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Alternative Methodology for Modeling Direct Steam Generation in Parabolic Collectors: A Study Case in Northweast Mexico

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Abstract:

The possible implementation of direct steam generation (DSG) in parabolic troughs in the Northweast region of Mexico is very limited, since there are no analyses of this type to try to define a first proposal for a solar plant. Recently, a methodology has been implemented to eliminate the convective coefficient h in order to model the DSG process in a simpler way. This methodology has been validated with experimental data from the DISS results from PSA. This methodology allows to evaluate the temperature evolution along the loop, the pressure drop and the flow pattern in a short time and with low computational resources. This allows generating the first evaluation of a solar field with parabolic troughs. Additionally, by applying a new practical efficiency limit for the conversion of solar radiation into work, the exergetic efficiency of the installation is evaluated. The results show that this first evaluation, seems to be adecuate acording to the results suggested in the open literature.

Keywords:

Direct steam generation modeling, parabolic collector, flow pattern, heat transfer, boiling

1. Introduction

The consequences of the socioeconomic model based on the consumption of fossil fuels have been so dramatic in recent years that no one denies that the current energy model is in crisis and, therefore, in the process of a fast and unstoppable transformation [1, 2]. It is urgent to shift from the current centralized system based on fossil fuels toward a system that is distributed, and based on renewable energies [3]. Among the energies that should make up an important part of the world's energy mix, solar energy undoubtedly stands out. Solar energy is clean, environmentally friendly, and freely available over the planet.

Concentrating solar power (CSP) plants, also known as solar thermal, is a commercial alternative to non-renewable energy sources such as oil, coal and nuclear power for the production of electricity, mainly with the use of parabolic trough solar fields. Most CSP facilities to date have implemented synthetic or mineral oil as a working fluid in the solar field. The main limitation of thermal oil is that at 400°C it begins to degrade, thus imposing a limitation on the maximum operating temperature of the thermodynamics power cycle, generally a steam Rankine cycle [4–6].

Alternatively, thanks to DISS loop at Plataforma Solar de Almería [7–10], it has been demonstrated that it is possible to work with the concept of direct steam generation (DSG), in which steam is generated directly in the solar field and then redirected to the power block turbine. The DSG has some technical advantages that must be considered [5, 7, 11]:

- There is no danger of contamination or fire due to the use of thermal oil
- Possibility of raising the maximum temperature of the Rankine cycle above 400°C, which is the limit imposed by the thermal oil currently used
- Reduction of the size of the solar field, thus reducing the investment cost
- Reduction of operating and maintenance costs, as thermal oil based systems require a certain amount of oil inventory that must be changed every year, as well as antifreeze protection when the air temperature is below 14 °C

However, the DSG process have several problems. For example, Almanza et al. operated a parabolic trough collector (PTC) plant to feed a low-power steam engine, highlighting the deflection problems suffered by low-temperature systems and controllability of the flow pattern [12, 13]. It is therefore necessary to carry out simulations to understand the thermohydraulics of the process. Up to date, there is considerable progress in

DSG analysis [11, 14, 15]. However, they all resort to the concept of classical heat transfer with the calculation of the convective coefficient h, in combination with the friction factor f, to understand the thermo-hydraulic behavior of the installation, even in CFD models [16, 17].

Nevertheless, the use of these parameters demands high computational power because the methodology to obtain these parameters requires an iterative process [18]. In this work, we apply the recently proposed and validated methodology for the analysis of DSG in PTC [19], with low computational times and high confidence, in which the use of the convective coefficients h is eliminated. The analysis is performed on a 10 MW conceptual plant located in northwestern Mexico.

1.1. Structure and scope

This article is organized in two parts. The first part outlines the thermohydraulic model and, subsequently, its application to the case study. This allows us to identify that the proposed methodology can be used without complications in the analysis of solar plants for direct steam generation in parabolic troughs.

