# Exergetic Sensitivity Analysis of ORC Geothermal Power Plant Considering Ambient Temperature

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## ABSTRACT

Binary cycles are the most common method to extract energy from moderate geothermal resources worldwide. This research investigates and compares the performance of Organic Rankine Cycle (ORC) for binary geothermal power plants from the thermodynamic point of view. A hypothetical ORC cycle which is using 150°C geothermal water as a heat source was considered in this study and its exergy and energy analysis were performed. A parametric study was also performed to understand the effect of ambient temperature and its effect on both energy and exergy efficiencies. The exergy and energy efficiencies, cooling water pump power consumption and also exergy destruction for each component are calculated within different ambient temperature and dead state as well.

The results indicated that energy efficiency of the plant varies by 1 and 7% in two different scenarios that were assumed based on different control volume description for the system.

Scenario (a) and (b) are based on brine exergy input rate to plant and exergy flow rate decrement across heat exchanger, respectively.

Meanwhile, exergetic efficiencies of the cycle face 70%; and 34% increment based on different exergy inflows scenarios. It was concluded that condensers are the most sensitive component to dead state temperature.

## 1. Introduction

Renewable energy plays a vital role in supplying the world's increasing energy demand [1, 2] Countries are trying to increase the share of renewable energy in their own future energy policies. In addition to the environmental benefits, the main motivation towards high interest in renewable energy, is growing electricity consumption in the last decades. Most of renewable resources are limited by their nature, but if there is a geothermal resource in one region, its availability is constant. Geothermal energy with temperatures varying from 50 to 350°C [3], can be utilized in wide range from direct uses to power production. District heating as direct uses of geothermal energy, both from economical and environmental point of view, has been found superior to other technologies in many situations. [4]

Geothermal power production works based on dry-steam, flash-cycle and binary-cycle. In this regard many studies are performed on performance of evaluation of these plants [5-7].

Jalilinasrabady et al. [8] conducted energy and exergy analysis of Takigami single–flash geothermal power plant using EES. Their Optimization was based on maximizing the net power output of the plant. In optimum condition the net power output and the overall first and second law efficiencies of the plant were calculated to be 24,992 kW, 6.73% and 28.77% respectively.

These studies developed geothermal power plants in two usual ways: the construction of new power plants with new fields and improvements in thermal efficiency of existing power plant [7].

The binary units based on ORCs because of their configuration simplicity, components availability and better economics as well as environmental points are becoming increasingly common [9-11]. Generally the technology of these cycles is similar to thermal power plants and except geothermal reservoir feeds the required steam for turbine rather than a boiler [8].

There are many researches that investigate ORC cycle performance based on different parameters. Investigation of ORC cycle's working fluid has been done in several papers [12-14].

Ronald DiPippo [15] concluded that comparisons among power plants must use an appropriate and consistent thermodynamic basis and conducted Second Law of thermodynamics as a best basis for ORC cycles comparison. Moreover he introduced a methodology to compare plant efficiencies on common input and environmental conditions.

V. Zare [16] compared the performance of three different configurations of ORC cycles for binary geothermal power plants in thermodynamic and economic viewpoints. This paper concluded that organic Rankine cycle in internal heat exchangers and simple ORC are the best cases from thermodynamic and the economical point of view, respectively.

Zvonimir Guzovic et al. [17] analyzed the replacement of a basic ORC with a dual-pressure ORC in order to further improvement of geothermal energy utilization. They concluded that dual-pressure ORC, works with lower thermal efficiency but higher exergy efficiency and net power.

Performance of the combined flash-binary geothermal power cycle for different geofluid temperatures is studied by Mehdi Zeyghami [18] and the suitable binary working fluids are identified from a list of thirty working fluid candidates. The results show that for low-temperature heat sources using refrigerants as binary working fluids result in higher overall cycle efficiency and for medium and high-temperature resources, hydrocarbons are more suitable.

Jalilinasrabady and Itoi [19] introduced a method to choose the most efficient energy conversion technology between a single flash plant as the main energy conversion system, double flash and Organic Rankine Cycle (ORC) as a bottoming unit of the single flash. Comparison of power output shows that ORC cycle is more efficient from the power output point of view.

