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THERMODYNAMIC ANALYSIS OF THE POWER GENERATION SYSTEM OF ORGANIC RANKINE WITH THE SOLAR HEATING SYSTEM

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Abstract. *Brazil has a great potential for electricity generation from the energy from the sun due to the high incidence of solar radiation over much of its territory, especially in the Northeast. Due to low energy conversion efficiency associated with the relatively low temperatures compared with the traditional thermal generation, it is necessary to review some assumptions of Rankine cycle using steam to the use of low temperature energy available is best used mainly in small-scale power generation. Thus, the use of the Organic Rankine cycle offers the possibility to use low temperature energy, as the energy available from the solar heating system. Thus, in this study, the coupling of an Organic Rankine cycle with a solar heating system for power generation will be analyzed. Types of fluids and operating the Rankine cycle configurations best suited for the coupling in Organic solar heating systems are analyzed. The thermodynamic analysis and exergy analysis will be used in the analysis settings.*

Keywords: *Organic Rankine Cycle, Solar Energy, Thermodynamic Analysis*

1. INTRODUCTION

In recent years, there has been a growing interest in the use of thermal energy readily available as process heat. Geothermal heat or energy from the sun, because of the growing need for better use of energy resources and gain in overall efficiency of systems that reject large amounts of thermal energy. However, the energy sources which do not involve combustion or nuclear decay processes are usually characterized by presenting relatively low temperatures, making traditional steam cycle using water (Rankine cycle) unfeasible for their use. Indeed, when the hot source temperature is below 370°C the water becomes unsuitable for power production, mainly due to their behavior during the expansion process, in which occurs partial condensation and problems associated with erosion of the materials (Gu, et al., 2007). In traditional Rankine cycles, this wet behavior of water can be avoided through overheating or steam reheat, which requires a larger area of heat exchange compared to using saturated steam.

There are cycles worked at a high maximum temperature and reject heat at relatively high temperature, and on the other hand, others have moderate maximum temperature and reject heat at a temperature close to the ambient temperature. A classification of different cycles according to their maximum temperature was done by Korobitsyn (1998) and is present in "Fig 1", where find theoretically and technically achievable combinations: Otto/Kalina, Rankine/Kalina, Fuel cell/Stirling, etc. Kalina and organic Rankine cycle operate at the lowest temperature, making them potentially suitable to recover the heat exhausted from high temperature fuel cells, gas turbines and diesel engines.

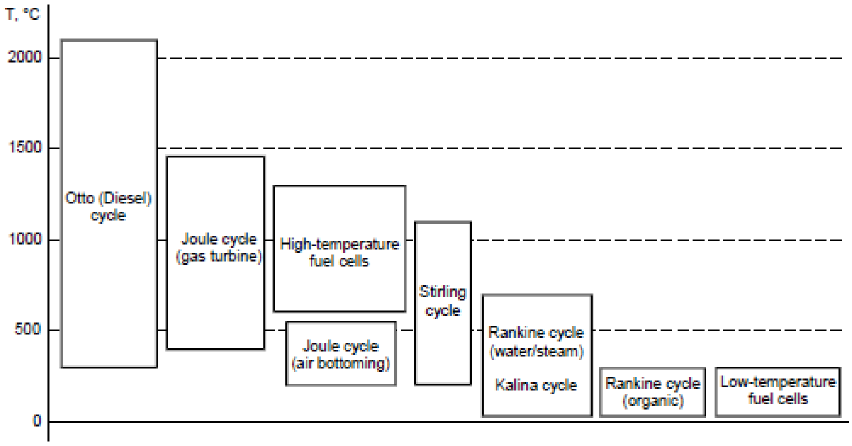

Figure 1. Basic power cycles classification with respect to the operating temperature (Korobitsyn, 1998).

The largest investments associated with an area of heat exchange and feedwater heater are justified by the power capacity of installations, which usually overcomes hundreds of MW, and the desirable properties of water in high temperatures, making it a suitable working fluid for large and centralized power system (Fankam Tachanche, 2011). However, in low temperature applications, the simplest configurations should be favored, an operational point of view and that of costs minimization. Thus, a suitable working fluid for the operating conditions can be the determining factor for the viability of a thermal recovery project. The practice of using a fluid different of water in power applications - the so called Organic Rankine Cycle (ORC) - dates back to the nineteenth century (Invernizzi, 2013) and since then the number of commercial operation systems increases, especially in geothermal applications, industrial waste heat recovery and biomass combustion (Velez et al., 2012).

