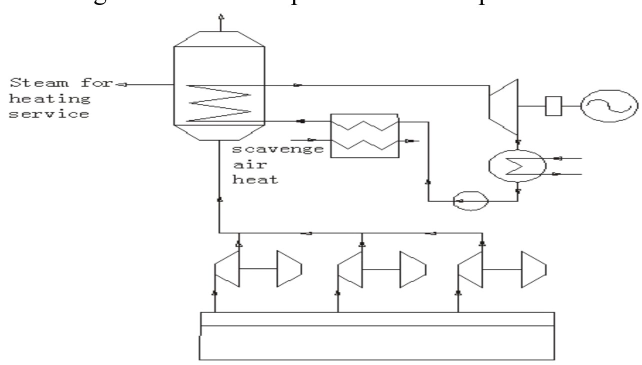

Figure 6 Organic Rankine Cycle (ORC) conceptual waste heat recovery system diagram

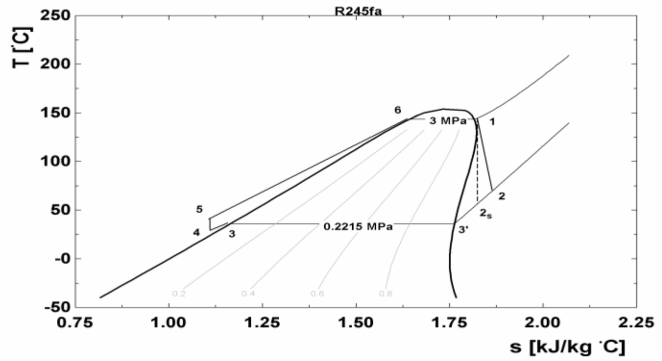

Figure 7 R245fa T-S diagram

4.1 Formulae of cycle

Formulae of the loop yielding the saturated steam for heating service.

Formulae are the same with Eq.(1)-(5) while

$$\dot{m}_{exh,B} = \dot{m}_{heating} \tag{20}$$

$$\dot{m}_{hw} = \dot{m}_{heating} \tag{21}$$

Formulae of the loop yielding electricity using organic rankine cycle. The temperature of organic fluid after condenser is low and it must be preheated before entering the exhaust gas boiler. Therefore, the waste heat of scavenge air is recovered to heat the organic fluid. The needed heat to preheat the organic fluid is calculated by formula (22) and according to the law of energy conservation, formula (23) can be obtained,

$$\dot{Q}_{need} = \dot{m}_{ORC} \times (h_{pre,ORC} - h_{pp,ex,ORC}) \tag{22}$$

$$\dot{m}_{ORC} \times (h''_{sat,ORC} - h_{pre,ORC}) = C_g \times \dot{m}_g \times (T_{g1} - T_{g2}) \times \eta_B \tag{23}$$

Objective Formulae. According to the above formulae, the organic fluid steam generation $\dot{m}_{ORC}$ of the WHRS and temperature T_{g4} can be expressed as Eq. (24)-(25)

$$\dot{m}_{ORC} = \frac{C_g \times \dot{m}_g \times (T_{g1} - T_{g2}) \times \eta_B}{h''_{sat,ORC} - h_{pre,ORC}} \tag{24}$$

$$T_{g4} = T_{g3} - \frac{(\dot{m}_{exh,B} + \dot{m}_{B1}) \times (h'_{sat} - h_{pre})}{C_g \times \dot{m}_g \times \eta_B} \tag{25}$$

The electric power yield of the steam turbine generation $W_{T,ORC}$, the net electric power of waste heat recover system $W_{net,ORC}$, the heat recovery efficiency η_{hr} can be written as Eq.(26)-(29)

$$W_{T,ORC} = \dot{m}_{ORC} \times (h_1 - h_{2s}) \times \eta_s \div 3600 \tag{26}$$

Where η_s is the isentropic efficiency of the turbine.