The main objective of present paper is energy and exergy analysis of a hypothetical binary cycle. This work includes energy and exergy efficiencies based on ambient temperature in ORC cycle which is working with Isopentane as a working fluid. The location of the geothermal field is assumed around the world to observe the effect of ambient temperature on system's performance in the coldest and warmest regions. The reservoir has two well; one for production and the other one for reinjection. The well head temperature and pressure are assumed to be 150°C and 4.7 bar respectively. There are only two pumps in the field, one for working fluid and the other for cooling water. The geothermal fluid is provided using natural pressure and reinjected by gravity, therefore, there is no need to consider pump for it. The mass flow rate of geothermal fluid that comes out and feeds heat exchanger is considered constant and equals to 50 kg/s.

**Table 1.** Important parameters of binary cycle.

| Operating parameters | Value | Unit |
|---|---|---|
| Source temperature | 150 | °C |
| Source mass flow rate | 50 | Kg/s |
| Working fluid upstream pressure | 10 | bar |
| Working fluid downstream pressure | 1.5 | bar |
| Working fluid mass flow rate | 20 | Kg/s |
| Pinch | 5 | °C |
| Turbine isentropic efficiency | 80 | % |
| Pump isentropic efficiency | 95 | % |
| Ambient temperature | 15 | °C |
| Ambient pressure | 1 | atm |
| Geothermal fluid | Water & Steam | |
| Cooling fluid | Water | |
| Working fluid | Isopentane | |

Cycle optimization consists of three parts. In the first part, the geofluid flows to HEX to heat the working fluid in ORC cycle, after heat exchange to working fluid in the secondary cycle, goes back to the reservoir through reinjection well. The plant was assumed to operate in a close loop and geo-fluid is being reinjected to the reservoir again. In the second part, superheated working fluid passes the turbine and completes the ORC cycle by passing through other equipments such as condenser and pump, respectively. Lastly, cooling water at ambient temperature absorbs heat from working fluid at turbine exit and changes its phase to liquid. Finally, the condensed fluid is pumped to the heat exchanger again to the closed cycle. The specifications of hypothetical cycle are presented in Table 1.

It was assumed that the binary cycle is located in vicinity of the well and there is no long pipeline between them, so the heat loss was neglected in this study.

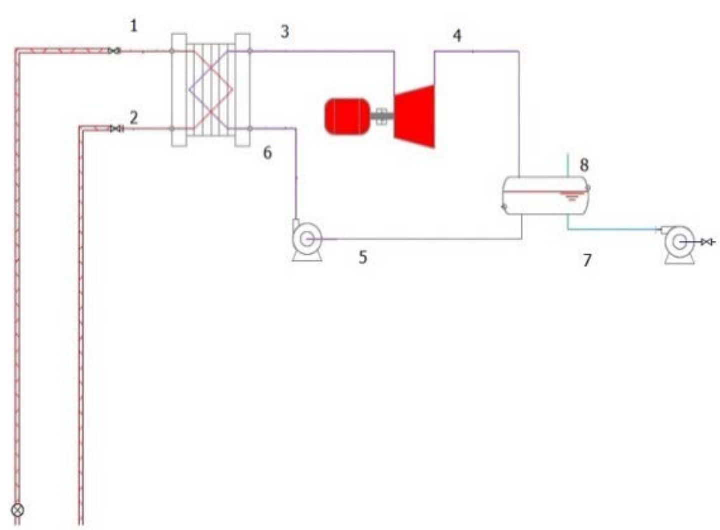


**Figure 1.** Simplified ORC diagram.

## 2. System Description and Assumptions

Given that cycle configuration affects ORC system performance [20], the simple ORC cycle is considered to use hot water as a heat source. Figure 1, shows the simplified diagram of proposed plan. The heat source for the plant is the flow of the geothermal water (brine) enters to the plant at 150°C with the total mass flow rate of 50 kg/s. This stream heats the working fluid and as a result of heat exchange process its temperature decreases to 102°C. Exhaust brine form the heat exchanger is being reinjected to the reinjection well where it goes back to the reservoir.

