

Designing a Sustainable Organic Rankine Cycle for Remote Geothermal Heat Sources in Pakistan [†]

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Abstract

This paper discusses a thorough analysis, as well as the design, of an environmentally friendly, single-stage Organic Rankine Cycle (ORC) system, particularly optimized for untapped geothermal applications in Pakistan that are secluded and off-grid, to tackle the severe energy crises choking this country and its resources. Keeping in mind its Global Warming Potential (GWP), as well as its performance in the ORC, R600a was chosen as the operating fluid. This study focuses on varying the temperature, pressure, and mass flow rate of not only the geothermal reservoir but that of the operating fluid in the ORC as well. The impacts of adjusting these parameters on the net power output, cycle efficiency, and component-wise exergy destruction, as well as the total exergy destruction, are examined extensively. Analyses of the component-wise exergy destruction found that the maximum exergy destruction occurred in the evaporator, whereas it was discovered that decreasing the condenser pressure below 350 kPa led to negative exergy destruction values, although the total exergy destruction remained positive.

Keywords: Organic Rankine Cycle (ORC); geothermal energy; exergy destruction; Global Warming Potential (GWP); thermodynamic analysis; remote and off-grid power generation



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1. Introduction

There is ever-growing demand for sustainable energy solutions, not only for rural areas, many of which are deprived of electricity, but also for urban areas in Pakistan. In addition, over-reliance on thermal energy puts enormous pressure on the country's economy, limiting its progress. However, one promising solution is geothermal energy, an abundant yet untapped resource in Pakistan, offering an encouraging future for sustainable and off-grid power production for this country. The Organic Rankine Cycle (ORC) is particularly suitable in using waste thermal heat from these geothermal sources for effective power production, thus tackling energy shortages and promoting rural growth [1–6].

One recent study warned about energy deficiencies in Pakistan and how these deficiencies force the country to rely on importing huge amounts of hydrocarbon products to satisfy its energy needs, putting enormous pressure on its economy and limiting its growth, despite it being gifted with abundant energy resources. It focused on investigating the characteristics of these geothermal sources comprehensively [1]. Another research work investigated the electricity access issues faced in rural areas as well as in urban areas and how these issues negatively impact the country's economy. It highlighted the problems faced by

both areas and underlines the significance of innovative and sustainable energy solutions to the future of Pakistan [5]. In March 2024, it was reported that thermal energy constitutes 59.4% of Pakistan's total power generation [6]. The problem with thermal energy is the import and use of fossil fuels, which are neither economical nor sustainable. For example, Pakistan must import one-third of its energy resources, putting enormous pressure on its economy [3]. By the end of November 2023, it was reported that the power section's circular debt had reached a troublesome Rs 5.7 trillion [2]. Moreover, it was also reported in March 2024 that hydroelectric power constituted 25.4% of the total energy generation in the country [6]. However, the problem with hydroelectric power is that the storage capacity of Pakistan's hydel dams is decreasing at a rapid rate. For instance, the storage capacity of Tarbela dam has been reduced by 40% since 1976 because of sedimentation, decreasing at an alarming rate of 0.94 percent per year in its total reservoir capacity, and this trend is expected to continue in upcoming years [7]. This means that hydroelectricity alone is not sufficient as a source of sustainable energy. Thus, the exploration of sustainable and economically friendly alternatives is crucial. Alternatives such as solar energy, low-grade industrial waste heat, and geothermal energy have the potential to be the sources of heat for the ORC [8,9]. However, the issue with solar energy is that it is dependent on sunlight, which is not available at night. Additionally, it depends on the weather conditions, whereas industrial low-grade heat is limited to only industrial areas. In comparison, many sources back geothermal energy as a sustainable alternative for solving Pakistan's energy crises [2,3]. Geothermal energy stands out, as it is available in vast amounts and is available in the remote areas of Pakistan [1], making it a suitable heat source for the ORC. Studies on the ORC's overall performance using various operating fluids have shed light on the best working fluids for the ORC [4].

