Open feed organic heater pressure analysis on single-stage regenerative organic Rankine cycle performance

Ghalya Pikra a,*, Nur Rohmah b, Rakhmad Indra Pramana a, Andri Joko Purwanto a

a Research Centre for Electrical Power and Mechatronics, Indonesian Institute of Sciences
Komp. LIPI Bandung, Gel 20, Lt 2, Bandung, West Java, 40135 Indonesia
b Research Unit for Clean Technology, Indonesian Institute of Sciences
Komplek LIPI Bandung, Gedung 50, Bandung, West Java, 40135, Indonesia

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Abstract

Single-stage regenerative organic Rankine cycle (SSRORC) is a system that is used for increasing the simple organic Rankine cycle (ORC) performance. Open feed organic heater (OFOH) addition in the ORC system increase power and efficiency of the system. This paper analyzes the SSRORC performance with a variation of $P_6/P_1$ ranges from 1.25 to 3.75 with an increment of 0.25, where $P_6$ is the OFOH pressure at the inlet side and $P_1$ is the pressure at the inlet pump 1, respectively. Hot water was used as the heat source with 100 °C and 100 l/min of temperature and volume flow rate as the initial data. R227ea, R245fa, and R141b were chosen as working fluids for performance analysis. The analysis was performed by calculating the heat input, heat loss, pump and turbine power, net power, and thermal efficiency through energy balance. Exergy input, exergy output, and exergy efficiency were analyzed through exergy balance. The results show that $P_6/P_1 = 2$ obtains the highest performance than the other pressure ratios for R227ea, while R245fa and R141b obtain the highest performance at $P_6/P_1 = 2.25$. R141b has better performance than the other two fluids with 10.97% and 11.96% for thermal and exergy efficiency. The results show that the ratio of OFOH pressure at the inlet side to the pressure at inlet pump 1 ($P_6/P_1$) in the middle value obtains the best performance.

Keywords: single-stage regenerative organic Rankine cycle; open feed organic heater; R227ea; R245fa; R141b.

I. Introduction

Electricity needs are increasing in this globalization era. On the other hand, the availability of fossil fuels is running low, so it is necessary to find other alternative sources before fossil fuels could no longer meet the world’s electricity demands. Renewable energy has begun to be developed in various parts of the world. During this time, the Rankine cycle has been known as one of the many power generation systems developed and used to generate electricity. Water is commonly used as a working fluid that can only generate electricity at high operating temperatures, whereas existing renewable energy such as geothermal, solar, and waste heat allows it to be used as a heat source to generate electricity at low and medium operating temperatures so that the use of water must be replaced for the renewable energy utilization. Therefore, organic fluids which have lower boiling temperatures than water can be used in the system to produce electricity at low and medium operating temperatures. This system is known as the organic Rankine cycle (ORC) which has the same components as the Rankine cycle but can produce electricity at low and moderate operating temperatures by using organic fluid as a working fluid.

Organic Rankine cycle (ORC) has been utilized in many heat sources, such as biomass [1], geothermal [2][3][4], solar [5], ocean thermal [6], and waste heat [7][8][9]. This system can also be combined with other cycles so that the use of heat sources can be maximized and the heat loss in the system can be reduced. However, because of the low operating temperature, the ORC system has a low performance.

Modification of the ORC configuration is one of many ways that can be used to increase the ORC performance. This paper analyzes the SSRORC performance with a variation of $P_6/P_1$ ranges from 1.25 to 3.75 with an increment of 0.25, where $P_6$ is the OFOH pressure at the inlet side and $P_1$ is the pressure at the inlet pump 1, respectively. Hot water was used as the heat source with 100 °C and 100 l/min of temperature and volume flow rate as the initial data. R227ea, R245fa, and R141b were chosen as working fluids for performance analysis. The analysis was performed by calculating the heat input, heat loss, pump and turbine power, net power, and thermal efficiency through energy balance. Exergy input, exergy output, and exergy efficiency were analyzed through exergy balance. The results show that $P_6/P_1 = 2$ obtains the highest performance than the other pressure ratios for R227ea, while R245fa and R141b obtain the highest performance at $P_6/P_1 = 2.25$. R141b has better performance than the other two fluids with 10.97% and 11.96% for thermal and exergy efficiency. The results show that the ratio of OFOH pressure at the inlet side to the pressure at inlet pump 1 ($P_6/P_1$) in the middle value obtains the best performance.

