

71st Conference of the Italian Thermal Machines Engineering Association, ATI2016, 14-16  
September 2016, Turin, Italy

## Modeling of Direct Steam Generation in Concentrating Solar Power Plants

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### Abstract

A simulation procedure has been developed to predict the performance of a concentrating solar power plant with direct steam generation (DSG) technology. A detailed modelling of the DSG solar field was conceived for both parabolic troughs (PTC) and linear Fresnel collectors within an integrated Octave-TRNSYS® environment. Predictions from this model were compared with published operational data of the first pre-commercial 5 MW DSG solar power plant, designed within the INDITEP project. Attention was paid to the plant performance under two different operation modes, i.e. “fixed” or “sliding” steam turbine inlet pressure.

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Peer-review under responsibility of the Scientific Committee of ATI 2016.

**Keywords:** Concentrating Solar Power; Direct Steam Generation; parabolic trough; part load modeling; simulation software.

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### 1. Introduction

Outcomes from EU-funded research projects have demonstrated that DSG is an interesting technological option for concentrating solar electricity production [1]. Compared to the prevailing CSP solution based on parabolic trough collectors with synthetic oil as heat transfer medium, there are several benefits if the water/steam can be used for both the solar field and the power block. First of all, the steam temperature limit of about 400°C for today's commercially available synthetic oils can be overcome [2]. Actually, an operating temperature of 500°C seems to be feasible for new generation receiver tubes [3, 4], thus increasing the attainable level of global efficiency.

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Secondly, there is no need for oil-water heat exchangers if water is directly evaporated and superheated in the solar field [5], resulting in a simpler plant configuration and lower investment costs. Finally, the use of demineralized water is ecologically uncritical: compared to oil, water is cheaper, non-toxic and non-flammable; compared to molten salt, water is less expensive and less corrosive [6]. Among DSG-related potential problems, the following critical issues are the most relevant [7-9]:

- Stability of the annular fluid inside the tubes to ensure internal cooling of the wall, preventing hot spots;
- Structural integrity of absorber tubes and interconnections between collectors and absorber tubes themselves, at high pressure and temperature;
- Plant control system, particularly during transitory periods of radiation;
- Integration with storage systems adapted to the special characteristics of the two-phase fluid water/steam, in the evaporation section (preheating and superheating part can be treated by conventional two-tank molten salt storage systems).

Within the DISS project, a DSG test facility was built at Plataforma Solar de Almeria in 1997 for testing start-up and shutdown procedures and monitoring thermal stress of the absorber tubes. The facility worked more than 3500 hours from 1999 to 2001 [10], paving the way for commercial DSG plants, ten years later. In fact, experience and know-how acquired in the DISS project were applied in the INDITEP project, to design the first pre-commercial DSG plant [11]. It is composed of a parabolic-trough solar field connected to a superheated steam (400°C-65bar) Rankine cycle, with rated power output of 5MWe. DSG is now applied commercially by Novatec Solar in its 30 MWe Puerto Errado 2 Fresnel plant commissioned in 2012 and by Solarlite with its 5 MWe parabolic trough plant TSE-1 in Thailand, commissioned in 2011, generating saturated and superheated steam, respectively [12].

In this paper, a modelling procedure has been proposed to simulate a DSG plant producing superheated steam. Data available from the INDITEP project have been taken as a reference, given the paucity of information concerning recent DSG installations. Focus has been put on electricity generation by the steam turbine, varying the inlet pressure control, over a one-year period.

## 2. DSG plant configuration

Figure 1 shows the layout assumed in the present study. It refers to the INDITEP DSG solar power plant with a rated power output of 5 MWe, whose theoretical feasibility has been assessed in [13]. The ET-100 PTC solar field consists of 7 loops connected in parallel, with 10 solar collectors in each loop. For a single loop of collectors, 8 collectors are designed for preheating and evaporating whereas the final 2 collectors are for superheating. A separator with an outlet steam quality of 0.85 is located between the evaporator and the superheater to guarantee a sufficient cooling of the absorber tubes during steady state operation. Both recirculation and injection nozzles are used for a better control on the solar field outlet temperature. Table 1 reports the design characteristics of the PTC [11]. As far as the power block is concerned (Table 2), the steam pressure and temperature at the turbine inlet are set at 65bar and 400°C, respectively. Given the small size of the plant, a single steam extraction line, whose pressure varies between 2.4bar and 5.6bar, was included to feed a deaerator. An air cooled condenser was considered with a design point operating pressure of 0.15bar.

