ENC-2020-0752
COST ANALYSIS OF A SOLAR ORGANIC RANKINE CYCLE WITH SCROLL EXPANDER

Hairton Júnior José da Silveira
Cristiana Brasil Maia
Pontifical Catholic University of Minas Gerais, Av. Dom José Gaspar, 500 - Belo Horizonte - MG Zip Code: 30535-901, Brazil
hairtonjjsilveira@gmail.com
cristiana@pucminas.br

Abstract. *This paper aims to analyze the costs and the generation of electrical energy from a Solar Organic Rankine Cycle (SORC). The research was made about the variations of Organic Rankine Cycles (ORC) proposed for thermal source applications. A Compound Parabolic Concentrating (CCP) collector was used as heat supply for the cycle. The equations of the CCP model, ORC model, and cost analysis are described. The analysis focusing on power levels, efficiency, and optimum operating points to choose the configuration that enables the higher cycle efficiency and low costs. A numerical simulation is held of the cycle in software EES for checking the environmental parameters that influence the performance. In this analysis, the main points of variation were the maximum inlet temperature of the scroll expander, also the evaporation pressure. In the design analysis, the maximum inlet temperature (T_{max}) varied within the range 125-150 ºC, while evaporation pressure range between 1000 kPa to 2100 kPa. The design point was selected with 125 ºC of maximum inlet temperature and an evaporation pressure of 1300 kPa. At this design point, the cycle achieves the highest performance, with 27.14% of cycle efficiency and 4.35 kW of power production. The increase of maximum inlet temperature and evaporation pressure rise the Investment Capital Cost, once the collector needs a larger area to supply this demand, representing a cost of 58% to 65% of the total cost. The increase in the evaporator pressure contributes to the reduction of the generation cost up to 1600 kPa. After that, the generation cost rises due to higher costs than power production. At the same time, the lowest temperatures do a better generation cost. It was possible to obtain the lowest costs at the design point, with $42203 of Investment Capital Cost and 0.088 $/kWh of Generation Cost which is a low cost compared to the bibliography.*

Keywords: *Solar Organic Rankine Cycle, Scroll Expander, Cost Analysis.*

1. INTRODUCTION

In the last decades, there has been observed a rapid increase in energy demand of the world, along with growing concerns regarding environmental pollution (Wang et al., 2020). The use of renewable energy sources is a potential solution to overcome these issues. Solar energy becomes an interesting alternative in regions with a high incidence of solar radiation with few resources for energy distribution, despite its high capital cost and low power density. New generation plants are designed based on thermodynamic and economic parameters, to achieve the maximum power at the lowest generation cost. The assessment of the electricity generation and investment cost of an electricity generation plant is one of the most important factors in engineering projects. The Solar Organic Rankine Cycle is an interesting cost-benefit solution of low and medium-temperature technologies of solar thermal power plants compared to other solar technologies because provides high thermal efficiency, high power, and no cost of power supply, is an interesting alternative for small-scale applications. It is a very efficient technique because of its simplicity and readily available components (Yadav and Sircar, 2019).

The thermal performance deviation in Solar Organic Rankine Cycles operating with a scroll expander has been studied by several authors during the last years. Sonsaree et. al (2018) studied a system using low-temperature heat (less than 100 ºC) of solar energy to generate electricity in a small-scale organic Rankine cycle. Operating on three different loads with the R-245fa fluid in conjunction with a water heating system. Composite parabolic concentrators, vacuum tubes, and flat plate solar collectors have been proposed. An economic analysis showed that the minimum LCE (Levelized Cost of Energy) was 0.20 $/kWh operating with 950 units of vacuum tubes when the initial investment cost was not considered, producing 113.5 MWh/year. Including this cost, the lowest LCE was identified as 0.67 $/kWh with 900 units of the same system, generating 110 MWh/year. Mehrpooya, Ashouri, and Mohammadi (2017) performed a thermoeconomic analysis of a regenerative organic Rankine cycle, using a parabolic solar collector as a source of heat supply for the cycle. The plant has a double pressure system in which the performance of two heat recovery boilers (economizer, evaporator, and superheater) are used between the expansion stages to lift the working fluid. The condenser cooling system was operated with liquefied natural gas, to absorb a larger portion of energy. At the end of the optimization, it was defined by a Pareto frontier as an optimum point in the cycle in which the annual production cost was $ 3.88 million, with an exergetic efficiency of 19.59%.