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Keywords: single-stage regenerative organic Rankine cycle; open feed organic heater; R227ea; R245fa; R141b.
system performance. Many ORC configurations have been studied and experimented by many researchers in order to enhance the system performance. Li et al. compared parallel and series of two-stage organic Rankine cycle and as a result, the series configuration has a better performance than the parallel [10]. K. Braimakis and S. Karellas investigated an open and closed preheater that resulting in better performance by using a closed preheater [11]. Li et al. optimized the ORC using dual-pressure evaporation and analyze nine different working fluids with 100-200 °C of heat source temperature. The optimization showed an increase in net power output between 21.4 - 26.7 % [12]. Sciubba et al. compared double stage ORCs and a recuperator addition that resulting in an electricity generation increment up to 8.11 % and 2.67 %, respectively [8]. Mosaffa et al. studied regenerative and recuperative ORCs for geothermal energy that obtain high efficiency [4]. Xi et al. optimized the ORC using single and double stage regenerative ORCs and shows that the double stage has the highest energy and exergy efficiency [13]. Zare compared three configurations such as a simple ORC, a recuperative ORC, and an open-type regenerative preheater ORC for binary geothermal power plants. The result shows that simple ORC obtains the highest power output and the lowest economic cost, and recuperative ORC observed the best energy and exergy efficiency [14]. Safarian and Aramoun evaluated a simple and regenerative-recuperative ORC and examined that the regenerative-recuperative ORC has the best thermal efficiency [15].

Single-stage regenerative organic Rankine cycle (SSRORC) is one of many configurations that can increase the ORC system performance. Inspired from our previous research [16] about the performance comparison of single SSRORC and double stage of regenerative organic Rankine cycle (DSRORC), this paper discusses the pressure analysis of an open feed organic heater (OFOH) in a single-stage regenerative organic Rankine cycle (SSRORC), since our previous research assumes that the pressure entering the OFOH is constant. Eleven different pressure values are investigated to determine the best performance. Those pressures were based on the ratio of pressure at inlet pump 1 and the pressure at inlet OFOH, that is, 1.25, 1.5, 1.75, 2, 2.25, 2.5, 2.75, 3, 3.25, 3.5, 3.75. Temperature and volume flow rate of 100 °C and 100 l/min are used as the initial data using water as the heat source. Three working fluids such as R227ea, R245fa, and R141b are compared and analyzed to determine the best performance. Because the pressure at inlet OFOH must be lower than the inlet turbine pressure, then R227ea can only be used until the maximum pressure ratio of 3.25.

II. Materials and Methods

A simple organic Rankine cycle consists of four main components, such as evaporator, turbine, condenser, and pump. A modification of the ORC system configuration is necessary to increase its performance. Open feed organic heater (OFOH) addition in the ORC system is one of many modifications that can enhance the system performance. The addition of OFOH in the system is commonly called as a single-stage regenerative organic Rankine cycle (SSRORC). Figure 1 shows the scheme of SSRORC.

The working principle of SSRORC that is shown in Figure 1 is almost the same with the simple ORC. The simple ORC only uses one pump and do not use an open feed organic heater (OFOH). In the simple ORC, the fluid from the condenser (1) is directly pumped to the evaporator (4) to be vaporized and expanded in the turbine (5). All of the expanded fluids are condensed in the condenser (7) to be pumped back to the evaporator (4). The SSRORC has an OFOH (6) and one additional pump (3) so that some of the expanded fluids from the turbine flow directly to the OFOH (6) while some other fluids are condensed in the condenser (7) prior to being pumped back to OFOH and finally be pumped to the evaporator. This configuration has a possibility to decrease the heat loss in the condenser and subsequently increase the system performance.