A parametric study of the cycle was performed based on thermodynamic facts. For each state of the geothermal fluid and ORC cycle, the temperature, pressure, energy and exergy rates were calculated by Engineering Equation Solver (EES) [21]. These calculations the pure water properties was used and the effects of salts and other mineral components in the geothermal fluid that can affect the thermodynamical specifications, were neglected.

In the ORC cycle, 20 kg/s of Isopentan as a working fluid at 40°C enters the heat exchanger and leaves at 139°C. During this process it reaches to the saturated condition and then becomes superheated steam. It was assumed that the heat exchanger consists of compact preheater and vaporizer in one unit. The working fluid then passes through the turbine and exits at about 99°C to a water cooled condenser where it condensates at 39°C. In the condenser, approximately 80 kg/s water at ambient temperature absorbs the heat from working fluid and reaches to 42°C. Figure 2, illustrates T-S diagram of ORC cycle with Isopentane.

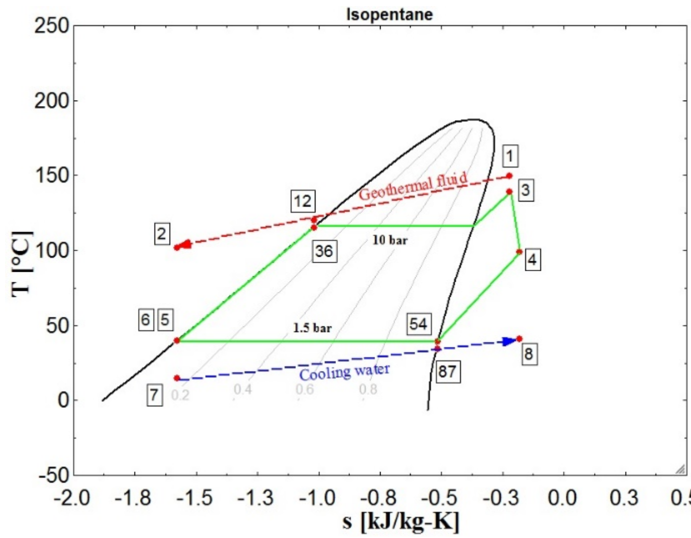


**Figure 2.** Exergy temperature (T-S) diagram of hypothetical cycle.

The heat exchanger process between geothermal brine and working fluid in the high pressure side and between working fluid and cooling water in the low pressure side of the cycle are shown in Figure 3 (a and b). An energy balance was applied to the heat exchanger by assuming 5°C as a pinch point [22].

$$\dot{m}_{gf} \times (h_3 - h_{36}) = \dot{m}_{wf} \times (h_1 - h_{12}) \tag{1}$$

$$\dot{m}_{gf} \times (h_{36} - h_6) = \dot{m}_{wf} \times (h_{12} - h_2) \tag{2}$$

$$T_{12} = T_{36} + 5 \tag{3}$$

$h_{36}$ is defined as saturated liquid enthalpy of working fluid at the saturation temperature (115.8°C) and $h_{12}$ is the enthalpy of the brine at the pinch point. The pinch point difference is the difference between the brine pinch point and the vaporization temperature of the working fluid. The amount of pinch point deference was set to 5°C; therefore, by solving these equations, four temperatures in the heat exchanger's sides can be defined.

