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EXERGY ANALYSIS OF A LOW-GRADE GEOTHERMAL SOURCE DRIVEN BY A ORGANIC RANKINE CYCLE (ORC).

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Abstract. A study on geothermal source driven by a Organic Rankine Cycle (ORC) is presented in this paper using conventional exergy analysis methods to provide information about system components interactions. Different operating conditions are performed for system components in order to obtain the magnitude of the destroyed exergy related to the various components of the plant and to determinate which components contribute most to the inefficiency of the plant. For this analysis different operating parameters were evaluated such as: superheating vapor level at the turbine inlet, the pinch temperature and the vaporization pressure, which directly affect the performance of the system. For this analysis 15 different working fluids were studied in a low-temperature geothermal source varying from 90 °C to 140 °C. For each temperature of the geothermal source there is a fluid where the best cycle performances are obtained. The highest net power obtained is 37.4 kW, working with R-152a. A wrong selection of the working fluid can lead to a reduction of more than 40% in power produced. For all working fluids the destroyed exergy has a similar behavior in the system. It is presented as the major responsible for the exergy destruction rate (approximately 80% of the total exergy destroyed) the evaporator, followed by the condenser, the turbine and finally the pump.

Keywords: *exergy, Organic Rankine Cycle, exergy destruction rate, working fluids*

1. INTRODUCTION

Brazil has geological resources and heat gradients above the world average in some specific places of some states such as: São Paulo, Bahia, Espírito Santo, and Rio Grande do Norte among others (Hamza et al. 2010). Although there is a geothermal study of the anomalous regions in Brazil (Hamza, Gomes, and Ferreira 2005), there is still no study directed to the production of electric power from these heat sources.

One of the main problems regarding the use of geothermal energy sources is that medium and high temperature sources are available in few places of the planet, and it is necessary to develop technologies to take advantage of low temperature sources. Among the available technologies are the Kalina Cycle and the ORC.

The Kalina Cycle and the ORC are technologies currently used to produce useful work from low temperature heat sources. These cycles are characterized by using work fluids or mixtures of fluids that have low vaporization temperatures, which makes them suitable for the use of low temperature heat sources (Chandrasekharam and Bundschuh 2008; Lolos and Rogdakis 2009; Saleh et al. 2007).

The existing unconventional Rankine cycles use different working fluids as well as different configurations. As the geothermal fluid is not in contact with the moving parts of the plant, the effects of erosion on the main components of the plant are minimized. They also have the advantage of being able to operate with geothermal fluids at temperatures lower than 150 °C and with a high content of dissolved or even corrosive gases (R. Dippipo 2007).

The Organic Rankine Cycle (ORC) has few differences from the conventional Rankine cycle. The main difference, besides the working pressures and temperatures, is that it uses an organic fluid as a working fluid (Chen, Goswami, and Stefanakos 2010; Guo, Wang, and Zhang 2011).

The selection of different working fluids allows to evaluate one of the most important properties of organic fluids, which is related to the saturation curve in the T-s diagram, which can be positive, negative or vertical. According to this characteristic, the fluids can be called: wet, dry and isentropic, respectively (Liu, Chien, and Wang 2004). Humid fluids, such as water, need to be superheated at the turbine inlet so that, after expansion, they do not fall into the mixing zone (with titers less than 85%), and thus prevent damage to the blades of this equipment. Isentropic and dry fluids do not need to be superheated. Another advantage is that the turbine in small capacity organic Rankine cycles requires a simple stage expander, resulting in a simpler and economical equipment in terms of costs and maintenance (Chen, Goswami, and Stefanakos 2010).

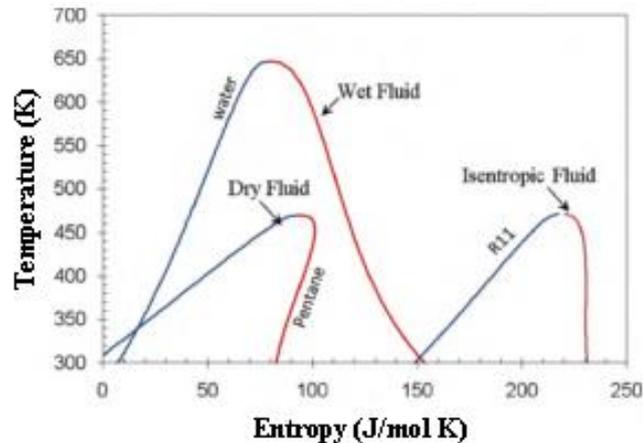


Figure 1: Three types of working fluids: dry, isentropic, and wet. (Chen, Goswami, and Stefanakos 2010)

According to (Campos Rodríguez et al. 2013; Lakew and Bolland 2010) for low temperature sources, where the amount of heat absorbed by the working fluid is critical to a good utilization of the geothermal source, the design of this heat exchanger is of vital importance. Three very important parameters are considered in this work for the analysis of the evaporator: effect of the superheat vapor at the turbine inlet, the pinch temperature differential and the vaporization pressure, which directly affect the performance of the system. This operations parameters are evaluated by means of 15 working fluids for different geothermal source temperatures.