Most applications of solar thermal power plants use water as a working fluid, such as the Solar Energy Generating System (SEGS) in the United States, a complex of nine plants using parabolic trough collectors, with a total installed capacity of 354 MW. There are few solar systems in commercial operation employing the ORC, one of them being the Saguaro plant in the United States, this plant started operating in 2006, n-pentane as working fluid in the ORC system, the turbine gross capacity is 1.16 MW, the turbine net capacity is 1.0 MW, turbine efficiency is 20.7% full load and design-Point Solar-to-Electricity Efficiency is 12.1% (NREL, 2017).

The characteristics of solar collectors coupled to ORC motivated this study, due to the temperatures below 400°C typical of these systems and due to its performance highly dependent on the working fluid, a third degree of freedom besides pressure and temperature as parameters important to maximize the overall efficiency of the combined system (Delgado-Torres; Garcia-Rodriguez, 2010).

2. METHODOLOGY

The concept of the Organic Rankine Cycle is the same as the traditional water cycle, in which an external heat source provides energy for the vaporization of the fluid, which expands in a turbine, is condensed and brought into circulation by a pump. This simple configuration can be modified through processes of reheating, regeneration and recuperation of the residual heat of the vapour exiting the turbine. The working fluid selection is an important step in analyzing the feasibility of a heat recovery project at low temperatures. This is due to the fact that at lower temperatures, the irreversibility occurring in the heat exchangers are very harmful to the overall efficiency of the cycle and the correct choice of the working fluid minimizes inefficiencies, which are highly dependent on the working fluid properties (Łukawski, 2009). In addition to performance, the choice of working fluid also determines the economic characteristics of the thermal unit (Fankam Tachanche, 2011).

One of the main advantages of the use of different working fluids is associated with a higher turbine efficiency, is even possible to use less complex and less costly single stage turbines (Gu; Weng; Cao, 2007). Another feature is to be able to choose the best fluid to recover the heat available in the given temperature and desired pressure, including pressures greater than atmospheric during the condensation, thereby avoiding the infiltration of air into the condenser. In "Fig. 2" it is emphasized that for a condensation temperature of 50 °C water and toluene operate at pressures below atmospheric, causing the condenser to operate at a partial vacuum.

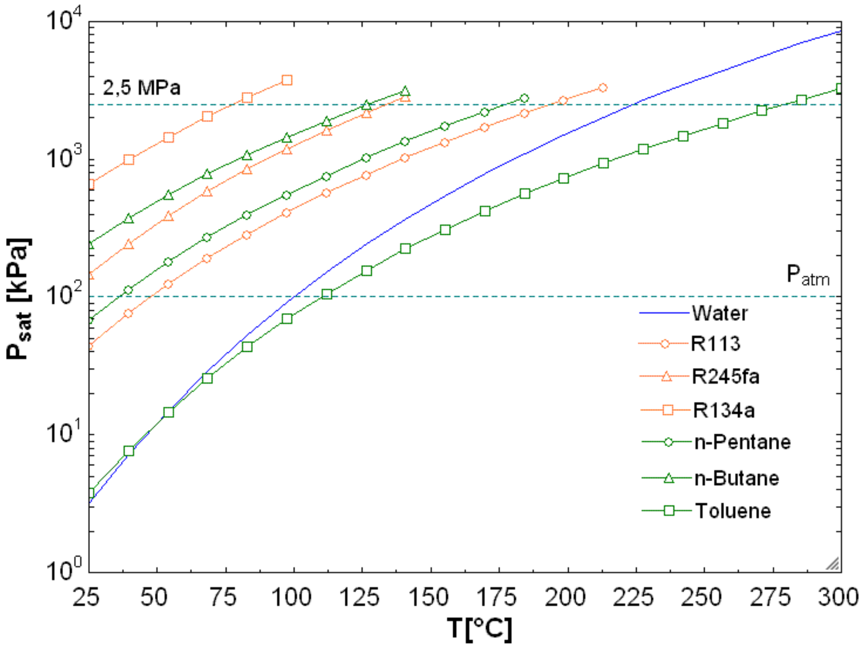

Figure 2. Condensate temperature for selected fluids.

The horizontal distance between the saturated liquid and vapour in the T-s diagram, “Fig. 3”, is also important because it is proportional to the enthalpy of vaporization per unit mass of the fluid. Thus, fluids such as water require less mass flow in the cycle to absorb a certain amount of heat and produce power while toluene requires higher mass flow, due to its smaller latent heat.

