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Methodology for the Thermal Characterization of Linear Fresnel Collectors: Comparative of Different Configurations and Working Fluids

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Abstract. Linear Fresnel collectors are becoming an attractive option to generate electricity from solar radiation. This paper is focused in the thermal performance of Fresnel collectors working with different heat transfer fluids: synthetic oil, water-steam, molten salt and air, also comparing the results of the Fresnel technology with those obtained in reference parabolic trough loops. Although there are two basic designs of the Fresnel receiver: multi-tube and single-tube with secondary concentrator, this work only studies in depth the single-tube option, as this design is more suitable for a proper comparison with parabolic troughs. The receiver in parabolic troughs has been modeled as an evacuated tube with a selective coating and a glass cover. For Fresnel receivers it has been simulated two different configurations: non-evacuated receiver, with a glass window at the cavity aperture and evacuated receiver, characterized by a tube with a glass cover and a selective coating.

INTRODUCTION

It can be found in the technical literature many comparative studies between Linear Fresnel Collectors (LFC) and Parabolic Trough Collectors (PTCs) [1]. The general conclusion is that the LFC annual efficiency is lower than that of the PTC, so Fresnel systems will only be competitive if they are capable of reducing investment costs enough compared to parabolic trough. Although this is true, this paper attempts to highlight the need for a proper characterization of the Fresnel receiver because it usually presents a better thermal performance than the PT receiver.

There are already several LFC power plants installed worldwide [2]. A detailed analysis of them shows that both the receiver design and the working fluid in the solar field have more degrees of freedom than in the case of PTCs, which often use an evacuated receiver cooled by synthetic oil.

Regarding to the receiver design, it can be found two basic configurations: the multi-tube (used in Kimberlina plant [2]) and the single-tube (used in Puerto Errado plant [2]). The thermal model described in this paper is referred to this last design, since, as said in the abstract, it can be easily compared to PT receiver.

Besides the receiver, there are also several alternatives for the Heat Transfer Fluid (HTF). Some plants used synthetic oil for coupling to an Organic Rankine Cycle (ORC), although the more extended option is the use of water-steam. Recently, it has been built a new prototype, using molten salts [3].

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In conclusion, the large number of alternatives in the design of Fresnel receivers requires of a detailed study of the thermal performance of those receivers, using different working fluids. This study can be accomplished by the thermal model described below.

METHODOLOGY FOR THE THERMAL COMPARISON OF FRESNEL COLLECTORS USING DIFFERENT WORKING FLUIDS

There are several thermal models applied to linear solar receivers. Most of them are referred to parabolic trough collectors [4, 5], although some recent ones perform the thermal behaviour of both multi-tube [6] and single-tube [7] Fresnel receiver. The model presented in this section is also based on energy balances, but presents the following features compared to other models: first, the model can be applied to both single-phase and two-phase fluid, using the corresponding correlations for the pressure drop and heat transfer in each case. Second, the heat transfer in the CPC (Compound Parabolic Concentrator) cavity is studied in depth. And third, the model calculates the energy and exergy balance for each of the configurations studied.

Thermal Model of the Single-Tube Receiver: Evacuated and Non-Evacuated Designs

The thermal model proposed is based on a two-dimensional, steady-state energy balance, in the receiver cross section and along its length. As seen in figure 1, two basic receiver single-tube Fresnel designs are used.