To reduce the investment cost for the expander, some authors have studied the implementation of a scroll expander, equipment that performs an excellent conversion of pressure energy into mechanical energy for small applications. Lemort et. al (2009) carried out a comparative study of a scroll expander applied to an organic Rankine cycle. The authors created a prototype of the expander and performed a numerical simulation to validate the model. First, experimental results showed that the maximum achieved pressure ratio was around 5.5. A machine with a larger built-in volume ratio than 4.05 would yield better performances, increasing pressure ratios. After a thorough investigation of the internal pressures and associated energy changes in the scroll expander, structural parameters were identified, and it was possible to predict the mass flow rate, the shaft power, and the exhaust temperature with good accuracy. The tested prototype achieved maximum isentropic effectiveness of 68%. Ibarra et. al (2014) analyzed an organic Rankine cycle system at load part operation wherein the scroll expander being the thermal machine responsible for generating energy. In this study, the authors varied the maximum temperature at the outlet evaporator (120 ºC to 150 ºC), the evaporation pressure (500 kPa to 3000 kPa), the condensation temperature (15 ºC to 35 ºC), and the expander speed (1000 at 5000 rpm) for two different fluids: R245fa and SES36. The SES36 fluid was shown to have a greater potential than the R245fa, and it was observed that the efficiency achieved by the scroll expander has a great influence on the thermal efficiency of the cycle. The design point of the ORC should be selected at the maximum high temperature (150 ºC), the minimum condensation temperature (15 ºC), and the maximum rotation speed (5000 rpm). However, the optimum point was defined as the design point, with a production of 5kW of electrical energy, with 16% thermal efficiency for the R245fa fluid, while this efficiency rises to 18% in the same operation for the SES36 fluid.

This paper aims to analyze the costs and the generation of electrical energy from a solar organic Rankine cycle for small-scale, operating with a scroll expander. A solar collector will be chosen as heat supply, designed for the city of Janaúba, state of Minas Gerais, Brazil, a place where there is high solar radiation. A cost analysis will be carried out to identify the investment cost and the generation cost, to verify the project's feasibility.

2. METHODOLOGY

The modeling cycle was performed in the software Engineering Equation Solver (EES). This software has a library of the most required thermophysical properties. Figure 1 shows the thermal cycle modeled in the software EES which will be the object of study in this present work. It is an Organic Rankine Cycle with a recuperator, using R245fa as the working fluid. Starting the cycle at point (1) the fluid enters the expander (ex) with high pressure and temperature, performing work, and leaving at point (2) with intermediate temperature and low pressure. Then, it goes in the hot stream of the recuperator (recu) to increase temperature an also enthalpy before the evaporator. In the exit of the hot stream (3), the low pressure and temperature fluid go to the condenser (cond), in which it will lose energy and come out in the saturated liquid phase at point (4). The fluid enters the pump (p) increasing its pressure at point (5), going to the cold stream of the recuperator to receive energy supply from the hot part. Then, enters the evaporator (evap) at (6) and receives energy from a solar collector to increase the temperature.

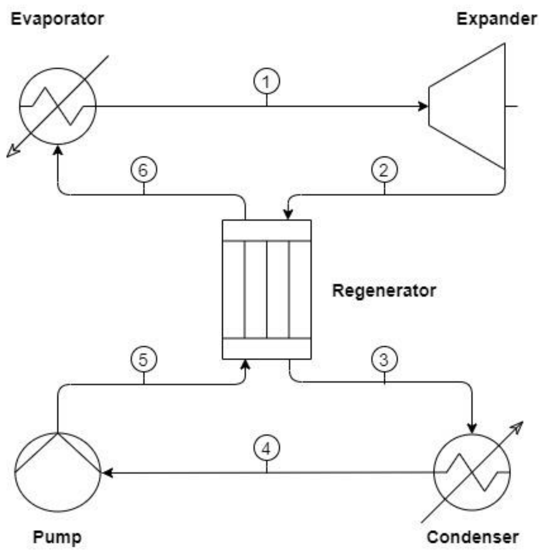

Figure 1. Organic Rankine Cycle with recuperator.
Available from: Adapted of Ibarra et. al (2014)