The exhausted electricity of organic fluid pump is

$$W_{PP,ORC} = \dot{m}_{ORC} \times (h_5 - h_4) \div \eta_{PP} \div 3600 \tag{27}$$

Where η_{PP} is the efficiency of organic fluid pump.

$$W_{net,ORC} = W_{T,ORC} - W_{PP,ORC} \tag{28}$$

The heat recovery efficiency is expressed as

$$\eta_{hr} = \frac{T_{exh,in} - T_{exh,out}}{T_{exh,in} - T_{amb}} \tag{29}$$

Where $T_{exh,in}$ is the exhaust gas temperature at boiler inlet and $T_{exh,out}$ is the exhaust gas temperature at boiler outlet and T_{amb} is the ambient temperature.

4.2 Exergy analysis of cycle

The environment reference state is as follows:

The enthalpy of R245fa at reference state: $h_{0,ORC} = 426.2kJ/kg$, the entropy of R245fa at reference state: $s_{0,ORC} = 1.787kJ/(kgK)$ ($T_0 = 299.15K$, $p_0 = 0.1MPa$).

Available exergy input into the ORC WHR system. The available exergy of exhaust gas $E_{exh,gas}$ is as follows:

$$E_{exh,gas} = (e_{exh,T_{g1}} - e_{exh,T_{g4}}) \times \dot{m}_g \times \eta_B = \\ [(h_{exh,T_{g1}} - h_{exh,T_{g4}}) - T_0 \times (s_{exh,T_{g1}} - s_{exh,T_{g4}})] \times \dot{m}_g \times \eta_B \tag{30}$$

Available exergy E is as follows:

$$E = E_{exh,\,gas} + \dot{Q}_{need} \tag{31}$$

Exergy loss of exhaust gas boiler. The exergy loss of exhaust gas boiler ΔE_B is as follows:

$$\Delta E_B = E_{exh,gas} + \dot{Q}_{need} - E_{sat,steam} - E_{sat,ORCsteam} = \\ (e_{exh,T_{g1}} - e_{exh,T_{g4}}) \times \dot{m}_g \times \eta_B + \dot{m}_{ORC} \times (h_{pre,ORC} - h_{pp,ex,ORC}) - \\ (e_{T_{sat,steam}} - e_{5,water}) \times \dot{m}_{heating} - (e_1 - e_5) \times \dot{m}_{ORC} \tag{32}$$

Exergy loss of turbine. Exergy loss of turbine ΔE_T is as follows:

$$\Delta E_T = (e_1 - e_2) \times \dot{m}_{ORC} - (h_1 - h_2) \times \dot{m}_{ORC} = \\ T_0 \times (s_2 - s_1) \times \dot{m}_{ORC} \tag{33}$$

Exergy loss of condenser. Exergy loss of condenser ΔE_C is as follows:

$$\Delta E_C = (e_2 - e_4) \times \dot{m}_{ORC} \tag{34}$$

Exergy loss of organic fluid pump. The irreversible loss in compression is ignored, so the exergy loss of organic fluid pump is zero.

$$\Delta E_{pp} = 0 \tag{35}$$

5. THERMODYNAMIC OPTIMIZATION

5.1 Ordinary Rankine Cycle (RC) conceptual waste heat recovery system

The exhaust gas temperature left boiler is higher than 166℃ and the temperature of water entering into boiler's preheater should be higher than 130℃ to avoid the risk of condensed sulfuric acid (the temperature of water entering into boiler's preheater is selected 130 ℃). The thermodynamic optimization aims at maxing the net power output W_{net} . According to Eq.(1)-(17), the net power output W_{net} varies with five variables—percentage of engine load x 、 exhaust gas boiler steam pressure p_1 、 condensation pressure p_2 、 superheated steam temperature T_{sh} and pinch point temperature PP . x varies from 0 to 100, p_1 varies from 0.7MPa to 1.0MPa, p_2 varies from 6kPa to 15kPa, T_{sh} varies from 180℃ to 185℃ and PP varies from 20℃ to 50℃. The NCR is 85%SMCR, i.e. $x = 85$ and WHR system is designed at this main engine load.

