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Parametric optimization and performance comparison of organic Rankine cycle, organic dual-pressure cycle and organic flash cycle

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ABSTRACT

Compared with the basic organic Rankine cycle (ORC), organic dual-pressure Rankine cycle (ODC) and organic flash Rankine cycle (OFC) can provide better temperature match between working fluid and heat source in the heat absorption process. However, heat exchanger area is also remarkably increased. Since, the net power output (W_{net}) and heat exchanger area per unit power output (APR) of the three cycles are optimized and compared in this study. The result shows that ODC's and OFC's maximum net power output and minimum APR are greater than ORC's when the initial temperature of exhaust gas without SO_2 is 120-200°C. When the initial temperature is 160°C and the flow rate of 62.15kg/s of heat source and R245fa as working fluids,The ODC system can achieve a better thermo-economic performance than ORC and OFC systems.

Keywords: Organic Rankine cycle, Organic dual pressure cycle, Organic flash cycle, thermodynamic performance, economic performance, performance comparison

1. INTRODUCTION

A great deal of waste heat energy with exhaust gas as the carrying medium is available in the industrial processes. To generate electricity based on waste heat can reduce the consumption of fossil energy and alleviate the strain on the environment caused by the burning of fossil fuels. Exploring the efficient utilization technology for renewable energy and waste heat resources has attracted the attention worldwide. Organic Rankine Cycle (ORC) is an important and promising heat energy conversion technology, which has been widely used in renewable energy utilization (such as solar energy, geothermal energy and biomass energy) and waste heat recovery (such as internal combustion engine exhaust gas, industrial exhaust gas and hot processing liquid) all over the world [1].

From the thermodynamic point of view, Rankine cycle can't be well matched with waste heat source. The temperature of exhaust gas decreases with its exothermic process, which is an open heat source, while the temperature of pure working medium keeps constant during the evaporation process of Rankine cycle, that is, isothermal phase change. This leads to an inherent heat transfer temperature difference (HTTD) between the heat source and the working fluid. The loss in the process of finite temperature difference heat transfer between working medium and heat source is usually the large. Reducing the loss between them is extremely important to improve the energy efficiency of conventional ORC system. This has great influence on the thermodynamic perfection of the cycle and the utilization of waste heat.

Improving the cycle structure of ORC system is a significant way to increase its energy efficiency. There are two widely studied cycle types, organic dual-pressure cycle and organic flash cycle, which can improve the temperature match of working fluid and heat source. ORC involving two evaporation processes with different pressures and one condensation process, called ODC, and ORC with a low-pressure flash evaporation process, called OFC, can make full use of available waste heat sources and decrease losses in the endothermic process through this modified cycle form [2,3].

However, the thermodynamic advantages obtained by ODC and OFC systems are based on reducing the heat transfer temperature difference between the cycle and the heat source and increasing heat absorption capacity of the system to enhanced the thermodynamic performance [3,4]. At the same time, due to the

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increase of heat exchanger area, the total purchase cost of heat exchanger also rises significantly, which will increase the total investment cost of the system. As a result, the thermodynamic performance of ODC and OFC systems may be better, but their economic performance may be worse. The published researches on organic dual-pressure Rankine cycle system and organic flash Rankine cycle system are mainly the comparison of single and dual-pressure evaporation system and the comparison of flash evaporation with carbon dioxide trans-critical cycle and organic working medium trans-critical cycle [5,6].

It can be known from previous studies that the two improved cycles are suitable for low-temperature waste heat sources. The study aims to optimize, compare and analyze thermodynamic performance (net power output, W_{net}) and economic performance (heat exchanger area per unit power output, APR) of basic ORC, OFC and ODC systems based on low-temperature exhaust gas without SO_2 and R245fa as working fluids. The effects of some parameters on cycle performance are analyzed. Parameter optimization is performed by means of genetic algorithm to maximize the net power output or minimize APR. Besides single-objective optimization, both objective functions are optimized simultaneously by Non-dominated Sorting Genetic Algorithm-II (NSGA-II). Considering that the variation of HTTD has an opposite effect on the net power output and APR, the heat absorption process pinch point temperature difference (ΔT_{PP}) is also taken as an optimization parameter.

