CONCEPTUAL DESIGN AND PERFORMANCE ANALYSIS OF WASTE HEAT RECOVERY SYSTEM FOR INTELLIGENT MARINE DIESEL ENGINES. PART 2: INTEGRATING POWER TURBINE INTO 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. For intelligent marine diesel engines, it is clear that the very low exhaust gas temperature after the turbocharger has led the impracticable installation of traditional WHR systems—ordinary Rankine Cycle (RC) conceptional waste heat recovery system and Organic Rankine Cycle (ORC) conceptional waste heat recovery system—onboard. To deal with this dilemma, the way out is obtained from the increased exhaust gas temperature by integrating power turbine into WHR systems. Therefore, Thermo Efficiency System (TES system proposed by MAN B&W), TES-Organic Rankine Cycle (ORC) system and TES-Screw Expander Generator (SEG) system have been developed to improve such vessels energy efficiency. In this paper, thermodynamic models for TES, TES-ORC and TES-SEG matching 10S90ME of MAN B&W—a typical intelligent marine diesel engine—are derived and numerically calculated. Numerical results indicate that the above three systems are more feasible than traditional WHR systems. Though the total power yield of TES-ORC system is the highest, TES system and TES-SEG system are more advantageous for their system simplicity and safety consideration.

1. INTRODUCTION

According to Part 1, the very low exhaust gas temperature after turbochargers brings a dilemma to install traditional WHR systems—ordinary Rankine Cycle (RC) conceptional waste heat recovery system and Organic Rankine Cycle (ORC) conceptional waste heat recovery system—for intelligent marine diesel engine onboard.

MAN B&W developed Thermo Efficiency System (TES) to overcome such weakness of intelligent marine diesel engines [1]. The principle of TES is to redistribute the exhaust gas heat from high amount/low temperature to low amount/high temperature—with changed timing and exhaust gas bypass—which increases the exergy of the exhaust gas heat. TES construction consists of a large exhaust gas boiler, steam turbine, power turbine, generator, condenser, feed water pump, pipes, and control systems. A certain amount of exhaust gas for the power turbine (up to 12% of the total exhaust gas amount at 100% SMCR, Specified Maximum Continuous Rating) is bypassed to increase mixed exhaust gas temperature to higher level up to 50℃, reduction of total exhaust gas amount is approximately 13% and excess fuel consumption is up to 1.8%. Both single-pressure steam system and dual-pressure steam system can be employed as exhaust gas boiler and steam systems. Total electric power production of the two systems is approximately 8.5% and 9.5% of the main engine power output, respectively. However, TES system is rather expensive, and its payback time depends very much on the size of the main engine and the trade pattern (main engine load and ambient temperatures) of the ships. Payback time varies from 5 years to 12 years. Therefore, the installation of the TES is normally only relevant for the large merchant ships, such as the large container vessels.

Another selection to recover heat from low grade waste heat sources is Organic Rankine Cycle (ORC) method. WHR ORC applications on both large and small scale are reported in published literatures[2-14]. References[2-5] mainly focused on ORC working fluid selection and pointed that ORC cycle efficiency is very sensitive to working fluid evaporating pressure. References[6-9] discussed several ORC cycle design and embodiment. References[10-12] focused on performance analysis of ORC systems and cycles, such as thermodynamic and thermo-economic optimization, modeling and dynamic simulation. However, the study of WHR ORC applied on ship is not that much. Yalcin Durmusoglu et al.[13] theoretically designed an energy saving and power solution using WHR ORC for a container ship, proposed three performance analysis criterion, but they did not discuss appropriate organic fluids for marine WHR ORC use and did not do thermodynamic optimization and thermo-economic analysis. Guoqiang Yue et al.[14] designed a marine Diesel engine waste heat recovery system with Organic Rankine Cycle, gave out the first law and second law analysis results and designed the most important components for ORC system—steam turbine.

This paper proposes three most promising conceptual WHR design—TES(Thermo Efficiency System，proposed by MAN B&W), TES-ORC(in which takes Organic Rankine Cycle instead of Rankine Cycle) and TES-SEG(in which employs screw-expander-generator instead of steam-turbine-generator) WHR system—for 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—to deal with the dilemmma of impracticable installation of traditional WHR systems onboard. 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—AFTER ADOPTING TES METHOD

Exhaust gas temperature after turbocharger and amount of intelligent marine diesel engine—10S90ME of MAN B&W—after adopting TES method—is shown in Figure 1. The temperature is 283.8℃ and the amount of the exhaust gas at NCR (85%SMCR) is 360 593kg/h.Specific Fuel Oil Consumption (SFOC) is shown in Figure 2. Compared with standard engine(without adopting TES method), exhaust gas temperature increases 60℃ and the amount of exhaust gas decreases 90149kg/h at NCR and SFOC increases 1.8%.

