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ELECTRIC POWER GENERATION FROM LOW-TEMPERATURE ENHANCED GEOTHERMAL SYSTEM IN BRAZIL USING ORGANIC RANKINE CYCLE.

Carlos E. Campos Rodríguez
UNIFEI - Universidade Federal de Itajubá
Av. BPS, 1303, bairro Pinheirinho, Itajubá – MG
eymelcampos@hotmail.com

Osvaldo J. Venturini
UNIFEI - Universidade Federal de Itajubá
Av. BPS, 1303, bairro Pinheirinho, Itajubá – MG
osvaldo@unifei.edu.br

Electo E. Silva Lora
UNIFEI - Universidade Federal de Itajubá
Av. BPS, 1303, bairro Pinheirinho, Itajubá – MG
esl43@yahoo.com

José C. Escobar Palacios
UNIFEI - Universidade Federal de Itajubá
Av. BPS, 1303, bairro Pinheirinho, Itajubá – MG
jocescobar@hotmail.com

Vladimir Melián Cobas
UNIFEI - Universidade Federal de Itajubá
Av. BPS, 1303, bairro Pinheirinho, Itajubá – MG
vlad@unifei.edu.br

Daniel Marques dos Santos
AES Tietê; Bauru, São Paulo – Brazil
danielmarques.Santos@aes.com

Fábio R. Lofrano Dotto
FAROL Pesquisa, Desenvolvimento e Consultoria - Brazil
fabio@farolconsultoria.com.br

Vernei Gialluca
Gênera Serviços e Comércio LTDA – Brazil
vernei@generatech.com.br

Abstract. *The Present work aims to study, by mean of the first and second law of the thermodynamic, the performance of low-temperature Enhanced Geothermal Systems (EGS) including efficiency, working fluids, mass flow rate consumption, electric power production and the irreversibilities of the ORC system to establish the thermodynamics conditions of the cycle and the thermo physical conditions of the working fluids to achieve the better performance of the conversion process of the thermal energy into electrical energy. In this work, the ASPEN-HYSYS software was used as a tool to simulate the thermal system and Peng- Robinson Stryek-Vera (PRSV) equations of state (EoS) for the 15 analyzed working fluids*

Keywords: *Thermodynamic analysis, exergy, working fluids, Organic Rankine Cycle, Enhanced Geothermal System*

1. INTRODUCTION

The geothermal resources comprise a wide variety of the earth´s heat sources, including not only the more easily developed, but also the thermal energy stored deep within the earth which is present anywhere. It is estimated that geothermal energy is equivalent to 50,000 times the energy obtained from all oil and gas resources.

Previous studies show that Brazil has geological resources similar to Australia and gradients of heat above the world average in some punctual places. Brazil, has carried out a comprehensive study on geothermal resources, (Gomes & Hamza, 2005), (Gomes & Hamza, 2008), (Hamza et al., 2005), (Hamza, et al., 2010). Here is presented an evaluation of these resources in some localities of Rio de Janeiro, with geothermal gradients found between 14 and 26 °C/km and a bit higher in sedimentary basins (Campos, Resende and Carapebus) between 19 and 33 °C/km. Measurements were

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made in 44 other Brazilian localities finding the maximum gradient, of 35 °C/km, in the Paraná Basin. Recent studies have identified several areas with geothermal gradients that are in the range 30-60 °C/km. These regions include the coastal zone of Santa Catarina, Midwest states of Paraná and São Paulo and southern Mato Grosso.

According to (Lund et al., 2005), until May 2005 the use of geothermal resources in Brazil was limited to the direct use, and had an installed capacity of 360.1 MWt. Of these, 355.9 MWt part is consumed by Petrobras, using the heat of the dry oil wells in Rio Grande do Norte to heat approximately 500 m^3 of natural gas per day, another part is used in thermal baths and swimming pools, the rest (4.2 MWt) in an industrial wood processing and pre-heating water for use in boilers at a facility producing coffee.

According to the Geothermal Technologies Program of the U.S. Department of Energy, reservoirs of steam and hot water are only a small part of the geothermal resource. The magma of the earth and the rocks without hot water will be a source of clean, cheap and almost unlimited geothermal energy as soon as the technology needed to use them will be developed. Meanwhile, regions of moderate temperatures underground are the regions where electricity frequently through the geotermeletric binary cycle plants. Globally the energy stored in the earth's crust up to a depth of 5000 m is estimated to be 140×10^6 EJ, (Bertani, R., 2012).

Enhanced geothermal systems (EGS) use the heat of the Earth, where there is not or there is insufficient steam or hot water and where the permeability is low. The EGS technology is focused on creating large areas of heat exchange in hot rocks. The process involves improving the permeability through the opening of pre-existing fractures and/or creating new fracture (Huengues, E., 2010).

Nowadays there are in the world only a few plants EGS, which are: Soultz in France, that produce 3.8 MWe using 100 kg/s of hot water at approximately 185 °C; Landau Plant in Germany, 2.9 MWe with 80 kg/s of water at 160 °C; Insheim plant in Germany, 4 MWe with 80 kg/s of water at approximately 160 °C and Unterhaching plant in Germany that produce 3 MWe with 80 kg/s of water at 125 °C. The vertical wells depths vary from 3000 to 5000 m for each project.

