THERMAL ANALYSIS OF A PARABOLIC TROUGH ORGANIC RANKINE CYCLE SOLAR POWER

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Abstract. *In this paper we develop a mathematical model for a parabolic trough organic rankine cycle solar power System. This system uses water as a heat transfer fluid and R245fa as working fluid for the thermal power cycle. The model consists of three parts. The first describes the movement of the sun in order to determine the angle of incidence, zenith angle, solar altitude angle, azimuth angle, etc.; the second determines the optical efficiency of the collector, the effective energy transfer to the heat transfer fluid, the temperatures at different points of the receiver, and the heat losses; and the third part models the thermal behaviour of the power block (ORC), from which can be calculated outputs such as the energy produced by the evaporator, the power generated by the turbine, the power consumed by the pump, and the heat rejected by the condenser to the environment. For a constant solar irradiation of 600W/m^2, ambient temperature of 300 K, wind speed of 3 m/s, collector width of 1.2 m, and collector length of 48 m, the heat transfer to the power block was 52.993 W, the power generated by the turbine was 3.973 W, the heat rejected by the condenser was 47.453 W, and the power consumed by the pump was 0.133 W, with an overall thermal efficiency of 7.23 %.*

Keywords*: Parabolic through, Solar energy, Organic rankine cycle, heat transfer, heat losses.*

1. INTRODUCTION

The ever increasing demand of energy for development of the society is fulfilled by a variety of energy sources. Large scale energy utilization has led to a better quality of life and faster all round development; it has also generated many critical problems (Sukhatme and Nayak, 2011). The most prominent of these is the harmful effect on the environment in various forms leading to global warming and climate change (Prisyazhniuk, 2008). At the same time, the fossil fuel resources are also fast depleting due to over exploitation. It is therefore vital to go for eco-friendly energy sources for the betterment of the future world (Alanne and Saari, 2006).Considering renewable energy sources such as solar energy, wind energy, hydropower and geothermal, is critically important in this sense as they are ecofriendly (Herzog et al. 2001). However, solar energy could be a best option for the future world because of several reasons: First, solar energy is the most abundant energy source of renewable energy and sun emits it at the rate of 3.8×10^{23} kW, out of which approximately 1.8×10^{14} kW is intercepted by the earth (Panwar eat al. 2011).Solar energy reaches the earth in various forms like heat and light. As this energy travels, majority of its portion is lost by scattering, reflection and absorption by clouds. Studies revealed that global energy demand can be fulfilled by using solar energy satisfactorily (Lewis, 2007).

The first solar plant to use an ORC system as power block refers to 1966, when the ORMAT successfully operated a pumping station in Israel and in North Africa. The plant was capable of producing 600 W of electric power, with an overall efficiency of 6%. The ORC system operated at temperatures between 90-125 ° C, using monochlorobenzene as the working fluid, using an area of the FPC collectors equivalent to 43m² (Bronicki, 1972). Solar ORCs have been studied both theoretically (Davidson, 1977; Probert et al., 1983) and experimentally (Monahan, 1976) as early as in the 1970s and with reported overall efficiencies varying between 2.52% and 7%. In (Quoilin, 2011), it is shown some about the solar ORC: (Kane et al. 2003) studied the coupling of linear Fresnel collectors with a cascaded 9- kWe ORC, using R123 and R134a as working fluids. An overall efficiency (solar to electricity) of 7.74% was obtained, with a collector efficiency of 57%. (Manolakos et al. 2007) studied a 2 kWe low-temperature solar ORC with R134a as working fluid and evacuated tube collectors: an overall efficiency below 4% was obtained. (Wang et al. 2010b) studied a 1.6 kWe solar ORC using a rolling piston expander. An overall efficiency of 4.2% was obtained with evacuated tube collectors and 3.2% with flat-plate collectors. The difference in terms of efficiency was explained by lower collector efficiency (71% for the evacuated tube vs. 55% for the plate technology) and lower collection temperature. Detailed models of such systems are also scarce in the scientific literature: (McMahan, 2006) proposed a detailed model and an optimization of the ORC cycle for solar applications, but this model was not coupled to a solar collector model; (Forristall, 2003) proposed a model of the solar collectors validated with the SEGS plants data, independent of a power cycle model. (Jing et al. 2010) developed a model of an ORC cycle using R123 as working fluid and coupled to CPC collectors: the predicted overall efficiency was about 7.9% for a solar insolation of 800 W/m^2 and an evaporating temperature of 147 C.

