Analysis of organic Rankine cycle based on thermal and exergy efficiency

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Abstract. The binary power plant is a geothermal power generation system used at low to medium temperature levels and used indirect system. In this system, the heat generated from the reservoir is channelled to the secondary working fluid which has a lower boiling point than water using a heat exchanger. In this case, the Organic Rankine Cycle (ORC) is a suitable system for use. This improves the performance and efficiency of the plants. However, in this system, the installation configuration of the ORC model is an important factor that can affect its performance. The selection of an improper ORC design reduces the thermal efficiency of the system. Therefore, it cannot utilize heat optimally. In this study, Engineering Equation Solution (EES) simulation program is used to run the system as in operation conditions. A comparative analysis is conducted using ORC, Regenerative ORC (RORC), and RORC with Internal Heat Exchanger (IHE). The results indicate that RORC with IHE has the greatest value for both energy (21.74%) and exergy efficiency (25.26%) while net power produced 5479 kW. This shows the addition of OFOH and IHE can increase these factors, improve performance and reduce energy degradation from the cycle.

1. Introduction
Indonesia is a country that passes through an arc known as Pacific fire or "ring of fire". Although it has great potential as a disaster-prone area, the country also saves a lot of natural resources, especially geothermal. It is proven that Indonesia has become one of the countries with the greatest heat potential in the world with resources of 11,073 MW and reserves of 17,506 MW. However, geothermal energy used as a power plant in 2018 has only reached 1,924.5 MW or about 11.03% of the total reserves obtained from 13 producing Geothermal Areas [1]. Besides, the acceleration of the development of renewable power plants is increasingly being intensified by the Indonesian government to obtain the urgency of national electricity needs. This advancement is a recommendation issued in Government Regulation No. 79 of 2014 concerning National Energy Policy to increase the total energy mix which is sponsored to contribute to 23% or the equivalent of 45 MW in 2025 [2].

The use of geothermal energy can be conducted with three types of systems, namely, dry steam, flash steam, and binary cycle power plants. The process involves using heat from the reservoir to drive the turbine. If the heat is in the vapor phase and has a temperature above 370°C, it employs dry steam power
plants where vapor from the geothermal resources is directly used to rotate the turbine. If it is a mixture of steam and liquid and between 170°C-370°C, it uses flash steam power plants where the fluid from the reservoir will be divided by a separator before being used to drive the turbine. In the binary cycle power plant, heat from the reservoir is used to warm up the working fluid in the secondary system on the heat exchanger and then used to drive the turbine. Furthermore, the combination of flash and binary cycle plants is being developed to improve the efficiency of current heat utilization. The Binary cycle power plant is a very effective generator in the utilization of low temperature geothermal energy and can be used to increase the utilization of heat from waste brine in the flash cycle power plant's separator before being inserted into the injection well. This technology works with low to medium enthalpies with temperatures of 120-180°C through working fluids that have lowest boiling points than water. In its development, the Binary power plant system is better known as the Organic Rankine Cycle (ORC). This is a development of the conventional Rankine Cycle; the difference is the working fluid used. It generally uses lower boiling working fluids and higher pressure than water.

The technology of Binary has been implemented in several geothermal power plants in Indonesia. One of them is Lahendong Unit III, located in North Sulawesi, which is designed to consume around 800 tons/day of a mixture of two phases of brine steam mixture from 4 production wells in cluster 5 [3]. The reservoir pressure in this unit is 13.4 bar and 10. 5 bar by separator produces a total of 175 tons/day of vapor in a steam manifold of 180 °C [3]. The relatively low temperature of geothermal fluid when compared to the current conventional power plants have higher temperatures but provide relatively low energy efficiency. To determine the efficiency of a geothermal power plant is not enough if it only refers to the energy (Thermodynamic Law 1) [4,5] because the method is less able to describe important aspects of its utilization. Therefore, it needs to be combined with an exergy approach based on the law of Thermodynamics II [5]. DiPippo conducted a study of second law efficiency on binary cycle geothermal power plant which showed that it has high exergy efficiency even though the geothermal fluid used has a low temperature. [6]. Exergy analysis can also be used to identify the types, causes, and locations of losses in thermal systems and sub-systems, thus, quality improvements can be made. Several different installation configurations were used in this study, namely basic ORC, Regenerative ORC (RORC), and RORC with Internal Heat Exchanger (RORC with IHE). This is a form of modification of the ORC system to compare differences in exergy and energy efficiency. Therefore, the ORC design which is more optimal can be seen.