From (IEA, 2011), the range of costs for EGS is between 100 USD/MWhe (for sources of 300 °C at 4 km deep) and 190 USD/MWhe (for sources of 190 °C at 5 km) in United States, while in Europe are estimated from 250 USD/MWhe to 300 USD/MWhe. In the United States, Australia, China, etc. are also been carried out research on EGS, 20 EGS projects were in development or under discussion in several European Union countries. United States included research on EGS in their recent clean energy initiatives as part of the national geothermal program reinstated. The largest EGS project in the world, a demonstration plant of 25 MWe, is under development in the Cooper Basin of Australia.

For a low-grade geothermal reservoir with a temperature of 150 °C or lower, the common type of power plant to be built is a binary plant, (Quoilin, S., *et al.* 2013), (Dippipo, R., 2007). This kind of plant use as a working fluid an organic fluids (ORC) or ammonia-water mixture (Kalina cycle), both with low boiling points to recover heat from a low-temperature heat source. Until 2010, binary plants were the most widely used type of geothermal power plants with 240 units in operation, generating 1199 MWe, making up to 45% of all geothermal units, but only 11% of the total power (Bertani, R., 2012). See fig. 1.

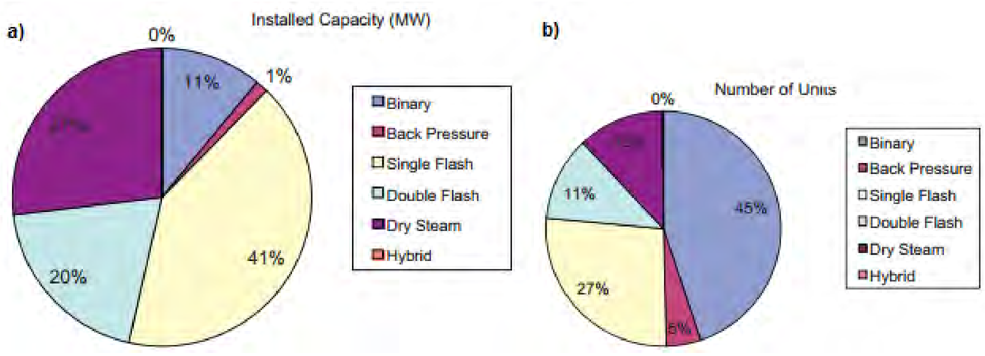

Figure 1. (a) Percentage of installed capacity by power plant type. (b) Percentage of units by power plant type. (Bertani, R., 2012).

In a binary cycle power plant the heat of the geothermal fluid is transferred to a secondary working fluid, usually an organic fluid or a mixture that has a lower boiling point and high vapor pressure compared to water at a given temperature. This type of geothermal plant has no emissions to the atmosphere. Thus, environmental problems that may be associated with the exploitation of higher temperature geothermal resources, like the release of greenhouse gases and the discharge of toxic elements, are avoided. Another advantage of the binary technology is that the geothermal fluids (or brines) do not contact with the moving mechanical components of the plant (like turbine), assuring a longer life for the equipment. Binary plants have allowed the exploitation of a large number of fields that may have been very difficult (or

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uneconomic) when using other energy conversion technologies, thereby increasing significantly the potencial of geothermal resources worldwide, (Rodríguez, C. E. C., *et al.,* 2013).

The selection of the working fluids for ORC system (figure 1) for diferents applications has been well discussed by many authors as (Papadopoulos, A. I., 2010), (Mikielewicz, D., and Mikielewicz, J., 2010), (Mago, P. J., *et al.,* 2008), (Maizza, V., and Maizza, A., 2001), (Tchanche, B. F., *et al*., 2009), (Quoilin, S., et al., 2010), (Heberle, F., and Brüggemann, D., 2010), (Chen, H., *et al.,* 2010), (Hung, T. C., *et al.,* 2010), (Liu, B. T., *et al.,* 2004), (Saleh, B., *et al.,*. 2007). The fluids properties performance, such as: thermal conductivity, the slope of the T-s curve, critical temperature and pressure, molecular mass, density, latent heat, etc. have a direct influence over the cycle.

2. TECNOLOGY DESCRIPTION AND SIMULATION

The scheme of an ORC system using low-grade energy sources is shown in Fig. 2. The system is composed by an evaporator, a turbine expander, a condenser, and a pump. The working fluid passes through the evaporator in which the high-temperature geothermal source is utilized. The steam enters the turbine expander and generates power. The fluid exit from the turbine expander then enters the condenser in which low-temperature cooling water is utilized to condense the fluid. Finally, a pump raises the fluid pressure and feeds the fluid into the evaporator to complete the cycle, (Rodríguez, C. E. C., *et al.,* 2013).

