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TECHNICAL AND ECONOMIC VIABILITY ANALYSIS OF AN ORGANIC RANKINE CYCLE FOR USE ASSOCIATED WITH COMPOUND PARABOLIC CONCENTRATOR SOLAR COLLECTOR

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Abstract: *Analyzed an Organic Rankine Cycle (ORC), for use associated with Compound Parabolic Concentrator (CPC) solar collector. This replaces the combustion of fossil fuels in the boiler, becoming a source of renewable energy, avoiding the emission of greenhouse gases. The temperature and pressure, as well as other operating parameters were used in the calculations referenced by manufacturers' catalogs and similar academic works.*
The software engineer equation solver (EES) was used for the mass and energy balances of the organic rankine cycle and the results obtained as a function of the fluid used were analyzed. We sought to use those fluids with higher critical point (pressure and temperature) as they are more suitable for higher temperature heat sources (as in the case of CPC).
In the analysis, a regenerator was used to take advantage of the residual heat at the turbine outlet and temperature gain before the evaporator inlet, obtaining a higher efficiency of the Organic Rankine Cycle. The heat exchange efficiency and its respective required heat exchange area were also calculated. In the condenser the cooling was the minimum necessary only to condense the working fluid, the thermodynamic optimization of the cycle was due to the increase of the heat exchange area in the evaporator.

Keywords: *Solar Energy, Rankine Cycle, Renewable Energy, Power Generation, Thermodynamic Analysis*

1. INTRODUCTION

The diversification of the energy matrix is a world need to bring greater energy security to the countries; prioritizing renewable energies has been a worldwide trend. In Brazil, about 45% of its energy matrix is renewable, highlighting large hydroelectric plants while the world average is 14%, and in the countries developed is only 6% (Lodi, 2011). Brazil has a high insolation countrywide throughout all year. On other hand, solar energy systems have high investment cost. Despite the opportunities, there are institutional and tributary obstacles to the development of this energy source in Brazil: The allocation of funds for research and development within the scope of the General Budget of the Union; Reduction of the tax burden for the acquisition of equipment for this purpose; Withdrawal of subsidies to other non-renewable energy sources (Silva, 2015).

The Organic Rankine Cycle (ORC) is identical to the traditional Steam Rankine Cycle, differing only in the working fluid, which is organic and can be hydrocarbons or halogenated hydrocarbons among others, with vaporization temperature smaller than the water, enabling mechanical energy to be obtained from heat at low and mid temperatures. Therefore, it is suitable for application in solar and geothermal power plants, as well as in waste heat recovery. The organic working fluid is characterized by a molecular mass higher than the water, which leads to lower pressures and slower rotation in the turbine, reducing erosion of its metal parts and blades and consequently its maintenance costs. It is normally used with a regenerator that preheats the working fluid before entering the boiler, increasing the cycle's thermal efficiency. In practice, the processes in the pump and turbine are not isentropic and the condenser and evaporator operate with pressure drop, decreasing the thermal efficiency of the actual cycle.

According to (Quoilin, 2011) , the success of ORC technology can be explained by its modular feature, that is can be used with various heat sources with few modifications in the sets. Table 1 shows the installed capacity of power plants using ORC, per application in 2017 according to (Tartiere, 2017). This manuscript aims at designing an ORC for usage associated with a compound parabolic concentrator solar collector system for power generation. The choice of the working fluid and its operational conditions was performed through computational simulation to find the most appropriate one to meet this application

Table 1. Total installed capacity per application

Source	Installed Capacity (MW)	Percent
Geothermal	2021	74.8%
Biomass	301	11.1%
Heat Recovery	375	13.9%
Solar	4	0.1%

from Tartiére et al. (2017)

2. METHODOLOGY

The pré-selection of working fluids was done based on (Quoilin, 2013), and international classifications such as Global Warming Potential (GWP), Ozone Depleting Potential (ODP) and ASHRAE Standard 34 (flammability) were considered. The minimum pinch point of 10^{o}C was adopted in all heat exchangers (engineering good practice). The Pinch Point corresponds in this study to the minimum difference between the temperature of hot fluid and cold fluid in the evaporator, condenser and regenerator. (Wakil, 1984) sayed when the Pinch Point is too small results in low global temperature difference and therefore lower irreversibilities but an exchanger big and expensive. Very large Pinch Point results in a small and inexpensive inefficient system.