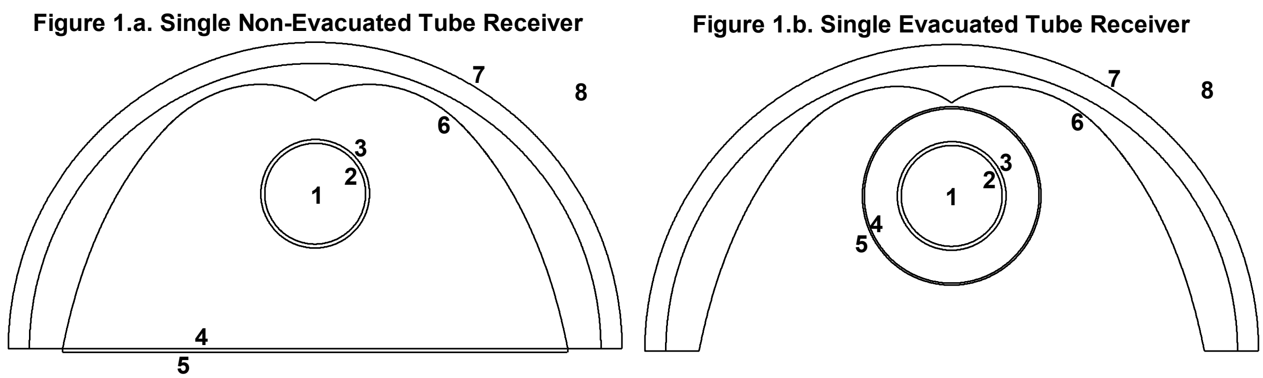

FIGURE 1. Cross section of a single-tube Fresnel receiver (1.a. Non-Evacuated; 1.b. Evacuated). (1) Heat transfer fluid; (2) absorber tube inner surface; (3) absorber tube outer surface; (4) glass inner surface; (5) glass outer surface; (6) secondary reflector (CPC); (7) outer insulation casing surface; (8) environment

Figure 1.a shows the single-tube non-evacuated receiver, with a glass window at the cavity aperture; this receiver is usually characterized by a tube coated with a black, non-selective paint, since vacuum cannot be ensured inside the cavity, being of complex geometry. Figure 1.b shows the single-tube evacuated receiver, provided by a glass cover and a commercial selective coating for the absorber, similar to that used in the PTCs.

As shown in figure 1, although the subscripts 4 and 5 refer to the inner and outer wall of a glass cover, respectively, this cover is flat and it is located at the aperture, if non-evacuated, while it is an absorber tube concentric cylinder, if evacuated. These different configurations introducing substantial differences in the heat transfer, which are summarized below.

- In the case evacuated, the heat is transferred by convection and radiation from the absorber tube to only the concentric glass cover. The correlations that characterize this heat transmission are widely referenced in the literature, as it is a standard geometry.
- The radiation heat transfer in the cavity is modelled in a similar way in both cases, considering the infrared radiosity of each of the walls that comprise the cavity. In the non-evacuated design, these walls are the absorber tube, the CPC and the glass flat cover at the aperture. In the evacuated design, those walls are the glass tube (with a greater diameter than the absorber tube), the CPC and the opening aperture, which is considered as a black body (thermal emissivity equal to 1). So, the view factors and the infrared emittances are different in each design.

- The convection heat transfer in the cavity also changes from one design to another. When the cavity is closed, the model described in [8] is used. For facing-down, opening cavities, there are several correlations, and it has been employed the one described in [9].

  Figures 2 and 3 show the thermal resistances schemes of both designs, in which there are evident differences.

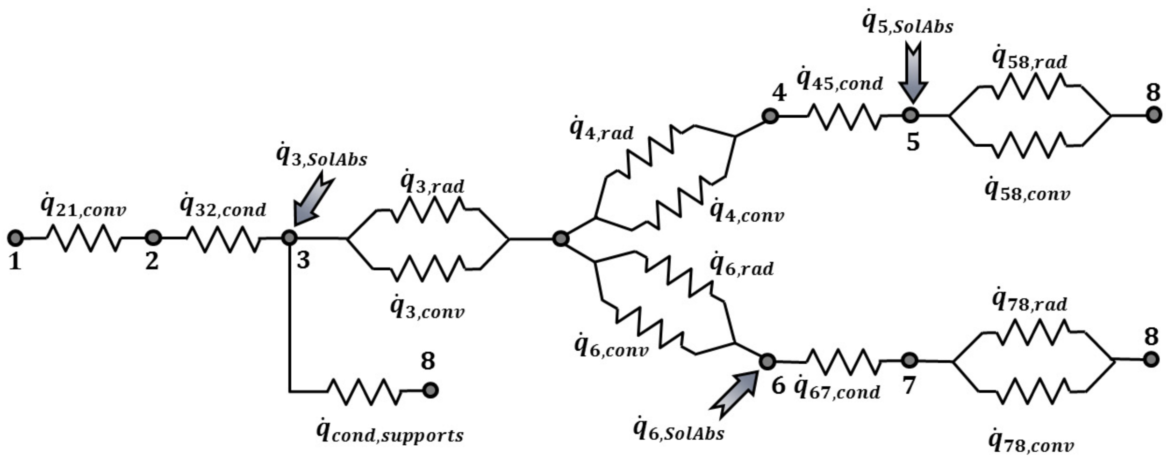

FIGURE 2. Thermal resistance model for non-evacuated single tube receiver with glass window at the cavity aperture