2. Thermohydraulic modeling for DSG in PTC

The model to be developed takes as a reference the one described by other authors [20–25]. The present, uses the typical 1D modeling technique based on a steady-state energy balance over the receiver. The receiver is divided into smaller parts called Heat Collector Elements (HCE). The balance equations comprises direct normal solar irradiation, optical losses, HCE thermal losses, and the working fluid gains. The distinction of this model lies for the use of the heat transfer methodology developed by Adiutori [26]. This methodology does not require iteration to solve the heat fluxes and associated temperatures in the thermal model.

2.1. Energy balance equations

The working fluid temperature increases as the energy is absorbed by the HCE. Heat losses will occur as a result of differences between the average fluid temperature in each cross section and the ambient temperature. With Fig. 1, the following balance equations can be written:

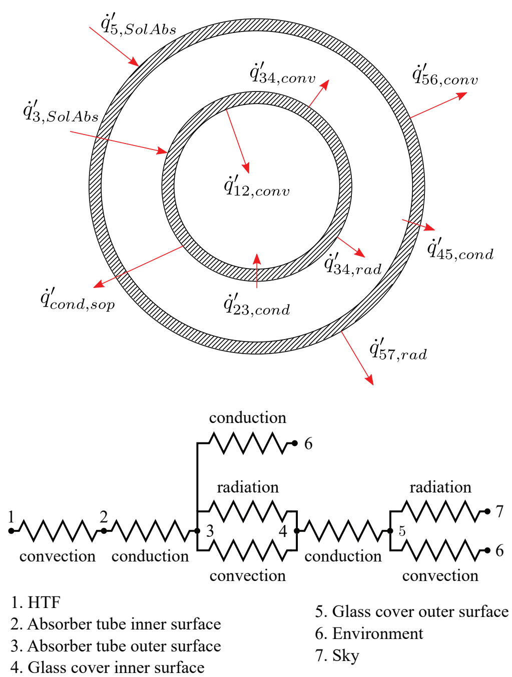

Figure 1: Heat fluxes over the parabolic trough.

$$\dot{q}'_{12,conv} = \dot{q}'_{23,cond} \tag{1a}$$

$$\dot{q}'_{3,SolAbs} = \dot{q}'_{34,conv} + \dot{q}'_{34,rad} + \dot{q}'_{23,cond} + \dot{q}'_{38,cond} \tag{1b}$$

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$$\dot{q}'_{45,cond} = \dot{q}'_{34,conv} + \dot{q}'_{34,rad} \tag{1c}$$

$$\dot{q}'_{45,cond} + \dot{q}'_{5,SolAbs} = \dot{q}'_{56,conv} + \dot{q}'_{57,rad} \tag{1d}$$

$$\dot{q}'_{12,conv} = \frac{\dot{m}}{L_{HCE}} \left(h_{in} - h_{out}\right) \tag{1e}$$

For the five equations of 1, the radiation absorption phenomena in the absorber tube and the transparent cover are treated as surface heat fluxes. In addition, the temperatures, heat fluxes and all thermodynamic properties are uniform around the cross-sectional area (1D model) [27]. These simplifications lead to theoretical values larger than the real ones, but with adequate results [21]. Under these conditions, each heat flux can be expressed as:

$$\dot{q}'_{i,SolAbs} = \eta_o W G_{bn} \tag{2a}$$

$$\dot{q}'_{ij,cond} = \frac{2\pi k_{ij}\left(T_i - T_j\right)}{\ln\left(\frac{D_j}{D_i}\right)} \tag{2b}$$

$$\dot{q}'_{ij,rad} = \frac{\sigma\pi D_i\left(T_i^4 - T_j^4\right)}{\frac{1}{\varepsilon_i} + \frac{\left(1-\varepsilon_j\right)D_i}{\varepsilon_j D_j}} \tag{2c}$$

$$\dot{q}'_{ij,conv} = f\left(\Delta T\right) \tag{2d}$$

where $\eta_o = IAM\Gamma\rho\tau\alpha$ is the optical efficiency, W is the trough width, G_{bn} is the incident solar radiation in the normal direction, k is the conductivity, T is the temperature, D is the diameter, σ is the Stefan-Boltzmann constant, and ε is the emittance. The subscripts i and j allow us to identify each of the surfaces, according to Fig. 1.