In this work, it is carried out the evaluation of the thermal performance of the Solar Organic Rankine System, for the city Itajubá, MG. The scientific literature about this kind of system is scarce and most of them simulated, in specifics software, the thermal behavior for the parabolic trough and the Organic Rankine Cycle separately, i.e., not considering both systems working integrated. Therefore, in this work it is developed a mathematical model to study the thermal behavior of these systems working integrated. This research is supported by the project a Hybrid Solar/Biomass Plant installed at the Federal University of Itajubá, Minas-Gerais, Brazil.

2. HEAT TRANSFER MODEL OF PARABOLIC TROUGH COLLECTOR

The receiver performance model uses an energy balance between the HTF and the atmosphere, and includes all equations and correlations necessary to predict the terms in the energy balance, which depend on the collector type, receiver condition, optical properties, and ambient conditions (Forristall, 2003). The figure 1 shows the one-dimensional steady-state energy balance for a cross-section of a receiver, with the glass envelope.

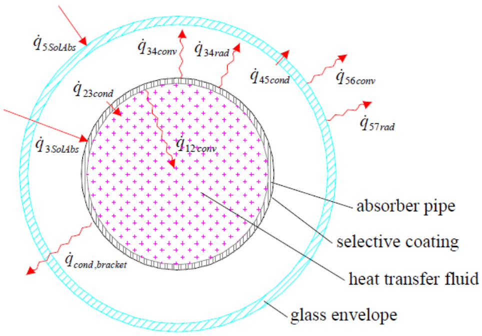

Figure 1. Steady state one-dimensional energy balance (Abdollahpour et al. 2014).

The effective solar energy absorbed by the glass envelope is $\dot{q}_{5SoLAbs}$ and absorbed by the selective coating is $\dot{q}_{3SoLAbs}$. Part of the energy absorbed into the selective coating is conducted through the absorber $\dot{q}_{23cond}$ and transferred to the HTF by convection $\dot{q}_{12conv}$; the remaining energy is transmitted back to the glass envelope by convection $\dot{q}_{34conv}$ and radiation $\dot{q}_{34rad}$. The energy from the radiation and convection then passes through the glass envelope by conduction $\dot{q}_{45cond}$ and along with the energy absorbed by the glass envelope $\dot{q}_{5SoLAbs}$ is lost to the environment by convection $\dot{q}_{56conv}$ and radiation $\dot{q}_{57conv}$ (Forristall, 2003).

2.1 Heat transfer by conduction through the wall of the absorber tube

The conduction heat transfer rate through the radial direction of the absorber pipe is obtained by applying the Fourier's law for heat conduction (Yilmaz and Soylemez, 2014).

$$\dot{q}_{23cond} = 2\pi k_{23}(T_2 - T_3)/ln(D_3/D_2) \tag{1}$$

Where: k_{23}: Thermal conductivity of the pipe absorber at $((T_2+T_3)/2)$ (W/m K);
T_2 e T_3: Temperature of inner and outer surface of the absorber pipe, respectively (K);
D_2 and D_3: Inner and outer diameter of the absorber pipe, respectively (m);

2.2 Heat transfer by convection between the absorber pipe and the heat transfer fluid

Using the Newton's law of cooling, it is possible to define the convection heat transfer from the absorber pipe to the heat transfer fluid, as such:

$$\dot{q}_{12conv} = h_1 D_2 \pi (T_2 - T_1) \tag{2}$$

$$h_1 = Nu_{D2}\frac{k_1}{D_2} \tag{3}$$

where: h_1: Heat transfer coefficient by convection at T_1 (W / m^2 K);
T_1: Heat transfer fluid temperature (K);

Nu_{D2}: Nusselt number at D_2;
k_1: Thermal conductivity of the heat transfer fluid at T_1 (W/m K).