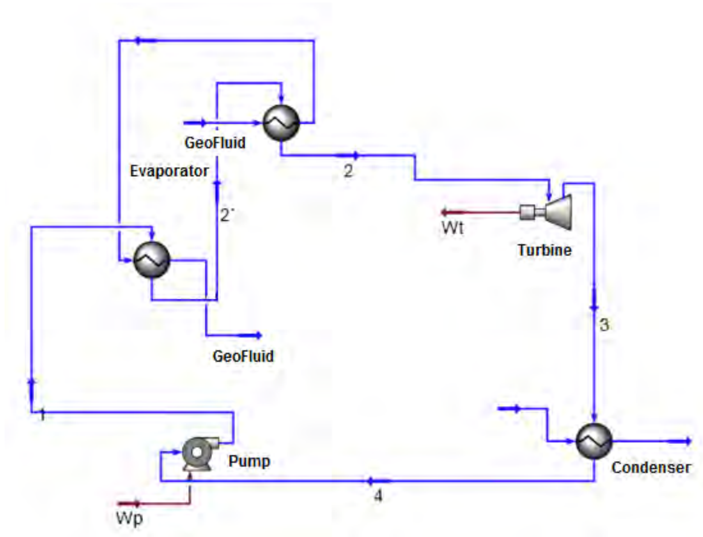

Figure 2. Schematic components of the ORC system for low-enthalpy geothermal source.

The performance of the technology Enhanced Geothermal System is studied in this work under Brazilian conditions, in order to reach a maximum specific work and efficiency. The configuration of the considered ORC is a subcritical cycle with saturated steam at the turbine inlet in order to achieve the maximum power output and efficiency, (Quoilin, S., *et al.* 2013). In this work were considered 15 organic working fluids to compare and to find the better performance of the ORC. For each fluid, main thermodynamic parameters are listed in Table 1, where T_c and P_c represents the critical temperature and pressure respectively, T_b, the boiling temperature, M, the molar mass, ω, the acentric factor and k_1 is a characteristic parameter of a pure component.

Table 1. Thermodynamic parameters of the pure working fluids considered in this study.

Working fluids	T_C (°C)	Pc (kPa)	Tb (°C)	M (Kg/Kmol)	ω	k_1	Type
n-Pentane	196.45	3375	36.06	72.15	0.25389	0.02227	Dry
I-Pentane	187.25	3334	27.88	72.15	0.22222	0.04451	Dry
n-Butane	152.05	3797	-0.5	58.12	0.2010	0.03951	Dry

I-Butane	134.95	3648	-11.73	58.12	0.18479	0.03781	Dry
R134a	100.99	4055	-26.22	102.03	0.3256	0.07076	Isentropic
R141b	116.95	4340	31.99	116.95	0.2211	0.05949	Isentropic
R142b	137.05	4120	-10.01	100.5	0.2360	0.00689	Isentropic
R290	96.66	4242	-42.08	44.10	0.1488	0.19724	Isentropic
R40	143.15	6700	-24.05	50.49	0.1530	0.03040	Wet
R152a	113.89	4444	-25	66.05	0.2557	-0.14590	Isentropic
R-11	198.05	4408	22.9	137.37	0.1910	0.02574	Isentropic
R-12	111.85	4124	-29.75	120.91	0.1760	0.02752	Isentropic
R-113	214.1	3436	47.57	187.39	0.245	-0.02468	Dry
R-114	145.89	3621	3.68	170.92	0.2502	0.05823	Dry
R-21	178.43	5184	8.9	102.92	0.2069	0.03808	Isentropic

The thermodynamic analysis of the considered cycle was carried out by using the commercial software Aspen-HYSYS and the cubic equation of state Peng-Robinson Stryjek-Vera, (Stryjek, R., & Vera, j. H., 1986), (Angelino, G., & Colonna, P. P., 1998), for the calculation of the working fluids thermodynamic properties (1-6). Main used equations follow:

$$\mathrm{P}=\frac{\mathrm{RT}}{\mathrm{v-b}}-\frac{a}{v(v+b)+b(v-b)} \tag{1}$$

$$a=(\alpha)4.55724\frac{R^2T^2}{P_c} \tag{2}$$

$$\alpha=\left[1+k(1-T_r^{0.5})\right]^2 \tag{3}$$

$$b=0.0777896\frac{RT_c}{P_c} \tag{4}$$

$$k=k_0+k_1(1+T_r^{0.5})(0.7-T_r) \tag{5}$$

$$k_0=0.378893+1.489715w-0.1713848w^2+0.0196544w^3 \tag{6}$$

2.1 Assumptions used in the analysis

1. Pressure drop and heat loss in pipe lines are neglected.
2. Ambient temperature, 25 °C.
3. The analysis was carried out for a low-temperature geothermal source (from 90°C to 140 °C).
4. The analyses were carried out for 1 kg/s of the geothermal fluid.
5. The organic fluid at the turbine inlet is saturated vapor.
6. The isentropic efficiency of the turbine is 85%.
7. The pump efficiency is assumed to be 80%.

