THERMODYNAMIC ANALYSIS AND COMPARISON OF AN ORC-OFC COMBINED POWER GENERATION SYSTEM

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ABSTRACT

In this work, an ORC-OFC combined power generation system is proposed to improve the energy conversion efficiency for low grade heat sources. Mathematical models of the system are established to simulate the system under steady-state conditions. Effects of two key thermodynamic parameters including evaporation pressure and flash pressure on the system performance are examined. The analysis indicates that there exists an optimal evaporation pressure and an optimal flash pressure that yield the maximal net power output and system exergy recovery efficiency for the proposed system. Parameter optimizations by genetic algorithm are conducted for ORC, OFC and the proposed system under same heat source and restrictions, and the optimization results of the three systems are compared, showing that the ORC-OFC combined power generation system, with maximal exergy recovery efficiency reaching 16.70%, performs better than the ORC and the OFC.

1. INTRODUCTION

In recent years, with the overuse of fossil energy and the damage to environment, energy saving and emission reduction have been an important strategy for the development of most regions. More and more attentions have been paid to the utilization of the industrial waste heat and the renewable energy. This kind of energy belongs to low grade energy and the traditional steam power cycle is hard to convert this energy to electricity efficiently. Organic Rankine cycle (ORC), as one of the potential power generation cycles to recover the low grade energy efficiently, was proposed and has attracted much attention of researchers. Figure 1 shows the schematic diagram and the *T-s* diagram of the ORC. Generally ORC employs pure organic fluids as working fluids. Thus the working fluids evaporate at constant temperature during the heat addition process, whereas the heat source releases energy at decreasing temperature. The temperature mismatch between the heat source and the working fluid is one of the major sources of irreversibility for ORC. It's significant to maintain a good temperature match between the heat exchanger streams to decrease this type of irreversibility. Two methods are proposed to improve the temperature match. One is using zeotropic mixtures as working fluids. Based on different evaporation temperatures of two different fluids under the same pressure, the mixed working fluids evaporate at variable temperatures, performing a good temperature match with the heat source.

Some investigations have been done on the ORC with zeotropic mixtures as working fluids. Angelino and Colonna di paliano (1998) evaluated the merits of zeotropic mixtures as working fluids of ORC and conducted an analysis and an optimization for the zeotropic mixture ORC. Yang *et al.* (2013) studied effects of eight kinds of zeotropic mixtures on the performance of ORC for exhaust energy recovery of vehicle engine. Bao and Zhao (2013) made a summary of zeotropic mixture working fluids and corresponding cycle types for ORC. Chys *et al.* (2012) examined the effect of using mixtures as working fluids on the system performance of ORC, discussed a mixture selection method and suggested the optimal concentrations. They concluded that the use of suitable mixtures as working fluids had a positive effect on the ORC performance. Heberle *et al.* (2012) conducted

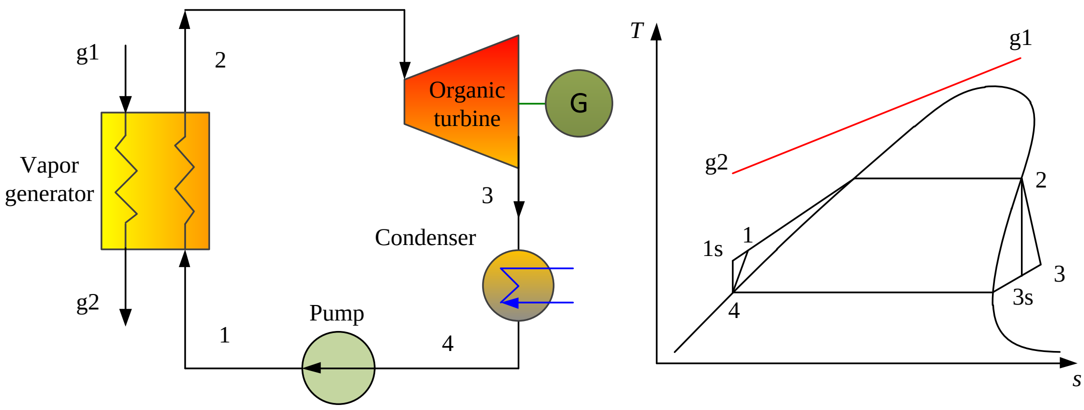

Figure 1: Schematic diagram and *T-s* diagram of organic Rankine cycle (ORC)