To account for the variable quality of different disordered energy forms in the analysis of thermal and chemical plants, a universal standard of quality is used. The most natural and convenient standard is the maximum work which can be obtained from a given form of energy using the environmental parameters as the reference state. This standard of energy quality is called exergy (T. J. Kotas 2012).

One of the main uses of this concept is in an exergy balance in the analysis of thermal systems. The exergy balance is similar to an energy balance but has the fundamental different that, while the energy balance is a statement of the law of conservation of energy, the exergy balance may be looked upon as a statement of the law of degradation of energy. Degradation of energy is equivalent to the irretrievable loss of exergy do to all real processes being irreversible (T. J. Kotas 2012).

Geothermal fluid temperatures are relatively low, so the First Law efficiencies of geothermal plants are also inherently low. Consequently, the differences between the First Law efficiencies of a good performance and a low performance geothermal plant located in similar locations is small. For this reason it becomes difficult to make a comparison only on the basis of the First Law (DiPippo 2004). This situation become more importance in binary power generation systems, since its working temperature is lower than that of simple flash and dual flash systems.

Any destruction of exergy causes a decrease in the useful work of the process or an increase in the consumption of its means of production. The potential for improvement of a component of the system is determined by the relationship between the irreversibilities generated by the system and the minimum generation of possible irreversibility, within the limits imposed by the physical, technological, and economic constraints.

The destroyed exergy is a parameter from which a plant and its components can be compared on a rational basis. By comparing the magnitude of the destroyed exergy to various components of a plant, it is possible to determinate which components of the system contribute most to the inefficiency of the plant.

The conventional exergy analysis plays an important role in the exergy determination in the mentioned researches. Conventional exergy claims that, the most important component is the one with the highest exergy destruction and suggests more attention to the mentioned component for system performance improvement (Nami, Nemati, and Fard 2017).

Thus, the objective of this study is to evaluate and compare the ORC system with respect to different types of work fluids, geothermal source temperature, Pinch Temperature Differential and vaporization pressure from a component exergy destruction analysis.

2. SYSTEM DESCRIPTION AND ASSUMPTIONS

The evaluated system is an ORC system, composed of its four main components: pump, evaporator, turbine and condenser, as shown in Figure 2a. The working fluid is a saturated liquid at the outlet of the condenser (6), then it is compressed (6-9), and goes to the vapor generator, where it gains energy from the geothermal source until it becomes vapor (9-4), the evaporator is divided into two components in order to determine the Pinch Temperature Differential. Finally it is expanded in the turbine to generate useful work (4-5). The representation of the T-s diagram of the thermal system is shown in Figure 2b (Rodríguez 2014).

The ORC is modeled using commercial software Aspen-HYSYS v. 7.0. For the calculation of the thermodynamic properties of the working fluids were used Peng Robinson-Stryjek Vera cubic state equations (PRSV) (Proust and Vera 1989) which is a modification of the Peng-Robinson equations (PR) and extends to original PR method. This method is designed to match the pressure curves of pure components and mixtures, especially at low steam pressures.