This research advances these works by designing and comprehensively analyzing a sustainable ORC, particularly suited to remote and off-grid areas of Pakistan with untapped geothermal potential. It offers an in-depth parametric analysis examining the effects of geothermal parameters and the parameters of the ORC, including the temperature, mass flow rates, and pressure conditions, on the net power, cycle efficiency, and both the component-specific and overall exergy destruction. This way, it provides a pathway to analyzing and optimizing the geothermal ORC by balancing the cycle efficiency, net power, and exergy destruction. Additionally, it provides a solution to the energy crises of Pakistan and, in turn, its economic crises as well. It also advances this field by offering an alternative method for power generation. Furthermore, by selecting r600a as the operating fluid due to its low GWP, this paper offers a well-rounded solution for green energy production. This work's novel integration of the ORC parameters and exergy analyses not only offers an efficient system for sustainable power optimizations, particularly in situations where resources are limited, but also discovers atypical thermodynamic behaviors, such as decreasing the condenser pressure below a certain limit resulting in negative exergy destruction.

2. Methodology

2.1. The System Overview

The ORC system under analysis, as shown in Figure 1, constitutes four main parts: a pump, an evaporator, a turbine, and, lastly, a condenser. The condenser is designed to exchange heat directly with the atmosphere. In the evaporator, a counter flow heat exchanger is used with an effectiveness of 0.8 [10]. Heated water from the geothermal reservoir is used as the source of heat to the evaporator, with the inlet temperatures analyzed ranging from 60 °C to 120 °C based on geothermal data [1]. The turbine's efficiency is assumed to be 0.85, whereas the pump's is assumed to be 0.8 [11–13]. Initially, three working fluids were considered, namely r600a, r245fa, and r141b. However, r600a was

chosen since it was nearly as efficient as its counterparts and had a far lower GWP [4]. Table 1 below shows the properties of these three fluids [14].

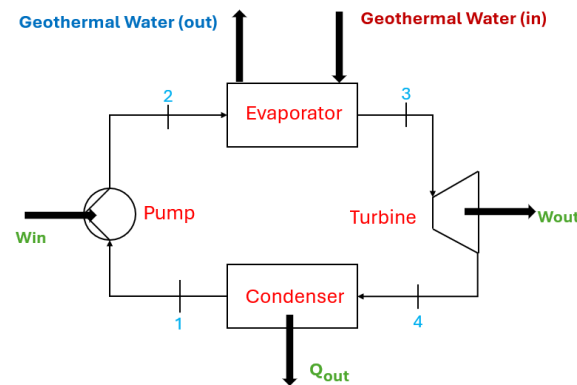


Figure 1. A schematic of a standard ORC power plant.

Table 1. Working fluid properties.

Working Fluid	Molar Mass (kg/kmol)	T_{cr} (°C)	P_{cr} (MPa)	GWP
r245fa	134.05	154.01	3.65	1030
r141b	116.95	204.35	4.21	600
r600a	58.122	134.66	3.62	3

2.2. Assumptions and Modeling

Steady-state operation and negligible heat losses in the piping were assumed in the model. Additionally, it was assumed that there were no pressure losses in the condenser or the evaporator. The parameters that were varied included the mass flow rate (3 kg/s to 7.5 kg/s), temperature (60 °C to 120 °C), and pressure (70 kPa to 340 kPa) of the hot water (geothermal water) entering the evaporator; the ORC fluid's mass flow rate (0.2 kg/s to 2 kg/s); and the pressures at the evaporator (500 kPa to 950 kPa) and condenser (350 kPa to 485 kPa) inlets. The thermodynamic model incorporates mass and energy balance equations with an exergy analysis to evaluate performance benchmarks such as the effective power output and cycle efficiency, as well as the exergy destruction of each component, i.e., the pump, evaporator, turbine, and condenser.