detailed simulations of ORC with two different groups of zeotropic mixture working fluids for low-enthalpy geothermal resources. The results showed that mixtures as working fluids lead to an efficiency increase. Some researchers made comparisons between pure-fluid ORC and mixed-fluid ORC. Garg *et al.* (2013) investigated the system performance of an ORC with isopentane, R245fa and their mixtures as working fluids for heat source temperature in the range of 385-425K. Aghahosseini and Dincer (2013) conducted a thermodynamic analysis and comparison of the low-grade heat source ORC with different pure and zeotropic mixture working fluids and the conclusions provided advice to selection of suitable working fluids for ORC. Andreasen *et al.* (2014) investigated the effect of mixed working fluids on the performance and the important design parameters of ORC by comparing using pure fluids and mixtures as working fluids. The results showed that mixed working fluid can increase the net power output of the cycle. But not every research proved the competitive advantages of using zeotropic mixture as working fluid. Li *et al.* (2011) evaluates the system performance of ORC with several pure fluids and one mixture as working fluids under different evaporation temperatures. The results showed the mixture-fluid ORC had lower efficiency than pure-fluid ORC. So whether using zeotropic mixtures as working fluid or not should depend on the specific cases.
The other method is making the working fluid heat transfer process at pressures above the critical pressure. The working fluid under supercritical state turns into some substance between liquid and vapor, which has no phase change during heat addition. Thus achieves a good temperature match with the heat source.
Several studies have been conducted on the supercritical or transcritical ORC. Gao *et al.* (2012) evaluated the system performance of a supercritical ORC using 18 different working fluids by several indicators and recommended two suitable working fluids according to some screening criteria. Karellas *et al.* (2012) investigated the influence of supercritical parameters of ORC on the heat

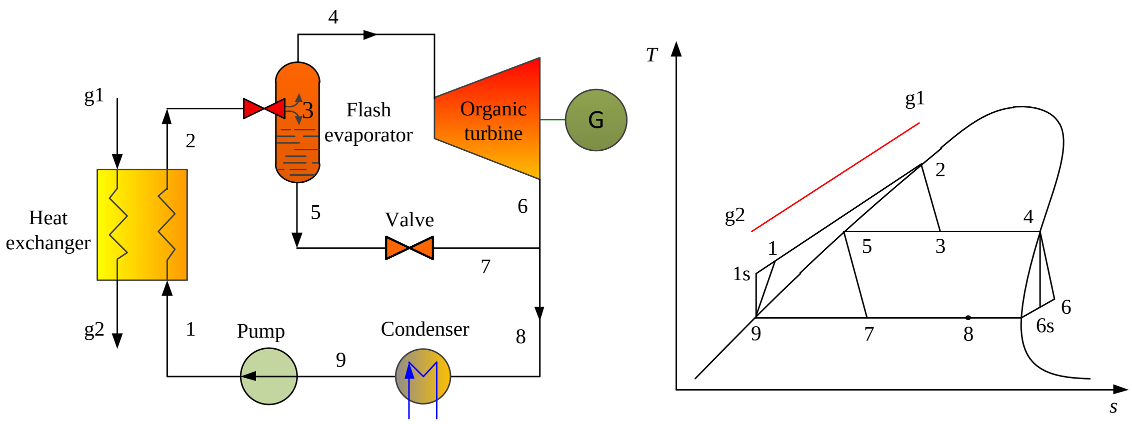

Figure 2: Schematic diagram and *T-s* diagram of organic flash cycle (OFC)

exchanger design. Some attention was paid to the comparisons between transcritical ORC and some other cycles, such as subcritical ORC, Kalina cycle and supercritical CO_2 cycle. Algieri and Morrone (2012) applied the transcritical ORC to the high-temperature biomass power generation case and made a comparison analysis with that using subcritical ORC. Shengjun *et al.* (2011) made a parameter optimization and a comparison of subcritical and supercritical ORC using different working fluids for low-temperature binary geothermal power system. Walraven *et al.* (2013) made a comparison among subcritical ORC, transcritical ORC and Kalina cycle under different pressure levels for low-temperature geothermal heat sources, and the results showed the transcritical ORC was one of the best. Schuster *et al.* (2010) conducted exergy analysis for sub- and supercritical ORC using different working fluids and the results indicated that supercritical parameters could improve the system efficiency of ORC. Baik *et al.* (2013) optimized a transcritical ORC using R125 and subcritical ORCs using several other pure fluids and compared the optimization results, which showed that the power output of the transcritical cycle was greater than that of subcritical ORCs under some conditions. They (Baik *et al.*, 2011) also demonstrated that the net power output of the R125 transcritical ORC produced more power than that of the CO_2 transcritical cycle by optimization and comparison. Chen *et al.* (2011) made a comparative study between a subcritical ORC using pure working fluids and a supercritical ORC using zeotropic mixture working fluids and the results showed the latter achieved higher thermal efficiencies than the former.