The Nusselt number depends on the type of flow inside the absorber pipe, during typical operation condition, the fluid is in turbulent, but during off solar hours the fluid becomes transitional or laminar. For laminar fluid the Reynolds number is lower than 2300 and then the Nusselt number will be consider constant, which a value of 4.36 (Incropera and Dewitt 1990). When the fluid is turbulent, it is possible to determine the Nusselt number as:

$$Nu_{D2} = \frac{(f_2/8)(Re_{D2}-1000)Pr_1}{1+12.7\sqrt{f_2/8}\left(Pr_1^{2/3}-1\right)}\left(\frac{Pr_1}{Pr_2}\right)^2 \tag{4}$$

$$f_2 = (1.82log_{10}(Re_{D2}) - 1.64)^{-2} \tag{5}$$

Where: f_2: Friction factor in the inner surface of the receiver tube;
Pr_1 and Pr_2: Prandtl number of the heat transfer fluid estimated at T_1 and T_2, respectively;

2.3 Heat transfer between absorber and glass envelope

The heat transfer mechanism between the absorber and the glass envelope involves simultaneous radiation and convection. Convection heat transfer is a function of pressure, when the pressure is less than 1 torr, heat transfer is through molecular conduction. The heat transfer can be defined by free molecular convection (Ratzel et al.1979).

$$\dot{q}_{34conv} = \pi D_3 h_{34}(T_3 - T_4) \tag{6}$$

$$\mathrm{h}_{34} = \frac{\mathrm{k_{std}}}{\left(\frac{\mathrm{D}_3}{\ln\left(\frac{\mathrm{D}_4}{\mathrm{D}_3}\right)}+\mathrm{b}\lambda\left(\frac{\mathrm{D}_3}{\mathrm{D}_4}+1\right)\right)} \tag{7}$$

Where: D_4: Inner glass envelope diameter (m);
h_{34}: convection heat transfer coefficient for the annulus gas at T_{34} (W/m^2 K);
T_4: inner glass envelope surface temperature (K);
k_{std}: thermal conductance of the annulus gas at standard temperature and pressure (W/m K);
b: interaction coefficient;
λ: mean-free-path between collisions of a molecule (cm);

When the pressure between the absorber and the glass envelope is over 1 torr, the natural convection became stronger. Then the convection heat transfer can be determined by the following equations (Bejan, 1995):

$$q_{34conv} = \frac{2.425k_{34}(T_3-T_4)\left(PrRa_{D3}/(0.86+Pr_{34})\right)^{1/4}}{\left(1+(D_3/D_4)^{3/5}\right)^{5/4}} \tag{7}$$

Where: k_{34}: thermal conductance of annulus gas at T_{34} (W/m-K);
Pr_{34}: Prandtl number;
Ra_{D3}: Rayleigh number evaluated at D3;

Now, the radiation heat transfer between the absorber and glass envelope is estimated with Eq (8) (Incropera, 1990):

$$\dot{q}_{34rad} = \frac{\sigma\pi D_3\left(T_3^4-T_4^4\right)}{\left(\frac{1}{\varepsilon_3}+(1-\varepsilon_4)D_3/(\varepsilon_4 D_4)\right)} \tag{8}$$

Where: σ: Stefan-Boltzmann constant (W/m^2 K^4);
ε_3 and ε_4: Absorber selective coating emissivity and glass envelope emissivity, respectively;

2.4 Conduction heat transfer through the glass envelope

The conduction heat transfer through the glass envelope can also be determined by Eq. (1). It should be noticed that the temperature distribution is linear and the glass thermal coefficient is 1.04 (Touloukian and Dewitt 1972).

2.5 Heat transfer between the glass envelope and environment

Heat transfer between the glass envelope and environment includes convection and radiation. The convection heat transfer from the glass envelope to the atmosphere is the largest source of heat loss, especially if there is a wind. From Newton's law of cooling (Forristall, 2003):

$$\dot{q}_{56conv} = h_{56}\pi D_5(T_5 - T_6) \tag{9}$$

$$h_{56} = \frac{k_{56}}{D_5} Nu_{D5} \tag{10}$$

Where: T_5 and T_6: glass envelope outer surface temperature and ambient temperature, respectively (K);
h_{56}: convection heat transfer coefficient for air at $(T_5+T_6)/2$ (W/m^2 K);
k_{56}: thermal conductance of air at $(T_5+T_6)/2$ (W/m K);
N_{uD5}: average Nusselt number based on the glass envelope outer diameter.