This paper analyzes the OFOH pressure influence on the performance of the SSRORC system. It analyzes eleven states of OFOH pressure from the ratio of pressure at inlet pump 1 (P1) and the OFOH inlet (P6). The P6/P1 values used in this study are 1.25, 1.5, 1.75, 2, 2.25, 2.5, 2.75, 3, 3.25, 3.5, 3.75.
1.5, 1.75, 2, 2.25, 2.5, 2.75, 3, 3.25, 3.5, and 3.75. The T-s diagram from Figure 2 shows all states at each component of SSRORC.

Figure 2 shows that the pressure at the state 1 (P₁) is the same as the pressure at 7 (P₇), the pressure at 2 (P₂) is the same as the pressure at 3 and 6 (P₃ and P₆), and the pressure at 4 (P₄) is the same as the pressure at 5 (P₅). The pressure at 6 as shown in Figure 2 is made varied for the analysis requirement. The performance analysis was carried out using the first and second laws of thermodynamics through energy and exergy balance from Moran et al. [17]. Figure 3 shows the flowchart of the analysis.

Figure 3 shows that the initial temperature and volume flow rate of the heat source (water) for all fluids and all states are assumed to be constant at 100 °C and 100 l/min. The inlet temperature of the turbine (T₉) is 90 °C which is at saturated vapor state, and the inlet temperature of the pump 1 (T₁) is 40 °C which is at saturated liquid state. Both states are assumed constant for various P₆. Isentropic efficiency of the pump and the turbine are assumed 0.75 and 0.85, while the potential and kinetic energy are negligible.

The analysis started with heat input (Qᵢᵢ) and heat loss (Qᵢᵢloss) calculation from energy balance at the evaporator and condenser. The calculation was continued with pump power (Wₚ) and turbine power (Wₜ) calculation from energy balance at the pump and the turbine to determine the net power output. The thermal efficiency (ηₜℎ) obtained from the ratio of net power output and the heat input. The calculation of Qᵢᵢ, Qᵢᵢloss, Wₚ, Wₜ, and ηₜℎ is shown in Equations (1) to (5)

\[
Qᵢᵢ = \dot{m}_{water} (h₅ - h₄) = \dot{m}_{water} C_p h_{water}(T₉ - T₈) \tag{1}
\]

\[
Qᵢᵢloss = \dot{m}_{water} (1 - y)(h₇ - h₁) \tag{2}
\]

\[
Wₚ = \dot{m}_{water} [(h₄ - h₃) + (1 - y)(h₂ - h₁)] = \dot{m}_{water}/\etaₚ [(h₄ₛₐₖ - h₃ₛₐₖ) + (1 - y)(h₂ₛₐₖ - h₁ₛₐₖ)] \tag{3}
\]

\[
Wₜ = \dot{m}_{water} [(h₅ - h₆) + (1 - y)(h₆ - h₇)]
\]