The two methods mentioned above all aim at forming a good temperature match in the heat exchangers, which may drive more working fluid for power generation. For the same purpose, Ho *et al.* (2012a) proposed an organic flash cycle (OFC) to potentially improve the energy conversion efficiency. The schematic diagram and the *T-s* diagram of the OFC are illustrated in Figure 2. As shown, the working fluid is just heated to saturated liquid under a high pressure in the heat exchanger, which avoids the constant-temperature evaporation and reduces the irreversibility. Then the working fluid is flashed to form two-phase fluids and separated to saturated vapor and saturated liquid in the flash evaporator. The saturated vapor is delivered into the organic turbine to produce power. Later Ho *et al.* (2012b) continued a further study that proposed several design enhancements to the basic OFC. Han and Kim (2014) also proposed an improvement that uses a two-phase expander instead of the throttle of the flash evaporator in the OFC, and carried out exergy analysis for their design.

This paper, based on ORC and OFC, proposes an ORC-OFC combined power generation system to recover the low-grade energy efficiently. The mathematical model of the system is established to simulate the system. Then the effects of several key thermodynamic parameters on the system performance are examined. Finally the system is parametrically optimized and compared with the ORC and the OFC under the same heat source and some same restrictions to verify the superiority of the proposed system.

2. SYSTEM DESCRIPTION

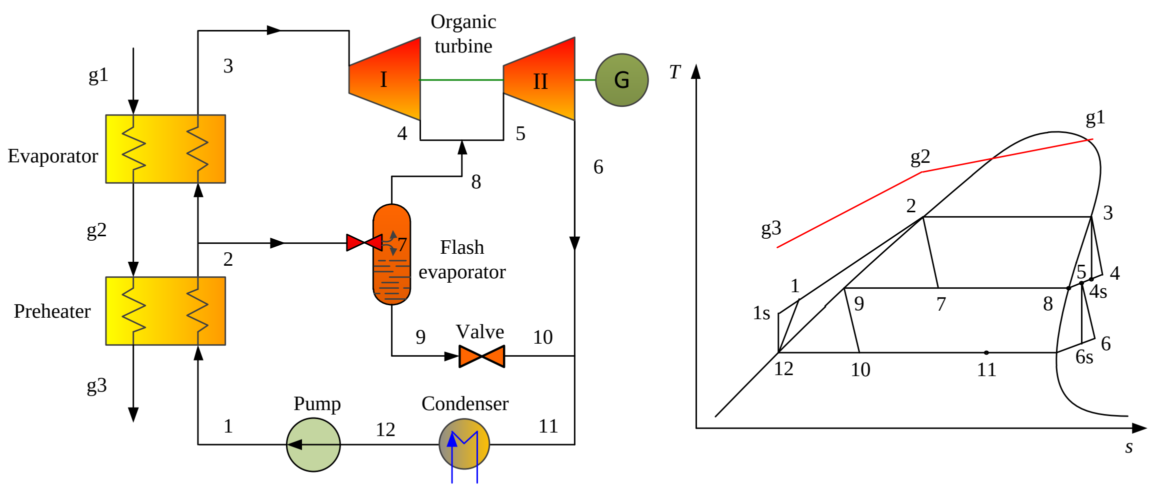

Figure 3: Schematic diagram and *T-s* diagram of ORC-OFC combined power generation system