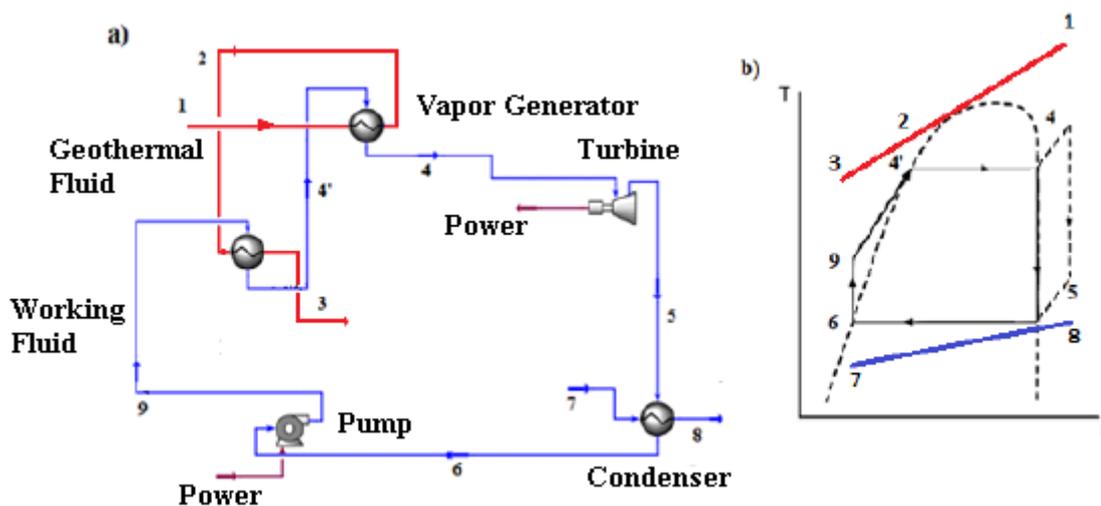


Figure 2: a) ORC cycle scheme, b) T-s diagram of the cycle (Rodríguez 2014).

To simplify the model, component pressure drops, with the exception of the turbine, are ignored. For each individual component of the thermal system, the First and Second Laws of Thermodynamics are applied to determine the work done or consumed, the heat added or rejected, exergy destruction of each components, as well as the thermal efficiencies and exergetic. In this analysis geothermal temperature source is considered between 90 °C to 140 °C, 15 different working fluids were studied, the pinch point temperature from 3 °C and 6 °C are considered and different levels of superheated vapor at the turbine inlet, as well as different working pressure. Other important considerations were taken into account in this model as follow:

- A geothermal mass flow rate of 1 kg / s was used as the calculation basis for the analysis, since the efficiency of the system is dependent only on the design points, and the power generated is directly proportional to the geothermal fluid flow.
- The condensation temperature, lower cycle temperature, is set at 37 °C.
- In case of expansion to the mixing zone, titles greater than 85% are accepted.
- The system operates at steady state, kinetic and potential energy changes are neglected and heat and friction losses are neglected.
- The isentropic efficiency of the pump and turbine are 80% and 85%, respectively.

2.1 Thermodynamic analysis

Considering each component as control volume undergoing a steady-state process the mass and energy conservation equations as long as exergy balance equations are applied to the system components as follow:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

$$\dot{Q} + \sum \dot{m}_{in} h_{in} = \dot{W} + \sum \dot{m}_{out} h_{out} \quad (2)$$

$$\dot{E}_Q + \sum \dot{m}_{in} e_{in} = \dot{W} + \sum \dot{m}_{out} e_{out} + \dot{E}_D \quad (3)$$

Where

$$\dot{E}_Q = \sum \dot{Q} \left(1 - \frac{T_0}{T} \right) \quad (4)$$

In these equations, subscripts *in* and *out* symbolize the component inlet and outlet, respectively. Also, \dot{Q} is the rate of heat transfer, \dot{W} is the exergy rate of mechanical power, \dot{E}_Q is the exergy rate associated with the heat transfer, and \dot{E}_D is the exergy destruction rate. Moreover, T_0 is the ambient temperature, \dot{m} is the mass rate, h is the specific enthalpy and e the specific exergy.

When it is assumed that the specific kinetic exergy and the specific potential exergy are zero, considering a system in equilibrium with the reference state and considering a pure substance in which the chemical portion of exergy is negligible because there are no chemical reactions and, therefore its chemical composition is not altered in the process, (Santos 2005) the calculation of the specific exergy (e) of a flow can be obtained from Eq. (5).

$$e = (h - h_0) - T_0(s - s_0) \quad (5)$$

Properties of the dead state (T_0 and P_0) are considered when the fluid is at atmospheric pressure and the temperature of the environment (25°C).

The exergy of the product generated by the plant is the net produced power (\dot{W}_{net}) and the exergy of the resource is the exergy of the heat flow rate contained in the geothermal source (\dot{E}_{in}), then, the exergetic efficiency of the ORC (\mathcal{E}), can be defined as follows:

$$\mathcal{E} = \frac{\dot{W}_{net}}{\dot{E}_{in}} \quad (6)$$

The exergy balance equations for each component of the ORC are outlined in Table 1.

Table 1: Exergy balance equations for the different components of the ORC.

