Experimental study of the organic rankine cycle under different heat and cooling conditions

Hong-Hu Zhang a, Huan Xi a, Ya-Ling He a,*, Yu-Wen Zhang b, Bo Ning a

a Key Laboratory of Thermo-Fluid Science and Engineering of Ministry of Education, School of Energy and Power Engineering, Xi’an Jiaotong University, Xi’an, Shaanxi, 710049, China
b Department of Mechanical and Aerospace Engineering, University of Missouri, Columbia, MO 65211, USA

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A B S T R A C T

A small-scaled organic Rankine cycle (ORC) system using R123 as working fluid was experimentally investigated. For ORC and regenerative organic Rankine cycle (RORC), the impacts of the evaporating and condensing temperatures on the performances of main components (i.e., the expander, the pump, and heat exchangers) and the system were tested. The comparison between these two systems under identical working conditions was also carried out. The results showed that the expander shaft power of ORC is greater than that of RORC under the identical evaporating temperature and condensing temperature. When the evaporating temperature is relatively low, the expander shaft power is more sensitive to the condensing temperature, and the thermal efficiency of ORC is higher than that of RORC. With the increasing of the evaporating temperature, the thermal efficiency of RORC exceeded that of ORC. Therefore, ORC is recommended for the low temperature heat source, while for the high temperature heat source, RORC is recommended for its higher thermal efficiency.

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1. Introduction

Along with the rapid development of science and technology, the consumption of non-renewable energy such as coal, oil, and nature gas increases rapidly, causing serious problems such as energy shortage and environmental pollution. In order to alleviate these problems, more attentions are paid to renewable energy utilization, energy conservation, and emission reduction. For high-grade energy utilization, such as nuclear energy and solar energy, supercritical CO2 Brayton cycle has been considered as one of the most promising power cycle because of good stability, compact structure and small device size [1]. For low-grade energy recovery, organic Rankine cycle (ORC) has been largely studied. Due to the low boiling point of organics, ORC is suitable to generate electricity by recovering waste heat. In the recent decades, many studies have been dedicated to theoretical research of advanced power cycles, such as working fluid selection [2–4], performance analysis [5–8], performance studies of critical components [9–11], and combined cycle analyses [12–14]. There are many researches concerning about the comparison of ORC and RORC. Li et al. [15] studied the influence of IHE and evaporating temperature on ORC using pure and mixture working fluid. They found that the thermal and exergy efficiency had significant improvement after adding an IHE to the system. However, the output power decreased when using IHE. Maraver et al. [16] conducted a research to study the effect of operating conditions on subcritical and transcritical ORC/RORC. They pointed out that RORC is not applicable if there is no adequate limitation upon the outlet temperature of heat source. Meinel et al. [17] presented a comparison of two-stage cycle with ORC and RORC utilizing the exhaust gas from an internal combustion engine. According to their results, dry fluids is suitable for RORC and isentropic fluids is suitable for two-stage cycle. Imran et al. [18] studied three different configurations of ORC for geothermal heat source. Their results showed that the basic ORC had the lowest thermal and exergy efficiencies but highest net power and economic performance. The two-stage ORC had the highest exergy efficiency, and lowest economic performance and net power. According to above studies, it is found that there are some disagreement about which is better between ORC and RORC. Further experiment research should

* Corresponding author. Key Laboratory of Thermo-Fluid Science and Engineering of Ministry of Education, School of Energy and Power Engineering, Xi’an Jiaotong University, Xi’an, Shaanxi, 710049, China.
E-mail addresses: xajd.zh@stu.xjtu.edu.cn (H.-H. Zhang), huanxi@xjtu.edu.cn (H. Xi), yalinghe@mail.xjtu.edu.cn (Y.-L. He), zhangyu@missouri.edu (Y.-W. Zhang), qf.she@163.com (B. Ning).

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be desired.