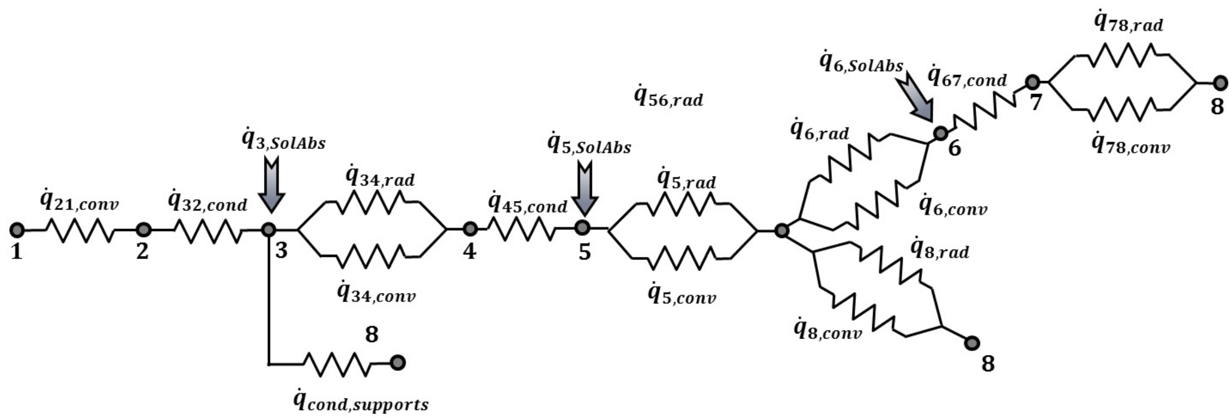

FIGURE 3. Thermal resistance model for evacuated single tube receiver

The detailed description of the correlations used, the energy and exergy efficiency definition, and the model validation are analysed in [10].

Design Parameters and Thermal Characterization of the Parabolic Trough and Fresnel Loop

In order to fix a similar framework for comparison, the absorber tube diameter is equal in both cases (OD = 70 mm). This diameter is usually employed in commercial PTC models [11], and it is also the diameter of the absorber tube in Solnova Fresnel plant [12].

The selected design point for this comparison has been fixed on the 21st June at solar noon (12:00 solar time). At this point, DNI (Direct Normal Irradiation) = 900 W/m^2 and ambient temperature is equal to 25ºC. For LFC with north-south orientation, the longitudinal and transversal angles are 13.21º and 0º, respectively, yielding to a mean concentration on the receiver's plane of 131.58 (see figure 4, section 2.3). For PTC with the same orientation, the incidence angle is equal to 13.21º (equal to longitudinal angle of Fresnel), so the incidence angle modifier equals 0.9616 [5]. The effective concentration (regardless optical parameters) based on these data is 79.12.

The design of the PTC loop for each of proposed fluid is determined according to references found in the literature review. The PTC model for all the cases is the SkalET model, with different length according to the thermal requirements.

When using water/steam, the Real Diss project have been taken into account as a reference [13]. The recirculation operating mode has been adopted. Every loop is composed of 6 collectors of 150 meters long: 4 collectors for water preheating and evaporation + 2 collectors for superheating steam. Inlet water temperature and pressure are equal to 280 ºC and 112 bar, respectively, yielding to an outlet steam conditions of 550 ºC and 103 bar. For this nominal conditions, the superheated steam produced at every row is 1.37 kg/s

The layout, for the case of using molten salts, is defined considering ENEA's experience in this technology [14]; each loop consists of six collectors, and each collector has eight modules of 12.27 meters long. The HTF temperature ranges from 290 °C to 560 °C. The inlet loop pressure is 10 bar and the mass flow per loop is 4.6 kg/s.