Thermodynamic optimization shows that the maximum of the net power output— $W_{net} = 988.8kW$ occurs at $p_1 = 0.7MPa$ 、 $p_2 = 6kPa$ 、 $T_{sh} = 185℃$ 、 $PP = 20℃$. It is clear that the bigger the superheated steam temperature、the smaller the exhaust gas boiler steam pressure and the condensation pressure and the pinch point temperature, the bigger the net power output.

5.2 Organic Rankine Cycle (ORC) conceptual waste heat recovery system

In many studies [6,7,8,9,10], it shows that the recommended working fluid is the one with the highest critical temperature so that the plant efficiency could achieve the maximum. The pre-selection of organic fluid is performed according to the following criteria:

•The working fluid should have a critical temperature near the exhaust gas temperature;

•The standard boiling point should near the ambient air temperature;

•It should be a well-known working fluid in the ORC field, i.e. a fluid that has been previously studied in the scientific literature or fluids that are used in commercial ORC power plants, such as solkatherm, n-pentane or R134a [11].

The final selection of working fluid candidates is described in Table 1.

For better compare and contrast, ORC WHR systems have the same condensation temperature—36.17℃—with RC WHR system, and the same temperature of organic fluid entering into boiler's preheater—130℃—with RC WHR system.The thermodynamic optimization also aims at maxing the net power output— $W_{net,ORC}$. What's more, superheating in ORC WHR systems does no good to increase the net power output and actually decreases the net power output. Therefore, the organic fluid enters into the turbine at saturated state and there is no need to worry about the liquid hammering which may occur in RC WHR system. Based on the above condition, the net power output— $W_{net,ORC}$ varies with only one variable—evaporation temperature T_{eva} . For each candidate, T_{eva} varies from 130℃ to the critical temperature.

Table 1 List of considered working fluids

Candidates	$T_{critical}$ ℃	$P_{critical}$ MPa	$T_{boiling}$ ℃
R113	214.1	3.439	47.59
R123	183.7	3.668	27.82
R141b	204.2	4.249	32.05
R245fa	154.1	3.639	15.14
HFE-7 000	164.5	2.478	35.23
n-pentane	196.5	3.364	35.87

5.3 Most promising candidate for Organic Rankine Cycle (ORC) conceptual waste heat recovery system

Table 2 shows the thermodynamic optimization results for RC and ORC WHR systems.

Table 3 shows with the same steam temperature—185℃—the net power output in RC WHR system and in ORC WHR systems.

According to Table 2, it is clear that the net power output maximum in ORC WHR systems are all bigger than in RC WHR system. According to Table 3, it is clear that the net power output in ORC WHR systems (except R123、R245fa、HFE-7 000) are all bigger than in RC WHR system with the same steam temperature—185℃. However, for candidates R123 and n-pentane, the net power output maximum occurs at the critical temperature, and for candidates R113 and R245fa, the net power output maximum occurs near the the critical temperature especially for R245fa. Different from other candidates, HFE-7 000 shows an opposing trend, i.e. the lower the evaporation temperature, the bigger the net power output.

By compare and contrast, the most promising candidate for ORC WHR system is R245fa.

Table 2 Thermodynamic optimization

Working fluid	$\dot{W}_{net}$ kW	T_{eva} ℃	T_{crit} ℃
water	988.8	185	374
R113	1 089	202.3	214.1
R123	1 247	183.7	183.7
R141B	1 112	193.5	204.2
R245fa	1 253	153.5	154.1
HFE-7 000	1 131	130	164.5
n-pentane	1 176	196.5	196.5

Table 3 Net power output with the same steam temperature—185℃

Working fluid	$\dot{W}_{net}$ kW	T_{eva} ℃	T_{crit} ℃
water	988.8	185	374
R113	1 081	185	214.1
R123	No exist	185	183.7
R141B	1 106	185	204.2
R245fa	No exist	185	154.1
HFE-7 000	No exist	185	164.5
n-pentane	1 114	185	196.5