In order to obtain the practical operating patterns and verify the model of cycle calculation, many experimental researches have been carried out. Lemort et al. [19,20] built and tested an ORC experimental apparatus. They proposed a method to calculate the parameters of ORC performance. Effects of the internal leakage, the supply pressure drop and the mechanical losses on expander isentropic efficiency were compared. They pointed out that internal leakage was the main factor of the expander performance. Li et al. [21] established a RORC system. They found that the thermal efficiency of RORC was almost 1.83% higher than that of ORC. The impacts of superheat in expander inlet and cooling water flow on the system performances were also studied [22]. Yang et al. [23] investigated the effects of pressure drop, the degree of superheating and condensing temperatures on system performance. They found that both the thermal efficiency and the generating efficiency of the system increased with increasing pressure drop, but condensing temperature exhibited a negative impact on system performance. Chang et al. [24] tested an ORC facility using a modified compressor as expander and R245fa as working fluid. The maximum thermal efficiency and power output were 9.43% and 2.3 kW, respectively. Shao et al. [25] investigated the impact of condensing parameters on ORC. It was found that the net power output, the pump consumption, and the heat transfer capacities of evaporator and condenser increased simultaneously with increasing cooling water flow. Zhou et al. [26] examined an ORC for heat recovery from low-temperature flue gas produced by a liquefied petroleum gas stove. The results showed that the system efficiency, power output, and exergy efficiency increased with the evaporation pressure. However, the increase of superheat will degrade the system performances.

The expander is the most important part of an ORC. Kang [27] presented a study using a radial turbine and R245fa as the working fluid. They analyzed the impact of evaporating temperature and expander pressure ratio on the system performance. Declaye et al. [28] conducted an ORC experiment and evaluated the performances of expander and system by using a modified open-drive scroll compressor as an expander. The maximum expander power output, isentropic efficiency, and thermal efficiency were 2.1 kW, 75.7%, and 8.5%, respectively. Cho et al. [29] studied the applicability of the impulse type turbine. Some nozzles were added at the inlet of the turbine to increase the power output. At the identical evaporating condition, the power output significantly increased as the number of nozzles increased. Zhang et al. [30] designed and tested a single-screw expander in ORC to recovery the waste heat from a diesel engine exhaust. The results showed that the single-screw expander was more suitable for ORC. Zheng et al. [31] tested an ORC using a rolling-piston expander. It was reported that the maximum power output was 0.35 kW, the isentropic efficiency was 40%, and the system efficiency was between 5% and 6%. The comparison of two expanders with different suction volumes are carried out by our recently work [32]. We found that the expander with high suction volume shows a higher shaft power and rotating speed. The range of optimal filling factor is 0.8–0.9.

Different working fluids were also adopted in the experimental researches. Borsukiewicz-Gozdur [33] designed and built an ORC facility using R227ea as working fluid, which the electrical efficiency could reach 4.88%. R245fa/R605mfc (48.5%/51.5% on a mole basis) was used as the working fluid, and evaluated by Jung et al. [34]. The maximum electrical power and thermal efficiency were 0.7 kW and 3.9%, respectively. Qiu et al. [35] established and tested a biomass-fired ORC-based micro-CHP with HFE7000 as the working fluid. A maximum electrical power of 860.7 W, a maximum electrical efficiency of 1.41%, and a maximum CHP efficiency of 78.69% were achieved. Desideri et al. [36] compared RORC working fluid with two different organics (i.e. SES36 and R245fa). When operating under the identical temperature difference between evaporator and condenser, the system using R245fa as the working fluid could generate more power than the one using SES36. The working fluids, expanders and working conditions in the above studies are summarized in Table 1. It can be found from Table 1 that R123 had been applied in many experimental researches [19,21,25,26]. Some theoretical studies also reported that better performance could be obtained by using R123 as the working fluid. Maizza et al. [37] calculated the thermodynamic efficiencies at different evaporating and condensing temperatures. They found that R123 and R124a show good system performance under varying operating conditions among the fluids they analyzed. Roy et al. [38] analyzed an ORC utilizing the waste heat of flue gas at 140 °C and 312 kg/s. The results showed that ORC using R123 as working fluid could obtain the maximum power and both first and second law efficiency among all the selected fluids. Wang et al. [39] investigated the effects of some parameters, such as heat source temperature, evaporation and condensing pressures, on the system performance under the optimal conditions. They suggested R123 as working fluid for ORC at the temperature ranges from 100 °C to 180 °C because of its shorter payback period. Therefore, R123 was used as the working fluid in this study.