Evaporator and Turbine

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The evaporator is one of the most important components of the cycle, depending on it the quantity of heat that can be transferred from the geothermal source to the working fluid. That´s why a good design of this heat exchanger can be translated in higher power output of the cycle. Several parameters must be considered for a good design of the evaporator, such as: the pinch point temperature and the terminal temperature differential (TTD) that affects directly the performance of the heat exchanger.

First, the pinch point temperature, which is the minimum temperature between the heat source and the working fluid, (as shown in fig. 3 for 1kg/s of geothermal fluid at 100 ^{o}C and the working fluid was R-290), affect directly the power production without affecting too much in the cycle efficiency, as can be seen in fig. 4, variations of the pinch point from 1 to 5 ^{o}C produce variations in the power production of the cycle from 11.6 kW to 6.4 kW, this phenomena occurs because the heat absorbed by the working fluid increase from 140.7 kW to 160 kW for the same heat source, (fig. 5). Values too smalls of the pinch point act over the mass flow rate of the working fluid that vary from 0.16 kg/s to 0.19 kg/s. Resuming, figures 3, 4 and 5 reveal that higher values of the pinch point affect the power production of the cycle and too small values affect the size of the equipments that means higher costs. That´s why, for the work, we assume values of 3 ^{o}C of the pinch point temperature.

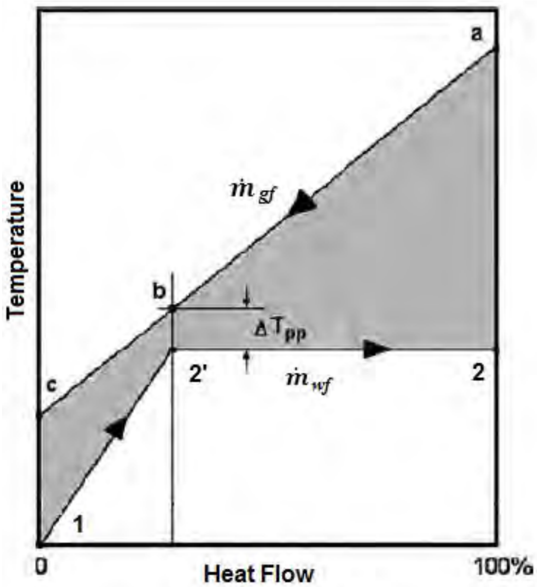

Figura 3. Heat transfer diagram for the evaporator.

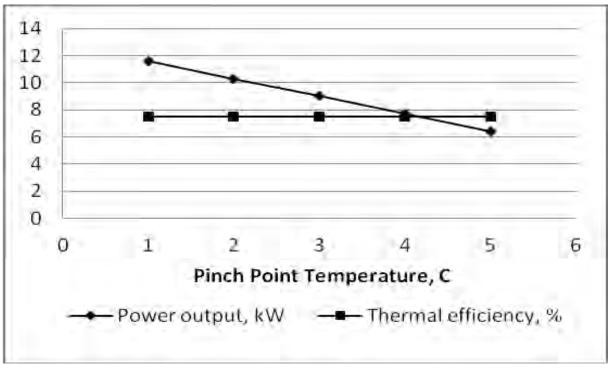

Figure 4. Power output and thermal efficiency of the cycle for different pinch points values.

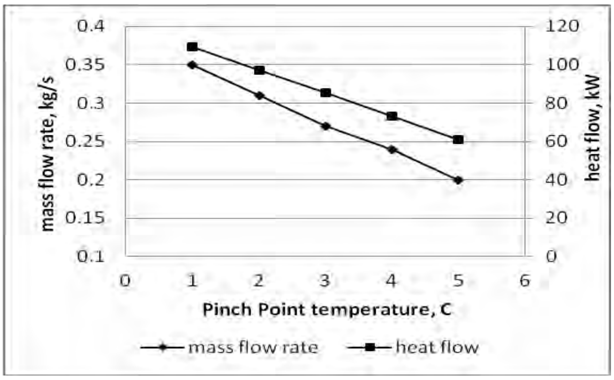

Figure 5. Heat flow and mass flow rate in the evaporator for different pinch points values.

Second, the TTD, (see fig. 6) which is the difference between the geothermal source temperature (**T_{s1}**) and the inlet temperature of the working fluid in the turbine (**T_{n1}**), have a great influence on the power output generated. When the TTD increase (or decrease the inlet pressure of the working fluid to the turbine), the outlet temperature of the geothermal fluid (**T_{s3}**) decrease, resulting in a higher quantity of heat flow absorbed by the working fluid. Then, lower values of the evaporation temperature result in a higher heat flow absorbed by the working fluid and a higher mass flow rate passing through the turbine, but lowest enthalpy drops. Hence, is necessary to find an optimum TTD to reach the higher power production of the cycle.

For each 15 working fluids, at low temperature geothermal sources, the evaporation temperature in which the best performance of the cycle is reached is about 60 to 70 % of the geothermal source temperature.