Notice that 2d, omits the use of the convective heat transfer coefficient h, and a particular function of the temperature difference should be used. These functions must be obtained from the expressions of the Nusselt number, by means of a transformation of the dimensionless groups, as explained in [26].

The annulus formed between the absorber tube and the glass envelope is modeled as free convection between two cylinders because the receiver is evacuated. Between the glass cover and the environment the convection can be forced or natural. In the first case, the Žhukauskas equation is used, while the Churchill-Chu equation is adequate for the second case [28]. For the convection between the absorber tube and the heat transfer fluid, there are two cases: the Pethukhov or Gnielisnki equation [28] for the single phase fluid and the Gungor and Winterton correlation for the two-phase flow [29]. The transformation of the two-phase flow equation is described in [27], resulting in the following:

$$\Delta T = C_1\dot{q}'_{12,conv}\left[C_2\left(C_3 + C_4\dot{q}'^{1.16}_{12,conv}\right)^2 + C_5 - \frac{C_6\dot{q}'^{0.67}_{12,conv}}{C_7\left(C_3 + C_4\dot{q}'^{1.16}_{12,conv}\right)^2 - 1}\right]^{-1} \tag{3}$$

where each C_i is a functional parameter of the temperature, obviously positive ($C_i > 0$), which can be obtained from any database of thermodynamic properties. This equation can be used with confidence, as validated in [27] with data recovered from the DISS loop [10, 16].

3. Study case

A conceptual regenerative Rankine cycle with parabolic trough collectors using the DSG concept is under consideration. The power block is designed to produce 10 MW of net power output, fixing the conditions at the turbine inlet 100 bar and 673.15 K (400 °C), and 0.08 bar at the turbine outlet. A generic turbine with a nominal isentropic efficiency of 80% is selected, as it is a value suggested by other similar analyses [30]. The Rankine cycle employs two internal regenerations (see Fig. 2), which allow the inlet temperature of the solar field to be higher (496.121 K) and consequently the size of the parabolic trough loops to be smaller. The tubine steam extractions have been determined with the premise of minimizing the exergy destruction and consequently maximizing the second law efficiency. The process of this optimization can be found in [31, 32].

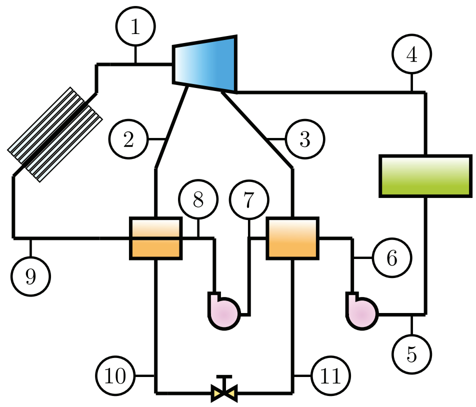

Figure 2: Schematic of the conceptual DSG Rankine power plant. Adapted from [31, 32].

With the operating conditions of the power block, 12.8058 kg/s of steam is needed to produce 10 MW of nominal power. Two conditions should be taken into account for DSG plants. The first is a mass flow rate that allows turbulent flow (Re $> 2 \times 10^5$) to ensure an appropriate heat transfer between the absorber and the water/steam [9, 21]. Second, the annular flow pattern should be sought to avoid overheating in the absorber. However, at the beginning of the boiling process, the flow pattern may be intermittent and annular, as in the DUKE loop [33]. Dividing the mass flow rate into a certain number of loops allows us to fulfill the first consideration. At the end of the thermohydraulic characterization, the second consideration can be verified.