2.3. Mathematical Modeling

The constant entropy efficiency of the pump (η_{pump}) was assumed to be 0.8, and that of the turbine was ($\eta_{turbine}$) assumed to be 0.85 [10]. The isentropic efficiency of the pump is $\eta_{pump} = w_{pump, ideal} / w_{pump, actual}$, where the actual and ideal work done by the pump are given by

$$w_{pump, actual} = (h_2 - h_1) \quad (1)$$

$$w_{pump, ideal} = (h_{2s} - h_1) = v(P_2 - P_1) \quad (2)$$

In the above equations, h_1 and h_2 represent the specific enthalpies at the inlet and outlet of the pump, while h_{2s} represents the isentropic enthalpy at the outlet of the pump. P represents the pressure, and v is the specific volume.

While the isentropic efficiency of the turbine is, $\eta_{turbine} = w_{turbine, actual} / w_{turbine, ideal}$, where

$$w_{turbine, actual} = (h_3 - h_4) \quad (3)$$

$$w_{turbine, ideal} = (h_3 - h_{4s}) \quad (4)$$

In the above two equations, h_3 and h_4 represent the specific enthalpies at the inlet and outlet of the turbine, while h_{4s} represents the isentropic enthalpy at the outlet of the turbine.

Using the formula for the heat exchanger's effectiveness (ε), $\varepsilon = \dot{Q}_{\text{actual}} / \dot{Q}_{\text{max}}$, we first assess the actual heat transfer rate in order to determine the rate of heat gained by the working fluid. The heat transfer rate is found using $\dot{Q} = \dot{m} c_p \Delta T$, where $\dot{m}$ is the mass flow rate, c_p represents the isobaric specific heat, and ΔT represents the difference in temperature.

The efficiency of the cycle is found using $\eta = \dot{W}_{\text{net}} / \dot{Q}_{\text{in}}$, where the net power $\dot{W}_{\text{net}} = \dot{W}_{\text{turbine}} - \dot{W}_{\text{pump}}$ and $\dot{Q}_{\text{in}}$ is the rate of heat gained by the operating fluid in the evaporator. The rate of exergy destruction is given by

$$\dot{X}_{\text{dest}} = T_0 \dot{S}_{\text{gen}} \quad (5)$$

where the rate T_0 is the ambient environment's temperature, and $\dot{S}_{\text{gen}}$ is the rate of entropy generation.

$$\dot{S}_{\text{gen}} = \sum \dot{m} s_{\text{out}} - \sum \dot{m} s_{\text{in}} + \frac{\dot{Q}}{T_0} \quad (6)$$

The entropy generation rate $\dot{S}_{\text{gen}}$ can be calculated utilizing the above equation, where $\dot{m}$ represents the mass flow rate of the ORC fluid; s_{out} and s_{in} are the specific entropies at the outlet and inlet, respectively; and $\dot{Q}$ represents the rate of heat transfer.

These equations underpin the simulation of the ORC cycle under varying operational conditions.

3. Results and Discussion

The subsequent sections discuss the findings obtained with respect to the net power, efficiency, and exergy destruction based on the geothermal parameters and ORC parameters.

3.1. The Geothermal Parameters

The temperature, pressure, and mass flow rate of geothermal reservoirs can vary depending on their location and depth. Therefore, we studied the effect of these variables on our performance indicators.

As shown in Figures 2 and 3, the rate of mass flow and pressure of the hot water which came from the geothermal energy source exhibited a negligible effect on the performance indicators, indicating that the system's response within the studied range was comparatively unaffected by these parameters. The reason for this behavior is that the effectiveness of the evaporator is fixed at 0.8. Additionally, as the heat capacity of the ORC is lower within the defined range, it is used to calculate $\dot{Q}_{\text{max}}$ for effectiveness. Hence, there is no direct influence of the rate of mass flow of the water.

Similarly, although the pressure increases, the effectiveness and the temperature of the geothermal source are constant, leading to insignificant changes. The efficiency remains at around 6%, while the net power is approximately 11.7 kW, and the total exergy destruction remains at around 156 kW for both cases.

In contrast, raising the hot water inlet temperature caused a sharp rise in the net power output, mainly due to the increased heat input in the evaporator. However, this increase came with the cost of a slight reduction in the cycle efficiency, as higher temperatures intensified the irreversibility, leading to an increase in the exergy destruction, particularly in the evaporator and condenser stages, as shown in Figure 4. Overall, it was observed that as the temperature increased from 60 °C to 120 °C, the net power increased by 23%, while the total exergy destruction increased by 29%.