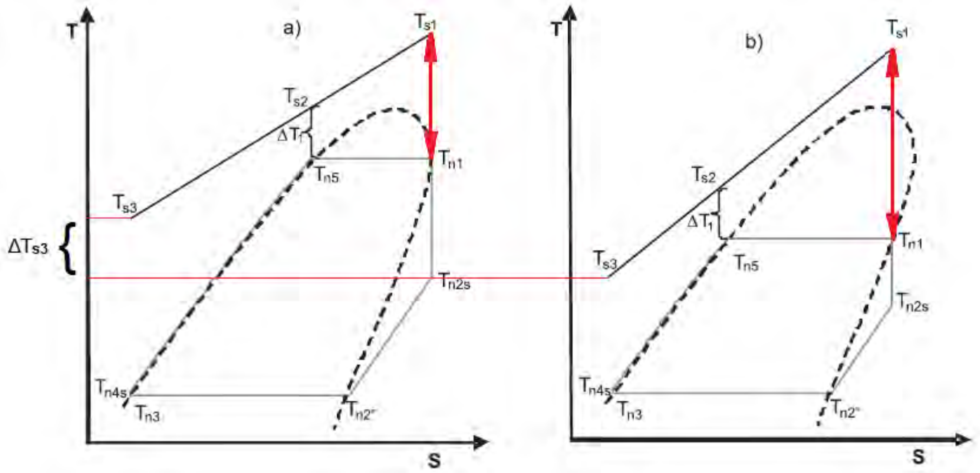

Figure 6. Thermodynamic cycles for the power plant, at the working fluid evaporation temperature $\mathbf{T_{n1}}$ of: a) relatively high value, b) relatively low value.

Turbine or expander in ORC plants is normally the most costly part. This equipment represents up to about 60% of total cost of the system; as a result, selection of the right turbine considering the application and system specifications is an important task.

In rather higher power ranges radial inflow turbines can be chosen. They have several advantages over axial turbines. They keep high efficiencies in small sizes and they are able to work in high pressure ratios. Since they have variable inlet nozzles, they can be well operated in variable load situations as the ones in geothermal plants. Dry gas seal with ability of recovering leakages can also be added to the system. This type of turbine cannot admit wet fluid due to significant decrease in turbine efficiency and erosion problems in turbine.

To obtain a higher power in MW scales, axial turbines can also be selected; however, in ORC applications of less than 1 MW they are not efficient enough as axial turbines that are designed for higher powers applications. (Rowshanzadeh, R., 2011).

In Equation 7, while v stands for volumetric flow in turbines and its unit is: (m^3/s), H is the isentropic enthalpy difference throughout the turbine, or specific work, in (kJ/kg). Turbine size factor is shown by $SF_{Turbine}$ here and the unit is meter. This can be used to compare different turbines sizes and is a proper indicator of its relative cost. Higher size factor means bulkier and more expensive turbines, (Rowshanzadeh, R., 2011), (Lakeu, A. A., & Bolland, O., 2010).

$$SF_{Turbine} = \frac{v^{0.5}}{H^{0.25}}\,(m) \tag{7}$$

Thermodynamics analysis

Mass and energy balances for each component of the ORC system can be calculated using equations 8 and 9.

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{8}$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out} \tag{9}$$

Exergy is the maximal work, attainable in given reference state. The objective of the exergy analysis is to determine the operating conditions of a system where less destructions of available work occur. Thus, the specific exergy for a given pure substances, neglecting the changes in kinetic, chemical and potential energy, can be calculated as follow.

$$e = (h - h_0) - T_0(s - s_0) \tag{10}$$

The properties in the dead state are evaluated at T_0 and P_0 when the fluid is in the liquid phase at the dead-state conditions, it is sufficiently accurate to take the dead-state enthalpy and entropy values as if the fluid is a saturated liquid at the dead-state temperature, for this case, 25 ^oC.

The exergy destruction rate can be calculated for each component of the cycle using the following exergy balance equation:

$$\dot{E}_d = \sum \dot{m}_{in} e_{in} - \sum \dot{m}_{out} e_{out} - \dot{W} \tag{11}$$

Finally, the energy and exergy analysis makes it possible to calculate the respectively thermal and exergetic efficiency of the cycle from the followings equations:

$$\eta_{th} = \frac{W_t - W_p}{Q_{in}} \cdot 100 \tag{12}$$

$$\eta_e = \frac{W_t - W_p}{E_{in} - E_{out}} \cdot 100 \tag{13}$$

Based on equations 8 to 13, the first and second law of the thermodynamic balance for the most important cycle components has been developed, according to (Cengel, Y. A., and Boles, M. A., 2002).

3. RESULTS AND DISCUSSION

The analysis was carried out for a geothermal mass flow rate of 1 Kg/s and temperature between 90 °C to 140 °C of the geothermal source. In order to obtain the optimum performance for Brazilian conditions, were analyzed the mass flow rate consumption of the working fluid, net power produced, pump consumption, turbine enthalpy drops, turbine size factor, thermal and exergetic efficiency and energy losses for each fluid.