The essence of TES method is to increase exhaust gas temperature which inevitably increases SFOC slightly. Three most promising conceptual WHR design—TES, TES-ORC and TES-SEG WHR system—are all based on TES method. TES and TES-ORC WHR system diagram is shown in Figure 3 and Figure 6, respectively. Temperature profiles of the exhaust gas and steam/water in the exhaust gas boiler for TES WHR system are shown in Figure 4. T-S diagram for TES and TES-ORC WHR system is shown in Figure 5 and Figure 7, respectively. Power turbine output is shown in Figure 8. Power turbine outout is 1924KW at NCR.

In this paper, detailed thermodynamic models have been built. TES WHR system has the same formulae with ordinary Rankine Cycle (RC) conceptional waste heat recovery system and TES—ORC WHR system has the same formulae with Organic Rankine Cycle (ORC) conceptional waste heat recovery system(see Part 1).

TES—SEG WHR system employs screw-expander-generator instead of steam-turbine-generator in MAN B&W TES. Screw expander generator is an alternative to steam turbine generator. The working fluid in screw expander could be saturated steam, saturated liquid and even unsaturated liquid and it does not occur liquid hammering while liquid hammering may occur in steam turbine so that there is no need to use superheated steam while using screw expander. Therefore, saturated steam is used in TES—SEG WHR system and there is no superheater so that no superheated steam is yielded. TES—SEG WHR system has the formulae mostly the same with ordinary Rankine Cycle (RC) conceptional waste heat recovery system. Readers could obtain the formulae easily while erasing Formulae of Superheater from the formulae of ordinary Rankine Cycle (RC) conceptional waste heat recovery system.

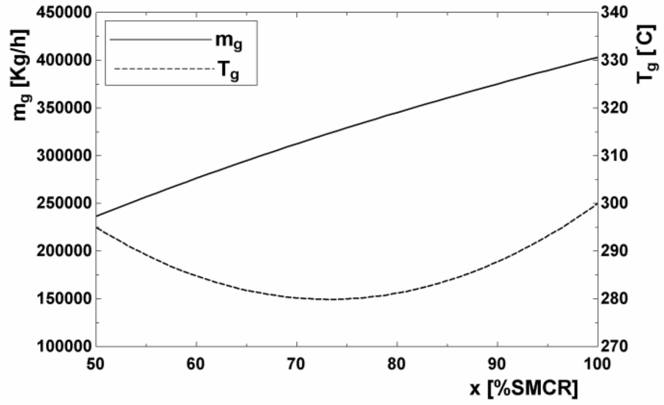

Figure 1 Exhaust gas temperature and amount after turbocharger

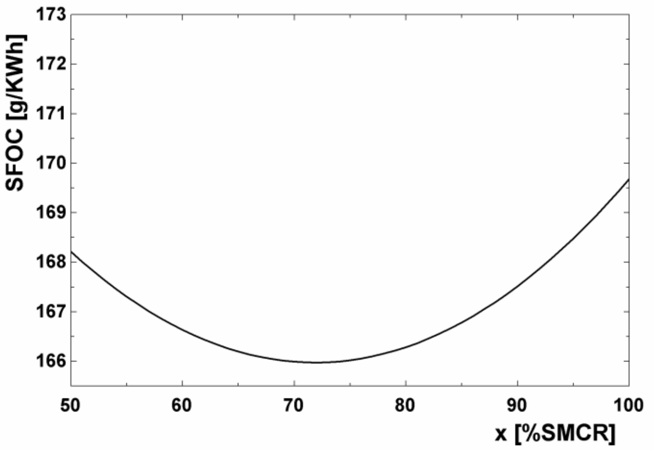

Figure 2 Specific Fuel Oil Consumption(SFOC)