The working fluids can be classified into dry, wet or isentropic, depending on the saturated vapour curve in the T-s diagram (“Fig. 3”). Fluids with a negative slope in the saturated vapour curve are called *wet*; with zero or very small slope are called *isentropic* and with positive slope are called *dry* fluids (Chen; Goswami; Stefanakos, 2010). Thus “Fig. 3” highlights that water is a wet fluid, R134a refrigerant is isentropic and the hydrocarbon toluene is a dry fluid. The behavior of the fluid during the expansion gives rise to its classification: the wet fluids condensate along an isentropic expansion process initiated from saturated vapor, forming droplets; isentropic fluids do not condense and dry fluids exit an isentropic process in a superheated state.

The shape of the saturation curve has influence in the choice of the type of cycle. Wet fluids necessarily need superheating and eventually reheating to avoid condensate formation at the end of the expansion, being water the typical case in power cycles. Dry fluids leave the expansion process in a superheated state, being possible to recover part of its thermal energy and transfer it to the fluid prior to entering the evaporator. If this energy is not recovered, the cycle efficiency can decrease due to the increase in average temperature of heat rejection in the condenser.

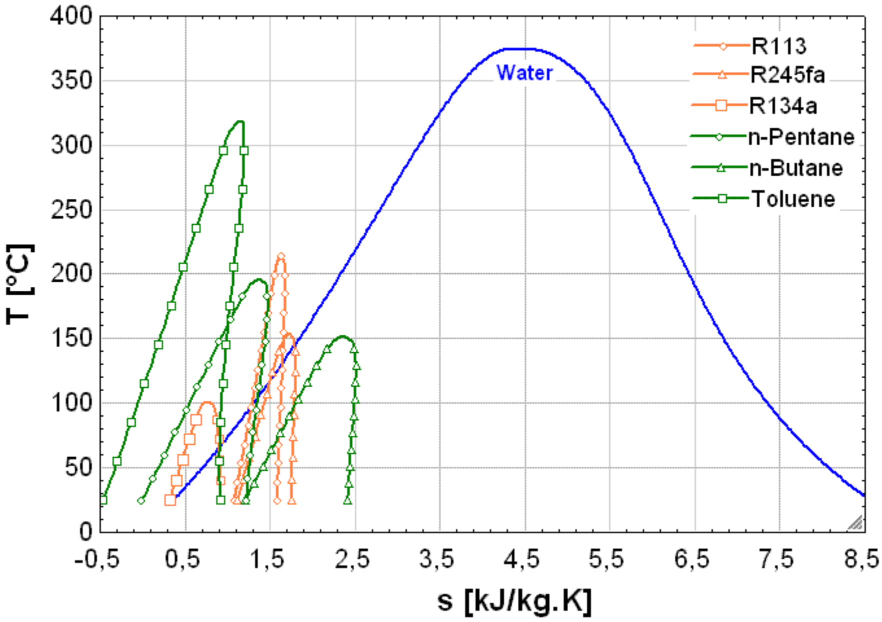

Figure 3. Saturation pressure and T-s Diagram for selected fluids

In addition to appropriate thermodynamic properties, the choice of a working fluid requires a review of the following factors, according to Fankam Tachanche (2011):

1. Safety: the fluid should not be toxic, causing damage to any leakage or be dangerous to handle;
2. Environmental impact: the fluid must have a low ozone depletion potential (ODP) and low global warming potential (GWP);
3. Stability: the fluid should not deteriorate in the operating temperature range of the cycle or in contact with the piping materials and insulation;
4. Cost: The fluid must have a low cost and be readily available commercially. Often, for a given application, the best working fluid identified by a thermodynamic analysis differs from the more economically viable (Quoilin et al., 2011). Therefore, there must be a compromise between efficiency and cost.
5. Good transport properties: for example, viscosity and thermal conductivity, are also of importance in order to properly design, simulate, and optimize components of the cycle, most especially heat exchangers.

2.1 Cycle Configuration

The most common configuration of Organic Rankine cycle employed in commercial installations is the recuperated cycle, as highlighted by McMahan (2006). Figure 4 shows the typical configuration of this cycle coupled with a solar thermal system. It is noteworthy that this type of configuration does not just apply to solar collectors, but also in other applications due to its simplicity, low cost, and modularity.

However, the heat recovery from the vapour exiting the turbine is only possible for dry fluids. In fact, the simple cycle configuration without regenerator is sufficient for isentropic fluids, while the presence of the regenerator can even be harmful if the working fluid is wet. In the case of dry fluids, the use of superheating is optional and must be accompanied by regeneration in order not reduce the cycle efficiency (Chen, et al., 2010).