The mass flow rate and the turbine enthalpy drop (see fig. 7 and 8) have an important influence on the power production of the cycle. Fluids that have bigger enthalpy drops generally are fluids with lower mass flow rate and, as can be seen in fig. 9, the net power production of the cycle is almost the same, especially for very low geothermal source temperature, between 90 ^{o}C and 110 ^{o}C. The net power productions vary from 6.6 kW at 90 ^{o}C (R-11) to 38.7 kW at 140 ^{o}C (R-152a). The variation of the net power production is small for the different working fluids, it should be evaluated, the influence of the mass flow rate and the pump consumption (see fig. 10), over the cost of the cycle, taking to account that higher mass flow rates and pump energy consumption have a important influence over the cycle cost.

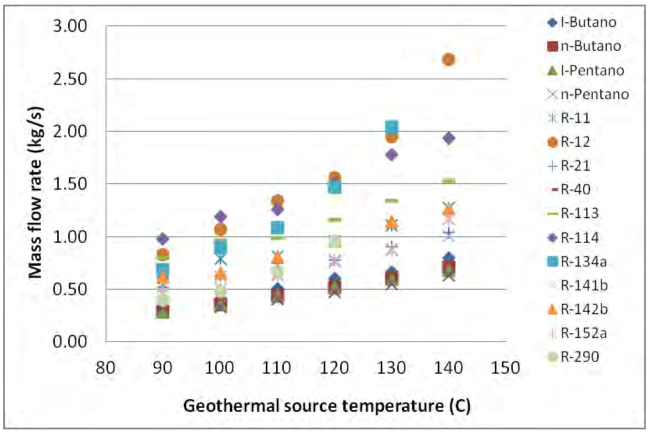

Figure 7. Mass flow rate for each working fluid at different geothermal source temperature.

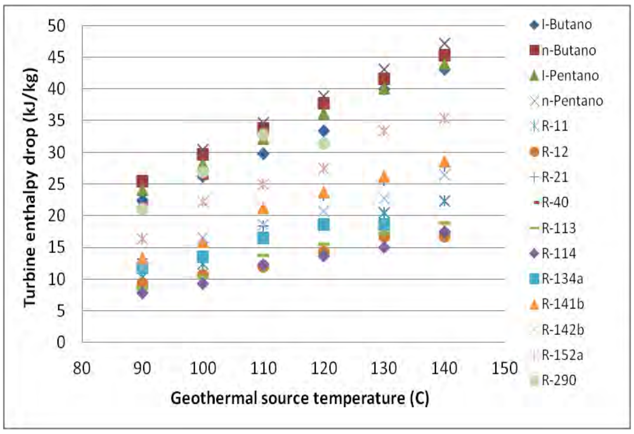

Figure 8. Turbine enthalpy drops for each working fluid at different geothermal source temperature

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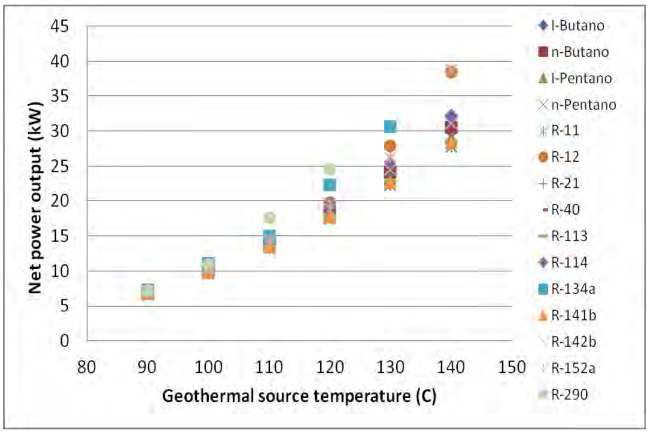

Figure 9. Net power output for each working fluid at different geothermal source temperature.

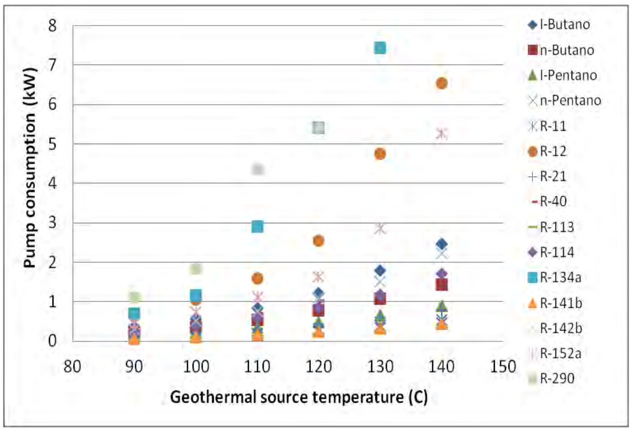

Figure 10. Pump energy consumption for each working fluid at different geothermal source temperature.

Figures 7 and 8 shows the thermal and exergetic efficiencies of the cycle, for different working fluids, at a geothermal source temperature from 90 °C to 140 °C. The thermal efficiency varies between 5.4 to 11.7 % while the exergetic efficiency varies between 37.7 and 58.8%.