2. Literature Review

2.1 ORC Generation System

Organic Rankine Cycle (ORC) generation system has 4 main components, namely, pump, turbine, evaporator, and condenser. The pump increases the working fluid pressure towards the evaporator which provides heat to be absorbed through the heat exchanger. This fluid changes from liquid to saturated vapor because of an isobaric heating process. High pressure and temperature working fluid stores energy to rotate the blade in the turbine and after expansion, the pressure would be lost and go through the condenser. This component has a lower temperature to absorb heat from the system towards the pump in a liquid state, which is compressed and then enters the evaporator to be circulated as a sustainable system. Modified components such as the Internal Heat Exchanger (IHE) are forced to accelerate the process of absorption or release of heat. It will reduce the burden on the main components of the ORC system.

The study uses a thermodynamic model of the utilization of geothermal brine before being injected into wells which still have medium to low temperatures using 3 different installation configurations, based ORC in Figure 1, RORC in Figure 2, RORC with IHE in Figure 3.
The design of the equations to be solved by EES referred to in the literature for ORC bases in Table 1, RORC in Table 2, RORC with IHE in Table 3.

| Component | Energy equation | Exergy equation |
|-----------|-----------------|-----------------|
| Pump      | \( \eta_p = \frac{v_3 (P_5 - P_3)}{h_4 - h_3} \) | \( \dot{E}_{D,P} = T_0 \dot{m}_{wf} (s_4 - s_3) \) |
|           | \( \dot{W}_p = \frac{\dot{m}_{wf} (h_4 - h_3)}{\eta_p} \) |               |
| Evaporator| \( \dot{Q}_E = \dot{m}_{wf} (h_1 - h_4) \) | \( \dot{E}_{D,E} = T_0 [\dot{m}_{wf} (s_1 - s_4) + \dot{m}_{Brine} (s_{brine.in} - s_{brine.out})] \) |
|           | = \( \dot{m}_{Brine} (h_{brine.in} - h_{brine.out}) \) |               |
| Turbine   | \( \eta_T = \frac{(h_1 - h_2)}{(h_1 - h_2)} \) | \( \dot{E}_{D,T} = T_0 \dot{m}_{wf} (s_1 - s_2) \) |
|           | \( W_T = \dot{m}_{wf} \eta_T (h_1 - h_2) \) |               |
| Condenser | \( \dot{Q}_C = \dot{m}_{wf} (h_2 - h_3) \) | \( \dot{E}_{D,C} = T_0 [\dot{m}_{wf} (s_2 - s_3) + \dot{m}_{CW} (s_{CW.out} - s_{CW.in})] \) |
|           | = \( \dot{m}_{CW} (h_{CW.out} - h_{CW.in}) \) |               |

**Figure. 1.** Configuration of Based ORC

**Figure. 2.** Configuration of RORC

**Figure. 3.** Configuration of RORC with IHE
This loss identification and qualification allows for evaluation and improvement of thermal system energy. According to Yogi et al., one of the uses of exergy is its balance in thermal analysis systems. Exergy is a measure of the availability of energy to do work. The exergy shows a large indication of the work that resources can do in a particular environment. This concept explicitly shows the quality of heat loss and the location of energy consumption during the conversion or transfer of energy.