Added by the restriction on the operating manometric pressure of the condenser being positive, since vacuum must be avoided. So, the condensing temperature was fixed at 10^{o}C above the design ambient temperature, 38.1^{o}C in case, obtained from the air conditioning Brazilian standard (16401, 2008) for the city of Rio de Janeiro, Brazil. Table 2 shows some properties for the pre-selected working fluids

Table 2. Some properties of the pre-selected working fluids

Fluid	Critical Temperature (^{o}C)	Critical Pressure (kPa)	Density *(kg/m^3)	Heat of Vaporization **(kJ/kg)
Toluene	318.6	4,126	862.2	361.3
n-Pentane	196.5	3,364	620.8	358
Benzene	560	4,406	873.7	433.7
CycloHexane	551.3	6,837	774.1	392.6
n-Hexane	500	2,752	657.3	365.7

*Density at Room Temperature (25^{o}C) 1 atm; ** Heat of Vaporization at 1 atm.

Figure 1 shows the regenerative organic Rankine cycle considered in the analysis. It comprised of an evaporator, turbine, regenerator, condenser and pump. In the regenerator heat exchanger, heat transferred from the superheated vapor at the turbine outlet to low temperature working fluid at the pump outlet in order to save energy. When using an isentropic or dry working fluid according to the classification based on the temperature-entropy diagram as presented by (Quoilin, 2011) and illustrated in Fig 1, the vapor exiting the turbine is superheated. Depending on the working fluid and the turbine pressure ratio, this temperature is higher than that of the liquid exiting the pump.

After the working fluid has left the regenerator, it enters the evaporator and absorbs heat from the heat source. The working-fluid phase varies from a sub-cooled liquid to a saturated or superheated vapor. Then the saturated or superheated vapor passes through the turbine linked to an electric generator, which converts the shaft work to electrical energy. The working fluid exiting the regenerator enters the condenser where heat is rejected to the environment. The ORC was modeled using the software Engineering Equation Solver (EES). The software EES, from the F-Chart (www.fchart.com) was appropriate for thermodynamics modeling since it has a full database of thermodynamics and transport properties of all candidate substances pre-selected in this study. It is an engineering program widely used because it contains a built-in library with several mathematical functions and thermophysical and transport properties of the most substances utilized in engineering.

The model was developed through mass and energy balance equations for control volumes around each cycle component, assuming steady state, with kinetic and potential energy variations negligible and flow without friction loss, according to (AY Cengel, 2015)

Other model assumptions are:

- No pipeline heat losses;
- Adiabatic heat exchanger model;
- Pump electricity consumption was negligible;