Regarding Therminol VP-1, the design scheme is selected considering a commercial PTC plant of synthetic oil; there are four collectors ET-150 in series per loop. Each ET-150 collector consists of twelve modules of 12.27 meters long. The oil mass flow is 8.5 kg/s, the inlet pressure is 30 bar, and the HTF temperature increment is equal to 100 ºC (from 293 ºC to 393 ºC) [5].

Finally, the loop scheme for air as HTF, is defined based on the experience of the PSA [15]; each loop consists of two collectors, and each collector has six modules of 12.27 meters long. The temperature ranges from 310ºC to 525ºC, the mass flow is 2.08 kg/s and the inlet pressure is 90 bar.

It is important to note that tube thickness in the case of using air or water/steam is greater (5.6 mm compared to 2.5 mm in case of using synthetic oil or molten salts), as working pressure is also greater.

TABLE 1. Parabolic trough collector loop: nominal thermal performance for each HTF technology

Thermal parameters	Therminol VP1	Solar Salt	Water/Steam	Air
Receiver energy efficiency (%)	76.82	71.05	75.94	71.68
Receiver exergy efficiency (%)	39.68	41.73	39.20	40.48
Collector energy efficiency (%)	70.60	65.3	69.79	65.87
Collector exergy efficiency (%)	36.46	37.4	36.06	38.35
Heat gain (MW_{th})	2.09	1.934	3.06	0.49
Heat loss (kW_{th})	183.60	340.54	345.9	80.88
Pressure drop (bar)	6.11	1.13	9	2.71

Table 1 shows the thermal performance of each HTF technology in the design point chosen. It is noted that the energy efficiency is higher in the technology operating at a lower average temperature (VP1) and it decreases as average temperature is increased. The exergy efficiency varies with the temperature in the opposite way although, in this case, the pressure is also taken into account, being penalized the case with higher pressure drop (water-steam).

Fresnel loops at nominal conditions are designed so that, for each fluid and temperature increment, the mass flow is adapted in order that the velocity is approximately the same than in PT. Thereby, as the diameter is the same, the tube refrigeration is approximately the same, allowing the comparison of the receiver performance in similar conditions.

For evacuated tubes, thermal emittance is the same that in PT, as the selective coating is the same [5], while for non-evacuated tubes there is not any commercial selective paint that presents a long term stability in air. According to recent investigations [15], a proper value for the thermal emittance is 0.4.

It has been obtained all the thermal parameters listed in table 1 for both single-tube non-evacuated receiver with glass window (table 2) and single-tube evacuated receiver (table 3).

TABLE 2. LFC with single-tube non-evacuated windowed receiver: nominal thermal performance for each HTF technology

Thermal parameters	Therminol VP1	Solar Salt	Water/Steam	Air
Receiver energy efficiency (%)	80	73.36	77.84	73.9
Receiver exergy efficiency (%)	43.91	44.92	40.41	41.46
Collector energy efficiency (%)	61.41	56.31	65.13	56.72
Collector exergy efficiency (%)	33.7	34.48	31.01	31.82
Heat gain (MW_{th})	1.99	1.812	3.17	0.4901
Heat loss (kW_{th})	274.9	419.8	436.4	111.5
Pressure drop (bar)	3.371	0.547	2.908	1.585

TABLE 3. LFC with single-tube evacuated receiver: nominal thermal performance for each HTF technology

Thermal parameters	Therminol VP1	Solar Salt	Water/Steam	Air
Receiver energy efficiency (%)	88.56	85.7	86.22	85.94
Receiver exergy efficiency (%)	45.89	48.9	44.79	48.27
Collector energy efficiency (%)	67.98	65.78	66.17	65.96
Collector exergy efficiency (%)	35.22	37.53	34.38	37.05
Heat gain (MW_{th})	2.092	1.954	3.134	0.488
Heat loss (kW_{th})	7.216	404.1	280.7	56.11
Pressure drop (bar)	3.199	0.5051	2.614	1.35

The first conclusion from tables 2 and 3 is that the energy and exergy efficiencies in a particular receiver design, but working with different fluids and temperature increments, follow the same trend as in PT. Considering the energy efficiency, the most favored technology is the one operating at a lower mean temperature while, from the point of view of the exergy efficiency, it is favored to work at higher temperatures and lower pressure drop, i.e, the molten salts.