2. SYSTEM DESCRIPTION

For the ORC， ODC and OFC systems using pure working medium, the schematics of systems and their thermodynamic processes are shown in Figs. 1,2 and 3. The basic ORC cycle will not be described too much. For dual-pressure system, working fluids leaving the low-pressure economizer is divided into two parts. One part enters the low-pressure evaporator and superheater and is converted into the superheated vapor (5–8), and another part is pumped to the high-pressure stage (5–6), then enters the economizer, evaporator and superheater in the high-pressure stage and is converted into the superheated vapor (6–1).And the generated high-pressure vapor and low-pressure vapor enter the turbine1 and turbine2 respectively (1–2,8–2pie). The turbine exhausts enter the condenser together to release heat (2,2pie–3). The working fluid is compressed to the evaporation pressure of the low-pressure stage using a low-pressure pump (3–4). and sent to the low-

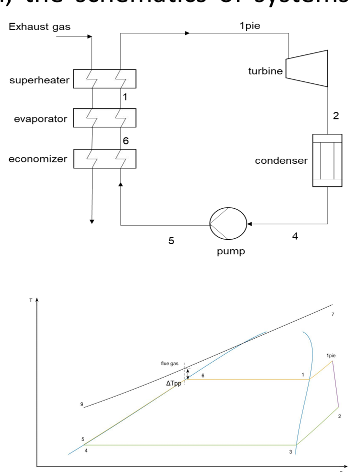

Fig.1. Schematic diagram of ORC system and diagram thermodynamic process for ORC

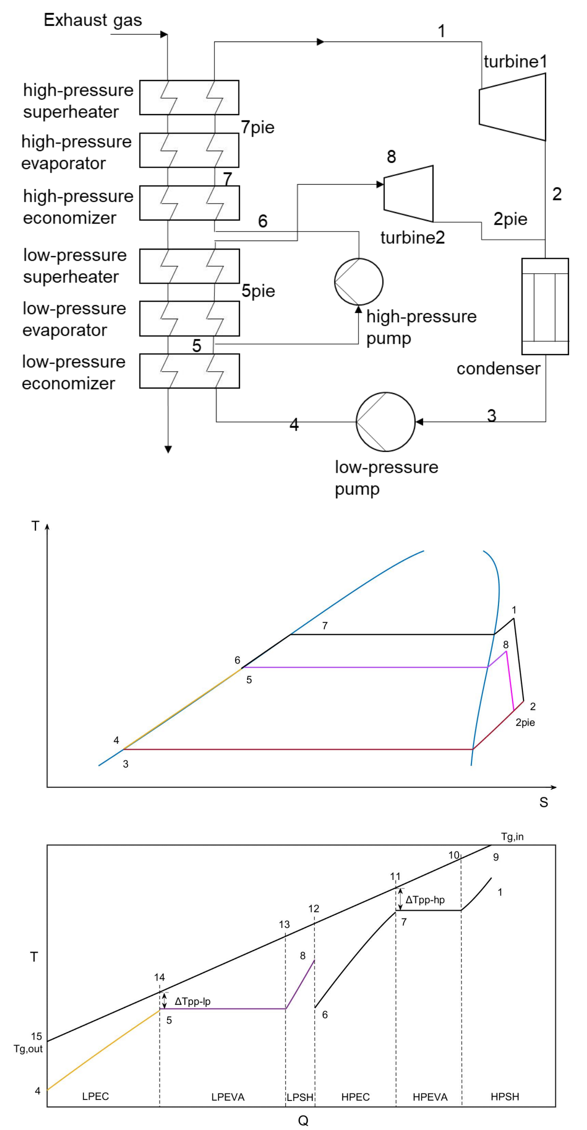

Fig.2. Schematic diagram of ODC system and diagram thermodynamic process for ODC

pressure economizer (4–5) to complete the cycle.