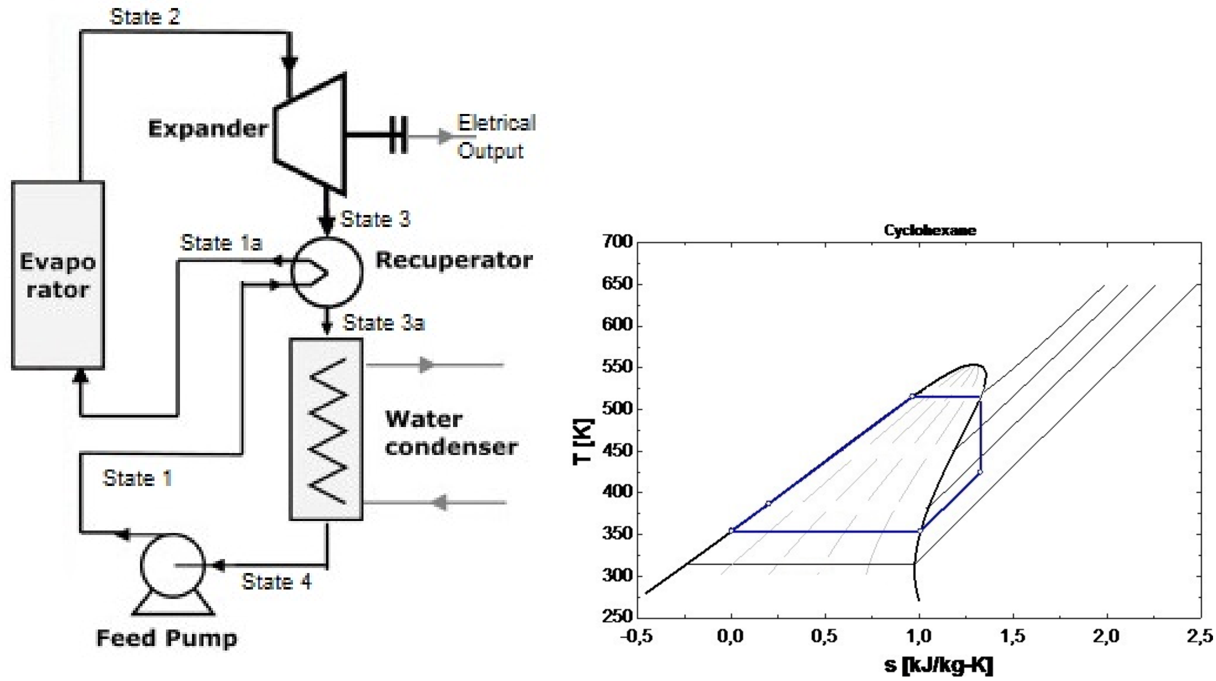

Figure 1. Organic Rankine Cycle and Temperature vs. Entropy Diagram of a Dry Fluid

- Heating produced by pump inefficiency was negligible;
- Organic Fluid has constant properties;
- Compression and pumping processes are isentropic.

The heating from the compound parabolic concentrator solar collector system is supplemented by natural gas burning whenever necessary in order to achieve the set point of the thermal oil at 240^{o}C, fixed based on (Ibrahim, 2015). Operation parameters surveyed from commercial ORC systems were used in this modeling, aiming to compare simulation results with those of existing systems. Therefore, the evaporator pressure was chosen at 2500 kPa as practiced by suppliers such as Turboden, Ormat and others mentioned in (Tartiere, 2017).

The overall heat transfer coefficient for evaporator and condenser were 100 W/m^{2}K, came from (Holman, 1999). It is worth remembering that these global coefficients already include the deposition factors. Conduction resistance is neglected as the materials involved are thin and have good thermal conductivity. It was also not necessary to consider pipe fins as the involved fluids (hydrocarbons, thermal oil and water) have good overall heat exchange coefficients, thus not making the design more expensive.

A single shell pass and with the inlet and the outlet at the opposite ends of the shell are modeled. The working fluid always flows on the shell side. The tube-side fluid (heat and sink source) will always be single phase in according to (Walraven, 2014). With these assumptions (single shell pass), the correction factor considered was 1. This factor influences the heat exchange area. It depends on the exchanger geometry and the cold and hot flow inlet and outlet temperatures. The mass flow rates for organic fluid and thermal oil are 0.25 and 1.3 kg / s respectively. Both came from (Faroni, 2016) and (Ibrahim, 2015). In time, it was found that the mass flow rate adopted meets the established pinch point criteria.