### Table 2. Summary of energy and exergy equations for RORC [7].

| Component       | Energy equation | Exergy equation |
|-----------------|-----------------|----------------|
| Pump            | $\eta = \frac{v7(P7 - P6)}{h7 - h6} = \frac{v5(P5 - P4)}{h5 - h4}$ | $\dot{E}D.P = T0mwf[(1 - X)(s5 - s4) + (s7 - s6)$ |
|                 | $\dot{W}p = \dot{m}wv/(1 - X)(h5 - h4) + (h7 - h6)$ | |
| Evaporator      | $\dot{Q}E = \dot{m}wv(h1 - h7)$ | $\dot{E}D.E = T0[\dot{m}wv(s1 - s7) +$ |
|                 | $= \dot{m}Brine(hbrine.in - hbrine.out)]$ | $\dot{m}Brine(sbrine.in - sbrine.out)]$ |
| Turbine         | $\etaT = \frac{(h1 - h2)}{(h1 - h2x)} = \frac{(h2 - h3)}{(h2 - h3x)}$ | $\dot{E}D.T = T0mwf(s1 - s2) + X(s2 - s3)$ |
|                 | $\dot{W}T = \dot{m}wv[(h1 - h2) + (1 - X)(h2 - h3)]$ | |
| Open Feed-Organic Hea | $X = \frac{(h2 - h5)}{(h2 - h5)}$ | $\dot{E}D.OFOH = T0mwf[s6 - Xs2 - (1 - X)s5]$ |
| Conden | $\dot{Q}C = \dot{m}wv(h3 - h4)$ | $\dot{E}D.C = T0[\dot{m}wv(s3 - s4) + \dot{m}CW(sCW.out - sCW.in)]$ |

### Table 3. Summary of energy and exergy equations for RORC with IHE [7].

| Component       | Energy equation | Exergy equation |
|-----------------|-----------------|----------------|
| Pump            | $\eta = \frac{v9(P9 - P8)}{h9 - h8} = \frac{v6(P6 - P5)}{h6 - h5}$ | $\dot{E}D.P = T0mwf[(1 - X)(s6 - s5) + (s9 - s8)$ |
|                 | $\dot{W}p = \dot{m}wv/(1 - X)(h6 - h5) + (h9 - h8)$ | |
| Evaporator      | $\dot{Q}E = \dot{m}wv(h1 - h9)$ | $\dot{E}D.E = T0[\dot{m}wv(s1 - s9) +$ |
|                 | $= \dot{m}Brine(hbrine.in - hbrine.out)]$ | $\dot{m}Brine(sbrine.in - sbrine.out)]$ |
| Turbine         | $\etaT = \frac{(h1 - h2)}{(h1 - h2x)} = \frac{(h2 - h3)}{(h2 - h3x)}$ | $\dot{E}D.T = T0mwf(s1 - s2) + X(s2 - s3)$ |
|                 | $\dot{W}T = \dot{m}wv[(h1 - h2) + (1 - X)(h2 - h3)]$ | |
| Open Feed-Organic Hea | $X = \frac{(h8 - h7)}{(h2 - h7)}$ | $\dot{E}D.OFOH = T0mwf[s8 - Xs2 - (1 - X)s7]$ |
| Internal Heat Exchanger | $\epsilon = \frac{T3 - T4}{T3 - T6}$ | $\dot{E}D.IHE = T0[\dot{m}2(s2 - s3) + \dot{m}8(s9 - s8)]$ |
| Conden | $\dot{Q}C = \dot{m}wv(h3 - h4)$ | $\dot{E}D.C = T0mwf[X(s3 - s4) + (s7 - s6)]$ |

### 2.2 Exergy Concept

Exergy is a measure of the availability of energy to do work. The exergy shows a large indication of the work that resources can do in a particular environment. This concept explicitly shows the quality of energy and substances as an additional use of its consumption during the conversion or transfer of energy. According to Yogi et al., one of the uses of exergy is its balance in thermal analysis systems. This loss identification and qualification allows for evaluation and improvement of thermal system design [8]. Exergy analysis methods show the quality and quantity of heat loss and the location of energy.
losses. Most cases of thermodynamic imperfection can be detected by these methods. Actual work and reversible work equations are often formulated in the exergy function equation for open and closed systems.