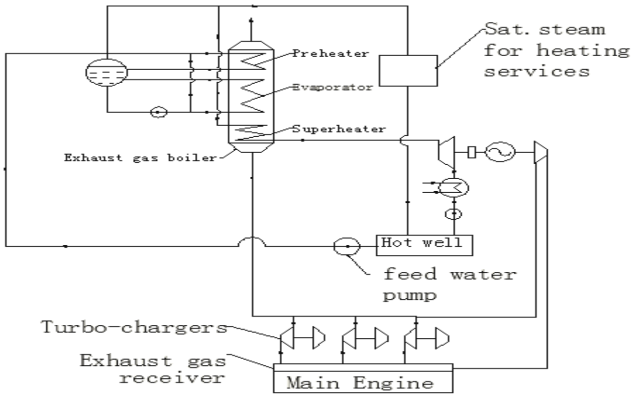

Figure 3 MAN B&W TES diagram

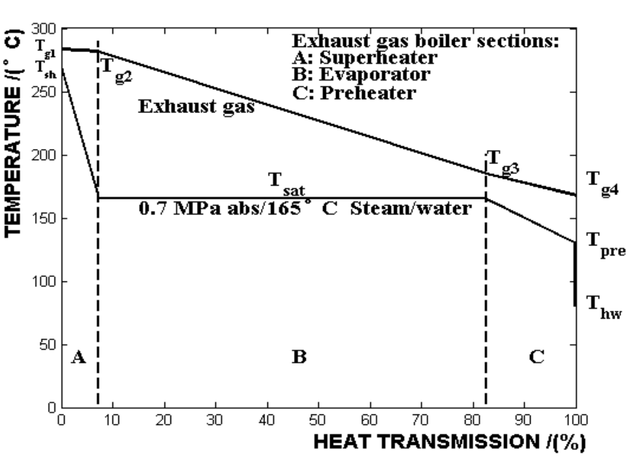

Figure 4 Temperature/heat transmission diagram of exhaust gas boiler

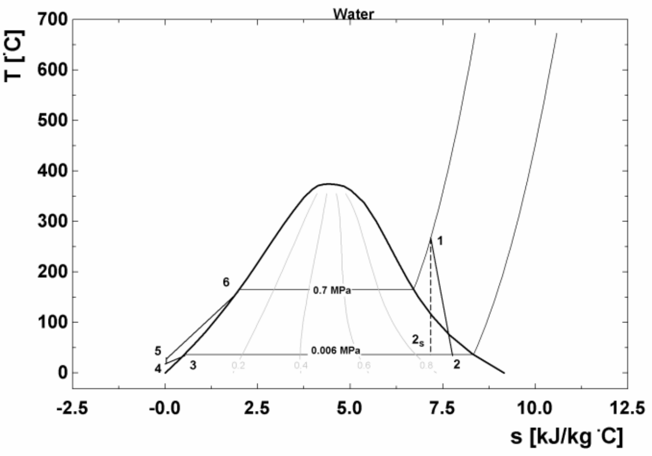

Figure 5 Water T-S diagram

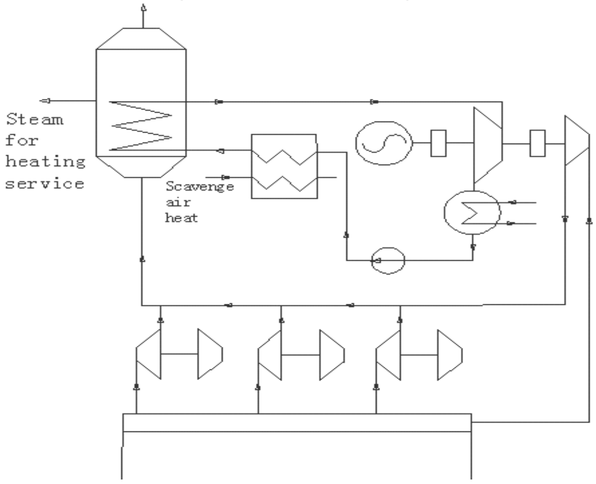

Figure 6 TES-ORC diagram

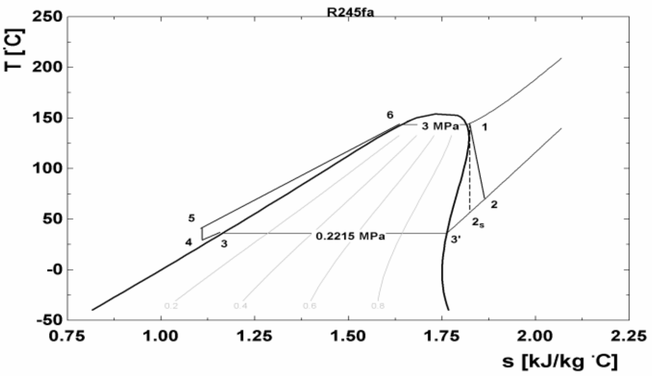

Figure 7 T-S diagram

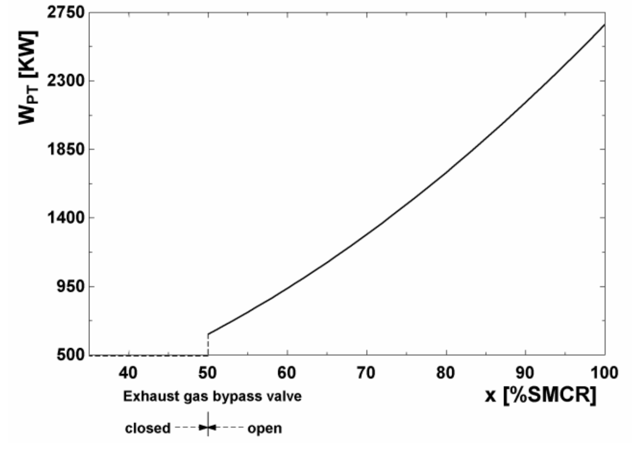

Figure 8 Power turbine output

3. RESUELTS AND DISCUSSION

3.1 Thermodynamic optimization and exergy analysis

The objective function of thermodynamic optimization, optimization method, and organic fluid selection have been studied in Part 1.

Table 1 shows the thermodynamic optimization and exergy loss results for TES, TES-ORC and TES-SEG WHR systems. It is clear that TES-ORC WHR system has more power output than TES and TES-SEG WHR system. R245fa is the most powerful candidate followed closely by R123 and the optimized evaporation temperature is 153.5 ℃ (very near critical temperature154.1℃) for R245fa and is 183.7℃ (critical temperature) for R123. TES and TES-SEG system have the relatively lowest power output and TES-SEG system has the mostly similar performance with TES system.

In this paper, exergy of scavenge air is considered one part of exergy entering the exhaust gas boiler while the other is exergy of exhaust gas. For TES-ORC systems, exergy analysis results show that R245fa is the most effective candidate to recover waste heat from scavenge air followed closely by HFE7000. Exergy loss consists of exhaust gas boiler exergy loss、turbine exergy loss and condenser exergy loss. For exhaust gas boiler exergy loss, HFE7000 is the biggest