For flash system, A portion of saturated working fluids from the high-pressure economizer expands in the flasher (6–7) into saturated vapor (9) and liquid (7pie) of lower pressure. The vapor expands in the turbine2(9-2pie) and the liquid is mixed with working fluids from low-pressure economizer and reenter the high-pressure economizer (12–5) after being pressurized by the high-pressure pump (8,7pie–12). The rest of vapor is heated to the superheated vapor by the exhaust gas (5–1) and enters the turbine1 (1–2). The turbine exhausts enter the condenser together to release heat (2,2pie–3). The working fluid is compressed to the flash pressure using a low-pressure pump (3–4). and sent to the low-pressure economizer (4–8) to start the next cycle.

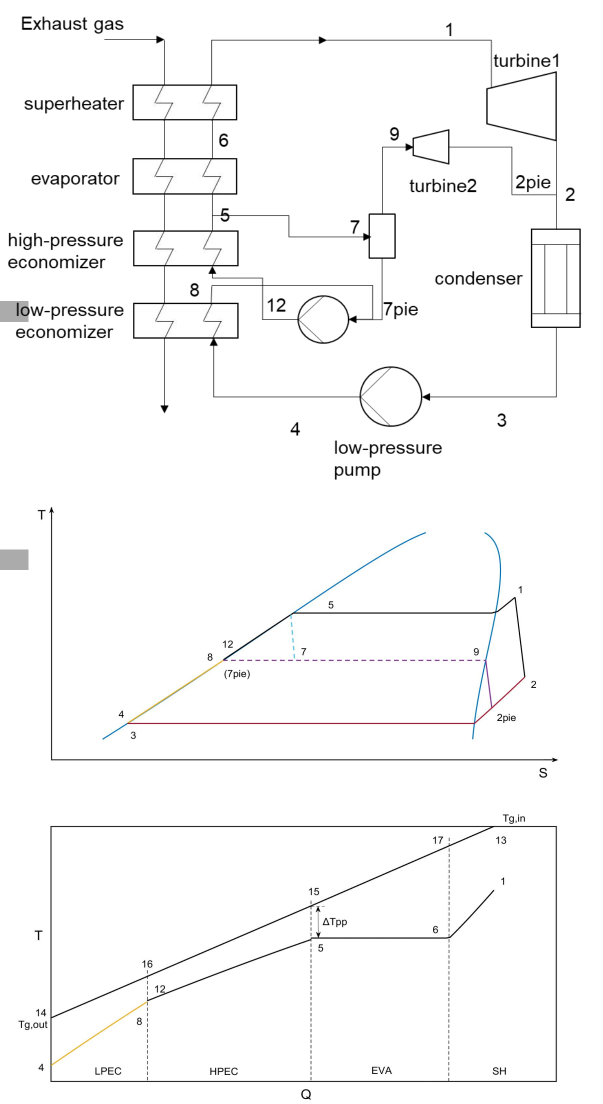

Fig.3. Schematic diagram of OFC system and diagram thermodynamic processes for OFC

3. ASSUMPTION AND OPTIMIZATION

Assumption parameters of three cyclic system models are listed in Table1.The thermodynamic model of net power output (W-net) refers to previous studies [7]. Heat transfer area in heat transfer process is calculated as:

$$A = Q/K\Delta T \tag{1}$$

Where A, Q, ΔT, and K present heat transfer area, heat absorption capacity, HTTD and coefficient of heat transfer. The economic model equations of APR are calculated as:

$$APR = A/Wnet \tag{2}$$

In the low-temperature waste heat recovery system,the heat exchange area required by the system is relatively large, and the investment cost is high, accounting for about 80% to 90% of the total system investment [8]. In order to improve system economy, the heat exchange area required for unit output power should be reduced. Therefore,the APR is taken as an economic performance indicator. The turbine inlet pressure and temperature, heat absorption process pinch point temperature difference and condensing temperature are selected as the optimization variables for the ORC. Parameters chosen for optimizing OFC are turbine inlet pressure and temperature,flash temperature,the mass flow ratio, heat absorption process pinch point temperature difference and condensing temperature. Variables chosen for optimizing the ODC are the turbine inlet pressure and temperature of the high-pressure stage, turbine inlet pressure and temperature of the low-pressure stage, condensing temperature and heat absorption process pinch point temperature difference.