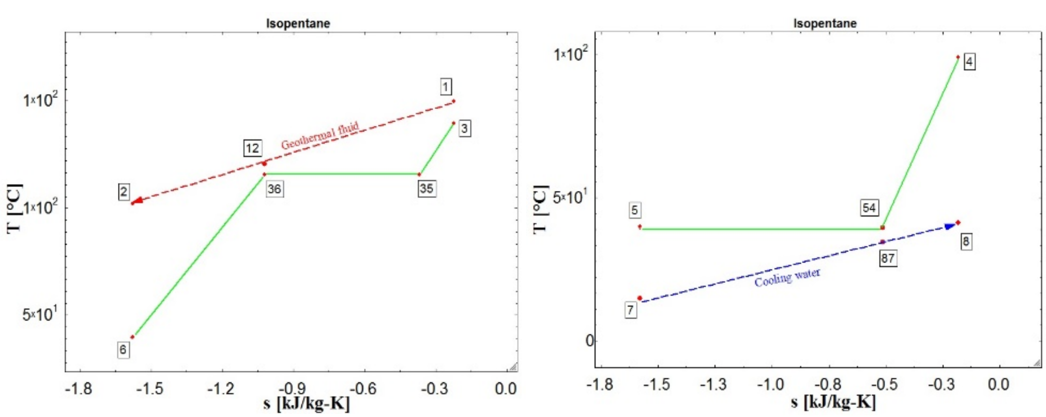


**Figure 3.** (a) Heat exchanger process between geothermal brine and working fluid, (b) between working fluid and cooling water.

The obtained results from the model, show that the actual turbine power output is at 993.6 kW while working fluid circulation pump is consuming 29.83 kW. The net power output of ORC cycle is 963.77 kW; approximately 1 kW is consumed by cooling water circulation pump. Therefore the net power output of the plant (962.9 kW) can be calculated by subtracting the pumps load from the net power of ORC cycle.

### 2.1 Energy Analysis

The energy efficiency of the ORC cycle was calculated based on two different scenarios. In scenario (a), the ratio of the net power output of the ORC cycle to the heat input rate that passes through the HEX, is considered as exergy efficiency and in the scenario (b), the difference between enthalpies of geothermal fluid in actual conditions and dead state was considered as heat input to the system.

$$\eta_{ORC,a} = \frac{W_{net,orc}}{\dot{m}_{gf}\left(h_1 - h_2\right)} \tag{4}$$

$$\eta_{ORC,b} = \frac{W_{net,orc}}{\dot{m}_{gf}\left(h_1 - h_0\right)} \tag{5}$$

### 2.2 Exergy Analysis

Exergy analysis was applied for each component of the plant such as evaporator, turbine, condenser and pump. Exergy is the maximum work when the stream of substance is brought from its initial state to the environmental state defined by $P_0$ and $T_0$ by physical processes involving only thermal interaction with the environment.

The specific flow exergy of the geothermal fluid at any state can be calculated as [23, 24]:

$$ex = \left(h - h_0\right) - T_0\left(s - s_0\right) \tag{6}$$

where h and s are the specific enthalpy and entropy of the geothermal fluid at the specific state, $h_0$ and $s_0$ are the properties at the dead state. For the mass flow rate, the exergy flow rate can be written as:

$$\dot{E}_x = \dot{m}\left(ex\right) \tag{7}$$

According to the above description of exergy concept, the exergy efficiency and exergy destroyed can be calculated for all components of the cycle (Table 2). For the heat exchanger and condenser, the exergy efficiency of a heat exchanger shows the exergy increase of the cold stream divided by the exergy decrease of the hot stream.

The exergy efficiency of the turbine shows how efficient the exergy of fluid passing through turbine is being converted to work and their difference is the exergy destruction rate in the turbine.

The difference between the exergy of the inlet and outlet streams, is the exergy destruction rate in the equipment.

### 2.3 Exergy Efficiency

In general, the exergy efficiency of a geothermal plant is the net electrical power output divided by total exergy flow rate of the brine [15, 24]. Values for exergy efficiency can vary according to definitions: in this study, the exergy output/ input from/ to the plant is considered. The work output from the cycle divided by the exergy value of steam entering the HEX and exergy flow from source are considered as exergy efficiencies in scenarios (a) and (b), respectively.

So, the exergy efficiency of whole cycle represents ratio of the net power to exergy decrement in steam flow across the HEX as well as main stream in two different scenarios. The total exergy loss is sum of the all exergy destruction in the plant.