The lower temperatures and pressures involved in the ORC allow a single design to be created and made available in commercial modules, requiring only some adaptations for each plant, which reduces the cost of implementing a project employing Organic Rankine cycle.

The maximum temperature at which the turbine operates in ORC is of the order of 400°C. The maximum operating pressure is of the order of 2 MPa, being possible the use of pressures up to 2.5 MPa, while the minimum pressure in the cycle is 5 kPa (Bao, et al., 2013).

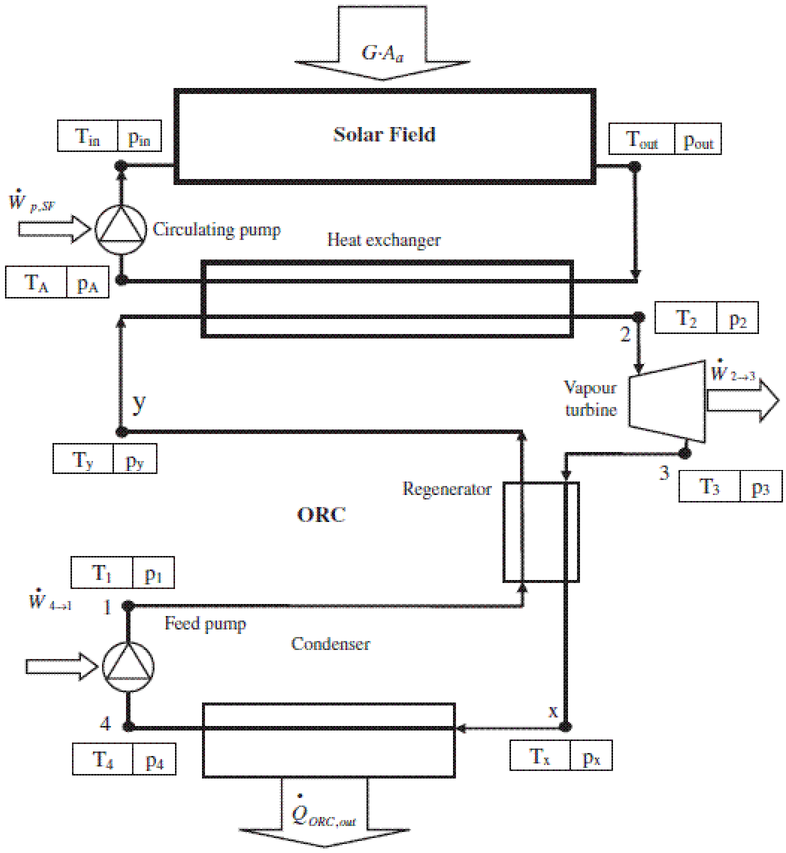

Figure 4. The most common configuration for commercial Organic Rankine Cycle. (Nafey and Sharaf, 2010)

2.2 Solar Collectors

Solar collectors are equipment that transforms the incident solar energy into thermal energy for the fluid flowing close or inside it, usually, some heat transfer fluids, increasing its temperature. They can be classified as concentrating or non-concentrating collectors.

The concentrating collectors focus the direct component of the solar radiation in a focal line or point, while non-concentrating collectors only reflect incident radiation, without focusing it. Although the Linear Fresnel collectors are widely used in practice to replace the parabolic trough, its temperature range depends on the arrangement of the individual linear collectors that composes the whole system. For this reason, it was not simulated in this study. Table 1 shows some common collectors for water heating or power production and its operating temperature range (Nafey and Sharaf, 2010).

Table 1. Types of solar collectors and operating temperature range.

Type of Collector	Collector	Temperature Range (°C)
Concentrating	Parabolic Trough Collector (PTC)	170 - 300
Concentrating	Compound Parabolic Collector (CPC)	120 - 170
Non-concentrating	Evacuated Tube Collector (ETC)	80 - 160
Non-concentrating	Flat Plate Collector (FPC)	80 - 100

The collector performance, except for concentrating towers, can be expressed by an equation of efficiency ("Equation 1"), defined by the ratio between the useful heat transferred to the heat transfer fluid (Q_u) and the heat absorbed by radiation (G_d) in the opening area (A_a). Experimental data for collectors are correlated by an "Eq. 1" (Nafey; Sharaf, 2010).

$$\eta_{col} = \frac{Q_u}{G_d \cdot A_a} = \eta_0 - a_1 \cdot \left(\bar{T}_f - T_{amb}\right) - a_2 \cdot \frac{\left(\bar{T}_f - T_{amb}\right)}{G_d} - a_3 \cdot \frac{\left(\bar{T}_f - T_{amb}\right)^2}{G_d} \tag{1}$$

Where: T_f is the mean temperature of the heat transfer fluid, T_{amb} is the ambient temperature, η_0 is the optical efficiency under normal incidence radiation and parameters a_1, a_2 e a_3 are experimental coefficients of the temperature-dependent heat loss coefficient for each solar collector model. (Delgado-Torres; García-Rodriguez, 2010).