2.1 Organic Rankine Cycle Equation

The energy balance applied to a control volume with a flow in steady state neglecting kinetic and potential energy variations can be written as:

$$\dot{Q}_{in} + \dot{W}_{out} + \sum_{in} \dot{m}(h) = \dot{Q}_{out} + \dot{W}_{out} + \sum_{out} \dot{m}(h) \tag{1}$$

where: $\dot{Q}$ is the heat transfer rate (kW), $\dot{W}$ is shaft power, kW, $\dot{m}$ is mass flow rate, kg/s, and h is specific enthalpy, kJ/kg

The subscripts "in" is the input and "out" is output. The thermal efficiency of the cycle is determined from:

$$\eta_{cyclo} = \frac{\dot{Q}_{in} - \dot{Q}_{out}}{Q_{in}} \tag{2}$$

where: $\dot{Q_{in}}$ heat received by the evaporator, $\dot{Q_{out}}$ is heat transferred from the condenser, $W_p\dot{u}mp$ is work realized by the pump

The exergy provided is that contained in the heat supplied to the ORC in the evaporator and pump input, and the recovered exergy is the turbine power output. The efficiency of the second law of the various devices with permanent flow may be determined on the basis of its general definition:

$$\eta_{exergy} = \frac{W_{net}}{W_{pump} + Q_{in}} \tag{3}$$

2.1.1 Turbine or Expander

System was adiabatic, so there was no heat transfer:

$$\dot{W}_{out} = \dot{m}(h_{in} - h_{out}) \tag{4}$$

2.1.2 Pump

The properties of at a fixed point remain constant over time, besides, the control volume was adiabatic and it consumes work:

$$\dot{W}_{in} = \dot{m}(h_{out} - h_{in}) \tag{5}$$

2.1.3 Evaporator, Regenerator and Condenser

Considering all exchangers with steady state (evaporator, regenerator and condenser), variations in kinetic and potential energy are small terms of work and heat transfer, and are generally neglected. The evaporator, regenerator and condenser do not involve any work and so the energy equation comes down to:

$$\dot{Q}_{in} = \dot{m}(h_{out} - h_{in}) \tag{6}$$

2.1.4 Heat exchange area and heat exchanger effectiveness (condenser, evaporator, regenerator)

This coefficient was defined as a function of the fluids involved in the heat exchange.

$$A_{exchanger} = \frac{Q_{in}}{FU\delta T_{lm}} \tag{7}$$

where: $A_{exchanger}$ is heat exchange area, m^2, U is Overall heat exchange, W/m^{2}K, δT_{lm} is logarithmic mean temperature difference, K, F configuration correction factor. The effectiveness of a shell tube flow changer is a function of the number of transfer units (NUT), dimensionless, and lower heat capacity between fluids (C_{min}).

$$e_{1NUT} = 2\left\{1 + C_r + (1 + C_r^2)^{1/2}\frac{1 + exp[-(NUT)(1 + C_r^2)^{1/2}]}{1 - exp[-(NUT)(1 + C_r^2)^{1/2}]}\right\}^{-1} \tag{8}$$

The number of transfer units (NUT), dimensionless, was:

$$NTU = \frac{UA_{exchanger}}{C_{min}} \tag{9}$$

3. ANALYS AND RESULTS

Table 3 shows efficiency and exergetic efficiency as well as net power (kW) of pré-selected working fluid.

Even without the highest vaporization energy and density, among the pre-selected fluids (see Tab. 2), n-hexane and cyclohexane presented the best thermodynamic results. In addition, they are not cancernogenic fluids like benzene, being the last one of the lowest performance.

The cycle efficiency was slightly higher than those in the manufacturers catalogs (approximated 12%). One possibility was that most manufacturer references are from combustion engine exhaust thermal sources. Another possibility is the simplifications considered in this study, which in practice do not occur, such as neglecting pressure losses, adiabatic and isentropic expansion and compression.

Table 3. Results modeled in EES

Fluid	Efficiency	Exergétic Efficiency	W-net (kW)
n-Hexane	0.3497	0.3394	44.9
Cyclo Hexane	0.3356	0.3275	44.98
Toluene	0.3069	0.2998	39.98
n-Pentane	0.3039	0.2849	39.02
Benzene	0.276	0.2701	37.5

Table 4 shows heat exchanger area and effectiveness calculations as a function of working fluid. N-hexane not only has the best thermodynamic performance but also requires one of the smallest heat exchange area, reducing costs. Fact that did not occur for the hexane cycle that had good thermodynamic performance. Toluene is widely used by manufacturers, had good thermodynamic performance (see Tab. 3), and the smallest heat exchange required area.