Without wind condition, the convection heat transfer from the glass envelope to the environment will be by natural convection and the following equations can be employed; valid for $10^5 < R_{aDo} < 10^{12}$ (Incropera and DeWitt 1990).

$$Nu_{D5} = \left\{0.6 + \frac{0.387 Ra_{D5}^{1/6}}{\left[1+(0.559/Pr_{56})^{9/16}\right]^{8/27}}\right\}^2 \tag{11}$$

Where: Ra_{D5}: Rayleigh number for air based on the glass envelope outer diameter, D5;
Pr56: Prandtl number for air at T_{56};

With wind: If there is wind, the convection heat transfer will be forced convection and the Nusselt number is estimated with the following correlation (Incropera and DeWitt 1990).

$$Nu_{D5} = C Re_{D5}^m Pr_6^n \left(\frac{Pr_6}{Pr_5}\right)^{1/4} \tag{12}$$

The temperature difference between the glass envelope and sky causes a radiation transfer, that can be obtained by Eq. (13). It is assumed that the cover is a small convex gray object in a large black body cavity, the sky:

$$\dot{q}_{rad} = \sigma D_5 \pi \varepsilon_5 (T_5^4 - T_7^4) \tag{13}$$

Where: D_5: Glass envelope outer diameter (m);
ε_5 : Emissivity of the glass envelope outer surface;
T_7: effective sky temperature (K). It is approximated 8 ^{o}C below ambient temperature (Abdollahpour et al. 2014).

3. SOLAR IRRADIATION

It is knowing, that is this kind of system, the solar energy is the primary source to generated work. For this reason, it is important to know the among of energy absorbed by the glass envelope and by the receiver pipe. The solar absorption in the glass envelope is a heat generation phenomenon and is a function of the glass thickness. An optical efficiency is estimated in order to calculate the solar absorption. So, the solar absorption in the glass becomes (Forristall, 2003):

$$\dot{q}_{5SolAbs} = \dot{q}_{si}\eta_{env}\alpha_{env} \tag{14}$$

The equation for the solar absorption in the absorber becomes (Forristall, 2003):

$$\dot{q}_{3SolAbs} = \dot{q}'_{si}\eta_{abs}\alpha_{abs} \tag{15}$$

Where: $\dot{q}_{si}$: Solar irradiation (W/m^2);
η_{abs} and η_{env}: Effective optical efficiency at absorber and the glass envelope, respectively;
α_{abs} and α_{env}: Absorptance of absorber and the glass envelope, respectively;

4. THERMAL ANALYSIS OF THE ORGANIC RANKINE CYCLE

For this model were used equations obtained from the mass and energy balance with these, it is possible to describe the performance of each components of the Organic Rankine Cycle (ORC). The energy balance, based on the first law of thermodynamics, is applied to each components of the system and considering the following hypothesis: (i) all the processes operated under steady state; (ii) the evaporator, the expander, the condenser and all pumps are adiabatic devices; (iii) pressure losses any of the ORC components and its piping connection negligible; (iv) there is no change in potential and kinetic energy of the working fluid. In the figure 2, is shows each component for the Organic Rankine cycle:

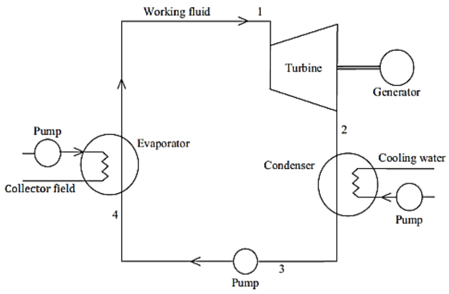

Figure 2. Thermal resistance model, adapted from (Madhawa et al. 2007).