and R113 is smallest. For turbine exergy loss, R245fa is the biggest and R113 is the smallest. For condenser exergy loss, HFE7000 is the biggest and R141B is the smallest. Therefore, HFE7000 has the biggest total exergy loss percentage followed by R245fa and the smallest is R141B. Compared with TES and TES-SEG systems, TES-ORC systems have relatively higher exergy loss percentages.

Also, only candidate R113 has the condensation pressure lower than ambient pressure and it means vacuum state needed in condenser the same with TES and TES-SEG systems and it will increase the cost. Therefore, in TES-ORC system, R245fa is chosen as the working fluid.

According to the above analysis, R245fa is the most promising candidate used in TES-ORC system because it could recover the most waste heat from exhaust gas and scavenge air and its exergy loss percentage is not the biggest and its condensation pressure is higher than ambient pressure so that no vacuum state needed in condenser.

The container ship studied operates in CSO conditions about 200 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. For TES , TES-SEG and TES-ORC system, total power output is slightly lower or higher than 5 000KW so that a generator with the capacity 5 000KW is needed. Therefore, the evaporation temperature of R245fa should be decreased so that its total power output could be lower than 5 000KW and this will also do good to avoid the supercritical state in evaporator and in turbine. Also, at the range of 130-153.5℃ for R245fa, the higher the evaporation temperature, the bigger the total power output. 145℃ has been chosen as the evaporation temperature while the total power output is 4 990KW.