From the above literature review, it can be found that although there are numerous experimental studies of ORC, most of them were carried out under certain working conditions. Some of them considered the impact of heat source or evaporating temperature on system performance. A few papers discussed the comparison of ORC and RORC. Little information can be found on experimental research on the impact of condensing temperature coupled with the evaporating temperature on ORC and RORC. In this paper, an ORC experimental system using R123 as the working fluid was built, and the system can be switched between ORC and RORC modes. The expander adopted in this platform is scroll expander, which was modified from a scroll compressor. In the section of Results and Discussions, the impacts of the evaporating and condensing temperatures with respect to the performances of the
main equipment, including the expander, the pump and the heat exchangers would be firstly investigated. Secondly, the variations of the system performance with different condensing and evaporating temperature are presented. In addition, the comparisons of the characteristics between ORC and RORC are also discussed.

2. ORC/RORC experimental system

Fig. 1 shows the schematic diagram of ORC/RORC experimental system. The system contains three subsystems: (a) ORC/RORC subsystem, (b) the heat source subsystem, and (c) the cooling water circulation subsystem. The picture of the system is shown in Fig. 2. The ORC/RORC subsystem consists of an evaporator, an expander, a regenerator, a condenser, a pump, two liquid storage tanks, several valves, and pipes. The scroll compressor using in automobile air conditioning system is modified as an expander because of its characteristics of low cost [40], compact structure, few moving parts, and usability in the gas-liquid two-phase region. The brazed plate heat exchangers are chosen due to its high heat transfer coefficient and compact structure. The pump adopted in this test system is a diaphragm-metering pump that the flow rate can be controlled by regulating the piston stroke. In the evaporator, the organic working fluid becomes high-pressure vapour by absorbing heat from heat-transfer oil (HTO). It then expands in an expander to generate power. In the regenerator, heat exchanging takes place between the working fluid from the outlet of the pump and expander. After that, it is cooled down in the condenser, pressurized in the pump, and returns to the evaporator. In this system, ORC or RORC can be switched by adjusting Valves 3 and 4. The main part of the heat source subsystem is an electrical heater. The HTO is heated in the heater and then is pumped into the evaporator to simulate a low-temperature heat source. The cooling water circulation subsystem consists of a water pump, a glass rotameter, a cooling tower, and a water storage tank.

<table>
<thead>
<tr>
<th>Author(s)</th>
<th>Working fluid</th>
<th>Type of expander</th>
<th>Temperature range</th>
</tr>
</thead>
</table>
| Lemort et al. [19,20] | R123          | Open-drive scroll expander | Expander inlet: 101.7–165.2 °C
|                    | R123          | Impulse axial flow type turbine with single stage | Heat source: 85, 95, 110, 130 °C
|                    | R245fa        | Open-drive scroll expander | Condensing: 25 °C
| Chang et al. [24]  | R245fa        | Scroll expander   | Heat source: 100 °C
| Shao et al. [25]   | R123          | Radial inflow turbine | Heat source: 72 °C
|                    | R123          | Scroll expander   | Condenser outlet: 26–51 °C
| Zhou et al. [26]   | R123          | Radial turbine    | Heat source: 90–220 °C
| Kang [27]          | R245fa        | Impulse turbine   | Evaporation: 77.1–82.3 °C
| Cho et al. [29]    | R245fa        | Rolling-piston expander | Condenser: 37.4–40.3 °C
| Zheng et al. [31]  | R245fa        | Radial inflow turbine | Expander inlet: 50–115 °C
| Jung et al. [34]   | R245fa/R365mfc (48.5%/51.5%) | Scroll expander | Heat source: 100–150 °C
| Qiu et al. [35]    | HFE7000       | Vane-type air motor | Condensing: 18–20 °C
| Desideri et al. [36] | R245fa, SES36 | Single screw expander | Heat source: 117.8–128.9 °C
| Desideri et al. [36] | HFE7000       | Vane-type air motor | Heat source: 125 °C
| Desideri et al. [36] | R245fa, SES36 | Single screw expander | Cooling glycol water: 14–43 °C

Fig. 1. Schematic diagram of ORC/RORC experimental system.
3. Parameters of the experiment

3.1. Operating conditions and theoretical analysis

Evaporating temperature (ET, temperature at the outlet of the evaporator) and condensing temperature (temperature at the outlet of the condenser) are the main experimental parameters in this study. The ET is controlled by adjusting the power of the heater, while the condensing temperature is adjusted through changing the flow rate of the cooling water. The measurement is conducted when the system is working under steady state. Each steady state is maintained at least 10 min. Table 2 shows the main operating conditions of the experiment.