In the present work, the intermediate heat transfer fluid is water. It is heated in the collectors, and after that, it pass through the evaporator, heating a secondary fluid (organic working). Vapor generated at the evaporator drives a turbine. The working fluid leaving the turbine is then condensed and pumped back to the evaporator (Madhawa et al. 2007).

Evaporator: Since the heat losses in heat exchangers are negligible, the amount of heat added to the working fluid equals the heat extracted from the heat source (Lukawski, 2009):

$$\dot{Q}_{evap} = \dot{m}_{ft}(h_1 - h_4) \tag{21}$$

Where: $\dot{m}_{ft}$: Working fluid mass flow (kg/s);
h_1: Enthalpy of the working fluid at the evaporator inlet (J/kg);
h_4: Enthalpy of the working fluid at the evaporator outlet (J/kg);

Turbine: its performance is defined as a function of the isentropic efficiency (Lukawski, 2009):

$$\eta_{isenT} = \frac{\dot{W}_{real}}{\dot{W}_{isen}} = \frac{h_1 - h_2}{h_1 - h_{2s}} \tag{22}$$

Where: $\dot{W}_{isen}$ *and* $\dot{W}_{real}$: Ideal and actual power generated by the turbine, respectively (W);
h_{2s} *and* h2: Isentropic and actual enthalpy of the working fluid at the turbine outlet (J/kg);

Condenser: In the case of the condenser, the energy balance produces the following expression (Lukawski, 2009):

$$\dot{Q}_{cond} = \dot{m}_{ft}(h_2 - h_3) \tag{23}$$

Where: h_3: Enthalpy of the working fluid at the condenser outlet (J/kg);

Pump: energy consumed by the feed pump can be calculated considering its isentropic efficiency (Lukawski, 2009):

$$\eta_{isenP} = \frac{\dot{W}_{isenP}}{\dot{W}_{realP}} = \frac{h_{4s} - h_3}{h_4 - h_3} \tag{24}$$

Where: $\dot{W}_{realP}$ *and* $\dot{W}_{isenP}$: Actual and ideal power consumed by the pump, respectively (W);
h_{4s}: Isentropic Enthalpy of the working fluid at the pump outlet (J/kg);

ORC thermal efficiency: The thermal efficiency of the cycle is defined (Lukawski, 2009):

$$\eta = \frac{\dot{W}_{real} - \dot{W}_{realP}}{\dot{Q}_{evap}} \tag{25}$$

5. RESULTS AND DISCUSSION

In this work was developed a mathematical model to study the thermal behavior of a Organic Rankine Cycle integrated with a parabolic trough collector for the ambient condition of Itajubá, Minas-Gerais/Brazil, using *Matlab* as computational tool. The figure 3 shows the behavior of the solar irradiation and the outlet temperature of the collector for the condition analyzed. Some of the parameters used in this work were taken from a system installed in Itajubá city, while the weather information was taken from the UNIFEI meteorological station. In this work it was modelled a solar field with the follow configuration: 4 parabolic trough collector forming two circuits with a collector length of 48m, the width of the collectors is 1.2m, the outlet and inlet diameter of the receiver pipe are 3.4cm and 2.5cm, respectively, while the outlet and inlet diameter of the glass cover are 5cm and 4.6cm.

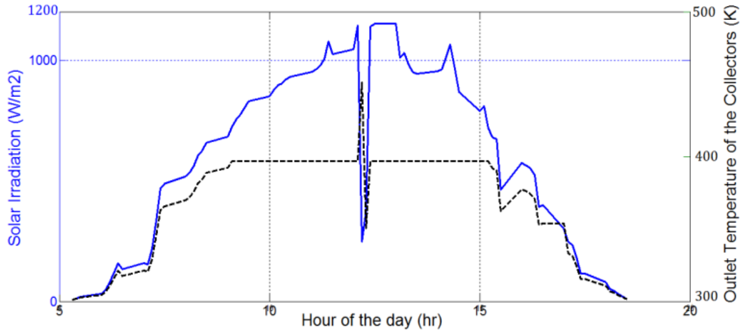

Figure 3. Behavior of the solar irradiation and the outlet temperature of the collector for the city of Itajubá.