Figure 3 illustrates the schematic diagram and the *T-s* diagram of the ORC-OFC combined power generation system. After being pumped to a high pressure state, the organic working fluid is fed into a preheater to absorb heat from the heat source to form saturated liquid. Then the saturated liquid working fluid at the outlet of the preheater is divided to two streams. One is delivered to a flash evaporator, and the other continues to be heated to saturated-vapor state by the heat source in an evaporator. The saturated organic vapor is expanded through the organic turbine I to generate power. The stream entering the flash evaporator is flashed to two-phase fluids firstly and then separated to saturated vapor and saturated liquid. The saturated vapor mixes with the exhaust working fluid from the turbine I and then is expanded through the organic turbine II. The saturated liquid from the flash evaporator mixes with the exhaust working fluid from the turbine II after being depressurized by a valve. Finally, the overall working fluid is condensed to saturated liquid in a condenser.
The proposed system, combining ORC and OFC, could drop the exhaust temperature of the heat source and push more working fluid to generate power, thus improving the energy conversion efficiency for the heat source.

3. MATHEMATICAL MODELS AND PERFORMANCE CRITERIA

3.1 Mathematical Models

For system simulation, the mathematical models are established based on the law of mass, momentum and energy conservations. Some assumptions are made to simplify the theoretical models.

(1) The system reaches a steady state.
(2) The pressure drops in preheater, evaporator, flash evaporator, condenser and connection pipes are neglected.
(3) There is no heat transfer between the equipment of the system and the environment.
(4) The working fluids at outlets of the preheater and the evaporator are saturated liquid and saturated vapor, respectively.
(5) The vapor stream and the liquid stream separated from the flash evaporator are saturated vapor and saturated liquid, respectively.
(6) The streams at the condenser outlet are saturated liquid.
(7) The turbine and the pump have a given isentropic efficiency, respectively.
(8) The flow across the valve is isenthalpic.

Based on the above assumptions, the formulas of each component in the ORC-OFC combined power generation system are shown as follows.

Preheater:

$$m_{\mathrm{hs}}(h_{\mathrm{g2}}-h_{\mathrm{g3}})=m_{\mathrm{wf}}(h_2-h_1) \tag{1}$$

Evaporator:

$$m_{\mathrm{hs}}(h_{\mathrm{g1}}-h_{\mathrm{g2}})=m_3(h_3-h_2) \tag{2}$$

Flash evaporator:

$$m_7=m_8+m_9 \tag{3}$$

$$m_7h_7=m_8h_8+m_9h_9 \tag{4}$$

Turbine:

$$\eta_{\mathrm{tb}}=\frac{h_3-h_4}{h_3-h_{4,\mathrm{s}}} \tag{5}$$

$$W_{\mathrm{tb,I}}=m_3(h_3-h_4) \tag{6}$$

$$\eta_{\mathrm{tb}}=\frac{h_5-h_6}{h_5-h_{6,\mathrm{s}}} \tag{7}$$

$$W_{\mathrm{tb,II}}=(m_3+m_8)(h_5-h_6) \tag{8}$$

Valve:

$$h_9=h_{10} \tag{9}$$

Pump:

$$\eta_{\mathrm{p}} = \frac{h_{1,\mathrm{s}} - h_{12}}{h_1 - h_{12}} \tag{10}$$

$$W_{\mathrm{p}} = m_{\mathrm{wf}}(h_1 - h_{12}) \tag{11}$$

Fluids mixing:

$$(m_3 + m_8)h_5 = m_3h_4 + m_8h_8 \tag{12}$$

$$m_{\mathrm{wf}}h_{11} = (m_3 + m_8)h_6 + m_9h_{10} \tag{13}$$

Net power output of the whole system:

$$W_{\mathrm{net}} = W_{\mathrm{tb,I}} + W_{\mathrm{tb,II}} - W_{\mathrm{p}} \tag{14}$$

3.2 Performance Criteria

In this paper, exergy recovery efficiency, based on the second law of thermodynamics, is employed to evaluate the system performance, being expressed as follows.

$$\eta_{\mathrm{exg}} = \frac{W_{\mathrm{net}}}{E_{\mathrm{in}}} \tag{15}$$

where E_{in} is the exergy input to system, i.e. the initial exergy of the heat source.