The dead balance phase occurs when a system and environment are in equilibrium suddenly changes in temperature and pressure. This equilibrium is in the standard atmosphere of 298.15 K and 1.01325 bar (1 atm). Exergy can be calculated by the following equation:

\[ X = \dot{Q}(1 - T_0/T) \]  

Exergy thermomechanics can be calculated by the following equation

\[ ex = h1 - h0 - T0(s1 - s0) \]  

Exergy loss can be calculated as follows,

\[ \dot{I} = \dot{W}_{rev.out} - \dot{W}_{out} \]  

Energy efficiency can be calculated as follows,

\[ \eta_I = \frac{W_{mbrine(hevap.in)}}{W_{mbrine(hevap.out)}} \]  

Energy efficiency can be calculated as follows

\[ \eta_{II, plant1} = \frac{W_{net}}{W_{net}} \]  

\[ \eta_{II, plant2} = \frac{W_{net}}{W_{net}} \]  

\[ \eta_{II, plant2} = \frac{W_{net}}{W_{net}} \]  

\[ W_{net} = W_{turbine} - W_{pump} \]  

2.3 Selection of Working Fluid

The working fluid is used in the binary cycle geothermal power plant to drive the turbine with the energy it carries. Brine fluid does not come in direct contact to drive the turbine because it contains compounds that damage this engine. Furthermore, the condition of the brine with a pressure and temperature that is not high enough cannot rotate the steam turbine properly. Thus, a working fluid is needed that has several criteria that need to be met in the condition of the unit III Lahendong geothermal power plant, such as critical temperature, pressure, and other desired thermodynamic properties. Latent, low or near-vertical fluid heat from the saturated liquid line is needed, therefore, the maximum is obtained throughout phase changes without the need for regenerative heat to achieve high cycle efficiency.

The working fluid used in this study is Isopentane, which has the following properties in Table 4:

| Organic Fluid | Tc (°C) | Pc (Mpa) | Wi (Watt) | Wr (Watt) | Wnet (Watt) | ηI (%) | ηII (%) | ηIII (%) | Iout (Watt) | Pout (Mpa) | Pلوم (Mpa) | Type |
|---------------|--------|---------|-----------|----------|-------------|--------|--------|---------|------------|-----------|-----------|------|
| Isopentane    | 187.2  | 3.37    | 37325     | 1223     | 36102       | 14.37  | 9.564  | 41234   | 0.109      | 1.311     | 12.024     | Dry  |

3. Methods

The primary data is taken on the condition of Lahendong geothermal plant unit III [3] and the secondary from several supporting literature [10]. While for data processing, EES is used to show the thermodynamic fluid properties and run the model under operating conditions. The assumptions used in this study are as follow:

- Environmental air temperature in Lahendong unit III of 28 °C.
- Atmospheric pressure in the lower field is 0.98 bar
- The pressure drop on the evaporator, condenser and pipeline is ignored.
- Isentropic turbine efficiency of 0.85 and pump of 0.75.
- The increase in temperature in cooling water in the condenser is 10 °C
- Changes in potential and kinetic energy are ignored
- There are no leaks in the system

4. Results and Discussion
This study uses several parameters taken from several previous studies as can be seen on Table 5.