2.1 Solar collector model

According to Giavonetti et. al (2016), a Compound Parabolic Concentrating (CPC) collector is the most popular technology with better thermal performance and optical efficiency, compared to other concentrating solar thermal technologies, suitable for the low and medium temperature ranges. For this reason, the CPC technology was specified for this article. The equations and assumptions were based on the model performed by Kuhre et. al (2020), in which the main collector efficiency equation is described for Eq. (1).

$$\eta_{col} = 0.6481 \times \cos\theta \times K_{\theta L} \times K_{\theta T} - 1.12\left[\frac{T_m - T_a}{I_g}\right] - 0.01\left[\frac{(T_m - T_a)^2}{I_g}\right] \tag{1}$$

where θ is the incident angle between beam radiation and normal of the collector plane, $K_{\theta L}$ and $K_{\theta T}$ are incident angle modifiers along the longitudinal axis and transverse axis respectively, T_m is mean of inlet and outlet temperature, T_a is the ambient temperature, and Ig is the global radiation.

Duffie and Beckman (2013) describe that the cos θ for the southern hemisphere can be calculated by Eq. (2), where the solar declination is described for Eq. (3).

$$\cos_\theta = \cos(\phi+\beta)\cos(\delta)\cos(w) + \sin(\phi+\beta)\sin(\delta) \tag{2}$$

$$\delta = 23{,}45^o \sin\left[360^o \frac{(284+n)}{365}\right] \tag{3}$$

where n is the day of the year, ϕ is the latitude, β is the slope and w is the hour angle. The time and the angle adjustment were done to reduce the significance of the incident angle modifier, even the cosine effect was minimum at this period. Thus, the $K_{\theta L}$ and $K_{\theta T}$ are approximated of 1, and for Eq. (4) when cos θ is approximated of 1.

$$\eta_{col} = \dot{m} \times Cp \times (T_{out} - T_{in}) / (I_g \times A_{col}) \tag{4}$$

where $\dot{m}$ is the mass flow of the fluid in the collector, C_p is the specific heat, T_{out} and T_{in} are the temperatures at collector outlet and inlet, and A_{col} is the aperture area of all collectors. The collector was simulated for a city of Janaúba, in the state of Minas Gerais, Brazil, because of the high solar incidence in this area. For this application, the data inlets and assumptions for the solar collector model was ϕ = -15.78º south, w = 0 for noon, $\beta = \phi$ for annual application, and I_g was specified by COMPANHIA ENERGÉTICA DE MINAS GERAIS (2012), with annual average solar global radiation I_g = 650 W/m².

2.2 Organic Rankine Cycle

Fig. 1 shows the schematic of the Organic Rankine Cycle evaluated in this work. All the equipment of the cycle can be calculated with a mass balance and an energy balance, in Eq. (5) to Eq. (7) below.

$$\dot{W}_{ex} = \dot{m}(h_1 - h_2) \tag{5}$$

$$\dot{W}_p = \dot{m}(h_5 - h_4) \tag{6}$$

$$\dot{Q}_{evap} = \dot{m}(h_1 - h_6) \tag{7}$$

$$\dot{Q}_{cond} = \dot{m}(h_3 - h_4) \tag{8}$$

$$\dot{Q}_{recu} = \dot{m}(h_2 - h_3) = \dot{m}(h_6 - h_5) \tag{9}$$

where $\dot{W}_{ex}$ is the power in the expander, $\dot{W}_p$ is the power of the pump, $\dot{Q}_{evap}$ is the heat transfer in the evaporator, $\dot{Q}_{cond}$ is the heat transfer in the condenser, and $\dot{Q}_{recu}$ is the heat transfer in the recuperator. The isentropic efficiencies of the expander (η_t) and pump (η_p) can be described by Eq. (10) and Eq. (11), respectively.

$$\eta_t = (h_1 - h_2)/(h_1 - h_{2s}) \tag{10}$$

$$\eta_p = (h_{5s} - h_4)/(h_5 - h_4) \tag{11}$$

To analyze the cycle on its effectiveness and its viability, some equations are governing the system and gives an idea of the behavior of the cycle as shown below at Eq. (12) and Eq. (13).

$$\dot{W}_{net} = \dot{W}_{ex} - \dot{W}_p \tag{12}$$

$$\eta_{cycle} = \dot{W}_{net} / \dot{Q}_{evap} \tag{13}$$

where $\dot{W}_{net}$ is the net power production of the cycle, and η_{cycle} is the thermal efficiency.