2.3 Solar Collectors coupled with Organic Rankine Cycle.

The coupling between the collector and the Organic Rankine cycle is presented in Figure 2, in which a heat transfer fluid (Therminol VP-1, widely used in solar applications) receives thermal energy from the solar collector and then provides heat energy to the ORC working fluid, which is then heated and vaporized. The regenerator extracts some thermal energy from the vapour exiting the turbine in order to preheat the liquid prior to enter the evaporator when the exit temperature is higher than that of the fluid leaving the pump. If this is not the case, the regenerator can be removed in the simulation by imposing its effectiveness to be equal to zero.

In this work it was analyzed subcritical fluid cycles operating with representative working fluids of wet (water), dry (toluene) and isentropic (R134a) type, widely used commercially and whose viability in commercial power cycles was already proven (Velez et al., 2012).

The coupled system (Fig. 2) was modeled in the software EES - Engineering Equation Solver (Klein, 2014). To develop the model were adopted mass and energy balance equations for control volumes, assuming a steady state, with kinetic energy and potential variations negligible and flow without friction loss, according to Cengel and Boles (2013).

From simplifying assumptions, the following performance parameters were defined for the solar system in order to check the behavior of the cycle with the selected fluids.

Aperture area (A_a) of the solar collector, Eq. (2).

$$A_a = \frac{Q_u}{\eta_{col} \cdot G_d} \tag{2}$$

The efficiency of the coupled system, disregarding the power for circulation of the heat transfer fluid, Eq. (3).

$$\eta_{system} = \frac{\dot{W}_{liq}}{G_d \cdot A_a} = \frac{Q_u}{G_d \cdot A_a} \cdot \frac{\dot{W}_{liq}}{Q_u} = \eta_{col} \times \eta_{cycle} \tag{3}$$

Exergy or Second Law Efficiency of the coupled system, Eq. (4).

$$\eta_{II} = \frac{\dot{W}_{liq}}{Ex_{rad}} \tag{4}$$

Exergy of solar radiation was calculated by the Petela's equation (Hepbasli, 2008).

$$Ex_{rad} = A_a \times G_d \times \left[1 + \frac{1}{3}\left(\frac{T_o}{T_{sun}}\right)^4 - \frac{4}{3}\frac{T_o}{T_{sun}}\right] \tag{5}$$

The term in brackets has a value of 0.934 for the sun surface temperature of 6000 K and room temperature of 298 K.

To validate the model developed a simulation was performed with the following working fluids: isobutane, isopentane, and R245fa chosen because they are the same used in the papers of Delgado-Torres and Garcia-Rodriguez (2010). The results obtained are shown in "Tab. 2", it can be seen that the developed model in EES has values close to those presented by Delgado-Torres and García-Rodriguez (2010), the source of errors being the use of different methods for the calculation of fluid properties in EES and their paper. This validation thus enables the coupled system performance to be evaluated in different operating conditions.

Table 2. Comparison between the model in EES and the paper of Delgado-Torres and García-Rodriguez (2010).

			Exit Regenerator Temperature (°C)			Cycle Efficiency (%)			Mass flow rate/Power (kg/kJ)		
Fluid	TIT (°C)	Evap. Pressure (kPa)	Ref	Model	Error (%)	Ref	Model	Error (%)	Ref	Model	Error (%)
Isobutane	95	1371	36,8	36,8	0,0	9,85	9,89	0,4	2,559	2,712	6,0
	145	2395	42,9	44,6	4,0	14,94	14,43	-3,4	1,488	1,691	13,6
Isopentane	95	491,6	37,7	37,6	-0,3	10,49	10,51	0,2	2,357	2,403	2,0
	145	1288	44,1	44,0	-0,2	16,40	16,46	0,4	1,343	1,388	3,4
R245fa	95	829,6	36,4	36,3	-0,3	10,14	10,18	0,4	4,444	4,558	2,6
	145	2087	41,7	41,7	0,0	15,46	15,52	0,4	2,614	2,739	4,8

TIT = Turbine Inlet Temperature

Delgado and García-Torres-Rodriguez (2010) conducted studies with other cycle configurations at various operating conditions and these researchers concluded, as McMahan (2006) did, that the recuperated cycle, though not having the highest efficiency among the cycles analyzed, is the one chosen by manufacturers because of the simplicity of manufacture and maintenance. For this reason, the recuperated cycle was selected for the evaluation of the coupled system.