| Input Parameter                     | Unit | Value and Explanation |
|------------------------------------|------|-----------------------|
| Working Fluid                      |      | isopentane            |
| Brine Input Temperature            | °C   | 180                   |
| Brine Input Temperature            | °C   | 146.7                 |
| Pressure brine input               | °C   | 10.23                 |
| Pressure brine output              | °C   | 10.23                 |
| Mass flow rate brine               | Kg/s | 173.56                |
| Temperature input Evaporator       | °C   | 164.6                 |
| Turbines Isentropic Efficiency     | %    | 0.85                  |
| Pump isentropic efficiency         | %    | 0.75 [12]             |
| P outlet Pump based ORC             |      | 1.68637               |
| P outlet pump RORC                 | Bar  | 5.812                 |
| P outlet pump RORC with IHE        |      | 4.945                 |
| Cooling water input temperature    | °C   | 28                    |
| Cooling water output temperature   | °C   | 38                    |
| Cooling water output pressure      | Bar  | 0.89                  |
| Cooling water input pressure       | Bar  | 0.89                  |
| Temperature ambient                | °C   | 28                    |
| Pressure ambient                   | Bar  | 0.89                  |
| Generator Efficiency               | %    | 1                     |

4.1 Energy Efficiency and Exergy Efficiency Analysis

4.1.1 Performance Analysis of Each Component
The overall exergy simulation results and each component of the three compared cycles are summarized in table 6 for ORC, table 7 for RORC, table 8 for RORC with IHE.

| Component           | Exergy degradation (kW) | Exergy Efficiency (%) | Degradation ratio | Energy efficiency (%) | Heat transfer (kW) |
|---------------------|-------------------------|-----------------------|-------------------|-----------------------|------------------|
| Evaporator          | 1884                    | 76.02                 | 8.686             | 0.85                  | 25196            |
| Condenser           | 951.1                   | 27.16                 | 4.385             | 0.85                  | 21054            |
| Turbine             | 467.8                   | 90.36                 | 2.157             | 0.85                  | 4386             |
| Pump working fluid  | 57.96                   | 76.19                 | 0.2672            | 0.75                  | 243.4            |
Large energy or work lost from each component for each cycle has the same tendency where the evaporator has the greatest value compared to the other main components. Heat exchange in the evaporator requires a large amount of energy because this process requires a significant increase in the temperature of the working fluid with high pressure. The importance of heat exchangers on the evaporator plays the role of individual component performance to be able to influence the overall performance of the cycle.

The results of the analysis of exergy efficiency at all cycles can be considered high and indicate that the performance of the heat exchanger of the cycle is good. The low efficiency of the exergy possessed by the condenser when compared with the other main components due to the low difference in temperature of the working fluid with the cooling fluid. Besides, it is caused by the absence of pressure changes in the condenser. Similar to the OFOH component which has a relatively low exergy efficiency, the heat exchange is not insignificant as an auxiliary component to accelerate heat exchange in the main component. It is proven by the higher exergy efficiency of the RORC configuration compared to the basic ORC configuration.

4.1.2 Exergy Degradation of Each Component
The exergy degradation ratio is the magnitude of exergy degradation of a component divided by the amount of exergy carried by geothermal fluid to enter into the cycle. The degradation ratio obtained from the simulation results is then plotted into the graph in Figure 4 to show a comparison of the loss of each component.

The impact of adding heat exchanger components such as OFOH and IHE significantly reduces energy degradation in each one. Some of the main components that have decreased degradation ratios such as evaporator and condenser show that the addition of OFOH and IHE improves heat absorption better so that the exhaust from the cycle decreases. On the other hand, turbines and pumps have increased the degradation ratio which is not significant. This is because the mass flow rate of the working fluid increases as it enters the turbine and also increases the work of its pump. However, the addition of these components also reduces turbine and evaporator load, where it has a heavier work to exchange heat and produce electricity through a generator. Therefore, it can be concluded that ORC has considerable waste heat which has not been utilized optimally.