Fig. 3 shows the $T$-$s$ diagrams of the experimental system in ORC and RORC mode. Eight measuring points are set up in this system. For points 1 to 6, the temperature and pressure are measured, but for point 7 and 8, only the temperature is measured. REFPROP 9.0 [41] is employed to calculate thermodynamic properties such as temperature, pressure, enthalpy, entropy, and density.

The system performances are quantified by the following parameters:

Evaporator heat transfer capacity:
\[ Q_{\text{ev}} = m \cdot C_P \cdot (T_7 - T_8) = \dot{V} \cdot \rho \cdot C_P \cdot (T_7 - T_8) \]  

(1) 

The mass flow rate of working fluid:

\[ \dot{m} = \frac{Q_{\text{ev}}}{(h_1 - h_6)} \]  

(2) 

Expander shaft power:

\[ W_t = \pi \cdot M \cdot n / 30 \]  

(3) 

Condenser heat transfer capacity:

\[ Q_{\text{con}} = \dot{m}(h_3 - h_4) \]  

(4) 

Pump power:

\[ W_p = \dot{m}(h_5 - h_4) \]  

(5) 

Pump isentropic efficiency:

\[ \eta_t = (h_5 - h_4) / (h_5 - h_4) \]  

(6) 

System thermal efficiency:

\[ \eta_{\text{cyc}} = (W_t - W_p) / Q_{\text{ev}} \]  

(7) 

**Table 3** 
Parameters of measuring equipment.

<table>
<thead>
<tr>
<th>Name</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type-T thermocouple</td>
<td>-200–350 °C</td>
<td>±0.5 °C</td>
</tr>
<tr>
<td>Pressure sensor</td>
<td>0–2.5 MPa</td>
<td>±0.2% Full scale</td>
</tr>
<tr>
<td>Glass rotor flowmeter</td>
<td>0.4–4 m³/h</td>
<td>±2.5% Full scale</td>
</tr>
<tr>
<td>Vortex shedding flowmeter</td>
<td>0–10 m³/h</td>
<td>±1.0% Full scale</td>
</tr>
<tr>
<td>Dynamometer and secondary instrument</td>
<td>rotation speed: 0–9999 rpm</td>
<td>rotation speed: ±0.2% Full scale</td>
</tr>
<tr>
<td></td>
<td>torque: 0–10 N m</td>
<td>torque: ±1.0% Full scale</td>
</tr>
</tbody>
</table>

**Fig. 4.** Effect of evaporating and condensing temperature at different torque for ORC.
Fig. 5. Effect of evaporating and condensing temperature at different torque for RORC.

Fig. 6. Expander pressure ratio versus condensing temperature at different ET.

Fig. 7. Variations of the working fluid flow rate with condensing temperature at different ET.
According to equation (1), $\dot{Q}_{eva}$ can be calculated by the specific heat at constant pressure ($c_{p,\text{oil}}$), the volumetric flowrate ($V_{\text{oil}}$), the density ($\rho_{\text{oil}}$) of the heat transfer oil, $T_7$ and $T_8$. Where, the $c_{p,\text{oil}}$ and $\rho_{\text{oil}}$ can be calculated by the average temperature of $T_7$ and $T_8$ according to the follow equations:

$$c_{p,\text{oil}} = -4.0732 \times 10^{-6}T_{\text{ave,oil}}^2 + 7.3 \times 10^{-3}T_{\text{ave,oil}} - 0.059776$$

(8)

$$\rho_{\text{oil}} = -0.49589T_{\text{ave,oil}} + 1009.2$$

(9)

$$T_{\text{ave,oil}} = (T_7 + T_8)/2$$

(10)

It can be found that the $\dot{Q}_{eva}$ is only dependent on $T_7$, $T_8$ and $V_{\text{oil}}$, in which the temperatures at states 7 and 8 are measured accordingly.

3.2. Instrumentation and uncertainty analysis

Type-T thermocouples and pressure sensors were adopted to measure temperature and pressure, respectively. A dynamometer is employed to measure the rotation speed and torque. A vortex shedding flowmeter and a glass rotor flowmeter are assembled in the heat source subsystem and the cooling water circulation subsystem, respectively. Table 3 shows the parameters of the measurement devices.