6. RESUELTS AND DISCUSSION

The main engine 10S90ME is assumed to be installed on a 10 000TEU container ship. The saturated steam flow rate for heating service is 2 500kg/h. Condenser is cooled by sea water. Subcooled condition occurred in condenser. The subcooling temperature of condenser is 1℃. Sea water inlet temperature is 26℃, sea water outlet temperature is 31℃. The following results and discussion are all based on NCR conditions and for ORC WHR system, R245fa—most promising candidate—is selected.

6.1 Thermodynamic performance

Exhaust gas exit temperature. Exhaust gas exit temperature in RC WHR system and ORC WHR system variation with main engine load is shown in Figure 8. At 30%—100% load range, the exhaust gas exit temperature are both higher than 166℃ so that the risk of condensed sulfuric acid could be avoided.

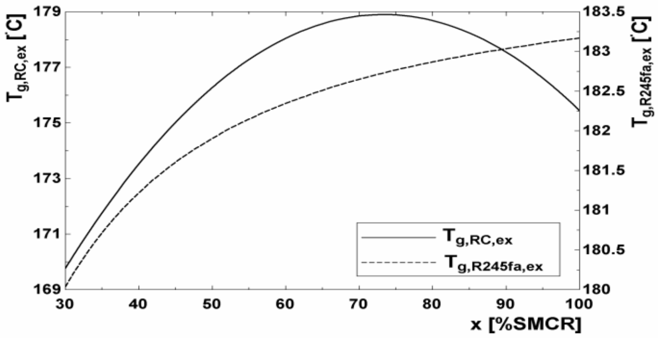

Figure 8 Exhaust gas exit temperature variation with main engine load

Heat recovery efficiency. Heat recovery efficiency of RC WHR system and ORC WHR system is shown in Figure 9. The minimum—23.65%、21.45%—occurs at 73%SMCR、74%SMCR for RC WHR system and ORC WHR system, respectively .

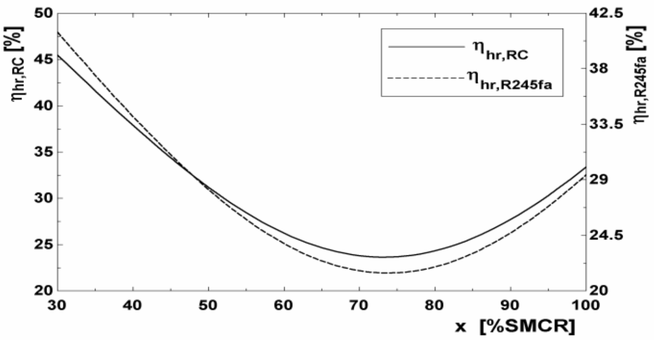

Figure 9 Heat recovery efficiency variation with main engine load

Exhausted electricity by pump. Exhausted electricity by pump in RC WHR system and ORC WHR system is shown in Figure 10. With the main engine load increasing, exhausted electricity by pump first decrease slightly then increase sharply. Due to the very small scale of exhausted electricity by pump in RC WHR system, the change could also be neglected. However, the sharp increase in ORC WHR system is so big that it could not neglected.

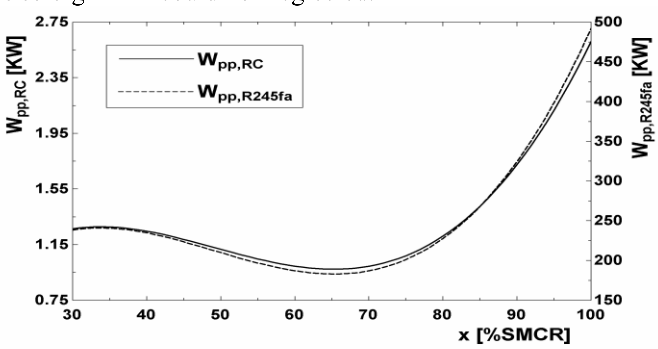

Figure 10 Exhausted electricity by pump variation with main engine load

Turbine power output. Turbine power output in RC WHR system and ORC WHR system is shown in Figure 11. With the main engine load increasing, turbine power output first decrease slightly then increase sharply. It is clear that at 30%—100% load range, turbine power output in ORC WHR system is bigger than that in RC WHR system.