To find the surface areas of the heat exchangers, further analysis must be performed. The plate heat exchangers are modeled employing the Logarithmic Mean Temperature Difference (LMTD) method for counter-flow heat exchangers. The Logarithmic Mean Temperature Difference is described in Eq. (14), and the heat transfer in Eq. (15).

$$\Delta_{LMTD} = (T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i}) / \ln((T_{h,i} - T_{c,o}) / (T_{h,o} - T_{c,i})) \tag{14}$$

$$\dot{Q} = U A_s \Delta_{LMTD} \tag{15}$$

where U is the overall heat transfer coefficient, As is the surface area, the subscripts 'h' and 'c' means hot and cold fluids, and the subscripts 'i' and 'o' means inlet and outlet of the flows.
The overall heat transfer is calculated by considering two convective heat transfer resistances in series, shows in Eq. (16), where the cold flow is analyzed internally, and the hot flow is studied externally.

$$1/U = 1/h_c + 1/h_h \tag{16}$$

Forced convection heat transfer coefficients are evaluated through the non-dimensional relationship internally for laminar flows, shows in Eq. (17), and turbulent flows, shows in Eq. (18).

$$Nu = 4.36 \tag{17}$$

$$Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.4} \tag{18}$$

where Nu is the dimensionless Nusselt number, Re is the dimensionless Reynolds number and Pr is the dimensionless Prandtl number.

For the externally flows, the Nusselt can be evaluated by Zukauskas and Jacob relationship, in Eq. (19) for 4000<Re<40000, and in Eq. (20) for 40000<Re<400000.

$$Nu = 0.193 \mathrm{Re}^{0.618} \mathrm{Pr}^{1/3} \tag{19}$$

$$Nu = 0.027 \mathrm{Re}^{0.805} \mathrm{Pr}^{1/3} \tag{20}$$

The inputs data for calculations based on Ibarra et. al (2014) model is shown in Tab. 1. In the design analysis, the maximum inlet temperature (T_{max}) varied within the range 125-150 ºC, while evaporation pressure range between 1000 kPa to 2100 kPa.

Table 1. Computational assumptions and parameters.

Input parameters	Units	Value
Pump efficiency	%	80
Expander efficiency	%	65.75
Inlet evaporator temperature	ºC	40
Condensation pressure	kPa	147.43
Outlet recuperator temperature (steam)	ºC	38.27
Mass flow rate	kg/s	0.1466
Internally diameter of heat exchangers	cm	2
Externally diameter of heat exchangers	cm	6

2.3 Cost Analysis

To analyze the project costs, the investment capital costs (ICC) of each equipment must be calculated, in \$, according to Nafey and Sharaf (2010) in the Eq. (21) to Eq. (24).

$$ICC_{col} = 639.5 \times A_{col}^{0.95} \tag{21}$$

$$ICC_{ex} = 4750 \times \dot{W}_{ex}^{0.75} \tag{22}$$

$$ICC_{p} = 3500 \times \dot{W}_{p}^{0.47} \tag{23}$$

$$ICC_{recu} = ICC_{cond} = 150 \times A_{s}^{0.8} \tag{24}$$

The Operating and Maintenance (O&M) costs are represented as 15% of the investment capital cost of the collector, and 25% of the investment capital cost of other equipment. The generation cost (Z), in \$/kWh of the cycle, can be evaluated by the Eq. (25).

$$Z = AF \times \left(\sum ICC + \sum O\&M \middle/ 8760 \times \dot{W}_{net} \right) \tag{25}$$

AF is the amortization factor is estimated based on the following relation shows in Eq. (26), where i is the long-term interest rate of 5%, and y is the plant lifetime and set as 20 years.

$$AF = i(1+i)^{y} / ((1+i)^{y} - 1) \tag{26}$$

3. RESULTS AND DISCUSSION

Once the detailed model of the cycle in the EES software has been performed, the maximum inlet temperature values and the evaporation pressure were changed to obtain the outlet parameters.

Fig. 2 shows the graph of power production as a function of evaporation pressure, varying the maximum inlet temperature. It can be seen that as the evaporation pressure increases, the power production rise because the high-pressure fluid will have a greater capacity to generate work in the expander. According to the increase in temperature, there is better reuse of heat energy in this cycle.