The analysis is carried out considering three days: June 21 (summer solstice), May 21 (highest insolation condition), and September 21 (lowest insolation condition). Therefore, there are a total of 24 simulations to be carried out. In all the cases, the HCE is fixed to 2 m. The weather conditions for the selected days in Agua Prieta (northwest Mexico) are described in Table 1. For the present analysis, the mass flow is divided into 8 loops; therefore, the mass flow rate per loop is 1.6007 kg/s, with Re = 3.699×10^5, fulfilling the first flow condition.

Table 1: Weather conditions for the selected simulation days in Agua Prieta, México. Data obtained from [34].

Parameter	21st june	21st may	21st september
DNI (W/m^2)	856.4815	889.6396	628.8580
Atmospheric pressure (bar)	0.886	0.884	0.885
Ambient temperature (K)	300.05	295.95	300.05
Sky temperature (K)	271.95	265.75	277.75
Wind speed (m/s)	4	4.1	3.2

For the two-steam extractions, the subcooled water must enter the solar field at 496.121 K. Once the fluid input conditions to the solar field are known, together with the normal weather conditions described in Table 1, it is possible to start with the simulations to thermo-hydraulically characterize the solar field, simultaneously solving 1. Knowing all the initial parameters and having defined the size of the HCE, the fluid starts its path along the loop, increasing its temperature and evidently suffering a pressure drop; until it reaches the saturated liquid condition. The fluid then undergoes a phase change from saturated liquid to saturated vapor in a non-isobaric process. An intermittent flow at the beginning of the boiling and annular in the rest of the process is expected. Finally, the fluid is superheated to a temperature of 673.15 K.

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Figure 3 shows the temperature evolution along the loop. It is important to note that, as expected, the temperature during the phase change does not remain constant due to the pressure drop (as shown in detail in Fig. 4). The pressure drop decreases slightly the saturation temperature, which also affects the vapor-liquid mixture quality. It is evident that in the three days, the evaporation zone has a larger size since the phase change processes require a high demand of thermal energy to achieve this task.

As seen in Figure 4a, the pressure along the loop decreases until it reaches the required conditions at the turbine inlet (100 bar and 673.15 K). Figure 4b shows that the phase change region presents the highest pressure drop. It is important to note that the curve obtained shows an abrupt change in its slope, as occurs in the DISS loop [35].