The other parameters discussed in this study are shown in Table 3. The numbers refer to state locations in Figure 1. States 0 and 0' refer to the restricted dead states for the geothermal and working fluid respectively. They correspond to an ambient

**Table 2.** Exergy relations of ORC cycle.

| Component | Exergy Equations | |
|---|---|---|
| | Efficiency | Destroyed |
| Heat exchanger | $\varphi_{HEX} = \dfrac{\dot{E}_{x3} - \dot{E}_{x6}}{\dot{E}_{x1} - \dot{E}_{x2}}$ | $\dot{I}_{HEX} = \left(\dot{E}_{x1} - \dot{E}_{x2}\right) - \left(\dot{E}_{x3} - \dot{E}_{x6}\right)$ |
| Condenser | $\varphi_{cond} = \dfrac{\dot{E}_{x4} - \dot{E}_{x5}}{\dot{E}_{x8} - \dot{E}_{x7}}$ | $\dot{I}_{cond} = \left(\dot{E}_{x8} - \dot{E}_{x7}\right) - \left(\dot{E}_{x4} - \dot{E}_{x5}\right)$ |
| Turbine | $\varphi_{tur} = \dfrac{\dot{w}_{tur,act}}{\dot{E}_{x3} - \dot{E}_{x4}}$ | $\dot{I}_{tur} = \left(\dot{E}_{x3} - \dot{E}_{x4}\right) - \dot{w}_{tur,act}$ |
| Pump | $\varphi_{pump} = \dfrac{\dot{E}_{x6} - \dot{E}_{x5}}{\dot{w}_{pump}}$ | $\dot{I}_{pump} = \dot{w}_{tur} - \left(\dot{E}_{x6} - \dot{E}_{x5}\right)$ |
| ORC cycle | $\varphi_{ORC} = \dfrac{\dot{w}_{net,ORC}}{\dot{E}_{x1} - \dot{E}_{x2}}$ | $\dot{I}_{ORC} = \dot{I}_{pump} + \dot{I}_{tur} + \dot{I}_{condenser} + \dot{I}_{HEX}$ |

temperature of 15°C and an atmospheric pressure of 84 kPa, which are hypothetical region's environmental conditions. In this section, the effects of ambient temperature variation on the exergy efficiency, will be observed.

Figure 4 shows the energy flow diagram of the plant.

In this paper, the exergy efficiency of cycle is calculated from two different points of view. $\varphi_{plant,a}$ (8) is formulating energy efficiency according to decrement in brine exergy flow rate across HEX and $\varphi_{plant,a}$ (9) is based on brine exergy input to the plant.

**Table 3.** Exergy rate at various locations of the plant.

| State | Fluid | Phase | Temp. T (°C) | Pressure P (bar) | Specific Enthalpy h (kJ/kg) | Specific Entropy s (kJ/kg K) | Mass flow rate $\dot{m}$ (kg/s) | Specific Exergy ex (kW) | Exergy rate $\dot{E}x$ (kW) |
|---|---|---|---|---|---|---|---|---|---|
| 0 | Brine | Dead state | 15 | 86 | 50.44 | 0.1804 | - | 0 | - |
| 0′ | Isopentane | Dead state | 15 | 86 | 265.8 | 1.082 | - | 0 | - |
| 1 | Brine | Liquid | 150 | 4.762 | 632.2 | 1.842 | 50 | 4729 | 94.57 |
| 2 | Brine | Liquid | 102.3 | 4.762 | 429.1 | 1.334 | 50 | 1895 | 37.89 |
| 3 | Isopentane | Steam | 139.6 | 10 | 193.1 | -0.2237 | 20 | 2150 | 107.5 |
| 4 | Isopentane | 2phase | 99.28 | 1.5 | 131 | -0.1816 | 20 | 665.4 | 33.27 |
| 5 | Isopentane | Liquid | 39.73 | 1.5 | -316.1 | -1.579 | 20 | -222.3 | -11.12 |
| 6 | Isopentane | Liquid | 40.09 | 10 | -314.6 | -1.579 | 20 | -193.9 | -9.694 |
| 7 | Water | Liquid | 15 | 1 | 62.92 | 0.2242 | 80.67 | -692.9 | -8.589 |
| 8 | Water | Liquid | 41.5 | 1 | 173.8 | 0.5922 | 80.67 | -304.8 | -3.779 |

$$\varphi_{plant,a} = \frac{\dot{W}_{net}}{\left(\dot{E}_1 - \dot{E}_2\right)} \qquad (8)$$