The influence of the maximum allowed heat exchanger area is investigated. Figure 2 shows the efficiency for Regenerate ORC for different working fluids as a function of the maximum allowed heat exchanger area. The efficiency increases with increasing the area, as expected. Therefore, more heat is added to the cycle and this heat is more efficiently converted to mechanical power when the total heat-exchanger surface increases.

Table 4. Results modeled in EES

Fluid	A_{cond}(m^2)	A_{rec} (m^2)	A_{evap}(m^2)	A_{Total} (m^2)	$e_{NUT,cond}$	$e_{NUT,rec}$	$e_{NUT,evap}$
n-Hexane	15.17	30.48	28.93	44.1	0.2402	0.2557	0.1924
Toluene	9.34	28.57	33.19	42.53	0.2198	0.2375	0.2597
Benzene	18.65	21.57	58.33	76.98	0.2922	0.2404	0.2906
Cyclo Hexane	12.86	29.58	68.17	81.03	0.2457	0.2423	0.2807
n-Pentane	78.55	27.53	14.03	92.58	0.3008	0.2482	0.1424

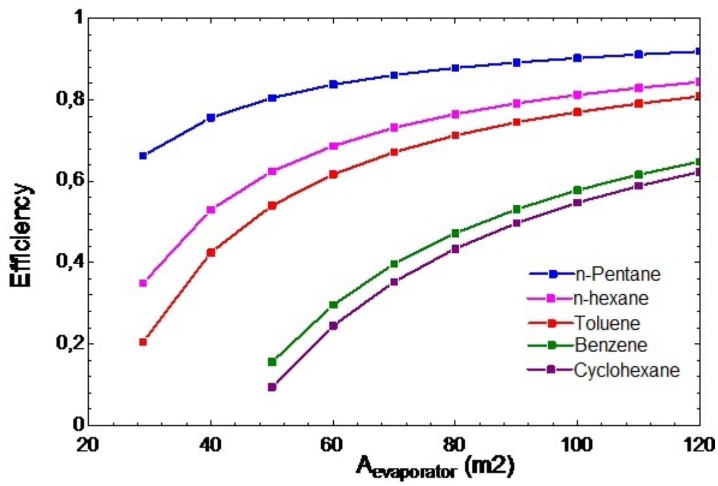

Figure 2. Energetic cycle efficiency for Regenerated ORC with all shell-and-tube heat exchangers for different fluids.

4. CONCLUSIONS

The heat exchange area minimum is oversized to take into account the deposition factor in the initial calculations, since the starting exchanger has no deposits or corrosion. Through this hypothesis, the efficiency of the cycle is possible to be higher than the one obtained.

As in the regenerator the maximum energy of the superheated was obtained came from turbine and in the condenser the cooling should be the minimum necessary only to condense the working fluid, the thermodynamic optimization of the cycle was due to the increase of the heat exchange area in the evaporator.

The high temperature and pressure levels eliminate the application for a series of fluids, such as n-decano, with lower critical point, but quite mentioned in academic articles for other temperature ranges. Those fluids with higher critical point will bring greater efficiency to the cycle, but these characteristics must be balanced so that the design does not have to develop robust equipment to withstand pressure and temperature levels.

The use of ORC with solar energy source, has lower utilization than the other energy sources (geothermal, heat recovery) as Tab. 1. This is due to the fact that the radiation energy is not constant all the time (at night and on cloudy days). For this reason, complementary sources of energy, such as natural gas flaring, will be needed for periods of low radiation, which increases costs.

5. ACKNOWLEDGMENT

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7. RESPONSIBILITY NOTICE