Figure 2. T-s Diagram of SSRORC

Figure 3. Flowchart of SSRORC performance analysis
where $Q_{in}$ is heat input (kW); $m_{of}$ is the organic fluid mass flow rate (kg/s); $m_{hw}$ is heat source mass flow rate (kg/s); $C_{p_{hw}}$ is heat source heat capacity (kJ/kg K); $T_8$ is heat source temperature at inlet evaporator (°C); $T_9$ is heat source temperature at outlet evaporator (°C); $W_{net}$ is net output (kW); $Q_{loss}$ is heat loss (kW); $W_i$ is turbine power (kW); $W_{t}$ is isentropic turbine power (kJ/kg); $h_1$ is organic fluid enthalpy at inlet pump 1/outlet condenser (kJ/kg); $h_2$ is organic fluid enthalpy at outlet pump 2/inlet evaporator (kJ/kg); $h_3$ is isentropic organic fluid enthalpy at inlet OFOH/outlet pump 1 (kJ/kg); $h_4$ is organic fluid enthalpy at inlet OFOH/inlet pump 1 (kJ/kg); $h_5$ is organic fluid enthalpy at outlet turbine/inlet condenser (kJ/kg); $h_6$ is organic fluid enthalpy at outlet turbine/outlet condenser (kJ/kg); $h_7$ is isentropic organic fluid enthalpy at outlet turbine/outlet condenser (kJ/kg); $h_8$ is isentropic organic fluid enthalpy at outlet turbine/outlet condenser (kJ/kg); $h_9$ is isentropic organic fluid enthalpy at outlet turbine/inlet condenser (kJ/kg); $m_{l}$ is mass flow rate (kg/s); $T_{amb}$ is ambient temperature (°C); $s_1$ is organic fluid entropy at inlet pump 1/outlet condenser (kJ/kg K); $s_2$ is isentropic organic fluid entropy at outlet pump 2/inlet evaporator (kJ/kg K); $s_3$ is organic fluid entropy at outlet evaporator/inlet turbine (kJ/kg K); and $s_7$ is organic fluid entropy at outlet turbine/inlet condenser (kJ/kg K).

R227ea, R245fa, and R141b are chosen as the working fluid for the analysis. They are chosen because they are suitable to be used at a low to medium heat source temperature [18][19]. The properties of the three fluids are shown in Table 1 [20][21].

### III. Results and Discussions

The performance analysis is divided into 5 sections, that is heat input and heat loss analysis, pump and turbine power analysis, net power output and thermal efficiency analysis, exergy input and exergy output analysis, and lastly, the exergy efficiency analysis. The five sections are depicted as the step of the energy and exergy analysis to obtain the system performance of SSRORC with various OFOH pressure.

#### A. Heat input and heat loss analysis

The results of heat input ($Q_{in}$) and heat loss ($Q_{loss}$) calculations for three fluids in all eleven pressure values at the OFOH inlet the OFOH are shown in Figure 4. Figure 4 shows the same heat input values for all fluids at different $P_6/P_1$, hence the constant initial data for all states and all fluids. Equation (1) shows that $Q_{in}$ is influenced by the mass flow rate ($m_{of}$), heat capacity ($C_{p_{hw}}$), and temperature differences ($T_9 - T_8$) of the heat source. Those three parameters are constant for all fluids and all $P_6/P_1$, hence the $Q_{in}$ becomes constant.

Figure 4 shows the heat loss ($Q_{loss}$) for R227ea obtains the highest value and R141b obtains the lowest $Q_{loss}$ among the three different fluids. This condition is connected to the properties of each fluid that is shown in Table 1, where R227ea has the lowest critical pressure and R141b has the highest critical pressure. The result determines that R227ea with the lowest critical pressure obtain a lower pressure at the same $P_6/P_1$ than the other fluids, thus made R227ea obtains the highest $Q_{loss}$ than others. R227ea obtains the lowest $Q_{loss}$ at $P_6/P_1 = 2$, and the highest result is obtained from $P_6/P_1 = 3.25$. R245fa and R141b obtain their lowest $Q_{loss}$ at $P_6/P_1 = 2.25$, and their highest result is obtained from $P_6/P_1 = 3.75$. Figure 3 shows that the middle value between the ratio OFOH pressure at the inlet side ($P_6$) and pump 1 ($P_1$) pressure at the inlet side shows the lowest heat loss for all fluids.