$$\varphi_{plant,a} = \frac{\dot{W}_{net}}{\dot{E}_1} \qquad (9)$$

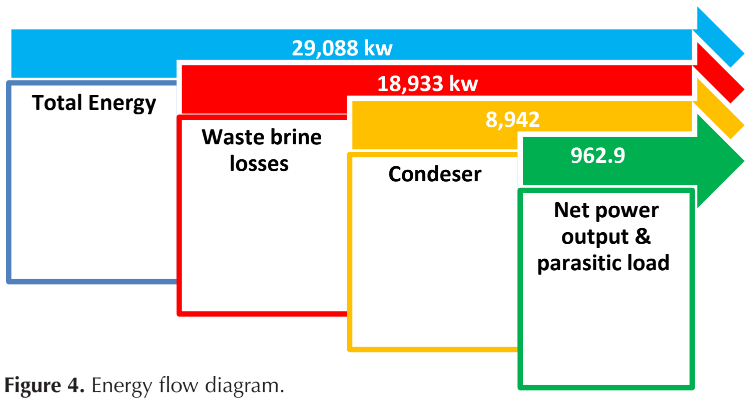


**Figure 4.** Energy flow diagram.

Table 4 shows the exergy flow of plant components and exergy and energy efficiencies, using values from this table Grassman diagram was drawn as shown in Figure 5.

This diagram (figure 5), shows that 79.62% of the exergy entering the plant is lost. The remaining 20.38% is converted to the power. The exergy efficiency of the plant is 20.98% based on the exergy input to the ORC cycle and 33.98%, based on the exergy flow passing through the heat exchanger.

Figure 5 shows exergy destruction at different stages of the plant. At stage 1,895 kw equal to 40.1% of total exergy stands for reinjection brine. It emphasizes on the important of reinjection in reservoir sustainability [8]

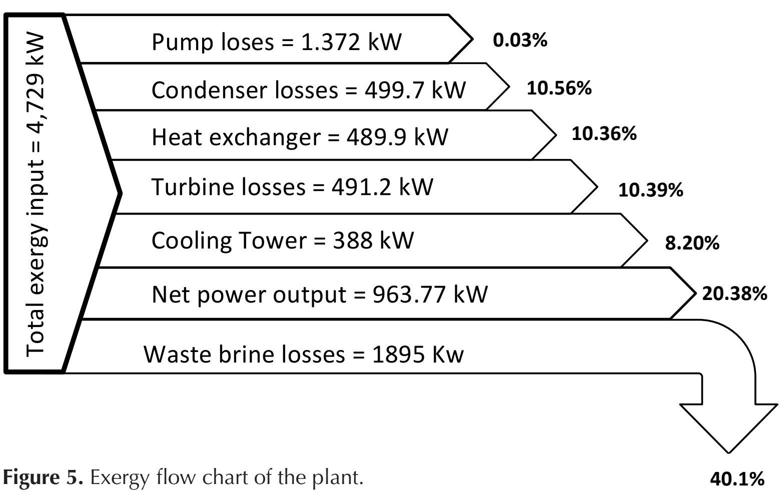


**Figure 5.** Exergy flow chart of the plant.

**Table 4.** Exergy flow in various components of the plant.

| Component | | Exergy Destruction Rate $\dot{I}$ (kW) | Exergy Efficiency φ (%) |
|---|---|---|---|
| Heat exchanger | | 489.9 | 82.71 |
| Condenser | | 499.7 | 43.71 |
| Pump | | 1.372 | 95.4 |
| Turbine | | 491.2 | 66.92 |
| ORC cycle | $\varphi_{plant,a}$ | 1482 | 20.36 |
| | $\varphi_{plant,b}$ | | 33.98 |

otherwise the second law efficiencies would be 60.46% and 74.08% in the scenarios (a) and (b) respectively.

## 3. Results and Discussion

A mathematical model was developed for the binary cycle and exergy and energy analysis were performed in mean ambient temperature.The effects of dead state temperature (ambient temperature) on net power output and the energy efficiencies of the system are shown in the Figure 6 and Figure 7, respectively. The reasonable ambient temperatures based on warmest and coldest states were considered.

As shown in Figure 6 by increasing ambient temperature, net power output decreased. In high ambient temperature, large quantities of water flow rate are required and it caused higher pump consumption of cooling tower.