Table 1 Thermodynamic optimization

	TES	TES-SEG	TES-ORC					
			R113	R123	R141B	R245fa	HFE7000	n-pentane
$\dot{w}_{net}$ KW	2480	2487	2720	3114	2777	3128	2825	2938
T_{eva} ℃	270	165	202.3	183.7	193.5	153.5	130	196.5
T_{crit} ℃	374	374	214.1	183.7	204.2	154.1	164.5	196.5
P_{con} MPa	0.006	0.006	0.0681	0.136	0.1168	0.2215	0.1047	0.1023
$\dot{W}_{total}$ KW	4404	4411	4644	5038	4701	5052	4749	4862
$\dot{m}$ Kg/h	13434	15112	220517	298479	178467	354216	368361	111974
E_g KW	4630	4684	4180	4180	4180	4180	4180	4180
E_{sca} KW	0	0	1779	2524	1845	3232	3086	2278
E_{total} KW	4630	4684	5959	6704	6025	7412	7266	6458
ΔE_B KW	929.9	964.7	1666	1967	1752	2322	2526	1895
ΔE_T KW	423.8	425	435.9	569.9	508.8	628.2	442.6	486.2
ΔE_C KW	267.7	279.1	561.8	474.8	400	670.8	913.8	569
w_{pp} KW	3.256	3.663	238	254.1	294.9	670.2	148.8	209.1
w_T KW	2483	2490	2958	3368	3072	3799	2974	3147
$E_{sat,steam}$ KW	528	528	528	528	528	528	528	528
$\frac{\Delta E_B+\Delta E_T+\Delta E_C}{E_{total}}$ %	35.02	35.63	44.7	44.92	44.17	48.85	53.44	45.68

3.2 Thermodynamic performance analysis

The main engine 10S90ME is assumed to be installed on a 10 00TEU 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 CSO conditions (85%SMCR) and for TES-ORC WHR system, R245fa—most promising candidate—is selected.

Turbine power output. Turbine power output in TES and TES-ORC system is shown in Figure 9. When main engine load is lower than 50%SMCR, the temperature and mass flow rate of exhaust gas are both low and exhaust gas bypass valve will be closed so that exhaust gas boiler could work normally. Therefore, no matter TES or TES-ORC system, they only work together with main engine when the load is higher than 50% SMCR. With the main engine load increasing, turbine power output increase sharply. It is clear that at 50%—100% load range, turbine power output in TES-ORC WHR system is bigger than that in TES WHR system.

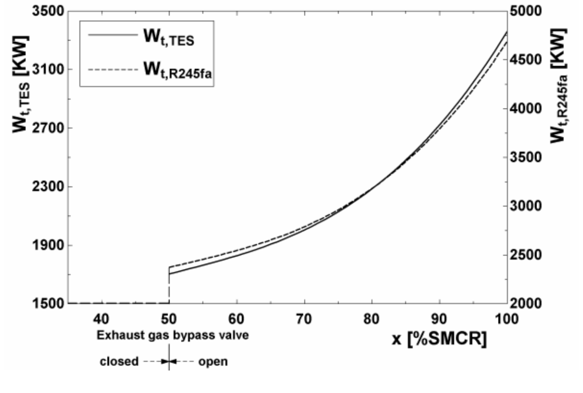

Figure 9 Turbine power output variation with main engine load

Exhausted electricity by pump. Exhausted electricity by pump in TES and TES-ORC system is shown in Figure 10. Due to the almost uncompressible characteristic of water, the very small scale of exhausted electricity by pump in TES WHR system could be neglected. However, the sharp increase in TES-ORC WHR system is so big that it could not neglected.

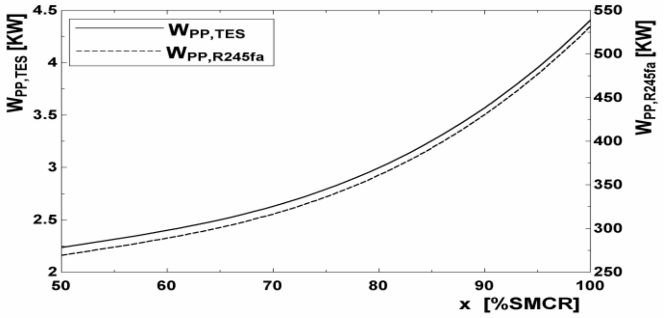

Figure 10 Exhausted electricity by pump variation with main engine load

Net power output. Net power output in TES and TES-ORC system is shown in Figure 11. Net power output means turbine power output minus exhausted electricity by pump. It is clear that at 50%—100% load range, net power output in TES-ORC WHR system is bigger than that in TES WHR system.