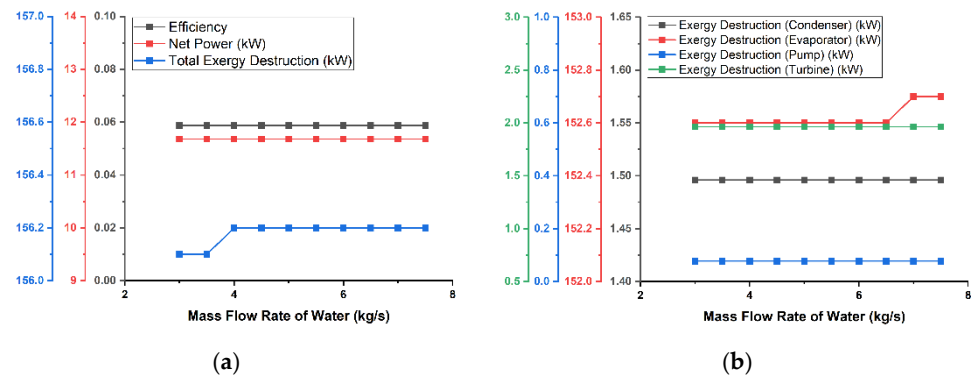


Figure 2. The effect of changing the mass flow rate of water. The variations in the efficiency, net power, and total exergy destruction are given in (a), while (b) represents the exergy destruction in each component, i.e., the condenser, evaporator, pump, and turbine.

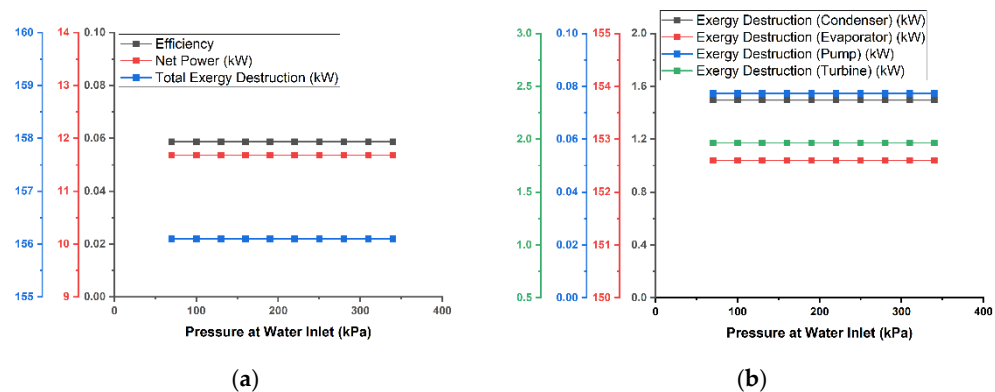


Figure 3. The effect of changing the water pressure. The variations in the efficiency, net power, and total exergy destruction are given in (a), while (b) represents the exergy destruction in each component, i.e., the condenser, evaporator, pump, and turbine.

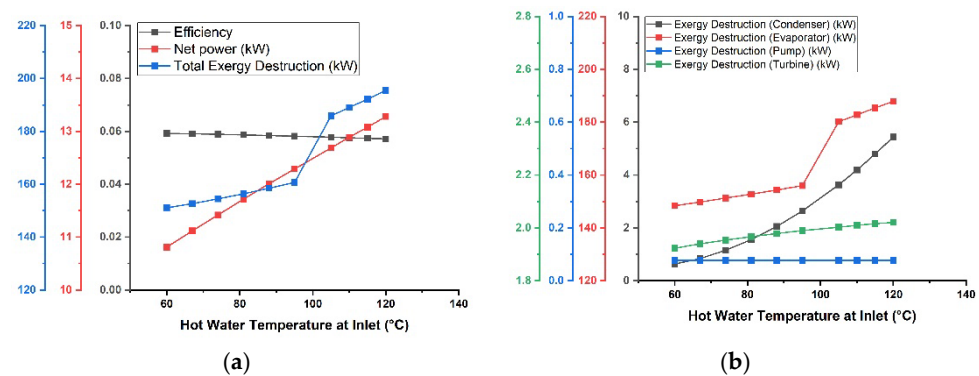


Figure 4. The effect of changing the water temperature. The variations in the efficiency, net power, and total exergy destruction are given in (a), while (b) represents the exergy destruction in each component, i.e., the condenser, evaporator, pump, and turbine.