It is also noted that the best receiver thermal performance is achieved in the case of evacuated receiver, regarding to non-evacuated, and both are higher than the PT receiver. If one focuses on the case evacuated, it can be seen that the difference in the receiver energy efficiency is more than 10 percentage points regarding to PT receiver. The global collector efficiencies in both technologies –Fresnel evacuated and PT- are almost the same because of Fresnel optics penalty, even the PT exceeds the Fresnel in the case of synthetic oil at lower temperature.

As the mean concentration in the Fresnel tube is almost twice that in PT tube (131.78 compared to 79.12, as cited above), and the Fresnel receiver designs present higher energy efficiencies than the PT receiver, it is required about half of the Fresnel collector length to get the same temperature increment than in PT. This yields to an improvement in the Fresnel exergy efficiency compared to PT. As in the case of the energy efficiency, the global collector exergy efficiency is the same (evacuated design), even worse (non-evacuated design) than in PT, due to the lower optical efficiency.

The thermal model presented in this section is a complex model to analyze the receiver performance under different design parameters and working conditions. However, for an annual analysis, it is necessary to obtain a simplified model, as described below.

Annual Simulation: Optical and Thermal Characterization

This last section is devoted to the annual simulation of a Fresnel loop working with molten salts and a temperature increment of (290-560) °C, in Seville (Spain). This annual performance is compared to the result calculated by the program SAM (Solar Advisor Model) [17]. The specific characteristics of the loop are summarized in table 4. Geometrical and optical parameters are very similar to those of Solnova [12]. The main difference is the loop length, as Solnova operates with water-steam in longer loops.

The annual simulation requires of both optical and thermal characterization of the collector in every time step. Hourly data for DNI and ambient conditions of a typical meteorological year in Seville were provided from SAM database [17].

The optical characterization has been done by means of the sun position and the mean concentration in the receiver. The sun position is defined in the center of every hour interval by the calculation of the solar azimuth and zenith angle [18].

TABLE 4. Geometrical data of the Fresnel loop working with Solar Salt as HTF

Geometrical parameter	Value
Loop length (m)	300
Module length (m)	42.8
Number of modules	7
Receiver height above the mirror field (m)	7.4
Mirror field width (m)	16
Reflective surface width (m)	12
Filling Factor	0.75
Mirror width (m)	0.75
Mirror length(m)	5.35
Receiver tube diameter (m)	0.07
CPC cavity aperture (m)	0.3255

The mean concentration values for the case of LFCs have been obtained by an own developed program based on Montecarlo raytracing. This program has already been explained and validated in other studies [19-22]. The mean concentration values, as function of the longitudinal and transversal incidence angles, are showed in figure 4. This optical calculation has been made accounting for the parameters of Solnova solar field, summarized in table 4.

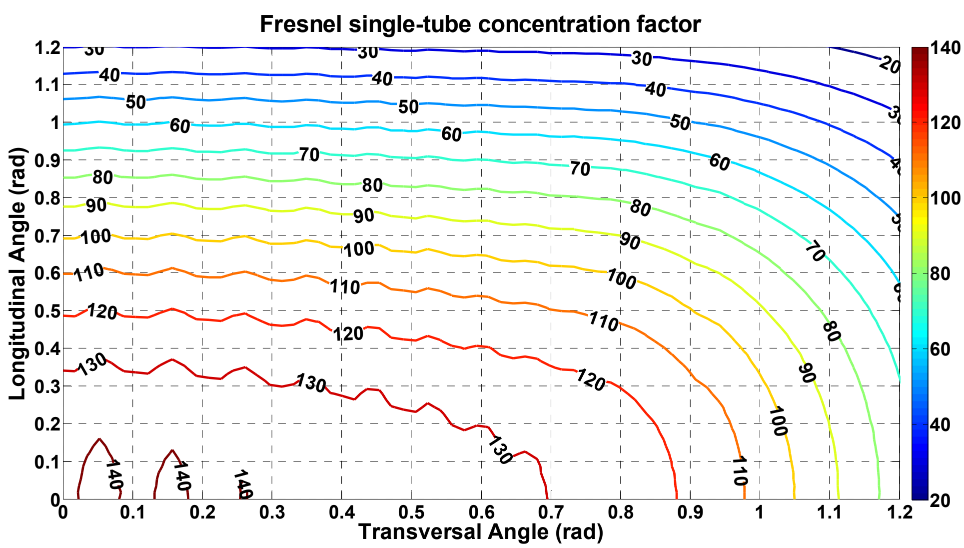

FIGURE 4. Mean concentration on the receiver's plane for Solnova solar field, N-S orientation, as a function of the longitudinal and transversal angles