Fig. 3 shows the graph of cycle efficiency as a function of evaporation pressure, varying the maximum temperature. As the maximum temperature decreases, there is an increase in the efficiency of the cycle, since the increase in the maximum temperature results in a greater heat supply to be introduced into the cycle. On the other hand, increasing the evaporator pressure has a positive effect on up to 1300 kPa. After that, there is a decay in the curve because with high pressures less power is generated in the expander concerning the amount of heat introduced.

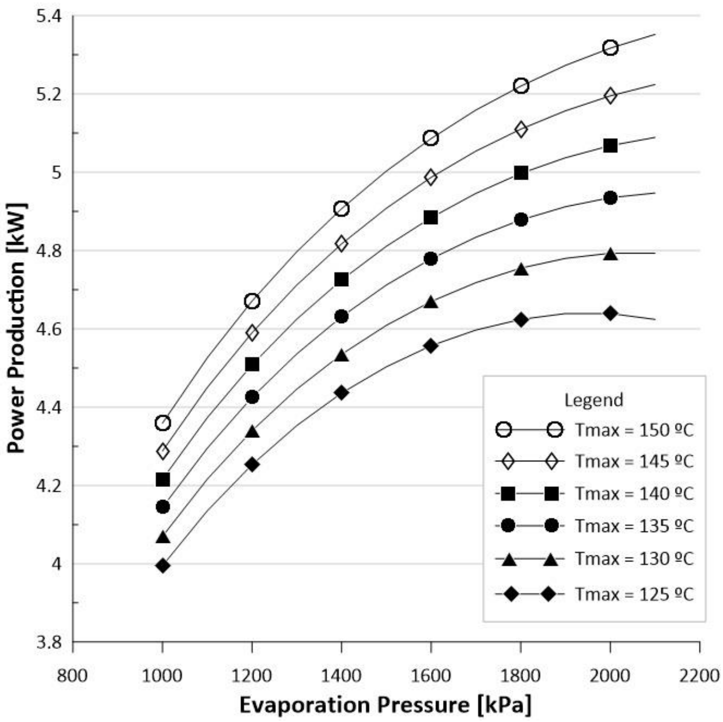

Figure 2. Power Production simulation for design point.

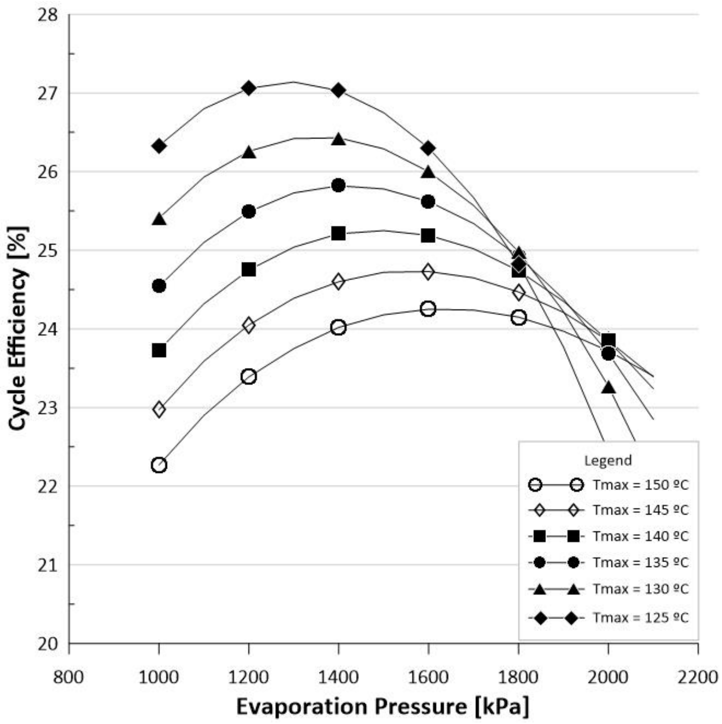

Figure 3. Cycle Efficiency simulation for design point.

Fig. 4 shows the graph of Investment Capital Cost (ICC) as a function of evaporation pressure, varying the maximum temperature. It was observed that with the increase of the evaporator pressure and the maximum inlet temperature the Investment Capital Cost rises. This is because, with the increase in the maximum inlet temperature and the pressure of the evaporator, the collector needs a larger area to supply this demand, representing a cost of 58% to 65% of the total cost.