For analysis of the coupled cycle the following parameters were used:

a. Working fluids: water, toluene, and R245fa
b. Turbine isentropic efficiency: 0.75
c. Pump isentropic efficiency: 0.80
d. Recuperator effectiveness: 080
e. Pinch point difference in evaporator and condenser: 10°C
f. Condensation temperature: 30°C
g. Heat transfer fluid: Therminol-VP1
h. Evaporator pressure range: 500 kPa up to 2,5 MPa
i. Ambient temperature: 25°C
j. Net electric power: 1 kW
k. Collector solar irradiance (G_d): 1000 W/m²
l. The pumping power of the heat transfer fluid has been neglected in the analysis
m. Temperature exiting the collector was assumed to be 300 ° C and 170 ° C for PTC for CPC.
n. In the analysis of the coupled system were used concentrating solar collectors, the data utilized are present in "Table 3".

Table 3. Fitting data for the selected concentrating solar collectors

Type	Collector	η_0	a_1	a_2	a_3
PTC	EuroTrough(1)	0.750	0.039	0.0003	0.000045
CPC	CPC Aosol 1.12X (2)	0.736	4.610	0.0000	0.000000

(1) Delgado-Torres and García-Rodriguez (2010); (2) Nafey and Sharaf (2010)

3. RESULTS AND DISCUSSIONS

The result of the simulation for the parabolic collector is shown in "Fig. 5", which shows the first and second law efficiency for the coupled system and the aperture area of the collector for water, toluene, and R245fa as working fluid as a function of evaporation pressure.

For the conditions studied, the regenerator was only necessary for toluene due to its dry characteristic and its thermal energy available to be recovered at the turbine exit. The presence of regeneration in the cycle with water and R245fa led to a negative entropy generation in the heat exchangers, indicating that this process was unnecessary for both fluids. Thus, the regenerator effectiveness was set to zero, resulting in the simple ORC without regeneration for water and R24f5a, while the recuperated cycle was possible for toluene.

The three fluids analyzed showed an increase in the efficiency with increased evaporation pressure, due to the increase in the average temperature of heat addition with saturation pressure. Moreover, the second law efficiency remained close to the first law efficiency, indicating that the maximization of the energy efficiency of the system is sufficient as the objective function because the exergy of solar radiation is a constant fraction of the value of the solar irradiance itself according to Eq. (6).

The best harnessing of the incident solar radiation occurred with toluene, which also presented the lowest aperture area required per unit kW, indicating that its adoption would result in a more compact solar field. The refrigerant R245fa presented lower efficiencies and an opening area per kW greater than toluene, while water showed intermediate results.