The heat transfer temperature difference directly affects the heat exchanger area, that is economic, so we use heat absorption process pinch point temperature difference(ΔT_{PP})as an optimization parameter to obtain more accurate results. At the same time, the parameter optimization process is also coupled with organic fluids turbine efficiency calculation, that is, calculating the corresponding turbine efficiency for a given working fluid and cycle parameters to further calculate the performance of the system. The isentropic efficiency is determined by the size parameter(SP), volume ratio(Vr), specific speed(Ns) and the number of stages.

$$\begin{gathered} \eta_{turb} = f(Vr, SP, Ns) \\ Vr = \frac{V_{out,is}}{V_{in}} \\ SP = V_{out,is}^{0.5}/\Delta h_{is}^{0.25} \\ Ns = \frac{RPM}{60}\frac{V_{out,is}^{0.5}}{\Delta h_{is}^{0.75}} \end{gathered} \tag{3}$$

Table.1. Main assumptions of the three cyclic system.

Parameters	Value
Heat source inlet temperature/ °C	120-200
Heat source mass flow rate/ kg/s	40,60
Heat absorption process pinch point temperature difference (ΔT_{PP})/ °C	15-75
Condenser pinch point temperature difference/ °C	5
Cooling water inlet and outlet temperature/ °C	15,23
Pump isentropic efficiency/ %	80

4. RESULTS AND DISCUSSIONS

Fig.4 presents the variation of the maximum net power output and their corresponding APR with the heat source inlet temperature for the three systems when the maximum net power output is taken as the optimization objective. The net power output depends on the thermal efficiency and heat absorption capacity of the system. The system thermal efficiency increases and heat absorption capacity also increases as the heat source inlet temperature increases, accordingly, the net output power also increases. The low pressure and temperature of ODC and the flash pressure and temperature of OFC are lower than those of ORC, therefore, the system efficiency of ODC and OFC is slightly lower than that of ORC. As the same time, ODC and OFC can be well matched with exhaust gas, the heat absorption capacity markedly increases, after synthetic effect, net power output is higher than ORC. It can be seen that the net power output of ODC and OFC is almost equal, whereas APR of OFC is significantly higher than that of ODC, When the heat source inlet temperature is 140 and 160 ° C, ODC achieves the minimum APR, ORC at other temperatures. When the heat source inlet temperature reaches 200 ° C, the maximum net power output of ODC and OFC system

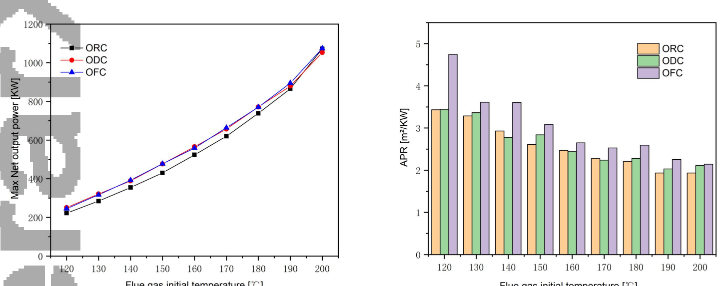

Fig.4. Maximum net power output and their corresponding APR of ORC, ODC and OFC systems for various heat source inlet temperatures

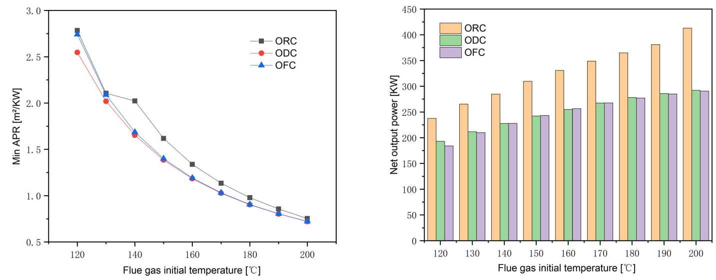

Fig.5. Minimum APR and their corresponding net power output of ORC, ODC and OFC systems for various heat source inlet temperatures

will be approximately equal to that of ORC system. ORC can be considered as a special form of ODC and OFC, which the low pressure and flash pressure is similarly equal to the condensation pressure. In a word, when the net power output is taken as the optimization objective, OFC and ODC are better than ORC systems in terms of thermodynamic performance, and ORC is better than OFC and ODC systems in terms of economic performance, since the better temperature matching between the exhaust gas and working medium means the higher heat absorption and smaller heat transfer temperature difference.