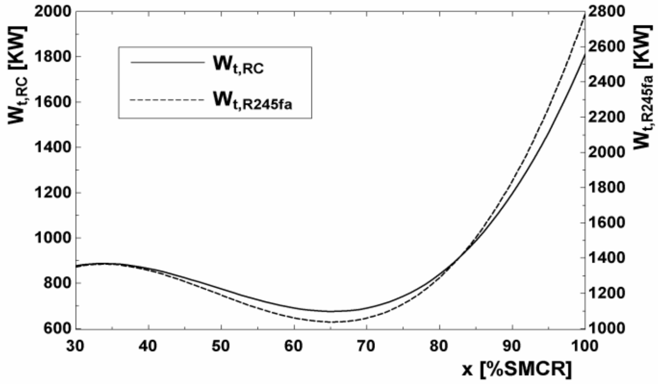

Figure 11 Turbine power output variation with main engine load

Net power output. Net power output in RC WHR system and ORC WHR system is shown in Figure 12. Net power output means turbine power output minus exhausted electricity by pump. It is clear that at 30%—100% load range, net power output in ORC WHR system is bigger than that in RC WHR system. The changing trend of net power output in RC WHR system and ORC WHR system is very similar.

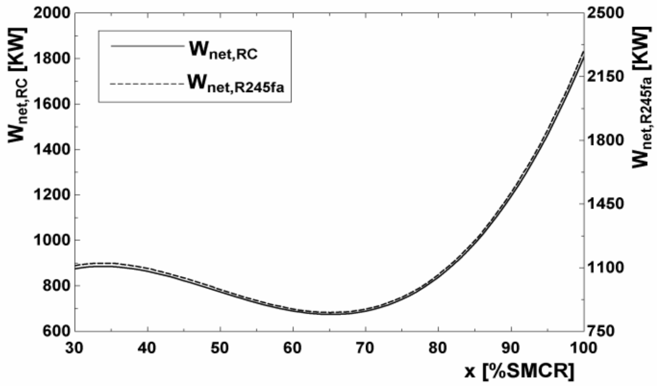

Figure 12 Net power output variation with main engine load

Evaporation temperature influence on net power output in ORC WHR system. The influence of evaporation temperature on net power output is shown in Figure 13. At 130℃—153.5℃ evaporation temperature range, net power output varies from 1197kW to 1253kW. It shows that net power output increses 2.38kW with per evaporation temperature increase.

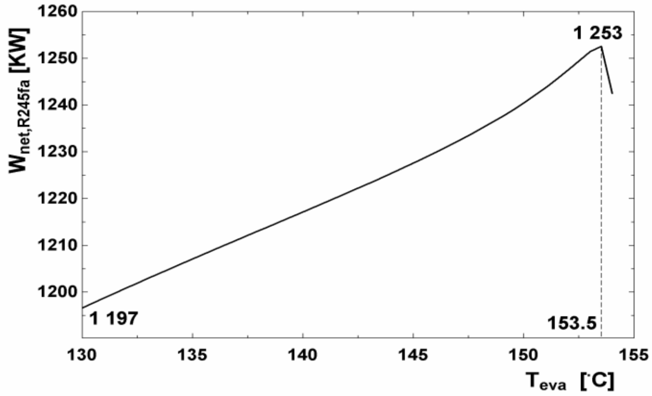

Figure 13 Net power output variation with evaporation temperature

6.2 Feasibility analysis

Electricity demand onboard.The container ship studied operates in NCR conditions about 280 days per year. The average number of the refrigerated containers is 450 and each one consumes about 11.4kW electricity with 64% loading rate. The total electricity consumption of the refrigerated containers is 5 130kW, the daily electricity consumption is 2 000kW and therefore the total electricity consumption onboard is 7 130kW. The ship should install 4×2 820kW diesel generators without any WHR system. In this case, three diesel generators operate normally while the left as standby.