Fig. 5 shows the graph of Generation Cost as a function of evaporation pressure, varying the maximum inlet temperature. Fig. 5 shows the generation cost is higher with increasing temperature since it will be possible to raise power production. The increase in the pressure of the evaporator contributes to the reduction of the generation cost up to 1600 kPa. This is because even this pressure is possible to generate more energy concerning the required heat. After this pressure, the cost of generation increases, due to higher costs than power production.

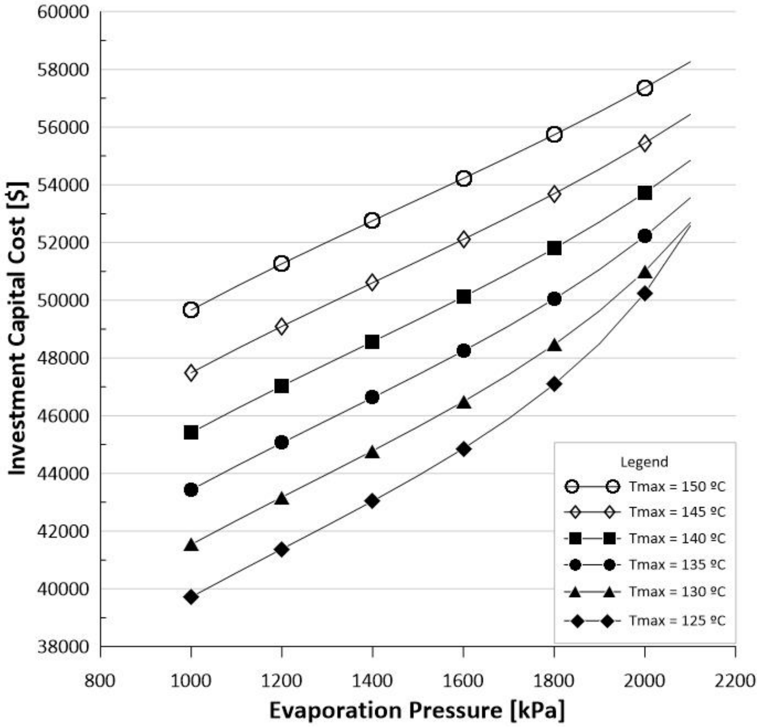

Figure 4. Investment Capital Cost simulation for design point.

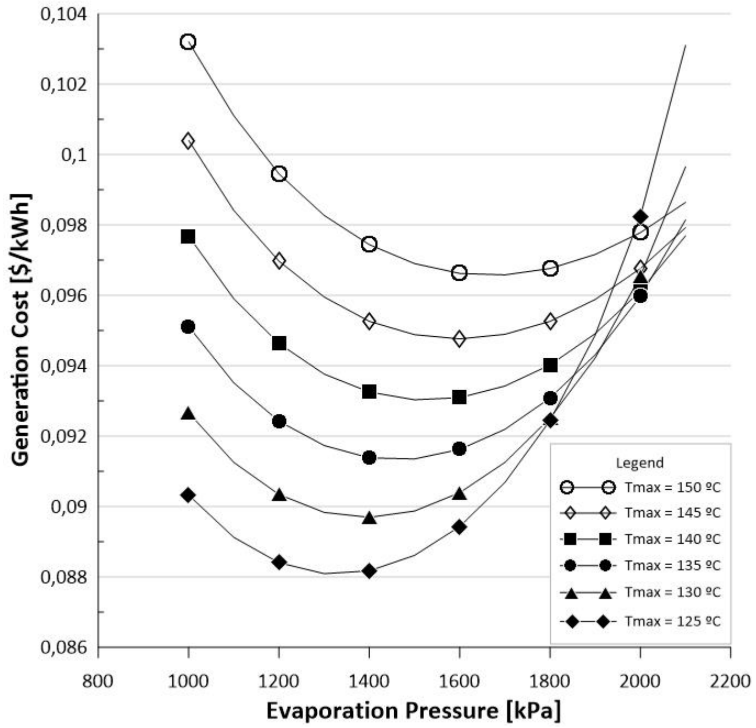

Figure 5. Generation Cost simulation for design point.

Considering metallurgical aspects, operational and performance were selected 125 ºC of maximum inlet temperature and an evaporation pressure of 1300 kPa for design condition. At this design point, the cycle achieves the highest performance, with 27.14% of cycle efficiency and 4.35 kW of power production. At the same time, it was possible to obtain the lowest costs, with $42203 of Investment Capital Cost and 0.088 $/kWh of Generation Cost which is a low cost compared to the bibliography.