3.2. Organic Rankine Cycle (ORC) Parameters

Under the assumed conditions, the ORC parameters produced more pronounced effects. Elevating the pressure in the evaporator improved the heat absorption, thereby increasing the effective output power and the cycle efficiency, which increased to a maximum of 0.0828 at the maximum pressure of 950 kPa, as indicated in Figure 5a. While the total exergy destruction decreased significantly from 165 kW to 147 kW as the pressure varied from 500 kPa to 950 kPa, as shown in Figure 5b, this was due to the reduction in exergy destroyed in the evaporator and the condenser, which could be attributed to the

high-quality energy available at the high pressure. The heat input to the operating fluid in the ORC was not affected by an increase in evaporator pressure.

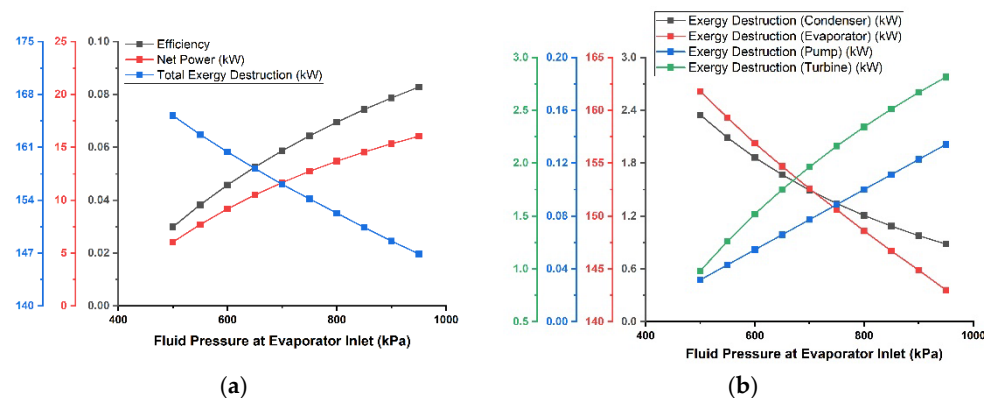


Figure 5. The effect of evaporator pressure. The variations in efficiency, net power, and total exergy destruction are given in (a), while (b) represents the exergy destruction in each component, i.e., the condenser, evaporator, pump, and turbine.

As shown in Figure 6a, a higher condenser pressure led to a decrease in efficiency, as well as net power. However, the total exergy destruction remained relatively constant. The efficiency decreased by almost 43% as the condenser pressure increased from 350 kPa to 485 kPa, while an approximately 50% decrease in the net power was observed. Although the exergy destruction in the evaporator decreased as the condenser pressure increased, as shown in Figure 6b, the exergy destruction in the condenser increased. Therefore, the total exergy destruction remained relatively constant. A detailed investigation of the pressure variations revealed that a pressure lower than 350 kPa leads to negative exergy destruction values in the condenser, indicating the lower limit of the condenser pressure, as negative exergy destruction is not possible. Similarly, the difference in pressure between the turbine's inlet and outlet should be large so that maximum power can be extracted.