Electricity yield by WHR ststem.According to the above thermodynamic performance analysis, net power output by RC WHR system and ORC WHR system in NCR conditions is 988.8kW and 1 253kW, respectively. It is obvious that net power output by WHR system is far from enough to meet the electricity demand (7 130kW) onboard. What's more, net power output by WHR system is very less than the rated capacity—2 820kW of per diesel generator. It means that no diesel generator could be displaced by WHR system, and the cost of diesel generators could not be cut down.

Waste heat of intelligent marine diesel engine—10S90ME could be theoretically recovered. However, due to the very low exhaust gas temperature, the electricity yield by WHR system could not meet the electricity demand (7 130kW) onboard. WHR system is actually impracticable.

The very low exhaust gas temperature brings a dilemma to install traditional WHR systems for intelligent marine diesel engine.

7. CONCLUSION

Based on derived thermodynamic models, WHR system performance analysis and feasibility analysis for intelligent marine diesel engine—10S90ME of MAN B&W—installed on a large container ship with very low exhaust gas temperature/high amount mass flow rate, the conclusions are as follows:

- ORC WHR system has a better performance due to more net power output;
- The most promising candidate for ORC WHR system is R245fa;
- Net power output in RC WHR system and ORC WHR system are both very less than the rated capacity—2 820kW of per diesel generator;
- RC WHR system and ORC WHR system are both impracticable;
- The very low exhaust gas temperature brings a dilemma to install traditional WHR systems for intelligent marine diesel engine.

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Nomenclature

C_g	specific heat of the exhaust gas	kJ/（kg K）
e	specific exergy	kJ/kg
E	available exergy	kW
ΔE	exergy loss	kW
h	specific enthalpy	kJ/kg
$\dot{m}$	mass flow	kg/h
NCR	Normal Continuous Rating	kW
p	pressure	MPa
PP	pinch point	K
$\dot{Q}_{need}$	the needed heat to preheat the organic fluid	kW
s	specific entropy	kJ/（kg K）
$SFOC$	specific fuel oil consumption	g/（kW h）
$SMCR$	Specified Maximum Continuous Rating	kW
T	temperature	K
W	work output	kW

Greek symbols

η	efficiency
η_B	exhaust gas boiler efficiency considering the radiation loss
η_{hr}	heat recovery efficiency
η_{pp}	pump efficiency
η_s	turbine isentropic efficiency

Subscripts

amb	ambient air
$back$	back pressure of steam turbine
B	exhaust gas boiler
B_1	part of saturated water from the boiler
C	condenser
exh	exhaust gas
exh,B	exhaust gas boiler
exh,in	exhaust gas at boiler inlet
exh,out	exhaust gas at boiler outlet
ex	exit
$g1$	exhaust gas at superheater inlet
$g2$	exhaust gas at evaporator inlet
$g3$	exhaust gas at economizer inlet
$g4$	exhaust gas at economizer outlet
$heating$	heating service onboard
hw	hot well
net	net electric power of waste heat recovery system
ORC	ORC system
pp	working fluid pump
pre	preheater
sat	saturated
sh	superheated
sup	superheated
T	steam turbine
0	reference state
1	superheater outlet state
2	expander outlet state
2_s	expander outlet isentropic state
3	saturated liquid state at condensing pressure
$3'$	saturated steam state at condensing pressure
4	condenser outlet state
5	pump outlet state

Superscripts

‘	saturated steam
‘‘	saturated water