4. CONCLUSIONS

Through the cost analysis of a Solar Organic Rankine Cycle with scroll expander, in this sense the results allow to express the following conclusions:

- The increase in maximum inlet temperature leads to an increase in power production. However, this leads to a decrease in the efficiency of the cycle, since more heat is needed in the evaporator;
- The evaporation pressure up to 1300 kPa increasing the cycle efficiency. After that there is a decay in the curve because with high pressures less power is generated in the expander concerning the amount of heat introduced;
- The increasing of maximum inlet temperature and evaporation pressure rise the Investment Capital Cost, once the collector needs a larger area to supply this demand, representing a cost of 58% to 65% of the total cost;
- The increase in the pressure of the evaporator contributes to the reduction of the generation cost up to 1600 kPa. After that, the generation cost rises due to higher costs than power production. At the same time, the lowest temperatures do a better generation cost;
- The design point was selected with 125 ºC of maximum inlet temperature and an evaporation pressure of 1300 kPa. At this design point, the cycle achieves the highest performance, with 27.14% of cycle efficiency and 4.35 kW of power production. At the same time, it was possible to obtain the lowest costs, with $42203 of Investment Capital Cost and 0.088 $/kWh of Generation Cost which is a low cost compared to the bibliography.

5. ACKNOWLEDGEMENTS

This study was financed in part by the Coordenação de Aperfeiçoamento de Pessoal de Nivel Superior – Brasil (CAPES) – Finance Code 001. The authors are also thankful CAPES and the Pontifical Catholic University of Minas Gerais – PUC Minas for the financial support for this work.

6. REFERENCES

COMPANHIA ENERGÉTICA DE MINAS GERAIS. Atlas Solarimétrico de Minas Gerais. Belo Horizonte, 2012. ISBN:978-85-87929-50-1.

Duffie, J.A., Beckman, W.A. Solar engineering of thermal processes. 4th ed., John Wiley & Sons, Hoboken, New Jersey, USA, 2013; p.910 pp.

Giovannetti, F., Horta, I.P., Marthy, H., Rittmann-Frank, M.H., Fahr, S.S., Kramer, K., ISE, F., Hertel, J., Martinez, V., Scarpellino, U.M., Palmieri, V., Comparison of process heat collectors with respect to technical and economic conditions. Technical Report A.2.1, 2016. IEA SHC Task 49, 36pp.

Ibarra, M.; Rovira, A; Alarcón-Padilla, D.; Blanco, J. Performance of a 5kWe Organic Rankine Cycle at part-load operation. *Applied Energy*. 2014;120:147-158.

Kuhre, N.; Pathak, A.; Deshpande, K.; Jadkar, S. Compound parabolic solar collector – Performance evaluation as per standard test method and actual field conditions for industrial process heat application in Indian context. *Energy for Sustainable Development*. 2020.

Lemort, V.; Quoilin, S.; Cuevas, C.; Lebrun, J. Testing and modeling a scroll expander integrated into an organic Rankine cycle. *Applied Thermal Energy* .2009;29:3094–102.

Mehrpooya, M.; Ashouri, M.; Mohammadi, A. Thermoeconomic analysis and optimization of a regenerative two-stage organic Rankine cycle coupled with liquefied natural gas and solar energy. Energy, v. 126, p. 899–914, 2017.

Nafey, A.S.; Sharaf, M.A. Combined solar organic Rankine cycle with reverse osmosis desalination process: Energy, exergy, and cost evaluations. *Renewable Energy*. 2010; 35:2571-2580.

Sonsaree, S. et. al. A small-scale solar Organic Rankine Cycle power plant in Thailand: Three types of non-concentrating solar collectors. *Solar Energy*, v. 162, n. January, p. 541–560, 2018.

Wang, S. et. al. Selection principle of working fluid for organic Rankine cycle based on environmental benefits and economic performance. *Applied Thermal Engineering* 178 (2020) 115598.

Yadav, K.; Sircar, A. Selection of working fluid for low enthalpy heat source Organic Rankine Cycle in Dholera, Gujarat, India. *Case Studies in Thermal Engineering*. Volume 16, December 2019, 100553.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.