When calculating the exergy value, it's assumed that only physical exergy of the steady flow is considered, whereas chemical exergy, macroscopic kinetic and potential energy are neglected. So the exergy input is expressed as

$$E_{\mathrm{in}} = m_{\mathrm{hs}}[(h_{\mathrm{g1}} - h_0) - T_0(s_{\mathrm{g1}} - s_0)] \tag{16}$$

4. PARAMETRIC ANALYSIS

The numerical simulation of the ORC-OFC combined power generation system is conducted by a program written in MATLAB and the thermodynamic properties of relevant fluids are calculated by REFPROP 9.01 (NIST, 2013). Before the simulation, some known conditions are assigned, as listed in Table 1. During the simulation, some restrictions should be imposed to ensure the simulation practically feasible. For example, the terminal temperature difference of the heat exchangers should be larger than 10℃ to avoid an oversize heat exchange area; The quality of working fluid at the outlet of organic turbines should be larger than 0.9 to prevent the two-phase fluid corroding the turbine blades. Table 2 lists the thermodynamic parameters of each node of the system for one of the feasible simulation results, which is the base for the following parametric analysis. Table 3 shows the corresponding system performance.

Variations of some key system parameters have significant effects on the system performance. For example, the working fluid's evaporation pressure in the evaporator has direct effects on the mass flow rate of working fluid in the evaporator and the inlet pressure of the organic turbine I. The flash pressure in the flash evaporator influences the mass flow rate of the organic vapor generated in the flash evaporator, the back pressure of the organic turbine I and the inlet pressure of the organic

Table 1: Known simulation conditions

Term	Value	Unit
Working fluid	R245fa	/
Ambient temperature	20	℃
Ambient pressure	101.3	kPa
Heat source (hot air) temperature	150	℃
Heat source (hot air) pressure	150	kPa
Mass flow rate of heat source	10	kg s^{-1}
Organic turbine isentropic efficiency	80	%
Pump isentropic efficiency	70	%
Pinch point temperature difference	10	℃

Table 2: Thermodynamic parameters of each node of the system

State	t/℃	P/kPa	h/kJ kg^{-1}	s/kJ kg^{-1}K^{-1}	Quality	m/kg s^{-1}
1	34.14	1500.00	245.00	1.1515	0	5.11
2	107.85	1500.00	352.34	1.4635	0	5.11
3	107.85	1500.00	478.62	1.7950	1	2.61
4	86.88	800.00	469.66	1.8012	1	2.61
5	85.53	800.00	468.07	1.7968	1	3.30
6	52.11	200.00	447.27	1.8129	1	3.30
7	80.54	800.00	352.34	1.4697	0.2782	2.50
8	80.54	800.00	462.10	1.7800	1	0.70
9	80.54	800.00	310.04	1.3501	0	1.80
10	33.35	200.00	310.04	1.3669	0.3586	1.80
11	33.35	200.00	398.76	1.6564	0.8375	5.11
12	33.35	200.00	243.59	1.1501	0	5.11
g1	150.00	150.00	428.15	7.1280	/	10.00
g2	117.85	150.00	395.25	7.0471	/	10.00
g3	64.07	150.00	340.44	6.8963	/	10.00

turbine II. These effects will directly or indirectly impact the system performance. Therefore, the two thermodynamic parameters mentioned above (i.e. the evaporation pressure in the evaporator and the flash pressure in the flash evaporator) are selected to analyze their detailed effects on the system performance. The parametric analysis will be conducted based on the operating condition listed in Table 2 and when one thermodynamic parameter varies, the other parameters keep constant.

Figure 4 shows the effect of evaporation pressure on the net power output and the exergy recovery efficiency. As the evaporation pressure rises, the heat transferred from heat source to working fluid in the evaporator declines, resulting in a decrease in the mass flow rate of working fluid in the evaporator. However, the enthalpy drop across organic turbine I achieves an increase. Combined effected by the decreasing mass flow rate of the flow and the increasing enthalpy drop, the power output of organic turbine I presents a variation that increases firstly, reaches a top and then decreases. Due to the decreasing mass flow rate of working fluid through organic turbine II, the power output of organic turbine II drops. Additionally the power consumption of pump ascends with an increase of the evaporation pressure. Adding algebraic values of the above three items up, it's obtained that the net power output of system increases firstly and then decreases, which means that there is an optimal evaporation pressure yields the maximal net power output of system. The inlet parameters of the heat source keep unchanged, so the exergy input to system is constant. Therefore, the exergy recovery efficiency of system shows a similar variation with the net power output of system according to its definition.