The figure 3, allows one to observe the behavior of the solar irradiation and the outlet temperature of the parabolic trough collector for 01/01/2015. For this day in Itajubá, there is solar irradiation since the 5:30 AM to 18:50 PM; the solar irradiation initially increases until achieve its maximum value at 12:50 of 1151 W/m^2, after this point it is observed a decrease in its value, from 12:20 to 12:30 due to the presence of clouds over the weather station. The last value registered by the meteorological station was 9 W/m^2 at 18:50. It is also observed that the outlet temperature of the collector initially increase with the increase of the solar irradiation but after a point, it remains constant until reach a maximum value temperature of 451.3 K, later the temperature decrease follow the behavior describe by the solar irradiation; for this day the least value of temperature reached in the collector was 301.2 K. In the model developed in this it was assumed that the maximum temperature has to be smaller than the organic fluid critical temperature.

Figure 4 shows the heat loss by convection from the glass enveloped to the ambient and the heat loss from the glass enveloped to the sky. The collector parameters are: outlet emissivity of 0.94, envelope transmissivity of 0.96, envelope absorptivity of 0.04, envelope emissivity of 0.86, coating absorptivity of 0.96 (Quoilin et al. 2011). Wind speed was assumed equal to 3 m/s, ambient temperature of 27 °C, with no vacuum between at the receiver.

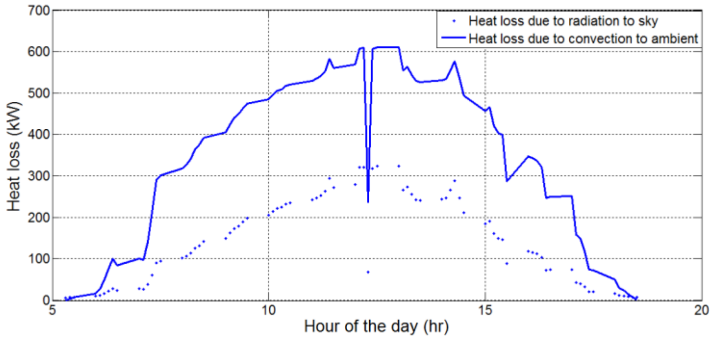

Figure 4. Behavior of the heat loss by convection and radiation from the glass cover to the surroundings.

As shown in Figure 4, the thermal losses from the collector receiver are functions of operating temperature. These depending on the receiver's configuration and operational conditions. For the conditions evaluated, the convection loss from the glass enveloped to the ambient is the largest, the next is the radiation loss from the glass enveloped to the sky. These heat losses increased their valued since the solar irradiation increased, once it reached the maximum value of irradiation, the slope of the curve change, decreasing until the sunset for the day analyze. The largest and lowest radiation heat loss was of 321.3 kW at 12:50 and 6.2 kW at 5:30, respectively; while the largest and the lowest convection heat loss was of 608.3 kW at 12:50 and 0.8 kW at 18:50, respectively.

In the figure 5 it is presented the behavior of the turbine inlet pressure, the power generated by the system and the power consumed by the pump. For this analyze was considered, a turbine efficiency of 0.8, pump efficiency of 0.8, generator efficiency of 0.9 and condensation temperature of 40 °C, the organic fluid used in the model was the R245fa.

Figure 5, shows the effect of the turbine inlet pressure on the power generated by the system and the power consumed by the pump. From this figure it is possible conclude that the power generated by the system increase as the turbine inlet pressure increase, but for this situation is important note that the power consumed by the pump to generate the required pressure also increase, causing a decrease in the cycle efficiency. The largest and lowest values of pressure reached in the inlet of the turbine were 1532.1 kPa and 0 kPa, respectively. After, the maximum turbine inlet pressure is achieved, there are two period of the day where is possible observed little changes in the pressure; for the maximum value of the power generated by the system of 9.02 kW, the power consumed by the pump is also maximum 0.407 kW. It is observed that the maximum power generated is reached for the maximum values of irradiation; the power increased follow the solar irradiation, but once it reaches the largest value, the power has little variation; later they decrease with the decrease of the irradiation, cause by the presence of clouds over the solar field area.