Based on the previous researches [23,24] and the error propagation theory, the root-sum-square method is carried out to calculate the measuring uncertainties:

$$\Delta F = \sqrt{\sum_i \left( \frac{\partial F}{\partial x_i} \Delta x_i \right)^2}$$

(11)

where $F$ denotes the parameters calculated by the measuring parameters $x_i$. $\Delta F$ and $\Delta x_i$ denote the uncertainty of $F$ and $x_i$, respectively.

4. Results and discussions

4.1. Expander performances

Figs. 4 and 5 present the variation of the expander shaft power at different shaft torques for ORC and RORC, respectively. It can be seen that when the shaft torque increases, the shaft power of expander first increases, achieves a maximum value, then decreases. This trend indicates that there exists an optimal shaft torque corresponding with the highest shaft power. The shaft power of ORC is higher than RORC. The maximum shaft power of ORC and RORC are 611.2 W and 577.6 W, respectively. Fig. 4(a) and (d) show the variations of shaft power with ET of 80°C and 110°C, respectively. When ET is fixed at 80°C, raising the condensing temperature from 23°C to 30°C leads the maximum shaft power to decrease from 261.0 W to 133.9 W, which reduces approximately 48.7%. However, when ORC is operated at an ET of 110°C, the shaft...
power reduces only 3.6% as condensing temperature increases from 23°C to 30°C. This illustrates that the condensing temperature has a great impact on the shaft power under lower ET. The effect of condensing temperature on the shaft power decreases with the increase of ET. When ET increases from 80°C to 100°C for ORC, the sensitivity of shaft power to condensing temperature decreases rapidly. The relative variation of maximum shaft power with different condensing temperatures is less than 5% when ET is greater than or equal to 100°C. However, for RORC, the relative variation is 17.4% when ET is 100°C; moreover, the relative variation is less than 5% until ET is 110°C. In conclusion, it is necessary to focus on the effect of condensing temperature on the expander performance, especially at low evaporating temperature. The upper limit of the evaporating temperature for RORC should be higher than that of ORC, when studying the influence of condensing temperature on expander performance of these two systems.

Fig. 6 illustrates the effect of condensing temperature on the expander pressure ratio under different ETs. It is clearly observed that the expander pressure ratio decreases with increasing condensing temperature, and increases with increasing ET for both systems. The pressure ratio of ORC is greater than that of RORC at the same ET. The back pressure of the expander for ORC is lower than RORC because of the regenerator. It is the main reason causing the reduction of the expander pressure ratio. The reduction of the expander pressure ratio is one of the main reasons that caused the decrease of the expander shaft power.

Fig. 7 shows the effect of condensing temperature on the mass flow rate of working fluid at different ETs. It is obvious that the mass flow rate of RORC is higher than that of ORC. With increasing condensing temperature, the pump outlet pressure and the expander inlet pressures increase. As a result, the density of the working fluid at the entrance of the expander increases. Therefore, the mass flow rate of the working fluid increases correspondingly. Some test results are not marked in Fig. 7, especially the results at a high condensing temperature and low evaporating temperature. The reason for this is that the working fluid goes into the two-phase state at the condition described above, which means that the thermodynamic parameters of working fluid could not be determined by temperature and pressure. Therefore, the specific enthalpy of the working fluid is uncertain, and the mass flow could not be calculated.
4.2. Pump performances

Fig. 9 shows the variations of inlet and outlet pressures of the pump with condensing temperature for ORC and RORC. The inlet pressure of the pump of ORC is higher than that of RORC, but the pump outlet pressure of ORC is lower than that of RORC. In RORC, the regenerator acts as a “pre-condenser”, which makes the overall cooling heat exchange capacity of RORC greater than that of ORC. The condensing pressure at condenser outlet for RORC are therefore slightly lower. Meanwhile, the flow resistance of the system increases due to the addition of a regenerator in RORC. In order to ensure the stability of the system, the pressure at the pump outlet increases accordingly.