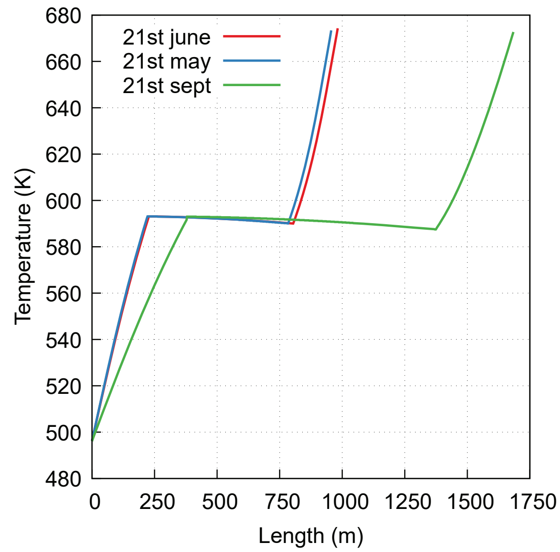
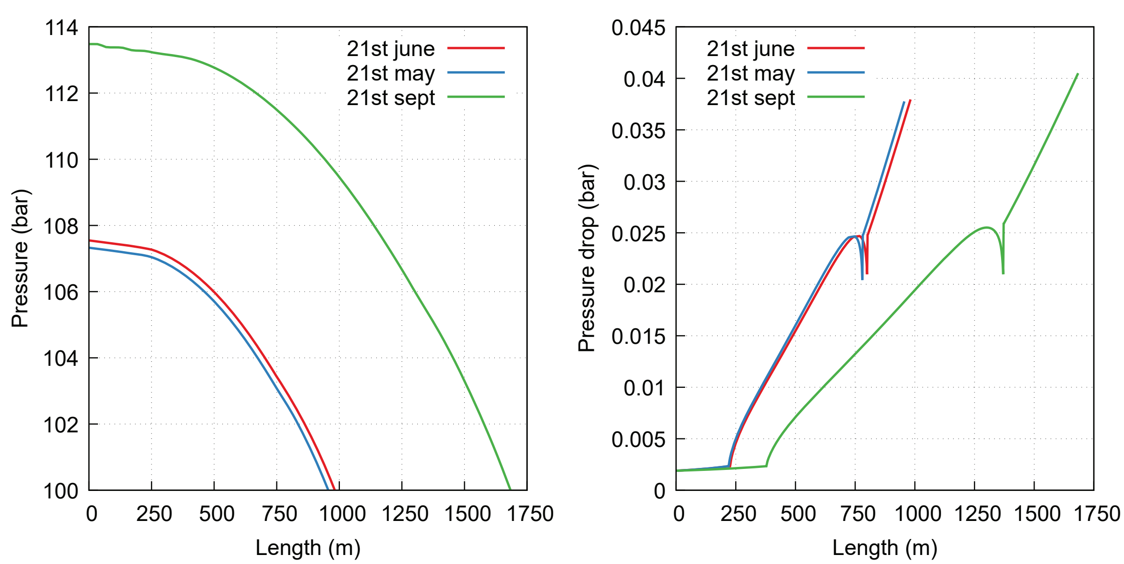


Figure 3: Evolution of the temperature increase along the loop.



(a) Pressure evolution along the loop.

(b) Pressure drop along the loop.

Figure 4: Loop pressure characterization.

The flow pattern in the boiling process along the loop must be verified, as stated previously. Figure 5 shows the flow pattern in the two-phase flow region in the Taitel and Dukler flow map. While analyzing the graph for the Martinelli parameter X, an intermittent flow (high values of X) is present at the beginning of the boiling process, then changes to an annular flow as the steam quality increases (low values of X). Doing this verifies that the mass flow is adequate and meets the two established conditions: a high Re number to ensure a high heat transfer rate and a flow pattern that reduces overheating in the absorber.

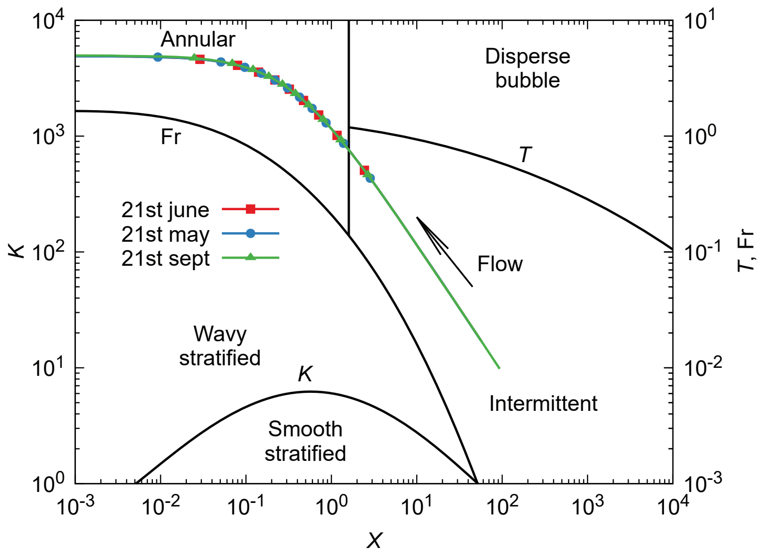


Figure 5: Flow pattern in the two-phase region.