Two different operation modes for the HP steam turbine have been tested: the former, named “sliding pressure” implies that the pressure at the turbine inlet will vary proportionally with the steam flow rate while the latter, named “throttle mode”, keeps the turbine inlet pressure constant at 65 bar even though the steam flow rate entering the turbine is lower than the design value. Furthermore, exhaust steam conditions were carefully calculated taking into account kinetic losses as a function of the exhaust volume flow (Fig. 2). The exhaust loss curve is an empirical correlation for an enthalpy difference, called exhaust loss (EL), which when added to the enthalpy at the expansion line end point (ELEP) allows for correctly computing turbine work per unit mass  $W$ , according with:

$$W = h_1 - (h_{ELEP} + EL) \quad (1)$$

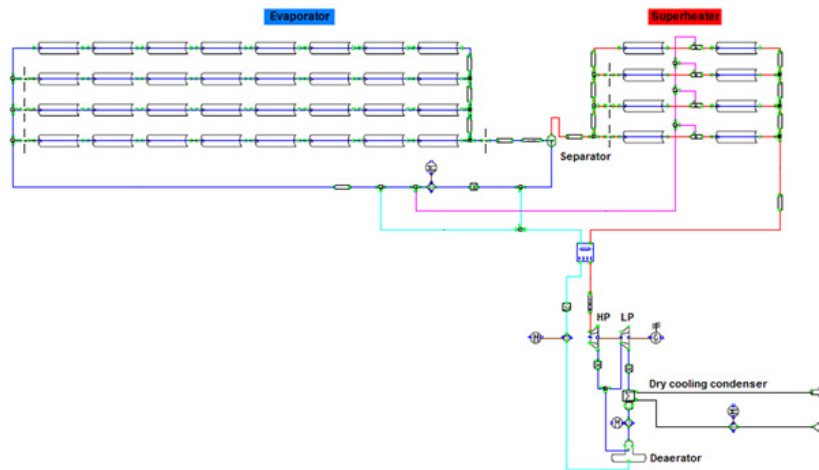


Fig. 1: Schematic of the modelled DSG plant [13]: only 4 loops are displayed.

To avoid power plant shutdown during night hours, an auxiliary gas-fired boiler is turned on when the steam flow rate is less than one third of the design value (8.5 kg/s), thus ensuring a minimum steam flow rate equal to 2.8 kg/s. The reference site is Seville. The yearly sum of the DNI for the site is 1625 kWh/m<sup>2</sup>, according with the weather data from Meteonorm database. Figure 3 shows clearly an increase in solar irradiance in summer months, when PTC can intercept a significant fraction of the DNI, due to lower cosine losses.

Table 1. Solar field parameters.

Overall length of a single collector (m)	98.5
Net collector aperture per collector (m <sup>2</sup> )	548.35
Outer diameter of steel absorber pipe (m)	0.07
Inner diameter of steel absorber pipe (m)	0.055
Number of 90° elbows	4
Relative roughness of the steel absorber pipe	7.23e-04
Peak optical efficiency	0.765
Solar field pressure drop (bar)	14

Table 2. Design power block parameters.

Pressure and temperature at turbine inlet	65bar-400°C
Pressure and temperature at solar field inlet	85bar-160°C
Deaerator pressure (bar)	5.6
Condenser pressure (bar)	0.15
HP/LP Turbine isentropic efficiency (%)	71/78
Gross power (MW)	5.9
Net power (MW)	5.0
Net electric efficiency (%)	22.3

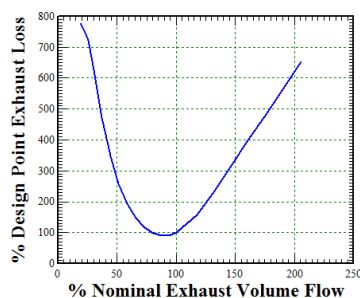


Fig. 2: LP turbine non-dimensional exhaust loss.