For the annual simulation, the receiver thermal performance is simplified by means of a polynomial expression that calculates the heat loss as a function of the difference between the receiver tube wall temperature and the ambient temperature, the wind speed and the concentrated flux on the receiver. These equations have been obtained by linear regression on a large sample of simulations under different conditions for both evacuated (equation 1) and non-evacuated (equation 2) Fresnel receiver.

$$\dot{q}_{l,evacuated}\left(\frac{W}{m}\right) = \left(0.014 \cdot v_{wind} - 1.09\right) \cdot \Delta T + 0.01716 \cdot \dot{q}_i + 0.004657 \cdot \Delta T^2 - 1.276 \cdot 10^{-6} \cdot \dot{q}_i \cdot \Delta T - 1.177 \cdot 10^{-7} \cdot \dot{q}_i^{\,2} \tag{1}$$

$$\dot{q}_{l,non-evac}\left(\frac{W}{m}\right) = \left(0.0212 \cdot v_{wind} - 0.6049\right) \cdot \Delta T + 0.0135 \cdot \dot{q}_i + 0.0072 \cdot \Delta T^2 - 1.046 \cdot 10^{-6} \cdot \dot{q}_i \cdot \Delta T - 9.582 \cdot 10^{-8} \cdot \dot{q}_i^{\,2} \tag{2}$$

In the above equations, v_{wind} is the wind velocity (m/s), ΔT (ºC) is the temperature difference between the tube wall and the ambient and $\dot{q}_i$ (W/m^2) is the concentrated flux impinging on the absorber tube. Regarding to other models [23], these simplified equations present the following features: heat loss depends on the wall temperature instead of the fluid temperature, so it can be used for different fluids (single-phase or two-phase), as convection heat transfer and pressure drop of the fluid is calculated with the particular equations for each case; the concentrated flux on the receiver takes into account not only the DNI, but also the sun position by means of the mean concentration (see figure 4); it can also be noticed that the wind velocity only appears in the first term, which depends linearly on the temperature difference, as the wind mainly affects convective heat loss.

SAM thermal models are particularized for water- steam and molten salts, but they cannot be used to study other fluids. There are two different models. First, a polynomial heat loss model, which is a function of the temperature difference between the fluid and the ambient. This equation does not take into account the concentrated incident flux or the wind velocity, although the latter parameter can be considered in a second polynomial equation; the default behaviour modelled in System Advisor does not include this sensitivity. Second, there is an evacuated receiver model based on a thermal resistances scheme, which allows changing the optical parameters of the different materials (i.e. the solar absortance or the thermal emittance). This model seems to be similar to the one proposed by Forristall [4]. Nevertheless, it has not been taken into account that the evacuated tube is not exposed to the environment, but within a CPC cavity, so its thermal performance is better than the PT receiver.

This last discrepancy becomes important when performing the annual analysis. As the objective is to compare the thermal performance, the optical efficiency matrix, function of the longitudinal and transversal incidence angles, has been imported into SAM from the own raytracing program [19-22].