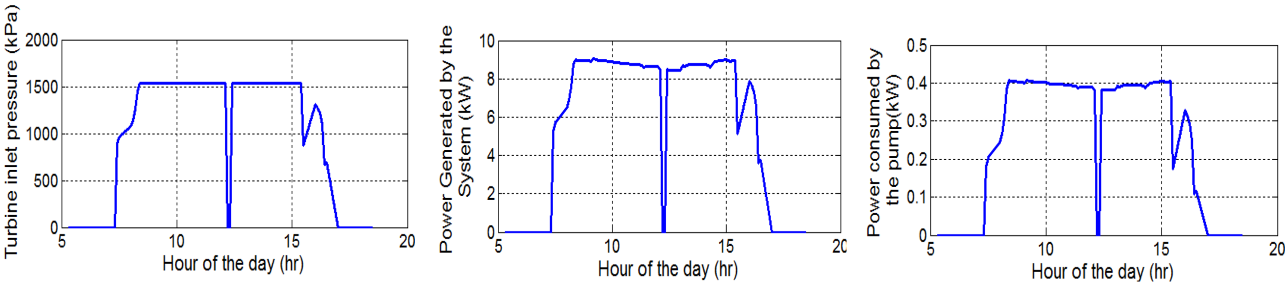

Figure 5. Behavior of the turbine inlet pressure, turbine power generated and the pump power consumed.

In the figure 6 it is show the ORC and the system thermal efficiency for each hour of the day 01/01/2015. In the figure 6, it is observed that the maximum ORC and system thermal efficiency are reached for the maximum values of irradiation. From this analyze is observed that initially the ORC efficiency increase follow the solar irradiation but once it reachs the largest value, the efficiency remains constant, later it decrease, cause by the presence of clouds over the solar field area, once the clouds are out of the area, the value of this parameters increase. The largest value of efficiency was 11.35 % and the lowest was 0 % (no power generation). For the system efficiency is observed that initially this is zero, later this increased to achieve its maximum value of 2.4 %.

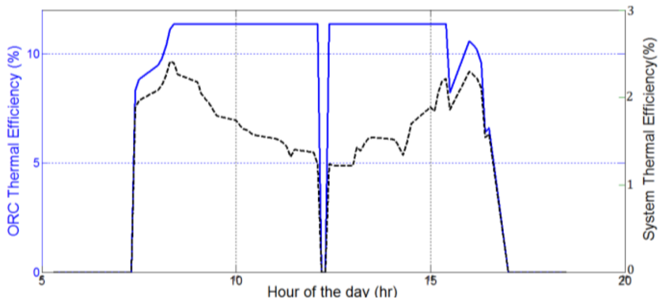

Figure 6. Behavior of the ORC and system thermal efficiency.

6. CONCLUSIONS

In this work, it was developed a mathematical model to study the thermal behavior of a Parabolic trough collectors working with an ORC system suitable for distributed power generation. This model can be used simulate the behavior of this systems, also helps to select the proper operating condition and detect the possible problems in the system design. In this paper it was examined the effect of system characteristics, such as heat losses from the glass envelope to the surroundings, collector temperature, turbine inlet pressure, power consumed by the pump on the system efficiency.

The power output and the ORC efficiency all decrease with the decrease of the solar irradiation, allowing one to conclude that this kind of system is quite sensitive to changes of the solar irradiation. This research allowed to determine that despite it is possible install small systems like this in cities with unfavorable weather conditions, during the winter and periods of rain, the system will not work properly, only achieving very small efficiencies.

Despite the R245fa is not the fluid with the best characteristics, it was used in this model because it has low ODP (Ozone Depletion Potential) and GWP (Global Warming Potential), making it environment friendly. For the condition evaluated in Itajubá, it was obtained a maximum ORC efficiency of 11.35 % and power generation of 9.02 kW. It was possible to determine that the operation temperature has an important influence on the heat losses. So, for this kind of technology, it is important to use materials that help to reduce this thermal loss assuring a good performance.

7. ACKNOWLEDGEMENTS

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