Fig. 10 shows the power consumption and isentropic efficiency of the pump in ORC and RORC as a function of evaporating and condensing temperature. As the condensing temperature increases, the isentropic efficiency of the working fluid pump in both ORC and RORC increase. Moreover, the increasing rate of pump isentropic efficiency decreases when the evaporating temperature changes from 80 °C to 110 °C, as the solid (for ORC) or the hollow (for RORC) symbols shown in Fig. 10 (a). It can also be found that the pump isentropic efficiency of ORC is higher than that of RORC. As shown in Fig. 10 (b), the pump power consumption decreases with the increasing of the condensing temperature. The reduction of the pump power consumption with condensing temperature decreases when the evaporating temperature increases from 80 °C to 110 °C. Nevertheless, when evaporating temperature is 110 °C, the power consumption of the working fluid pump shows a fluctuating variation. It is because when the evaporating temperature is at a low level, the isentropic efficiency of the pump increases with the obvious increase of the condensing temperature. The isentropic efficiency is the main factor in influencing the pumping power, so the pump power consumption shows a trend of decline. However, when the evaporating temperature increases, the increment of the pump isentropic efficiency reduces. Meanwhile, the mass flow rate of the working fluid increases with increasing condensing temperature. The higher isentropic efficiency gives a lower specific enthalpy difference and hence a less pumping power consumption, but the greater mass flow causes a higher pump consumption.

Fig. 11 shows the variations of subcooling at pump inlet with evaporating and condensing temperatures. The subcooling at pump inlet decreases slightly with the increase of the condensing temperature, and higher ETs could correspond to greater subcooling. Furthermore, it can be found that the subcooling of ORC is higher than that of RORC. It is because that RORC has lower condensing pressure, as shown in Fig. 9, its saturated liquid temperature and the subcooling is also lower as a result. The range of subcooling in this work is about 7−10.5 °C. According to the results in Ref. [42], the recommended subcooling should be greater than 20 °C to avoid cavitation. It is difficult to obtain such a large subcooling due to condensing conditions in this work. Therefore, the cavitation occurs in the pump, which was the main factor causing the low efficiency of the working fluid pump. Pressure fluctuation caused by cavitation during the experiment had also caused damage to the diaphragm of pump, as shown in Fig. 12.

4.3. Heat exchangers performances

Fig. 13 shows the variations of inlet/outlet specific enthalpies and enthalpy differences of the evaporator with condensing temperature in ORC and RORC when the ET is 110 °C. It can be seen that for ORC, inlet specific enthalpy of the evaporator increases with the condensing temperature, but the specific enthalpy at evaporator
outlet has a slighter variation than the inlet. Therefore, the enthalpy difference decreases with increasing condensing temperature. For RORC, due to the buffer effect of the regenerator, the variation of condensing temperature has less influence on the inlet specific enthalpy of the evaporator. The outlet specific enthalpy of the evaporator decreases due to the increase of the outlet pressure. The enthalpy difference of the evaporator decreases consequently. Comparing the results of ORC with that of RORC, the evaporator enthalpy difference of RORC is significantly lower than that of ORC. The presence of the regenerator makes the evaporator inlet specific enthalpy of RORC higher than that of ORC. Meanwhile, the outlet pressure of the evaporator in RORC is higher than that of ORC. The evaporator outlet specific enthalpy of RORC is lower as a result.

Fig. 14 illustrates the variation of inlet/outlet specific enthalpies and enthalpy differences of the condenser with condensing temperature in ORC and RORC when the ET is 110°C. The condenser inlet specific enthalpy shows little variation with the increase of the condensing temperature, but the specific enthalpy of the outlet has increased significantly, which makes the enthalpy difference of the condenser exhibited a trend of decrease. Due to the effect of the regenerator, the condenser inlet specific enthalpy of RORC is significantly lower than that of ORC. Moreover, the outlet pressure of the condenser as well as the specific enthalpy of the working fluid for both systems are relatively close at the same condensing temperature. These are the reasons why the condenser enthalpy difference in RORC is significantly lower than that in ORC.

4.4. System performances

Fig. 15 indicates the effect of condensing temperature on the thermal efficiency of the two systems at different ETs. The thermal efficiency increases with increasing ET for both systems. Through the comparison of the experiment results of both systems, it can be found that the thermal efficiency of ORC is slightly higher than that of RORC when the ET is lower than 90°C. The thermal efficiency of RORC gradually exceeds that of ORC with the increase of the ET.