Components	Exergy Destruction
Pump	$\dot{W}_P + \dot{E}_6 - \dot{E}_9$
Turbine	$\dot{E}_4 - \dot{E}_5 - \dot{W}_T$
Evaporator	$\dot{E}_1 + \dot{E}_9 - \dot{E}_3 - \dot{E}_4$
Condenser	$\dot{E}_5 - \dot{E}_6 - \dot{E}_8$

2.2 Results and Discussions

Table 2 show some thermodynamics properties and mass flow rate for a specific case analysis, which was determined considering 140 °C of the geothermal source temperature, 1824 kPa of evaporation pressure, 0 °C of superheated vapor at the turbine inlet, 500 Kpa of condensation pressure of the cycle and pinch point 3 °C, using I-Butane as a working fluid.

It is important to remark that the figures below were elaborated with the operation parameters of the system that offer the production of higher net power and efficiency of the thermal system. Several levels of superheated vapor at the turbine inlet were evaluated, different operating pressures and pinch point for each working fluid and geothermal source temperature where evaluated in this study.

Table 2: Thermodynamic properties and mass flow rates for a specific case.

Point	Fluid	mass (m) (kg/s)	Enthalpy (h) kJ/kg	Entropy (s) kJ/kg K	Exergy (E) kW
1	Geothermal Fluid	1	-15381.92	4.411629	73.74
3	Geothermal Fluid	1	-15688.85	3.593215	10.70
9	I-Butane	0.80	-2624.103	1.560055	2.29
4	I-Butane	0.80	-2239.092	2.642031	52.18
5	I-Butane	0.80	-2282.202	2.665562	12.23
6	I-Butane	0.80	-2627.203	1.557656	0.39
7	water	6.81	-15853.34	3.077134	0.00
8	water	6.81	-15812.93	3.21045	4.64

$\dot{W}_T = 34.36kW$; $\dot{W}_p = 2.47kW$; $\eta = 6.77$; $\varepsilon = 43.3$

According to Figure 3 for a given temperature of the heat source there is a working fluid where the best performance of the system is obtained. For the cases of 90 °C and 100 °C of the geothermal source, the net power produced using the different working fluids are very similar, due to the low energy potential of the heat source, prevailing a slight improvement in the generation with the use of R- 114, which produces 7.45 kW and R-134a, which produces 11 kW at 90 °C and 100 °C of the geothermal source respectively. For 110 °C the highest power obtained is 17.6 kW working with R-290. For the cases of 120 °C and 130 °C, the highest net power are obtained with the use of R-134a, generating 26.5kW and 33kW respectively. And for the case of 140 °C of the geothermal source the highest net power obtained is 37.4 kW, working with R-152a.

It can be concluded that for each temperature of the geothermal source there is a fluid where the best cycle performances are obtained. A wrong selection of the working fluid can lead to a reduction of more than 40% in power produced.

Figure 4 shows the mass flow rate of evaporated working fluid for each organic fluid and temperature of the heat source. It can be observed that there is a great difference in the flow rate values for each working fluid. Since the flow is directly related to the system costs, a variation of the operations parameters can cause a decrease in the efficiency, power produced and the heat exchange areas and the system can improve the energy production costs.

The efficiency calculated from the second principle of thermodynamics offers a more realistic analysis of the thermal system, with efficiency around 50 % for a geothermal source temperatures of 140 °C, which places these cycles at the same efficiency level as a conventional Rankine cycle, which uses fossil fuels and high operating pressures and temperatures. The best exergetic efficiency reach 59% working with the R-152a at 140 °C of the heat source.

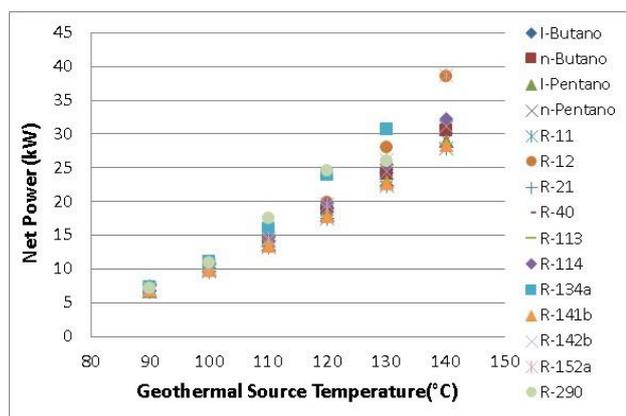


Figure 3: ORC net power produced for each working fluid at different geothermal source temperature.