Fig.5 presents the variation of the minimum APR and their corresponding net power output with the heat source inlet temperature for the three systems when the minimum APR is the optimization objective. As can be seen that ORC is better than OFC and ODC systems in terms of thermal performance, and OFC and ODC are greater than ORC systems in terms of economic performance.

Therefore, multi-objective optimization of three systems is conducted considering both W_{net} and APR simultaneously by means of NSGA-II. It definitely reveals the trade-off between the two objectives, that is, as the net power output increases, the APR increases accordingly[9]. Fig.6 shows the Pareto frontier of the ORC, ODC, and OFC systems under the conditions of the initial heat source temperature of 160°C and the flow rate of 62.15 kg/s heat source. It is impossible to obtain maximum net power output and minimum APR simultaneously,as one improves, the other deteriorates. Each point introduced by the Pareto frontier solution can be selected as an optimum point. The decision maker can select the most preferred according to the weights of the objective functions [10]. When the initial temperature of heat source is 160 ° C, all optimum points of ODC are located in the lower right of ORC, that is, the APR with the same net power output is better, and the net power output with the same APR is great. OFC shows advantages when net power output is higher. Meanwhile, Both ODC and OFC can reach higher and lower optimum points, with a wider range than ORC.

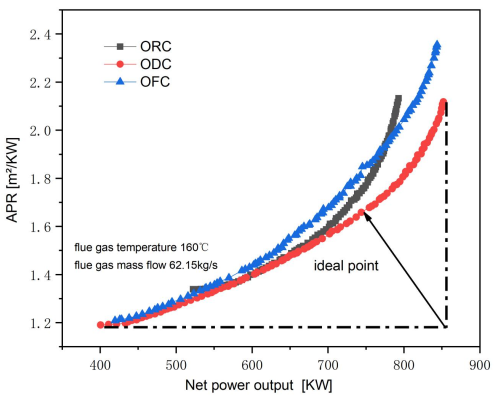

Fig.6. The Pareto frontier of APR with net power output for ORC, ODC and OFC systems for various heat source inlet temperatures (62.15kg/s of mass flow rate)

Now the Pareto solution set of fig.6 is optimized based on the ideal point assistance method. The optimal decision parameters and results are shown in Table 2 and 3. The comprehensive performance of ODC is the best, followed ORC, finally OFC.

Table.2. Multi-objective optimization parameters (°C)

Cycle	t6	t1pie	t3	ΔT_{PP}	-	-
ORC	104.32	105.60	31.32	15.15	-	-
	t7	t1	t5	t8	t3	ΔT_{PP}
ODC	112.57	114.53	90.47	93.11	32.74	17.67
	t5	t1	t7	t3	MFR	ΔT_{PP}
OFC	105.19	106.21	90.91	31.61	0.94	15.12

Table.3. Multi-objective optimization results

Cycle	W_{net} (KW)	APR(m^2/KW)
ORC	690.43	1.57
ODC	692.65	1.55
OFC	674.49	1.61

Fig. 7 shows Effects of heat absorption process pinch point temperature difference (ΔT_{PP}) on performance of ORC, ODC and OFC. It is evident that APR, the net power output and total heat exchange area decrease with the increase of pinch point in evaporator. The main reason is that higher pinch point results in higher average temperature difference between waste heat and working fluid. And it results in less heat exchanger area. Meanwhile, as the pinch point