In the Figure 7 variation of ambient temperature had a different effect on the energy efficiencies. In scenario (a) decreasing net power output as dominant parameter, caused decreasing rate for $\eta_{ORC,a}$ . In scenario (b) the effect of increasing $h_0$ was stronger than net power decrement; so as a result, $\eta_{ORC,b}$ had a growing trend.

Figure 8 shows the effect of dead state temperature on the system exergy efficiencies and its component. Different dead state temperatures make changes in the value of the exergy analysis in different ratios. According to equation 8 and 9 by increasing the ambient temperature, exergy efficiencies were improved. Also changing the ambient temperature as condenser inlet temperature had a very huge effects on its exergy efficiency.

The effect of ambient temperature changes on exergy efficiency of working fluid pump, heat exchanger and turbine, were negligible. Because changing ambient temperature has same effect on exergy of inlet and outlets of equipment.

Figure 9, illustrates sensitivity of plant components to energy destruction, as it can be seen that, the condenser is the most sensitive component that ambient temperature affects its exergy destruction. Afterwards, the heat exchanger, pump, and turbine are less sensitive respectively.

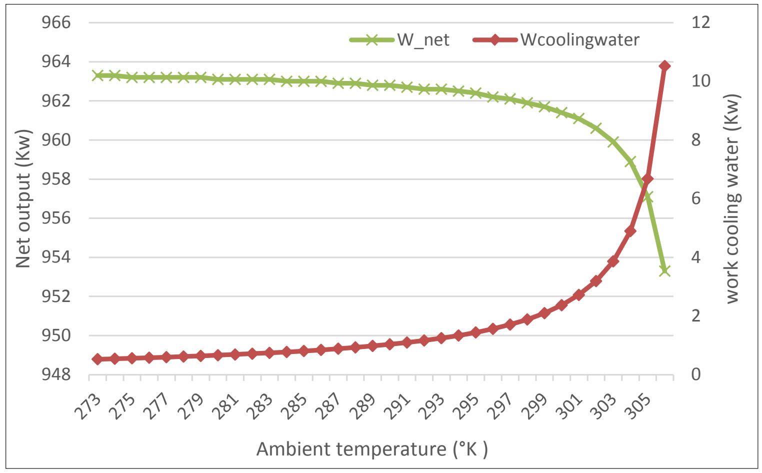


**Figure 6.** Cooling water pump consumption and net output in the different ambient temperature.

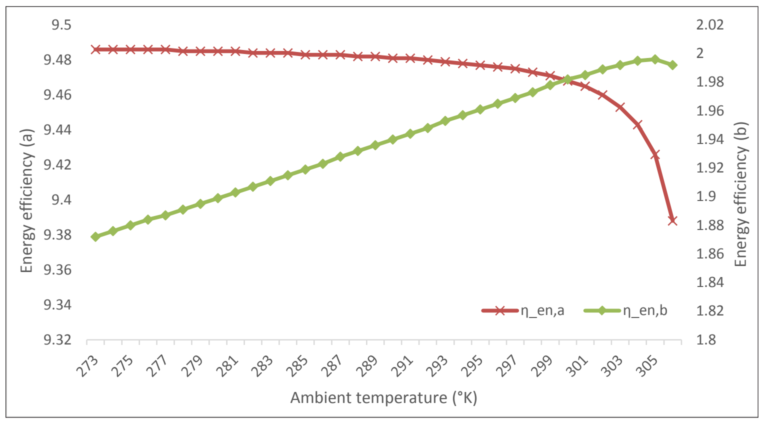


**Figure 7.** Effect of ambient temperature on energy efficiency in scenario (a) and (b) using equations (4) and (5) respectively.