When analyzing Figure 3, and as expected, the solar field has a longer length on the day of less insolation (September 21), being shorter for the day of greater insolation (May 21) although there is no substantial difference concerning the summer solstice (June 21). Since the total mass flow rate of the Rankine cycle was divided into eight loops, the total length of the PTC field must be multiplied by 8. It is possible to define the total concentration area for the solar field. These will allow us to define the type of arrangement of the loops in the solar field, since for areas greater than 40 ha, an "H" arrangement is recommended, while for less than 40 ha an "I" arrangement can be used [21, 36]. In Table 2, the precise loop length is shown, including the total area of the solar field, in addition, the pressure at the inlet of the solar field is also shown. In these three cases, the arrangement "H" is recommended.

Table 2: Solar field description.

Parameter	21st june	21st may	21st september
Preheater length per loop (m)	226	220	378
Evaportor length per loop (m)	578	570	1016
Superheater length per loop (m)	178	164	290
Total length per loop (m)	982	954	1684
Total concentrating area (ha)	4.53	4.41	7.78
Inlet pressure (bar)	107.55	107.33	113.48

The pressure evolution along the loop presents a similar behavior (Figure 4a). Since the September 21 loop is the longest, it will have higher pressure drops, so the solar field must be operated at a higher pressure (almost 114 bar). Figure 4b shows that the highest pressure drop occurs in the two-phase flow zone. The behavior at the phase change interfaces (saturated liquid and saturated vapor) presents an abrupt change in its slope (nondifferentiability condition), although as reported in [37], it is continuous.

Once all the operating parameters in the solar field (temperature and pressure) are known, it is possible to identify the flow pattern in the phase-change region. Although the loops have different lengths and pressure drops in the three cases of analysis, the flow at the beginning of boiling is intermittent. As the liquid transforms

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into steam, the flow behaves as an annular. This behavior is desired in plants with the direct steam generation, so the feasibility of this loop is adequate.

Four additional parameters of great interest are usually computed for CSP plants: solar multiple (SM, 4), thermal energy storage size ($\dot{Q}_{TES}$, 5), energy (η_I, 6) and exergy (η_{II}, 7) efficiency of the PTC solar field. Considering that on September 21st, the loop is the largest, this day is considered as the design condition. If $SM > 1$, it is possible to include thermal energy storage (TES). For PTC, the SM must be below of 2 [6]. To date, there are no commercial TESs for DSG. However, a generic TES size system can be defined and rated accordingly. These four parameters are defined as:

$$SM = \frac{\dot{Q}_{solar\ field,design}}{\dot{Q}_{solar\ field}} = \frac{A_{ap}G_{bn}}{A_{ap,september}G_{bn}} \tag{4}$$

$$\dot{Q}_{TES} = (SM - 1)\,\dot{Q}_{solar\ field} \tag{5}$$

$$\eta_I = \frac{\dot{Q}_{HTF}}{\dot{Q}_{inc}} = \frac{\dot{m}\,(h_{out} - h_{in})}{A_{ap}G_{bn}} \tag{6}$$

$$\eta_{II} = \frac{\Delta \dot{E}x_{HTF}}{\dot{E}x_{inc}} = \frac{\dot{m}\,(ex_{out} - ex_{in})}{A_{ap}\dot{E}x_{solar}} \tag{7}$$

The exergy of solar thermal radiation is calculated with González-Mora formula [38]:

$$\eta = \left(1 - \frac{1}{\xi}\left[\frac{T_r}{T_{sun}}\right]^4\right)\left(1 - \frac{\lambda_c T_{amb}}{\lambda_c T_r + \xi\sigma T_{sun}^4 - \sigma T_r^4}\right) \tag{8}$$

where the optimum receiver temperature T_r must be obtained by solving $4\sigma^2 T_r^{11} - 8\sigma\lambda_c T_r^8 - 8\sigma^2\xi T_{sun}^4 T_r^7 + 4\lambda_c^2 T_r^5 - T_r^4\left(4T_{amb}\lambda_c^2 - 8\lambda_c\sigma\xi T_{sun}^4 - \lambda_c^2 T_{amb}\right) - \lambda_c^2 T_{amb}\xi T_{sun}^4 = 0$. In [38], it has been demonstrated that the exergy of solar radiation is $0.8391G_{bn}$. Table 3 shows the solar multiple, the TES size, the energy efficiency and the exergy efficiency of the PTC solar field.