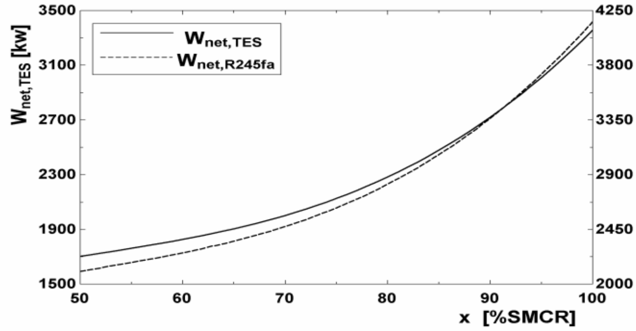

Figure 11 Net power output variation with main engine load

Total power output. Total power output variation with main engine load is shown in Figure 12. Total power output means net power output plus power turbine power output. It is clear that at 50%—100% load range, total power output in TES-ORC WHR system is bigger than that in TES WHR system.

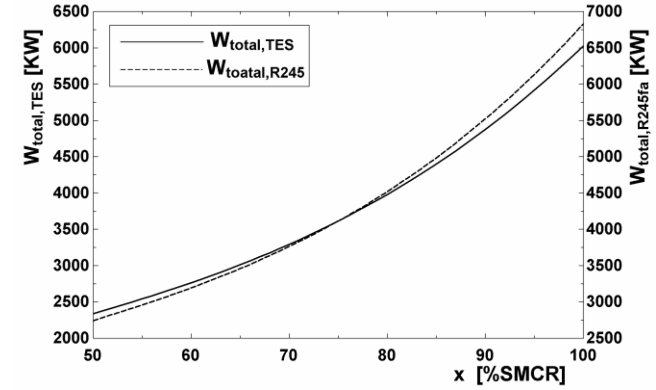

Figure 12 Total power output variation with main engine load

Evaporation temperature influence on total power output in TES-ORC WHR system. The influence of evaporation temperature on total power output is shown in Figure 13. At 130℃—153.5℃ evaporation temperature range, net power output varies from 4 913KW to 5 052KW. It shows that net power output increses 5.91KW with per evaporation temperature increase.

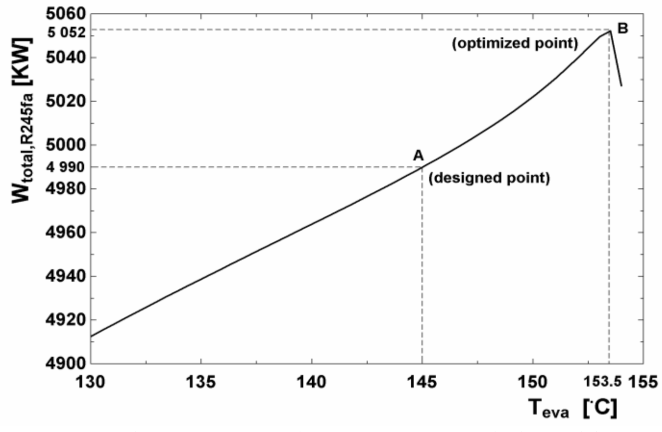

Figure 13 Total power output variation with evaporation temperature

Exhaust gas exit temperature. Exhaust gas exit temperature in TES and TES-ORC system variation with main engine load is shown in Figure 14. At 50%—100% load range, the exhaust gas exit temperature are both higher than 166℃ so that the risk of condensed sulfuric acid could be avoided.

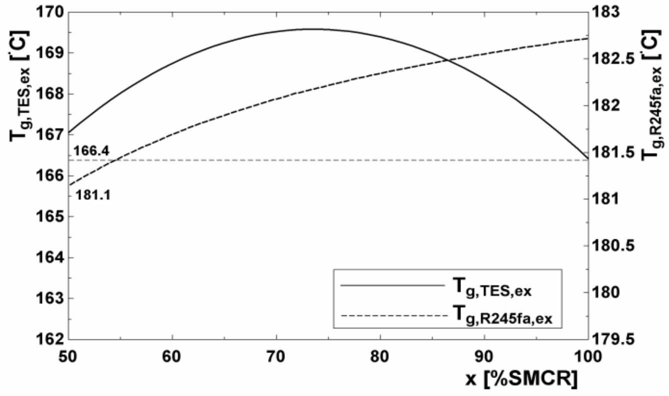

Figure 14 Exhaust gas exit temperature variation with main engine load

Heat recovery efficiency. Heat recovery efficiency of in TES and TES-ORC system is shown in Figure 15. The minimum—47.39%、42.02%—occurs at 73%-74%SMCR、74%SMCR for in TES and TES-ORC system, respectively . Heat recovery efficiency means the ratio of the actual temperature drop (between boiler inlet and boiler outlet) and the theoretical temperature drop (between boiler inlet and ambient). At specific main engine load, boiler inlet temperature is constant, so that a higher heat recovery efficiency means a higher actual temperature drop between boiler inlet and outlet and more exhaust gas waste heat is recovered. It is clear that TES system could recover more exhaust gas waste heat.