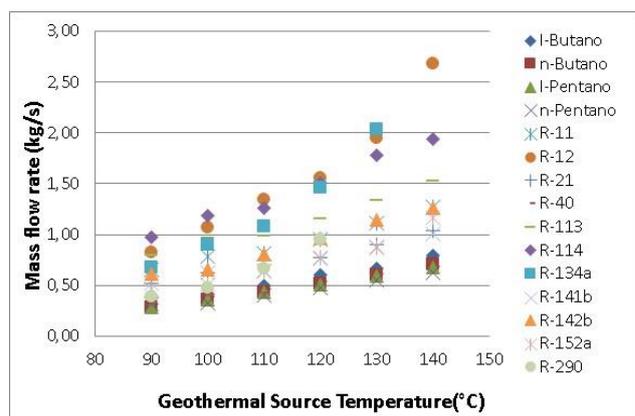


Figure 4: ORC mass flow rate of the working fluids in relation to the geothermal source temperature.

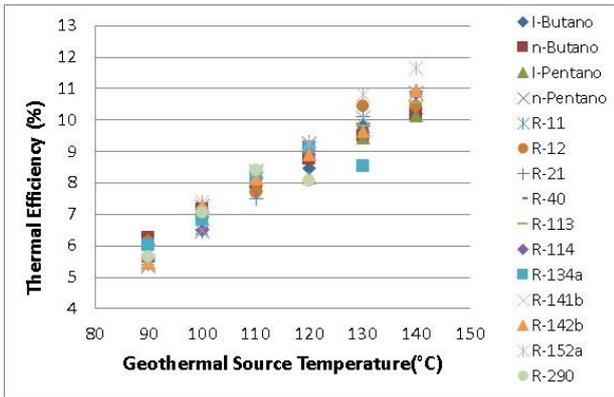


Figure 5: ORC thermal efficiency for each working fluid at different geothermal source temperatures.

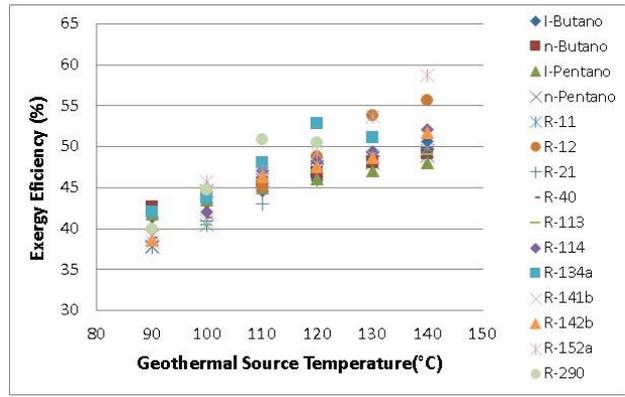


Figure 6: ORC exergetic efficiency for each working fluid at different geothermal source temperature.

The destroyed exergy rate is a parameter from which a plant and its components can be compared on a rational basis. By comparing the magnitude of the destroyed exergy to various components of a plant, it is possible to determine which components contribute most to the inefficiency of the plant. Through the exergetic analysis, the location and the magnitude of the exergy destruction rate of each component of the systems were identified. The ratio of the exergy destruction rate to the total exergy destruction rate for each component of the systems are presented in Figure 7 to Figure 21 using different working fluids, geothermal source temperature and operations parameters.

In general, for all working fluids the destroyed exergy has a similar behavior in the system. It is presented as the major responsible for the exergy destruction rate (approximately 80% of the total exergy destroyed) the evaporator, followed by the condenser, the turbine and finally the pump.

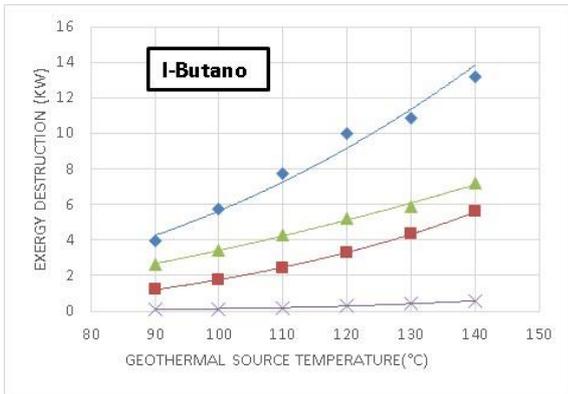


Figure 7: ORC exergy destruction rate of each component with increasing geothermal source temperature.

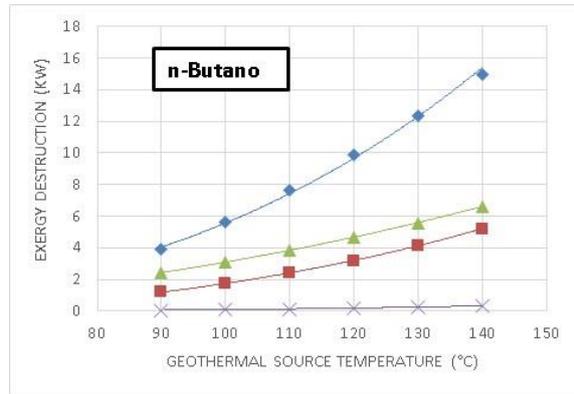


Figure 8: ORC exergy destruction rate of each component with increasing geothermal source temperature.