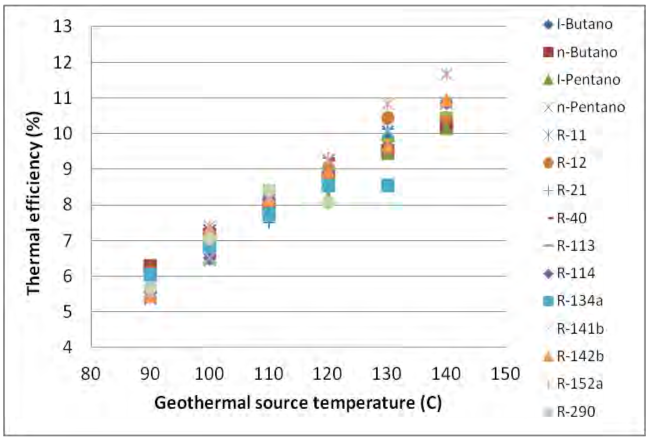

Figure 11. Thermal efficiency for each working fluid at different geothermal source temperature.

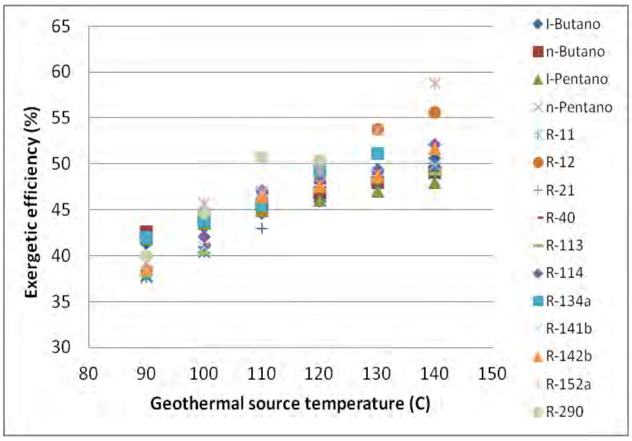

Figure 12. Exergetic efficiency for each working fluid at different geothermal source temperature.

In Fig. 13 is shown the turbine size factor as mentioned in equation 7. Notice that there is a relative big different in size factor for the different working fluids, and it means in cycle equipment cost. Then, to select the working fluid that offers the best performance of the cycle, a economic analysis is mandatory.

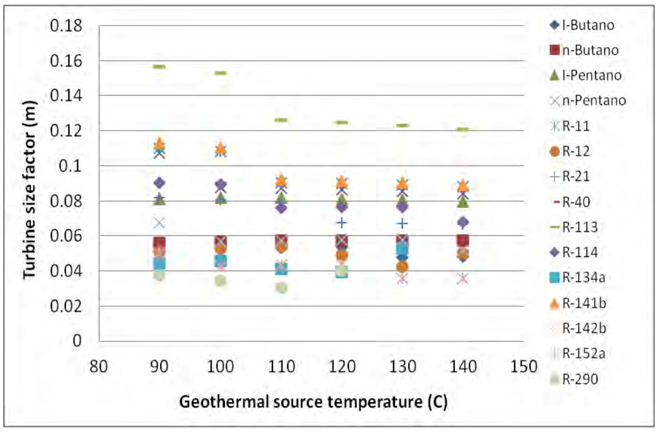

Figure 13. Turbine size factor for each working fluid at different geothermal source temperature.