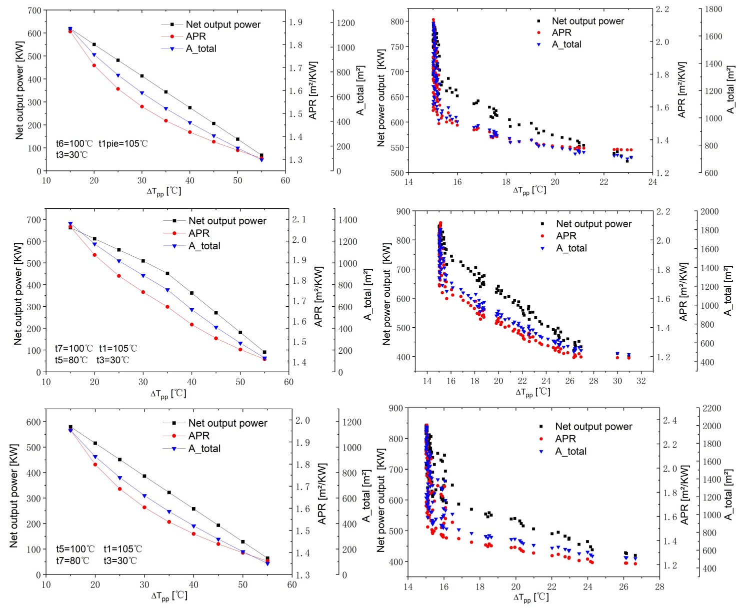

Fig.7. Effects of ΔT_{PP} on performance and distribution of ΔT_{PP} and system performance for the Pareto fronts of ORC, ODC and OFC.

temperature difference increases, the system efficiency will decrease, resulting in a decrease in net power output. The degree of net power output reduction is more obvious than the heat exchange area, and under the combined influence, APR decreases with the pinch point temperature difference increases. As can be seen that the increase of the pinch point temperature difference has disadvantages for thermal performance, but has advantages for economic performance. The variation range of the ΔT_{PP} for the Pareto fronts of the ORC,ODC and OFC systems is as follows 15-23 °C,15-31 °C and 15-27 °C. And the systems performance distribution regulation accord with the variation regulation of the left picture. And more points are distributed at the ΔT_{PP} of 15 °C, it can be seen from Table 2 that our optimal ΔT_{PP} is correspondingly close to 15 °C.

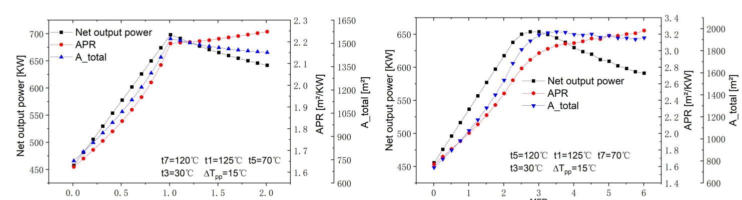

Fig.8. Effects of mass flow ratio on performance of ODC and OFC systems.

Effects of mass flow ratio on performance of two cycles are shown in Fig. 8. For both cycles, heat absorption capacity increases with the increasing of mass flow ratio, on the contrast, heat exchange temperature difference decreases, therefore, the heat exchange area increases. Meanwhile, cycle efficiency decreases as mass flow ratio increases, the change of the pinch point' location, the heat absorption gradually increases smoothly, an optimal value of output power is obtained, eventually, result in the addition of APR.

5. CONCLUSIONS

The purpose of this study is to investigate, analyze and compare the thermodynamic performance(net power output, W_{net}) and economic performance(heat exchanger area per unit power output, APR) of the ORC, ODC, and OFC systems. The parameter optimization is performed by means of genetic algorithm and the effect of parameters on the performance of the cycle is analyzed. Main results are detailed as follows.

(1) The increases of heat source temperature can increase maximum W_{net} and reduce the minimum APR of ORC, ODC and OFC systems. When the maximum W_{net} is taken as the optimization objective, OFC and ODC are better than ORC systems in terms of W_{net}, and ORC is better than OFC and ODC systems in terms of APR. When the minimum APR is the optimization objective, ORC is better than OFC and ODC systems in terms of W_{net}, and OFC and ODC are greater than ORC systems in terms of APR.

(2) When the initial temperature is 160°C and the flow rate of 62,15 kg/s of heat source and R245fa as working fluids, the minimum APR and the maximum net power output (Wnet) are optimized simultaneously, the comprehensive performance of ODC system is the best, followed ORC, finally OFC system. The ODC system can achieve a better thermo-economic performance than ORC and OFC systems.

(3) When considering economic performance, ΔT_{PP} is a quite important parameter. The power output is increases with the decrease of ΔT_{PP}, however the APR is also increase. Therefore, ΔT_{PP} should be considered when performing cycle optimization. Increasing ΔT_{PP} in heat transfer processes can improve the thermo-economic performance of systems.

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