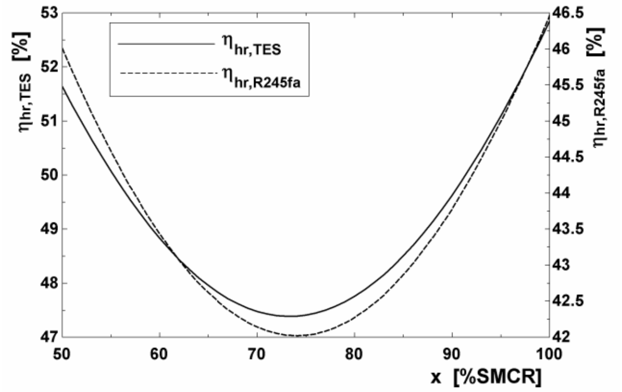

Figure 15 Heat recovery efficiency variation with main engine load

3.3 Feasibility analysis

According to the above thermodynamic performance analysis, total power output in TES and TES-ORC system in CSO conditions is 4 404KW and 4 990KW, respectively. Total power output by TES and TES-ORC system are much more than the rated capacity—2 820KW of per diesel generator. It means that one diesel generator could be displaced by WHR system, and the cost of diesel generators could be cut down. Therefore, with TES or TES-ORC WHR system, the ship could install 3×2 820KW diesel generators with one opreating normally while the other two as standby.

3.4 Payback time

The container ship operates in CSO conditions about 200 days per year. Total electricity requirement onboard is 7 130KW. Total power output in TES and TES-ORC system in CSO conditions is 4 404KW and 4 990KW, respectively.

Therefore, after adopting TES and TES-ORC system, diesel generator power output will be reduced from 7 130KW to 2 726KW and 2 140KW, i.e. only one diesel generator operates normally. Table 2 shows fuel oil cost calculation results.

The current average fuel oil price is 750$/t, suppose the cost of TES and TES-ORC system are the same—90 000 000$, the annual net income ratio is 6%, and Figure 16 shows payback time variation with fuel oil price. In this analysis, only total saved fule oil cost has been taken into account while saved maintenance cost and saved lubricating oil cost have been neglected. According to Figure 16, for TES and TES-ORC system, the payback time is approximately 4.4years and 3.7years. Though the cost of TES and TES-ORC system is very expensive, the higher average fuel oil price makes the payback time less than five years. Container ship has long-period service life—normally 25years, and it is clear that the installation of TES and TES-ORC system will bring huge benifits to the ship owners as long as the average fuel oil price is high.

Table 2 Fuel oil cost calculation results

		Standard Engine	TES	TES -ORC
Main Engine	Power output KW	49385	49385	49385
	SFOC g/KWh	164	166.952	166.952
	Fuel oil price $/t	750	750	750
	Saved fule oil cost $/year	524824	0	0
Diesel Generator	Power output KW	7130	2726	2140
	SFOC g/KWh	183.8	183.8	183.8
	Fuel oil price $/t	750	750	750
	Saved fule oil cost $/year	0	2914039	3301783
Total saved fule oil cost $/year		0	2389215	2776959

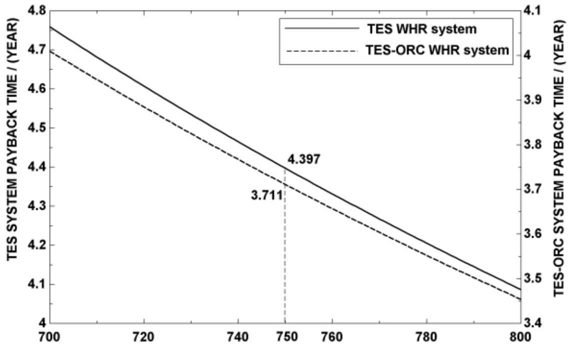

Figure 16 Payback time variation with fuel oil price

4. CONCLUSION

- TES-ORC system has a better performance than TES and TES-SEG system while TES and TES-SEG sytem have the mostly similar performance ;
- The most promising candidate for TES-ORC system is R245fa and the designed evaporation temperature is 145℃;
- Total power output in TES and TES-ORC system are both more than the rated capacity—2 820KW of per diesel generator so that one diesel generator coule be replaced;
- TES and TES-ORC system are theoretically and actually practicable;
- The payback time of TES and TES-ORC system are both less than 5years and will bring huge benifits to the ship owners as long as the average fuel oil price is high.