#### B. Pump power and turbine power analysis

Figure 5 shows the result of pump power and turbine power calculations for three fluids in eleven states of pressure at inlet the OFOH. Figure 5 shows the highest $W_p$ obtained by R227ea, and the lowest

| Properties | R227ea | R245fa | R141b |
|------------|--------|--------|-------|
| Molecular Weight (g/mol) | 170.03 | 134.05 | 116.95 |
| Boiling Temperature (°C) | -16.19 | 15.29 | 32.2 |
| Critical Temperature (°C) | 101.9 | 154.16 | 204.5 |
| Critical Pressure (bar) | 28.7 | 36.1 | 42.1 |
| Type | Dry | Dry/Isentropic | Isentropic |
\( W_p \) obtained by R141b. Moreover, R141b obtains the highest \( W_t \) and R227ea obtains the lowest \( W_t \). \( W_p \) for R227ea obtains the lowest and the highest result at \( P_6/P_1 = 3.25 \) and \( P_6/P_1 = 2.25 \), while \( W_t \) obtains its lowest and highest result at \( P_6/P_1 = 3.25 \) and \( P_6/P_1 = 2.25 \). R245fa and R141b obtain the highest and the lowest \( W_p \) at \( P_6/P_1 = 2.5 \) and \( P_6/P_1 = 3.75 \). Furthermore, the \( W_t \) value for R245fa and R141b obtain at \( P_6/P_1 = 2.25 \) and \( P_6/P_1 = 3.75 \) for the highest and the lowest \( W_t \). The \( W_p \) value for each state and each fluid are not significantly different, while for \( W_t \), the middle ratio of \( P_6 \) and \( P_1 \) obtain the highest value, and the lowest value obtain from the highest \( P_6/P_1 \), which is close to inlet turbine pressure, thus made it gain the lowest \( W_t \) and \( W_p \). The result of \( W_p \) and \( W_t \) are then used to determine the net power output (\( W_{\text{net}} \)) of the system, thus will obtain the energy performance of the system.

C. Net power output and thermal efficiency analysis

Net power output and thermal efficiency calculation are the final energy analysis. The result is shown in Figure 6. Figure 6 shows that \( W_{\text{net}} \) for R227ea obtains the highest result at \( P_6/P_1 = 2 \), R245fa and R141b at \( P_6/P_1 = 2.5 \). On the contrary, R227ea obtains the lowest \( W_{\text{net}} \) at \( P_6/P_1 = 3.25 \), and R245fa and R141b at \( P_6/P_1 = 3.75 \) obtain the lowest \( W_{\text{net}} \). \( W_{\text{net}} \) value is influenced by the difference between \( W_p \) and \( W_t \), which means that the higher \( W_t \) and the lower \( W_p \) obtain a high \( W_{\text{net}} \). Figure 5 shows that \( W_p \) for R227ea obtains the lowest result at \( P_6/P_1 = 3.25 \), while \( W_t \) obtains its highest result at \( P_6/P_1 = 2 \). The result shows that \( W_{\text{net}} \) value is more influenced by \( W_t \) than \( W_p \) because \( W_{\text{net}} \) value results in the highest value at \( P_6/P_1 = 2 \), which is the same as \( W_t \).
Furthermore, $W_p$ value obtains a very low value compared to $W_t$, thus made $W_{net}$ value more influenced by $W_t$ than $W_p$. R245fa and R141b also obtain the highest $W_{net}$ at $P_6/P_1 = 2.25$ because they obtain the highest $W_t$ at the same $P_6/P_1$. However, R141b obtains the highest $W_{net}$ value than R227ea and R245fa. The lowest $W_p$ value obtained by R141b than R227ea and R245fa made R141b obtain the highest $W_{net}$ value for each fluid almost obtain the same result in the same condition.