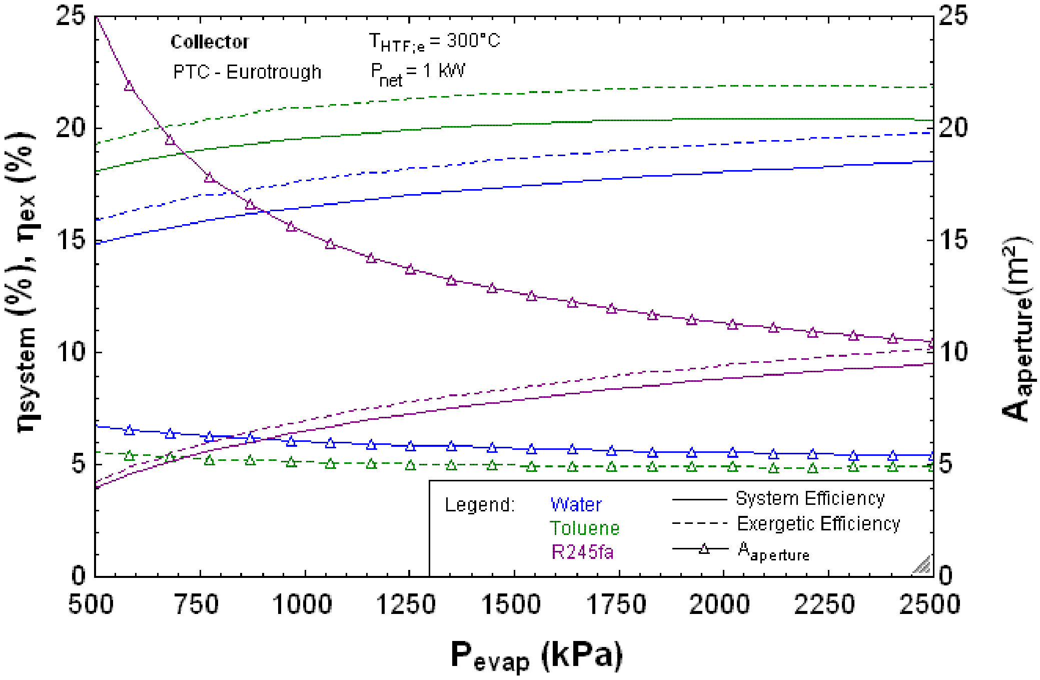

Figure 5. Performance parameters as a function of evaporation pressure.

"Figure 6" shows the mass flow rate of the working fluid and heat transfer fluid as a function of the evaporation pressure. The mass flow rates are directly related to system size and the required pumping power.

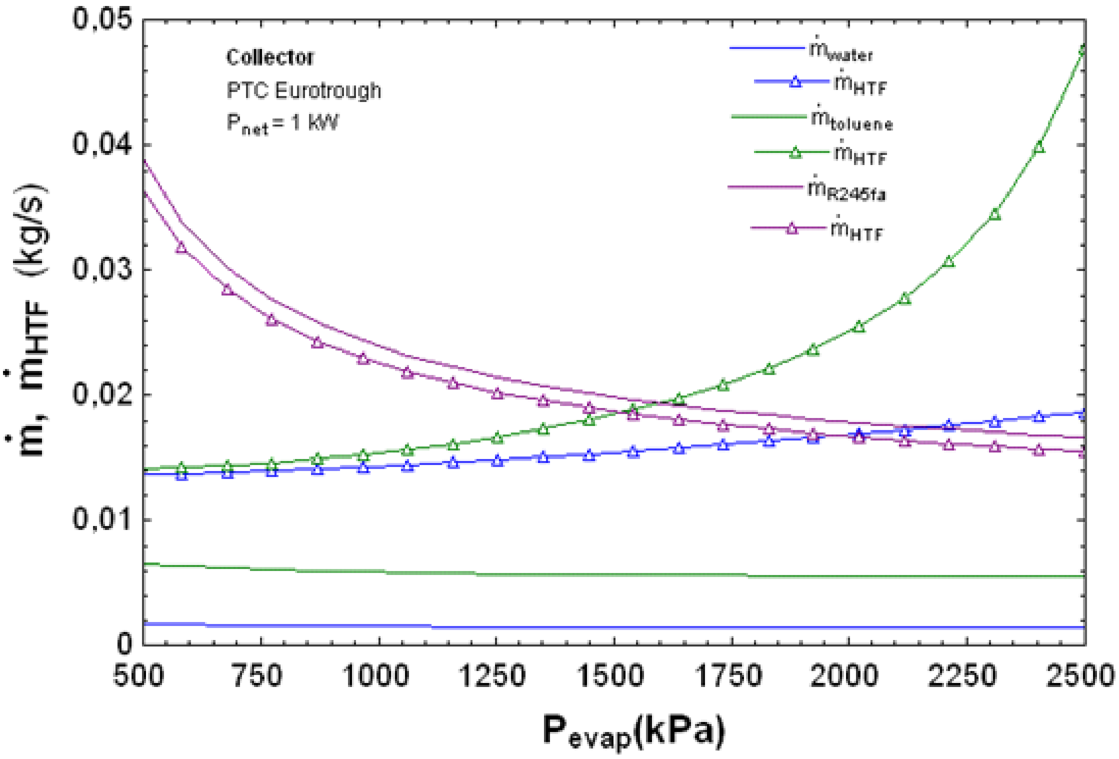

Figure 6. Mass flow hates as function of evaporation pressure.

The lower mass flow rate shown in "Fig 6" was obtained for water, followed by toluene and R245fa. This result was due to the greater enthalpy of vaporization of water, which requires less mass flow to absorb the same amount of heat as compared to the other fluids. For higher pressures, which are associated with a greater efficiency, toluene showed the highest mass flow required for heat transfer fluid. However, the higher efficiency for toluene tends to compensate the pumping power associated with the higher flow rates, and due to its higher specific mass in comparison to water, the resulting volumetric flow rate is reduced, leading to smaller equipment.