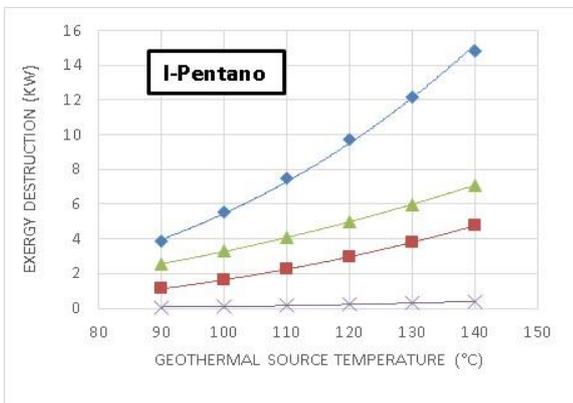


Figure 9: ORC exergy destruction rate of each component with increasing geothermal source temperature.

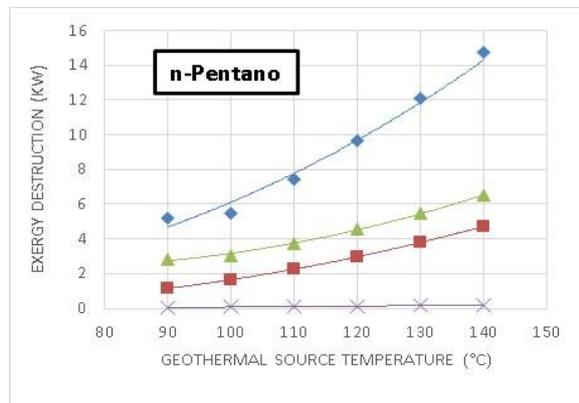


Figure 10: ORC exergy destruction rate of each component with increasing geothermal source temperature.

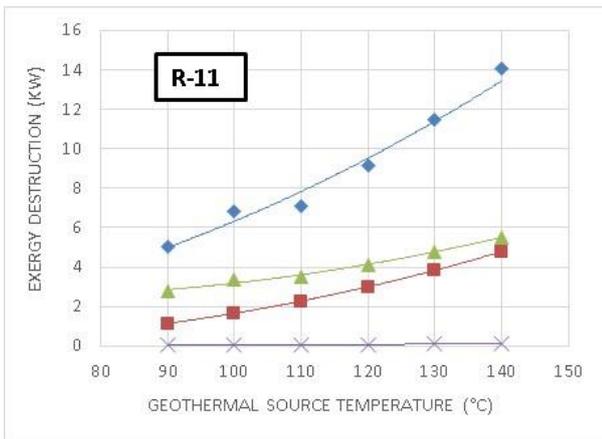


Figure 11: ORC exergy destruction rate of each component with increasing geothermal source temperature.

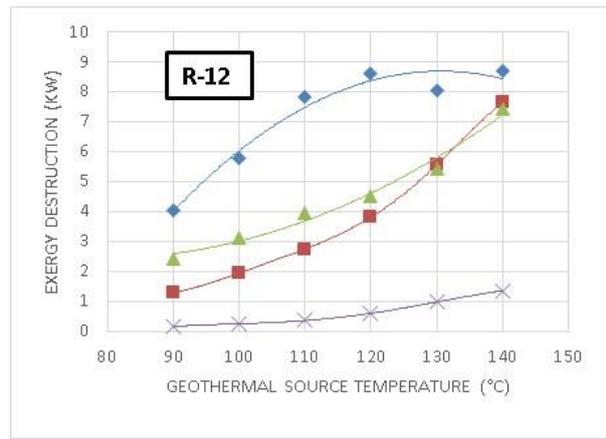


Figure 12: ORC exergy destruction rate of each component with increasing geothermal source temperature.

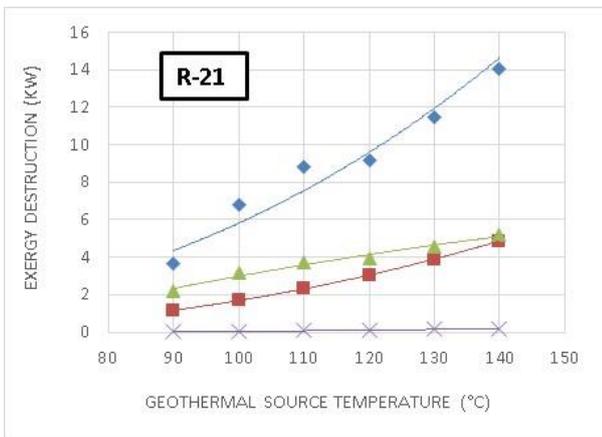


Figure 13: ORC exergy destruction rate of each component with increasing geothermal source temperature.