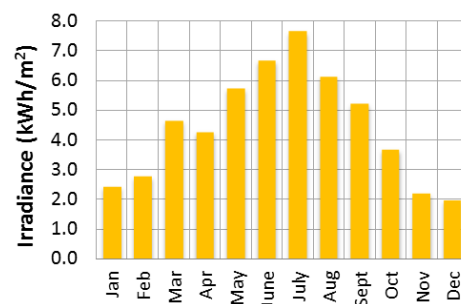


Fig. 3: Daily average irradiance for Seville.

### 3. Simulation method and assumptions

A detailed modelling of the DSG solar field was conceived for both parabolic troughs and linear Fresnel collectors within an integrated Octave-TRNSYS® environment. In the former, a homemade code computes the solar field steam production and pressure drop, depending on solar radiation and desired steam quality. The latter, allows for including advanced control strategies such as auxiliary boiler switch-on during night hours and defocusing. Both programs work together to provide a first guess of the solar field performance according to an iterative method (Fig. 4). In detail, Octave requires input data from TRNSYS® (beam radiation  $I_b$ , dew point  $T_{dew}$ , sky temperature  $T_{sky}$ , wind velocity  $V_{wind}$ , ambient temperature  $T_{amb}$ , Incidence Angle Modifier IAM, number of PTC loops  $N_{loop}$ , PTC optical efficiency  $\eta_{op}$ , steam quality  $x$ ), features of collectors and pipes (internal diameter  $d_{in}$ , thickness and absolute roughness of pipes, collector area, absorber thermal conductivity  $k_{ab}$  and emissivity  $\varepsilon_{ab}$ ) as well as first attempt values of pressure and temperature at the inlet of the solar field ( $P_{in}$ ,  $T_{in}$ ).

Absorber total heat loss was computed in terms of local absorber temperature  $T_{ab}$  and ambient temperature as:

$$\begin{aligned}
 Q_{loss} &= (a + cV_{wind})(T_{ab} - T_{amb}) + \varepsilon_{ab}b(T_{ab}^4 - T_{sky}^4) \\
 a &= 1.91 \cdot 10^{-2} \text{ WK}^{-1} \text{ m}^{-2} \\
 b &= 30 \cdot 10^{-9} \text{ WK}^{-4} \text{ m}^{-2} \\
 c &= 6.608 \cdot 10^{-3} \text{ JK}^{-1} \text{ m}^{-3} \\
 \varepsilon_{ab} &= 0.00042 T_{ab} - 0.0995 \quad (T_{ab} \text{ in K})
 \end{aligned} \tag{2}$$

Coefficients  $a$ ,  $b$ ,  $c$  for conduction, radiation and convection heat losses were evaluated by curve fitting this simplified  $Q_{loss}$  correlation with heat loss data resulting from a complex PTC model composed of 16 equations [14].

With the aim of assessing the heat transfer to the working fluid, the internal heat transfer coefficient was computed through experimental correlations. Gnielinski and Dittus-Boelter were used for the single-phase fluid whereas Chen correlation was applied to the two-phase fluid. In the interest of an accurate result, each module of the solar field (preheater, evaporator, superheater) was divided into integer number of segments, having length  $dx$ .