Figure 5 shows the effect of flash pressure on the net power output and the exergy recovery efficiency. As the flash pressure increases, the back pressure of organic turbine I rises, leading to a decrease in the power output of organic turbine I. The increased pressure in the flash evaporator reduces the mass flow rate of the generated saturated vapor, so the mass flow rate of the working fluid entering organic turbine II declines. The enthalpy drop across organic turbine II increases with an increase of the flash pressure. Since the effect of the increased enthalpy drop overweighs the effect of the decreased mass flow rate of working fluid, the power output of organic turbine II increases. The power consumption of pump keeps constant. To sum up, the net power output of system shows a variation of increasing firstly and then decreasing. So does the exergy recovery efficiency of system.

Table 3: System performance

Term	Value	Unit
Power output of organic turbine I	23.35	kW
Power output of organic turbine II	68.68	kW
Power consumption of pump	7.20	kW
Net power output of system	84.83	kW
Exergy input	560.69	kW
Exergy recovery efficiency	15.13	%

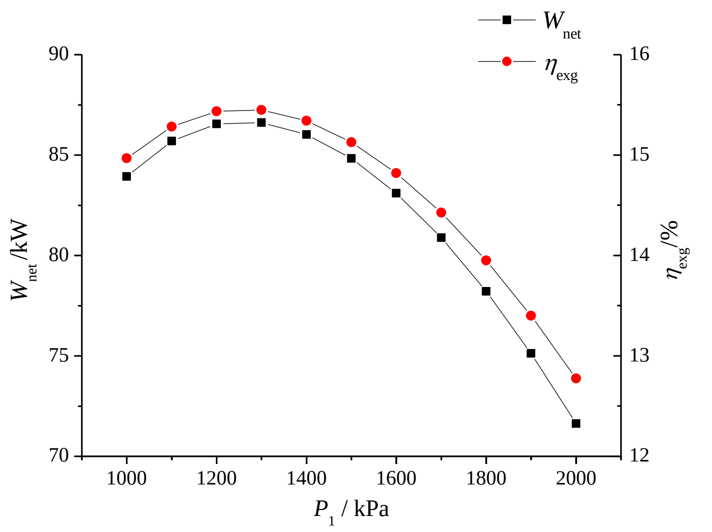

Figure 4: Effect of evaporation pressure on net power output and exergy recovery efficiency

5. OPTIMIZATION AND COMPARISON

Judging a power generation system being good or not, it should be compared with other types power generation systems. This paper would compare the proposed ORC-OFC combined power generation system with ORC and OFC power generation systems, respectively. Before comparison, the three systems would be optimized to their optimal performance state. The mathematical models of ORC and OFC, similar with parts of the mathematical model of ORC-OFC, will not present in this section. The system exergy recovery efficiency is selected as the objective function for the optimization. Different systems have different thermodynamic parameters needing optimization to obtain the maximal system exergy recovery efficiency. The ranges of optimization parameters of the three systems are listed in Table 4.

During the optimization, the three systems employ the same known simulation conditions listed in Table 1. Some restrictions, same with those mentioned in the parametric analysis section, are also set in the optimization process. The genetic algorithm (GA) (Dai *et al.*, 2009) is adopted as the optimization method to obtain the maximal system exergy recovery efficiency and find the corresponding optimal optimization parameters for the three systems. The operation parameters of GA are listed in Table 5.

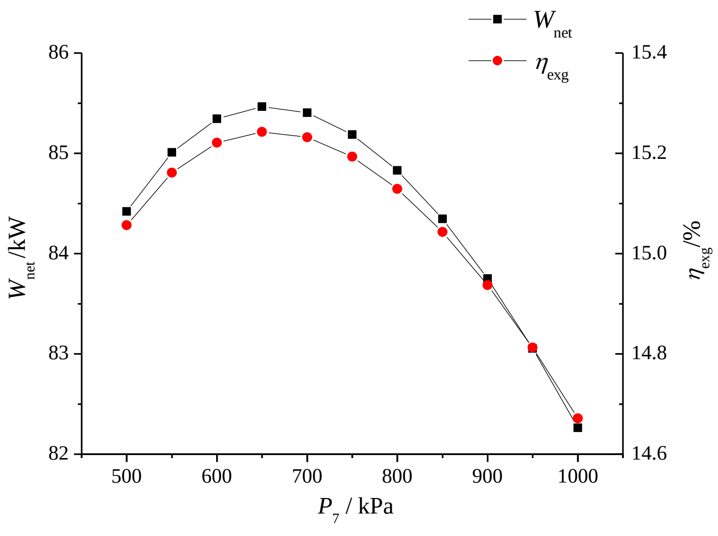

Figure 5: Effect of flash pressure on net power output and exergy recovery efficiency