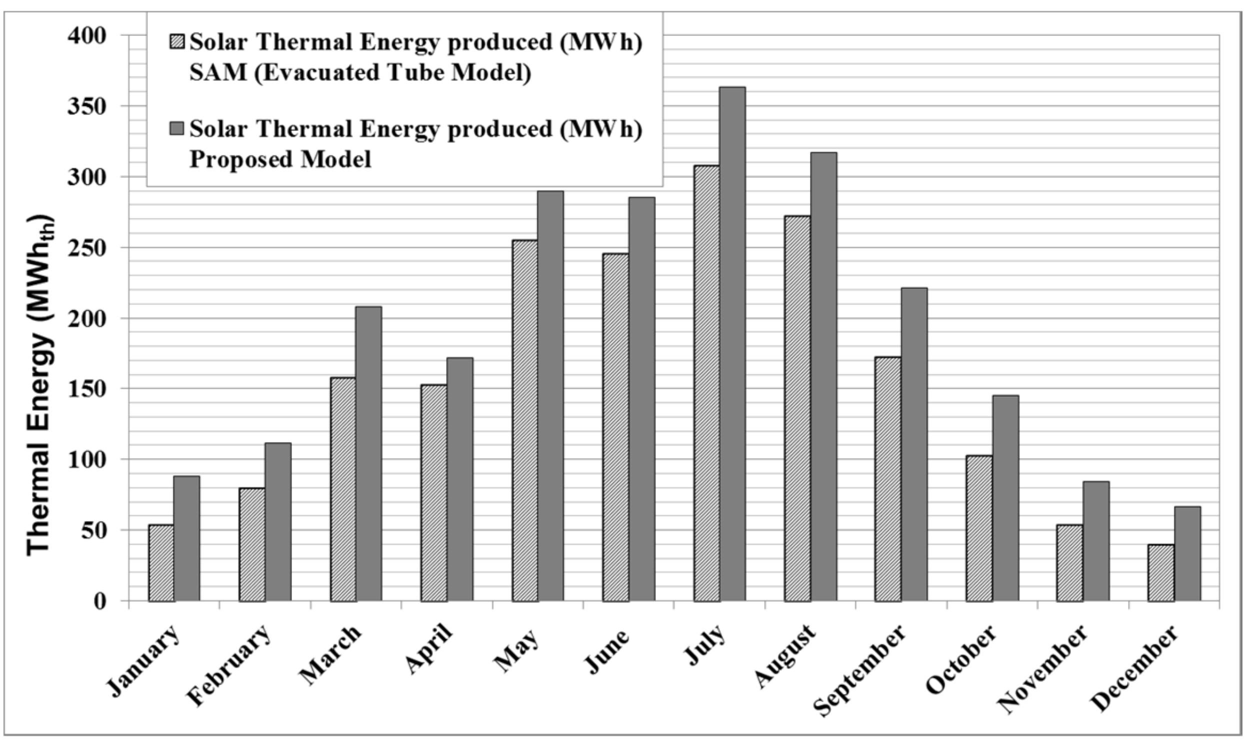

FIGURE 5. Monthly thermal energy produced by a Fresnel loop working with molten salts. Results from SAM and from the proposed model.

As shown in figure 5, the thermal energy produced by the Fresnel loop is substantially greater in the own model than in the SAM model; for the annual production, this difference becomes 20% ($1.9 \cdot 10^3$ MWh compared to $2.35 \cdot 10^3$ MWh). This is because, as already said, the SAM evacuated model underestimates the thermal performance of the Fresnel evacuated receiver, not considering the improvement introduced by the effect of the CPC cavity.

CONCLUSIONS

It has been performed a thermal model of the Fresnel receiver, which takes into account the effect of CPC cavity for two different versions: evacuated single-tube and non-evacuated single-tube with a glass window at the aperture. The model presents different characteristics depending on the design.

These thermal models can be used to compare the performance of Fresnel collectors refrigerated by different fluids. It has been shown that the energy efficiency is higher at a lower mean working temperature, while the exergy efficiency is favored by high temperatures and reduced pressure drop (in these terms, the better fluid is molten salt).

It has been also highlighted that the receiver having higher performance is the Fresnel evacuated, then the Fresnel non-evacuated, and finally the PT receiver.

For the annual analysis, the model is simplified by a linear regression taking into account the concentrated incident flux on the receiver, the temperature difference between the tube wall and the ambient, and the wind velocity. By employing the tube temperature instead of the fluid temperature, it has decoupled the thermal losses model from the convective heat transfer to the fluid, allowing the study of different fluids. It is also necessary to consider the incident flux, as the heat absorbed by the different elements of the receiver (CPC, glass and tube) becomes an important factor. Finally, it has been found that the wind velocity only affects the term that depends linearly on the temperature difference tube-ambient, i.e., the convective heat loss.

The comparison of the proposed model with a conventional evacuated tube model has highlighted the need to consider a particular model for the Fresnel receiver, which takes into account the effect of CPC cavity in the thermal receiver performance, since if it is used an evacuated tube directly exposed to the environment, the thermal output of the receiver is underestimated.

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