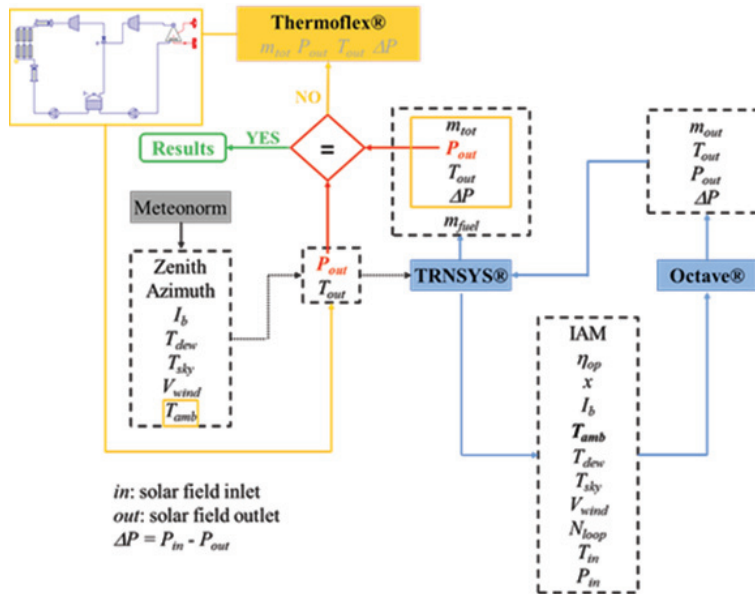


Fig. 4: Schematic diagram of the hour-by-hour modelling procedure of the DSG plant.

Thermal input and flow rate balances were checked at every  $dx$ . Control variables are fluid temperature in the preheater and enthalpy in the evaporator/superheater. The *X-Steam* function allowed for computing thermodynamic properties of water and steam, at every condition of pressure, temperature and steam quality.

In addition, concentrated and distributed pressure losses were computed for one phase and two-phase modules, separately, as suggested by Uçkun [15]. Concentrated losses take into account 4 standard 90° elbows whereas distributed losses are calculated at every  $dx$ , since they include shear losses along the entire length of the module. In case of two-phase flow, pressure losses are multiplied by a factor depending on steam quality, as defined by Friedel [16]. Outputs from the Octave code are pressure and temperature at the solar field outlet ( $P_{out}$ ,  $T_{out}$ ), together with the delivered steam flow rate  $m_{out}$ . The latter variable is transferred to TRNSYS® for further adjustment: if  $m_{out}$  exceeds the maximum flow rate  $m_{tot}$  then defocusing is activated; on the other hand, if  $m_{out}$  is lower than minimum flow rate, the auxiliary boiler is turned on and the fossil fuel consumption is taken into account (see  $m_{fuel}$  in Fig. 4). The reader is referred to [17] for a full report of the equations implemented in the solar field model. The present study does not include thermal energy storage. Defocusing has been deactivated.

Thermoflex® was chosen to simulate the power block to take advantage of accurate predictions of steam turbine and air cooled condenser behavior at part load. The hour-by-hour performance of the whole solar power plant is the result of an iterative procedure. In fact, the solar field outlet pressure  $P_{out}$  calculated by Octave-TRNSYS® is iteratively adjusted to meet the requirements of the steam turbine inlet, according to Thermoflex®. After establishing the correct value of  $P_{out}$ , Thermoflex® provides a user defined list of important outputs pertaining to the Rankine cycle hourly operation. At that point the whole procedure is repeated for the next hour, until the whole simulation period is completed. In this way, the DSG solar field is totally coupled to the power cycle to ensure realistic simulations under off-design operating conditions.

Microsoft Excel provided the interface through which Octave, TRNSYS and Thermoflex exchange information. In particular, the Visual Basic programming language made the codes interact with each other, transferring inputs and outputs. Convergence errors of the iterative interaction process were below 4% over the whole simulation period.

#### 4. Results and discussion

Annual simulation campaigns have been carried out to evaluate DSG plant performance under different turbine operation modes. Results are presented by focusing on a typical winter and summer day, in order to discuss the details of part-load behavior. Then, outputs from one-year simulations are compared with each other to quantify the impact of turbine operation mode on electricity production and solar-to-electric efficiency.

##### 4.1. Simulation outputs for representative days

Meteorological conditions in terms of ambient temperature and DNI profiles are reported in Fig. 5 for the selected days: one in winter (January 18<sup>th</sup>) and one in summer (July 20<sup>th</sup>). The coldest ambient temperature is about 6°C in winter whereas the hottest conditions (45°C) are achieved on summer afternoon, when DNI gets the pick at 900 W/m<sup>2</sup>. In January the maximum solar radiation is limited to 550 W/m<sup>2</sup>. Figure 6 displays the DSG net electric power and net electric efficiency, evaluated as the ratio of the net power output to the solar heat input. Four profiles are represented over time to account for winter/summer weather conditions and throttle/sliding turbine modes.