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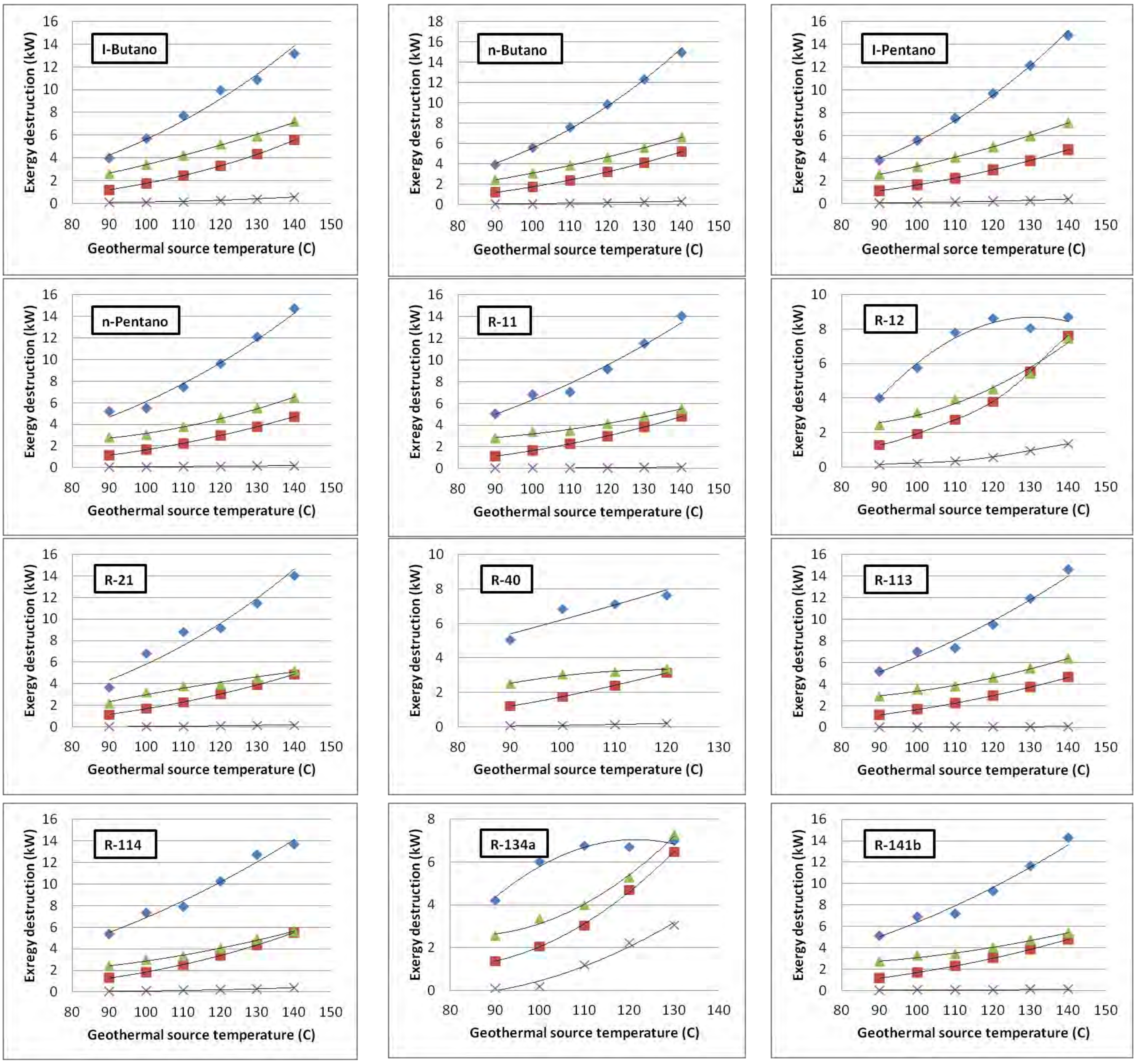
Figure 14 shows the exergy destruction in each component of the cycle for different working fluids. The exergy destruction in the cycle behaves as follow: the evaporator is responsible for the biggest irreversibility, followed by the turbine, condenser and finally the pump. For working fluids in which the geothermal source temperature exceeds it´s critical point, it can be observed how the slope of the exergy destruction in the evaporator varies. This can be explained because the sensible heat in the evaporation process is much bigger than the latent heat, in that case the evaporation process seem like a “transcritical cycle” were the exergy losses are smaller. In wet fluids, as R-40, to avoid qualities of the vapor lower than 0.9 (for this analysis without a super heater), the study was performed until a geothermal temperature of 120 ^oC.



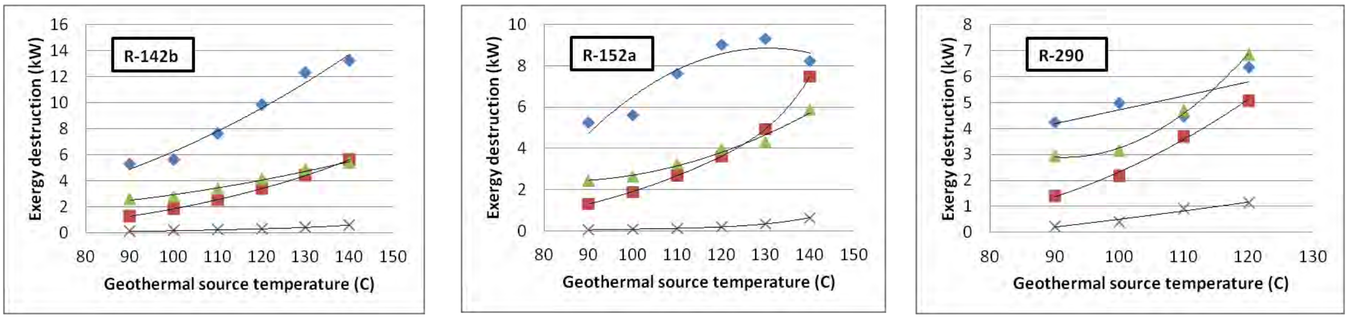


Figure 14. Exergy destruction for the different components and working fluids of the ORC at different geothermal source temperature.

4. CONCLUSIONS

In this work, the performance of the ORC was investigated by using the first and second law analysis, with an emphasis in low-temperature geothermal sources. Starting from power generation, mass flow rate consumption of the working fluid to thermal and exergetic efficiency and irreversibilities of 15 working fluids were investigated. In summary, the following conclusions can be drawn:

1. The pinch point temperature and the TTD have an important influence over the performance of the cycle.
2. Exergy analysis is necessary to find where the energy losses are concentrated.
3. An economic analysis is necessary to improve the energy analysis in order to have a complete vision over thermal and economical aspects of the cycle.
4. Special attention in turbine size factor is required taking to account that turbine is responsible for the biggest part of the cycle cost.

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