Table 4 shows the performances of the two systems, when the evaporating temperatures are 90°C and 110°C, the condensing temperatures are 24°C and 25°C, respectively. The heat transfer capacity of the evaporator of ORC is greater than that of RORC under the same operating condition, due to the impact of the regenerator. The expander shaft power of ORC is also greater than that of RORC under the same condition. It can also be found that, under the same condensing temperature, the relative variation of the expander shaft power of the two systems at lower ET are greater than that at higher ET. When the ET is 90°C, the relative variation of the expander shaft power is greater than that of the heat transfer capacity of the evaporator. Therefore, the expander shaft power is the major factor of the system performances. When the ET is 110°C, the relative variation of the expander shaft power is lower than that of the evaporator’s heat transfer capacity. The heat transfer capacity of the evaporator becomes the major factor of the system performances for this case, and the thermal efficiency of RORC exceeds that of ORC.

5. Conclusions

In this study, the performances of an ORC/RORC to utilize the low-temperature waste heat is experimentally studied. A modified scroll compressor is chosen as the expander of the system, and R123 is used as the working fluid. The system performances under different heat and cooling conditions are investigated. The main conclusions are as follows:

1. The expander shaft power of ORC is greater than that of RORC under the fixed evaporating temperature and condensing temperature. Within the range of temperature studied in this paper, the maximum shaft power of RORC and ORC are 611.2 W and 577.6 W, respectively. The effect of the condensing temperature on the performance reduces with the increase of evaporating temperature. For ORC, the condensing temperature has less impact on the expander shaft power (with a relative variation < 5%) when the evaporating temperature is greater than or equal to 100°C. However, for RORC, the evaporating temperature should be greater than or equal to 110°C when the impact of condensing temperature is insignificant.

2. Under the given working conditions in this work, the mass flow rate of RORC is higher than that of ORC. Thermal efficiencies of both systems increase with the increasing of evaporating temperature. When the evaporating temperature is 90°C, the thermal efficiency of ORC is slightly higher

Table 4

<table>
<thead>
<tr>
<th>Conditions</th>
<th>Systems</th>
<th>Evaporator capacity/W</th>
<th>Shaft power/W</th>
<th>Pump power/W</th>
<th>Net power/W</th>
<th>Thermal efficiency%</th>
</tr>
</thead>
<tbody>
<tr>
<td>ET: 90°C</td>
<td>ORC</td>
<td>13345</td>
<td>375</td>
<td>144</td>
<td>231</td>
<td>1.73</td>
</tr>
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<td>Condensing temperature: 24°C</td>
<td>RORC</td>
<td>12499</td>
<td>325</td>
<td>124</td>
<td>201</td>
<td>1.61</td>
</tr>
<tr>
<td>ET: 110°C</td>
<td>ORC</td>
<td>16571</td>
<td>590</td>
<td>161</td>
<td>429</td>
<td>2.59</td>
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<td>RORC</td>
<td>15228</td>
<td>578</td>
<td>147</td>
<td>431</td>
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<td>ET: 90°C</td>
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<td>383</td>
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</tr>
<tr>
<td>Condensing temperature: 25°C</td>
<td>RORC</td>
<td>12793</td>
<td>315</td>
<td>93</td>
<td>222</td>
<td>1.73</td>
</tr>
<tr>
<td>ET: 110°C</td>
<td>ORC</td>
<td>16557</td>
<td>594</td>
<td>168</td>
<td>426</td>
<td>2.57</td>
</tr>
<tr>
<td>Condensing temperature: 25°C</td>
<td>RORC</td>
<td>15187</td>
<td>572</td>
<td>122</td>
<td>450</td>
<td>2.97</td>
</tr>
</tbody>
</table>
that of RORC. With a higher evaporating temperature, the thermal efficiency of RORC would exceed that of ORC.

(3) Adding a regenerator in the organic Rankine cycle may not improve the thermal efficiency. When the evaporating temperature is low, the regenerator would reduce the pressure ratio of the expander, thus reducing the output power and the thermal efficiency of the system. However, the influence of the regenerator on the power output of the expander is reduced when the evaporating temperature is high, and the thermal efficiency of RORC is higher than that of ORC. Therefore, it is recommended to adopt ORC when the heat source temperature is low but RORC when the heat source temperature is high to ensure the high thermal efficiency of the system.

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