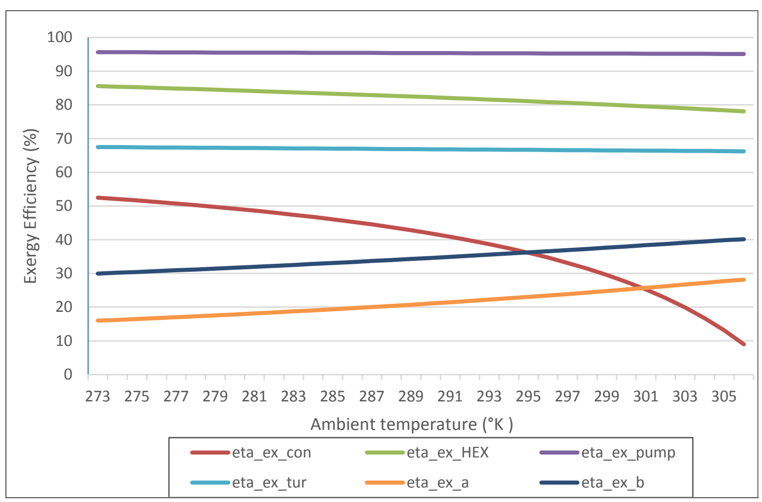


**Figure 8.** Exergy efficiency for system and its components in various ambient temperatures.

## 4. Conclusions

Geothermal ORC power plant with Isopentane as a working fluid was studied based on exergy and energy analysis. Several assumptions were made and the effects of different ambient conditions were investigated.

Obviously, pump consumption for cooling water increased by increasing ambient temperature and caused the net output of the system to decrease. Energy efficiency in scenario (a) decreased from 9.49% to 9.39% by increasing ambient temperature due to net power output decrement, and in scenario (b), increasing dead state enthalpy ($h_0$) as result of ambient temperature increment, caused energy efficiency increased from 1.87% to 2%.

Total available exergy was calculated to be 4,729 kW and exergy efficiency of ORC cycle in two different scenarios were estimated to be 20.36% and 33.98% considering parasitic loads.

The exergy efficiency for both scenarios increased from 15.99% to 28.15% and 29.96% to 40.11% in scenarios (a) and (b) respectively. Furthermore, for the heat exchanger and condenser, these values decreased. In condenser, values changed in a relatively big range from 52.5 to 8.99%. For the pump, the reduction was very limited compared with the other components. It changed from 67.5% to 66.24% for the turbine, 85.56% to 87.1% for the heat exchanger and finally, for the pump, from 95.64% to 95.11%. Net power produced by plant and energy efficiency of ORC cycle are estimated as 962.9 kW and 9.482%, respectively.

The following conclusions can be considered from this study:

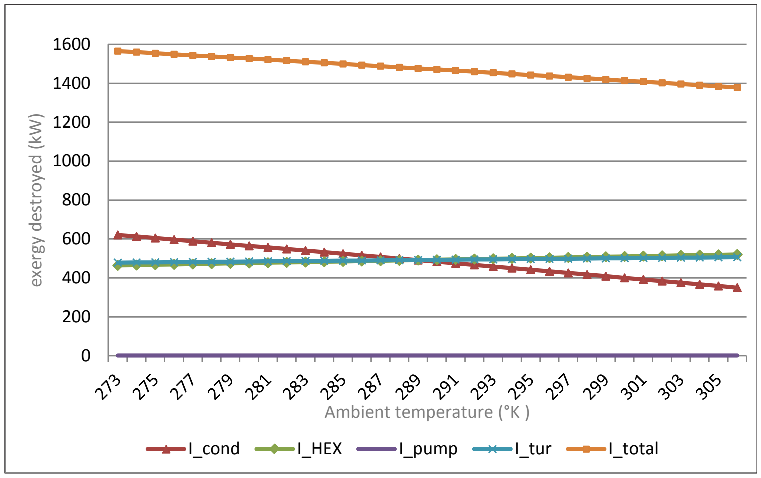


**Figure 9.** Variation of exergy destruction in the cycle and all components.

- The effect of ambient temperature on energy efficiency depends on definition of efficiency. The results indicated that considering two different energy input flow to the system showed different results.
- Energy efficiency fluctuation in two scenarios, confirms that the effect of ambient conditions on dead state enthalpy ($h_0$) is higher than that of increasing cooling water pump consumption.
- The second law of thermodynamic proved that condensers are the most sensitive components to dead state.
- The results showed that there is considerable variation in exergy efficiencies (70% and 34% for scenarios (a) and (b), respectively) in the range of 34°C. It can be concluded that the ambient temperature plays an important role in ORC cycles performance and needs more attention on designing of these cycles.

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