Table 3: Solar multiple and efficiencies of the solar loop.

Parameter	21st june	21st may	21st september
SM	1.72	1.77	1
TES (MW_t)	3.49	3.76	0
Energy efficiency (%)	70.38	69.49	56.05
Exergy efficiency (%)	40.71	40.78	32.23

4. Conclusions

The modeling of parabolic trough solar plants in direct steam generation remains crucial to understanding the operating behavior and, consequently, in their implementation. In this work, an alternative methodology for thermo-hydraulic characterization of these types of systems is used to characterize the solar field. The developed model, unlike other models, eliminates the use of the convective coefficient h and the friction factor f. This allows a fast computation, with confident results.

The thermohydraulic model was applied for the analysis of the conceptual direct steam generation plant in Agua Prieta (Northwestern Mexico). The 10 MW Rankine cycle was previously optimized to minimize the destruction of exergy. As a result, 12.8058 kg/s of steam at 100 bar and 673.15 K (400 °C) is required. With the restriction of maintaining a turbulent flow, the mass flow rate has been divided into 8 loops; each loop circulating 1.6007 kg/s of water/steam entering at 496,121 K, with a maximum pressure of 113.48 bar.

Three days have been analyzed for the conditions of highest insolation (21st may), minimum insolation (21st september) and the summer solstice (21st june), considering their respective weather conditions for each day. As a result, the total length of each loop has been defined first, where it turns out that the longest loop is for

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21st september (1684 m, 7.78 ha in total). This loop is considered as the design field. Once the loop length has been defined, the flow pattern type in the two-phase zone has been characterized. It is identified that for the three days of analysis, the flow at the beginning of the boiling is intermittent and later evolves towards an annular flow, which is the one suggested in DSG plants.

Finally, the solar multiple has been calculated, showing that for the design condition, the solar multiple is not excessively large, so the loop is adequate, with a capacity of 3.76 MW. Similarly, the PTC energy efficiency has a maximum value of 70.38% (and 56.05% minimum), while the PTC exergy efficiency has a maximum value of 40.78% (and 32.23% minimum).

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Nomenclature

Letter symbols

- A area, m^2
- C temperature functional parameter, -
- D diameter, m
- ex specific exergy, kJ/kg
- $\dot{E}x$ total exergy rate, W
- G irradiation, W/m^2
- h heat transfer coefficient, W/m^2K; specific enthalpy, kJ/kg
- IAM incidence angle modifier, -
- k conductivity, $W/m \cdot K$
- L lenght, m
- $\dot{m}$ mass flow rate, kg/s
- $\dot{q}'$ heat flux per unit length, W/m
- $\dot{Q}$ heat transfer rate, W
- Re Reynolds number, -
- SM solar multiple, -
- T temperature, K
- W parabolic collector aperture, m
- X Martinelli parameter, -

Greek symbols

- α receiver absorptivity
- Γ intercept factor
- ε receiver emittance
- η efficiency
- λ thermal conductance per unit length, $W/m \cdot K$
- χ concentration acceptance product
- ρ mirror reflectivity
- σ Stefan-Boltzmann constant

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Subscripts and superscripts

amb ambient

ap aperture

bn beam normal

cond conduction

conv convection

HCE heat collector element

i generic counter

I first law

II second law

j generic counter

in inlet

inc incident

o optic

out oulet

r receiver

solar solar

SolAbs slar absorption

Solar field, design solar field in design conditions

Solar field solar field

rad radiation

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