Table 4: Ranges of optimization parameters for different systems

System	Thermodynamic parameters needing optimization	Range
ORC-OFC	Evaporation pressure/ kPa	[500,2800]
	Flash pressure/ kPa	[300,2500]
ORC	Evaporation pressure/ kPa	[500,2800]
OFC	Preheating pressure/ kPa	[500,2800]
	Flash pressure/ kPa	[300,2500]

Table 5: Operation parameters of GA

Term	Value
Population size	100
Crossover probability	0.8
Mutation probability	0.01
Stop generation	200

Table 6 shows the comparison results of the three power generation systems. It can be seen from the table that the ORC-OFC combined power generation system obtain the maximal exergy recovery efficiency, reaching 16.70%. Comparing the results of ORC and OFC, OFC pushes more working fluids to generate power and the enthalpy drop across turbine in OFC is also larger, so the power output of organic turbine in OFC is more than that in ORC. However, since the working fluids in OFC should be pumped to a much higher pressure before being heated to saturated liquid, OFC consumes much more power by pump than ORC. Therefore the net power output of OFC is less than that of ORC. Comparing with ORC and OFC, ORC-OFC combines the advantages of ORC and OFC that avoiding high power consumption of pump, pushing more working fluids for power generation and dropping the heat source exhaust temperature as low as possible, so ORC-OFC obtains the maximal net power output of the system. In conclusion, the ORC-OFC combined power generation system shows the best performance in the three systems.

6. CONCLUSIONS

Based on organic Rankine cycle (ORC) and organic flash cycle (OFC), an ORC-OFC combined power generation system is proposed to improve the energy conversion efficiency. By establishing the mathematical model to simulate the system under steady-state conditions, we analyzed the effects of two key thermodynamic parameters on the system performance, including evaporation pressure and flash pressure. Parametric optimizations with exergy recovery efficiency as the objective function are conducted for ORC, OFC and ORC-OFC respectively and the optimization results of the three

Table 6: Comparison results

Term	ORC-OFC	ORC	OFC
Evaporation/Preheating pressure / kPa	1289.5	1033.2	2800
Mass flow rate of fluid entering evaporator / kg s^{-1}	3.00	3.50	/
Flash pressure / kPa	596.56	/	1132.0
Mass flow rate of fluid entering flash evaporator / kg s^{-1}	4.05	/	6.61
Mass flow rate of saturated vapor generated in flash evaporator / kg s^{-1}	1.20	/	3.63
Condensation pressure / kPa	200	200	200
Heat source exhaust temperature / ℃	44.17	73.15	44.93
Power output of organic turbine / kW	I: 33.45 II: 68.51	84.44	92.69
Power consumption of pump / kW	8.33	3.16	18.60
Net power output of system / kW	93.63	81.28	74.09
Exergy input / kW	560.69	560.69	560.69
Exergy recovery efficiency / %	16.70	14.50	13.21

systems are compared. Some main conclusions drawn from the study are summarized as follows.

(1) The ORC-OFC combined power generation system, having advantages of dropping heat source exhaust temperature and pushing more working fluids for power generation than ORC, meanwhile avoiding high power consumption of pump like that in OFC, shows great potential to improve the energy conversion efficiency for low grade heat sources.

(2) There exists an optimal evaporation pressure and an optimal flash pressure that yield the maximal net power output and system exergy recovery efficiency for the ORC-OFC combined power generation system.

(3) The optimization results show that the ORC-OFC combined power generation system, with maximal exergy recovery efficiency reaching 16.70%, performs better than the ORC and the OFC.

NOMENCLATURE

E	Exergy	kW
h	Enthalpy	kJ kg^{-1}
m	Mass flow rate	kg s^{-1}
P	Pressure	kPa
s	Entropy	kJ $kg^{-1}K^{-1}$
T	Temperature	℃
W	Power output/consumption	kW

Greek symbol

η	Efficiency

Subscrips

hs	Heat source
p	Pump
s	Isentropic
tb	Turbine
wf	Working fluid

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