Thermal efficiency ($\eta_{th}$) value depends on heat input ($Q_{in}$) and $W_{net}$ value. Since $Q_{in}$ is the same for all fluids and all states, then $\eta_{th}$ value depends on $W_{net}$. Equation (5) shows that the higher the $W_{net}$ obtain a higher $\eta_{th}$. It is clear that Figure 6 shows the highest $\eta_{th}$ value for R227ea obtained from $P_6/P_1 = 2$, the lowest $\eta_{th}$ obtained from $P_6/P_1 = 3.25$, which is the same states with $W_{net}$. The result also the same with R245fa and R141b, where $\eta_{th}$ value obtains its highest value at $P_6/P_1 = 2.25$, and they obtain the lowest $\eta_{th}$ at $P_6/P_1 = 3.75$, the same with $W_{net}$ value. Furthermore, R141b obtain the highest $\eta_{th}$ than R245fa and R227ea because R141b gain the highest $W_{net}$ than other fluids. This result shows that R141b obtains the best energy performance than R227ea and R245fa.

D. Exergy input and exergy loss analysis

Exergy input and exergy loss are two main parameters for exergy analysis. The result of both parameters for three fluids in eleven states is shown in Figure 7. Figure 7 shows that all fluids obtain the highest $E_{xin}$ at the highest $P_6/P_1$, which is 3.25 for R227ea, and 3.75 for R245fa and R141b. In addition, the lowest $E_{xin}$ obtained from the lowest $P_6/P_1$, the same with $W_{net}$.
which is 1.25. It can be analyzed that the highest pressure difference from P1 and P6 obtain the highest $E_{li}$. However, $E_{li}$ value for all fluids at the same states obtain almost the same because of the same $Q_{in}$ value for all fluids at all states. Figure 7 shows that R227ea obtains its highest and its lowest $E_{li}$ at $P_6/P_1 = 3.25$ and at $P_6/P_1 = 1.75$. Similarly, R245fa and R141b obtain their highest and lowest $E_{li}$ at $P_6/P_1 = 3.75$ and at $P_6/P_1 = 2.25$. $E_{li}$ has a similar curve with the $Q_{loss}$, where it reaches the lowest $P_6/P_1$ in the middle and the highest $P_6/P_1$ at the highest value. The exergy input is then used to determine the exergy efficiency of the system.

E. Exergy efficiency analysis

Exergy efficiency is determined to perform the exergy performance of the system. Figure 8 shows the results for the three fluids in eleven states of OFOH in the SSRORC system. Figure 8 shows that $\eta_{ex}$ obtains the highest value at $P_6/P_1 = 1.75$ for R227ea, nearly the same as at $P_6/P_1 = 2$, and at $P_6/P_1 = 2.25$ for R245fa and R141b. On the contrary, $\eta_{ex}$ has the lowest value for R227ea at $P_6/P_1 = 3.25$, and for R245fa and R141b at $P_6/P_1 = 3.75$. The result is the same as $\eta_{th}$, where they obtain the highest value at the middle $P_6/P_1$ and the lowest value at the highest $P_6/P_1$. Although $\eta_{ex}$ depends on $W_{net}$ and $E_{in}$ value, in this analysis the $W_{net}$ result is more dominant than $E_{in}$ for $\eta_{ex}$. Since $E_{in}$ value is almost the same for all fluids and all states, the $\eta_{ex}$ value has the same maximum and minimum value as $W_{net}$ and $\eta_{ex}$.

IV. Conclusion

Open feed organic heater (OFOH) pressure analysis using eleven states obtain the best performance with thermal and exergy efficiency of 10.97 % and 11.96 %, thus made R141b is the most recommended fluid to be used for 100 °C of heat source temperature rather than R227ea and R245fa. Since in this paper it is assumed that a constant heat source temperature is used in analyzing the OFOH pressure influence to the SSRORC performance, the variation of the heat source temperature will be done in the future to complete the analysis of OFOH pressure to the performance of SSRORC.

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Declarations

Author contribution

G. Pikra is the main contributor of this paper. All authors read and approved the final paper.

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Conflict of interest

The authors declare no conflict of interest.

Additional information

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