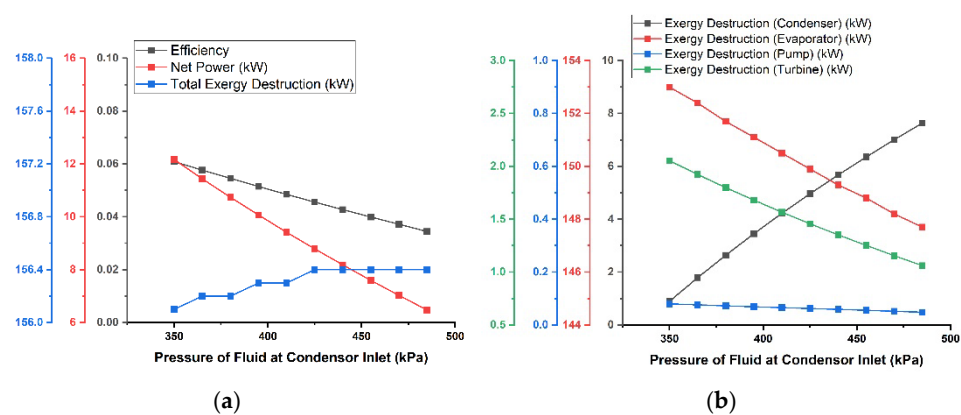


Figure 6. The effect of changing the pressure of the condenser. The variations in efficiency, net power, and total exergy destruction are given in (a), while (b) represents the exergy destruction in each component, i.e., the condenser, evaporator, pump, and turbine.

From Figure 7a, it is observed that the rate of mass flow in the ORC is directly proportional to the effective power and total exergy destruction. Both increase linearly as the mass flow rate increases. The exergy destruction of all of the components increases linearly with an increasing mass flow rate, as shown in Figure 7b, indicating that a high value for the mass flow rate is not suitable.

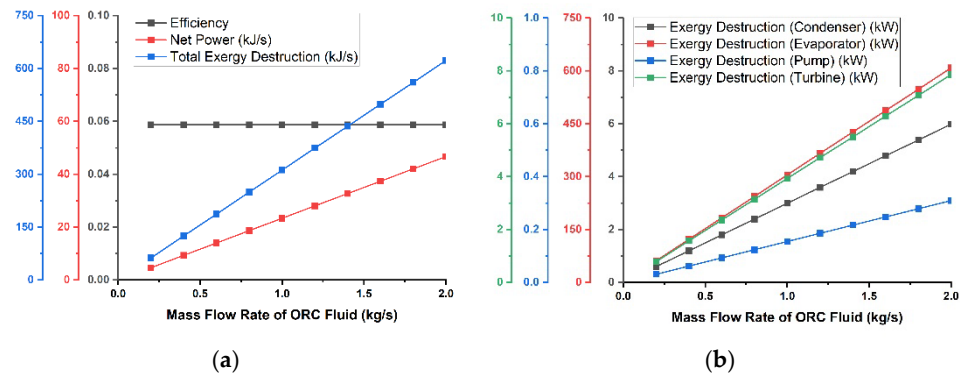


Figure 7. The effect of changing the mass flow rate of the ORC. The variations in efficiency, net power, and total exergy destruction are given in (a), while (b) represents the exergy destruction in each component, i.e., the condenser, evaporator, pump, and turbine.

4. Conclusions

This study analyzed the Organic Rankine Cycle (ORC)'s performance through an investigation of the cycle efficiency, as well as the exergy destruction. Both the geothermal source parameters and the ORC parameters were investigated. The results can be summarized as follows:

Raising the geothermal source's pressure and mass flow rate showed negligible effects on the cycle efficiency and the exergy destruction. However, increasing the temperature of the geothermal water leads to a greater power output but at the expense of slightly lower efficiency due to an increase in exergy destruction. Moreover, the ORC operates between two pressure limits. Raising the evaporator pressure increases the power output and efficiency, while the exergy destruction decreases as the quality of the energy increases. Meanwhile, increasing the condenser pressure has the opposite effect on net power and efficiency. In addition, raising the mass flow rate in the ORC raises the power production as well as the exergy destruction, suggesting a tradeoff is required. In addition, the maximum exergy destruction in all cases was at the evaporator, which was because of the presence of heat transfer. Furthermore, the lower limit of the condenser pressure was found to be 350 kPa, as decreasing the pressure further led to negative exergy destruction, which is impossible. This comprehensive analysis confirms that an effective balance between different parameters is required for an effective performance. These results provide insights into the impact of various variables, on exergy destruction and efficiency, which are crucial parameters for the design of the ORC.

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