**Table 7. RORC Simulation Results**

| Component         | Exergy degradation (kW) | Exergy Efficiency (%) | Degradation ratio | Energy efficiency (%) | Heat transfer (kW) |
|-------------------|-------------------------|-----------------------|-------------------|-----------------------|-------------------|
| Evaporator        | 1005                    | 87.21                 | 4.633             | 25196                 |
| Condenser         | 787.9                   | 35.85                 | 3.632             | 20170                 |
| Turbine           | 228.9                   | 86.01                 | 1.055             | 0.85                  | 5346              |
| Pump working fluid| 68.45                   | 83.7                  | 0.3156            | 0.75                  | 301.1             |
| OFOH              | 302.2                   | 4.134                 | 1.393             | 10198                 |

**Table 8. RORC with IHE Simulation Results**

| Component         | Exergy degradation (kW) | Exergy Efficiency (%) | Degradation ratio | Energy efficiency (%) | Heat transfer (kW) |
|-------------------|-------------------------|-----------------------|-------------------|-----------------------|-------------------|
| Evaporator        | 878.5                   | 88.82                 | 4.05              | 25196                 |
| Condenser         | 676.7                   | 35.55                 | 3.12              | 19739                 |
| Turbine           | 253.2                   | 108.6                 | 1.167             | 0.85                  | 5800              |
| Pump working fluid| 72.49                   | 84.05                 | 0.3342            | 0.75                  | 321.8             |
| OFOH              | 57.81                   | 136.8                 | 0.2662            | 321.8                 |
| IHE               | 100                     | 82.58                 | 0.461             | 4626                  |
4.1.3 Comparison of Energy and Exergy Efficiency

The power produced and the comparison of energy efficiency and exergy of the cycles with different configurations are compared in Table 9.

| Performance Parameters | Based ORC | RORC | RORC with IHE |
|------------------------|-----------|------|---------------|
| Net Power (kW)         | 4143      | 5045 | 5479          |
| \( \eta_{\text{L,plant}} \) (%) | 16.44     | 20.02 | 21.74         |
| \( \eta_{\text{L,plant,1}} \) (%) | 19.1      | 23.26 | 25.26         |
| \( \eta_{\text{L,plant,2}} \) (%) | 52.72     | 64.2  | 69.72         |
| \( \eta_{\text{L,binary}} \) (%) | 69.34     | 73.62 | 78.49         |

Consecutive energy efficiency at ORC, RORC, and RORC with IHE were 16.44%, 20.02%, and 21.74%. This means that more than 78% of the energy is discarded and untapped. The exergy efficiency produced by the three cycles is still quite low, while to increase efficiency, cycle performance, and turbine power, the ability to utilize exergy is very important. The addition of OFOH and IHE components is an effort to improve the performance and efficiency of the ORC system. This can be seen from the significant difference between ORC and RORC with IHE. The increase in turbine net power is accompanied by a rise in energy and exergy efficiency of the cycle therefore, it can be considered in the design of the power plant.

4.2 Analysis Effect of Condenser Pressure

In addition to the analysis of the effect of turbine inlet temperature, the impact of condensation pressure is also made on net power and efficiency of the cycle.
The data obtained from the simulation results show that the turbine inlet temperature is proportional to the net power as seen in figure 5. If this temperature increases, energy efficiency increases as seen in figure 6. At a certain point, the increase reaches a limit, experience a decrease and then stop. Approaching the critical temperature of the isopentane fluid which is 187.2 °C causes the cycle not to take place because the working fluid has high pressure and cannot change phase.

4.3 Analysis Effect of Condenser Pressure

In addition to the analysis of the effect of turbine inlet temperature, the impact of condensation pressure is also made on net power as seen in table 7 and efficiency of the cycle as seen in Table 8.
5. Conclusion

Binary cycles aim to increase the efficiency of the energy conversion process of geothermal power plants by utilizing the brine disposed. Optimization was carried out in this study by adding IHE and OFOH components. Based on the simulation using the EES program, the exergy and energy analysis can be concluded RORC with IHE has the greatest value, both energy efficiency values (21.74%) and exergy efficiency (25.26%) and net power produced 5479 kW. Addition of OOH and IHE can increase energy efficiency and exergy, net power, and better performance and reduce energy degradation from the cycle. Higher turbine inlet temperature values will increase both net power and energy efficiency and exergy. Decreasing condensation pressure increases both net power and energy efficiency and exergy and this can be proven by previous research [13].

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