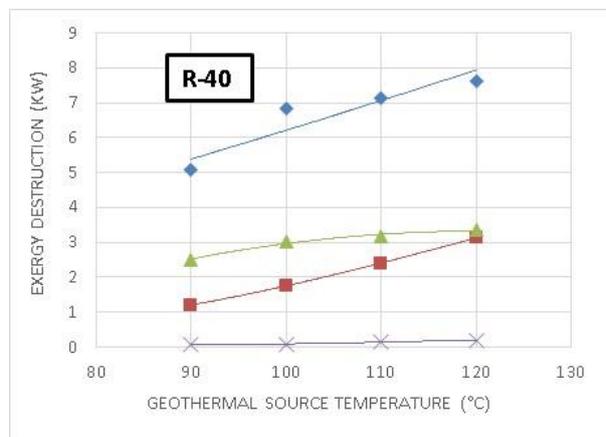


Figure 14: ORC exergy destruction rate of each component with increasing geothermal source temperature.

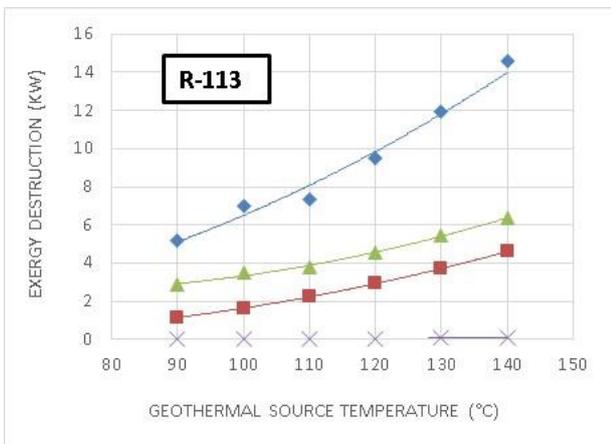


Figure 15: ORC exergy destruction rate of each component with increasing geothermal source temperature.

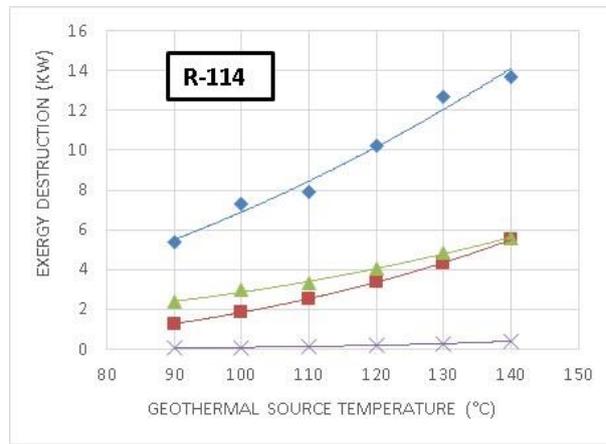


Figure 16: ORC exergy destruction rate of each component with increasing geothermal source temperature.

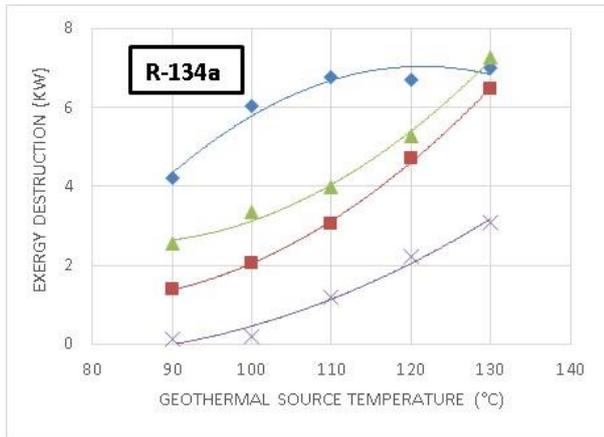


Figure 17: ORC exergy destruction rate of each component with increasing geothermal source temperature.