The thermodynamic analysis of the CPC collector showed a negative entropy generation in the heat exchangers for water and toluene, indicating the impossibility in utilizing these working fluids within the selected pressure range, at the maximum temperature of 170°C reached by this type of collector. The possibility of using the refrigerant R245fa in these operating conditions is a fact consistent with the literature, especially because this temperature is common to geothermal applications (Velez et al., 2012).

"Figure 7" shows the behavior of R245fa coupled to CPC collector, indicating that although being the only fluid applicable in the simulated conditions, the greater area associated with it make its practical use unfeasible for solar power applications. The use of another type of collector with higher temperatures can make more sense for obtaining higher efficiency and lower aperture area, as indicated in "Fig 3". Nevertheless, it can notice the presence of a minimum area of 32 m² which corresponds to a maximum efficiency of 3% for this working fluid, and this should be the operating point selected in the case of using this working fluid coupled to CPC.

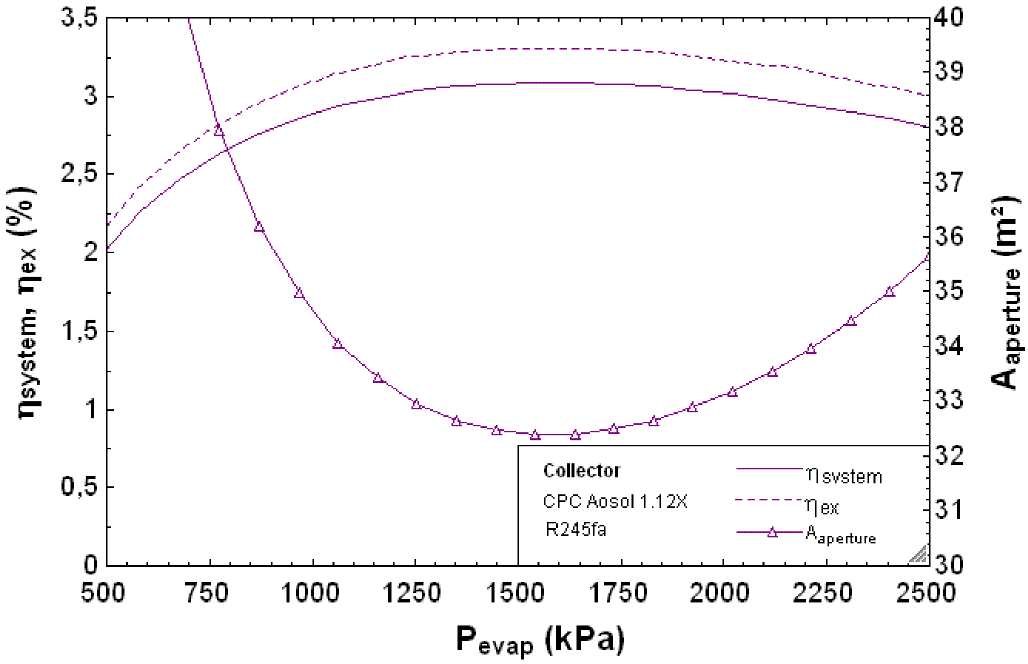

Figure 7. CPC operating with R245fa.

4. CONCLUSION

Results obtained for the solar regenerative organic Rankine cycle (ORC) show that, in general, dry fluids analyzed yield lower values of the unit aperture area than wet fluids. For the conditions studied, the regenerator was only necessary for toluene due to its dry characteristic and its thermal energy available to be recovered at the turbine exit. The three fluids analyzed showed an increase in the efficiency with increased evaporation pressure, due to the increase in the average temperature of heat addition with saturation pressure. The best harnessing of the incident solar radiation occurred with toluene, which also presented the lowest aperture area required per unit kW, indicating that its adoption would result in a more compact solar field.

The lower mass flow rate was obtained for water, followed by toluene and R245fa, which requires less mass flow to absorb the same amount of heat as compared to the other fluids. For higher pressures, which are associated with a greater efficiency, toluene showed the highest mass flow required for heat transfer fluid. However, the higher efficiency for toluene tends to compensate the pumping power associated with the higher flow rates, and due to its higher specific mass in comparison to water, the resulting volumetric flow rate is reduced, leading to smaller equipment.

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