6. REFERENCES

1. MAN B&W Diesel A/S, Thermo efficiency system for reduction of fuel consumption and CO_2 emission, MAN B&W brochure, Denmark, 2007.
2. O. Badr and P.W. Ocallaghan and S.D. Probert, Rankine-Cycle systems for harnessing power from low-grade energy-sources, Applied Energy, vol. 36, pp. 263-292, 1990.
3. T.C. Hung and T.Y. Shai and S.K. Wang, A review of Organic Rankine Cycles(ORCs) for the rcovery of low-grade waste heat, Energy, vol. 22(7), pp. 661-667, 1997.
4. W. Gu and Y. Weng and Y. Wang and B. Zheng, Theoretical and experimental investigation of an Organic Rankine Cycle for a waste heat recovery system, Part A: Journal of Power and Energy, vol. 223, pp. 523-533, 2009.
5. Y.P. Dai and J.F. Wang and Lin Gao, Parametric optimization and comparative study of Organic Rankine Cycle (ORC) for low grade waste heat recovery, Energy Conversion and Management, vol. 50, pp. 576-582, 2009.
6. Z. Gnutek and A. Bryszewska-Mazurek, The thermodynamic analysis of multicycle ORC engine, Energy, vol. 26, pp. 1075-1082, 2001.
7. Y. Chen and P. Lundqvist and A. Johansson and P. Platell, A comparative study of the carbon dioxide transcritical power cycle compared with an Organic Rankine Cycle with R123 as working fluid in waste heat recovery, Applied Thermal Engineering, vol. 26, pp. 2142-2147, 2006.
8. J. Larjola, Electricity from industrial waste heat using high- speed Organic Rankine Cycle (ORC), International Journal of Production Economics, vol. 41, pp. 227-235, 2009.
9. Sylvain Quoilin and Vincent Lemort and Jean Lebrun, Experimental study and modeling of an Organic Rankine Cycle using scroll expander, Applied Energy, vol. 87, pp. 1260-1268, 2010.
10. Wei D. and Lu X. and Lu Z. and Gu J., Performance analysis and optimization of Organic Rankine Cycle (ORC) for waste heat recovery, Energy Conversion and Management, vol. 48, pp. 1113-1119, 2007.
11. Husband W.W. and Beyene A., Low-grade heat-driven rankine cycle, A Feasibility Study, International Journal of Energy Research, vol. 32, pp. 1373-1382, 2008.

12. Sylvain Quoilin and Sebastien Declaye and Bertrand F. Tchanche and Vincent Lemort., Thermo-economic optimization of waste heat recovery Organic Rankine Cycles, Applied Thermal Engineering, vol. 31, pp. 2885-2893, 2011.
13. Yalcin DURMUSOGLU and Tanzer SATIR and Cengiz DENIZ and Alper KILIC., A novel energy saving and power production system performance analysis in marine power plant using waste heat, 2009 International Conference on Machine Learning and Applications , pp. 751-754, 2009.
14. Guoqiang Yue and Shiwei Dong and Qun Zheng and Jiarui Li, Design of diesel engine waste heat recovery system with Organic Rankine Cycle, Applied Mechanics and Materials, vol.148-149,pp. 1264-1270, 2012.

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
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）
T	temperature	K
W	work output	KW

Greek symbols

$\varepsilon_{e,B}$	boiler exergy loss coefficient
$\varepsilon_{e,T}$	turbine exergy loss coefficient
$\varepsilon_{e,C}$	condenser exergy loss coefficient
$\varepsilon_{e,pp}$	pump exergy loss coefficient
η	efficiency
η_B	exhaust gas boiler efficiency considering the radiation loss
$\eta_{e,B}$	boiler exergy efficiency
$\eta_{e,T}$	turbine exergy efficiency
$\eta_{e,C}$	condenser exergy efficiency
$\eta_{e,pp}$	pump exergy efficiency
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 working fluid state at condensing pressure
4	condenser outlet state
5	pump outlet state

Superscripts

‘	saturated steam
“	saturated water