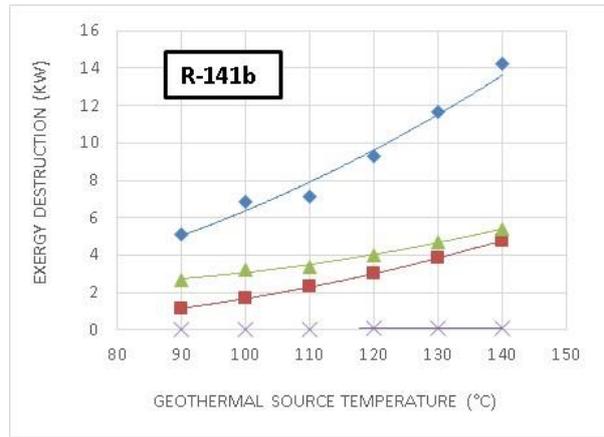


Figure 18: ORC exergy destruction rate of each component with increasing geothermal source temperature.

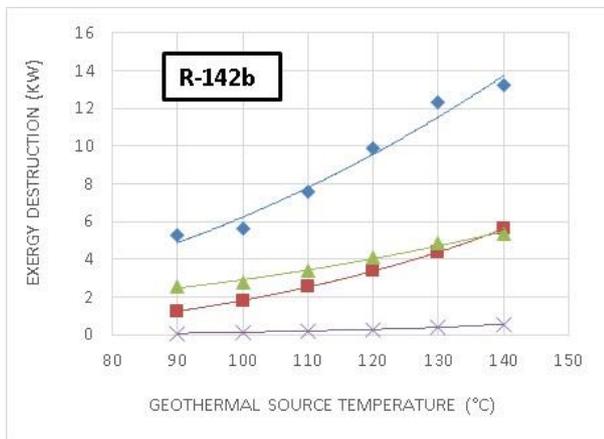


Figure 19: ORC exergy destruction rate of each component with increasing geothermal source temperature.

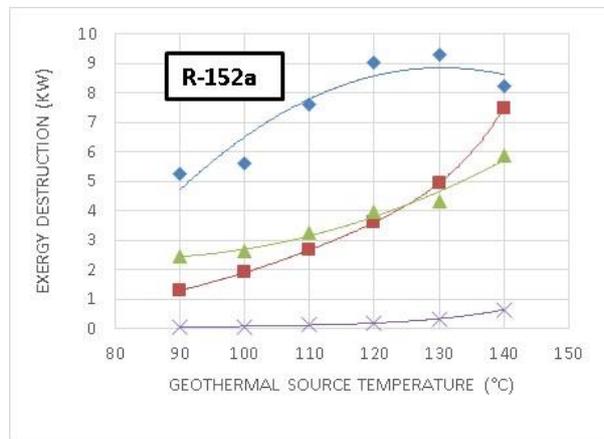


Figure 20: ORC exergy destruction rate of each component with increasing geothermal source temperature.

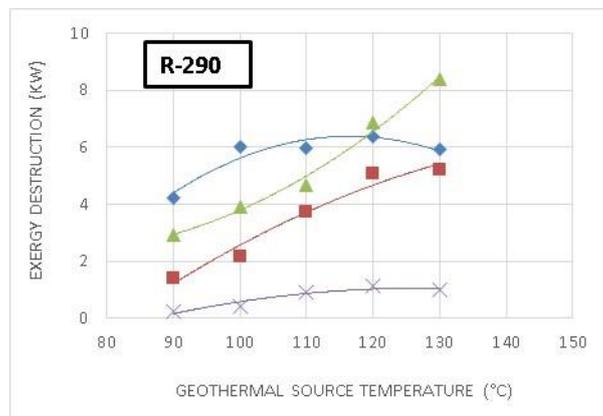


Figure 21: ORC exergy destruction rate of each component with increasing geothermal source temperature.

Legends for Figure 7 to Figure 21 are listed below:

◆ Exergy destruction rate of the evaporator; ▲ exergy destruction rate of the condenser; ■ exergy destruction rate of the turbine and X the exergy destruction rate of the pump.

3. CONCLUSIONS

This paper presents a detailed energy and exergy analysis of different parameters combinations of Organic Rankine Cycles (ORCs) operation. The results show the higher exergy efficiency was 59 %. The biggest part of the system exergy loss is related to evaporator in each study cases.

For low temperatures heat sources where the amount of heat absorbed by the working fluid is critical to a good utilization of the geothermal source, the design of the vapor generator is of vital importance. Three very important parameters are considered in this work for the analysis of this component: the effect of the steam superheat on the turbine inlet, the Pinch Temperature Differential and the Vaporization Pressure, which directly affect the performance of the system and can affect by more than 40% the power generated by the cycle.

A analysis of heat exchanger area is necessary to determinate the relations of high power production and system cost in order to improve the criteria of the selection of the operation parameters.

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