Starting from the left, the “sliding” profiles of net power put in evidence that the design point of 5 MWe is exceeded for 8 hours on July 20<sup>th</sup>. Conversely, in January, the power output is less than half the rated value all day. Reasons for this are related to: high level of solar radiation combined with high optical efficiency of PTC in summer; low amount of solar energy, coupled with poor PTC performance in winter. In fact, the fraction of solar radiation collected by PTC decays in winter, because of cosine effect and low ambient temperatures. On January 18<sup>th</sup> the electricity production reaches its maximum (2.3 MW) at 11<sup>h</sup>, with a second peak later at 17<sup>h</sup>. For the rest of the day, the minimum power output to avoid plant shutdown is guaranteed by the auxiliary boiler, as documented by the fossil fuel consumption plotted in Fig. 7. It can be deduced that a DNI higher than 500 W/m<sup>2</sup> is required to operate the DSG plant without fossil fuel backup.

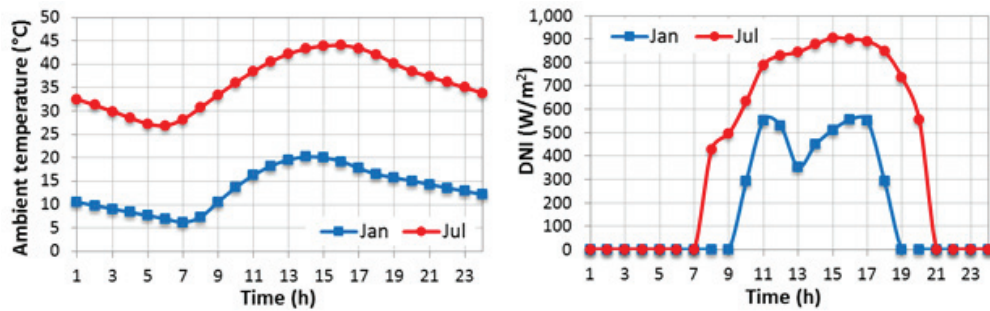


Fig. 5: Daily ambient temperature (left) and solar radiation (right).

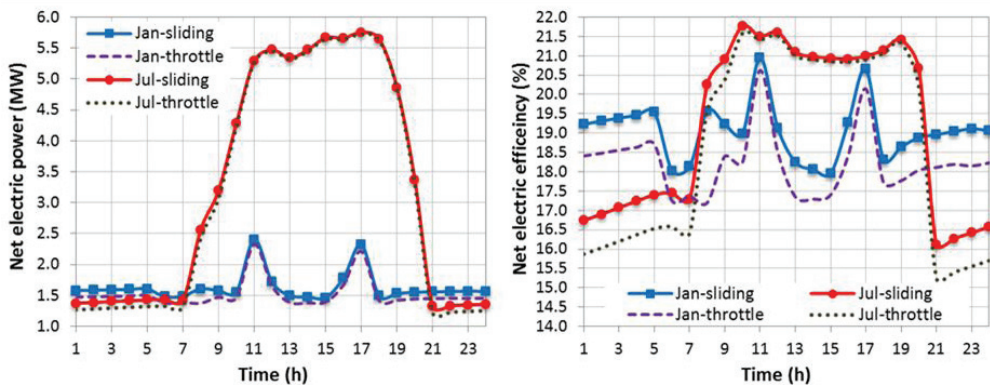


Fig. 6: Daily net electric power (left) and net electric efficiency (right).

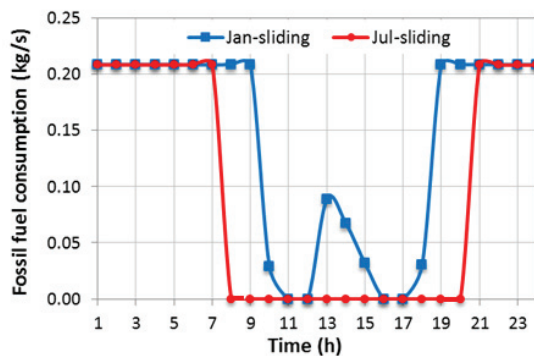


Fig. 7: Fossil fuel consumption.

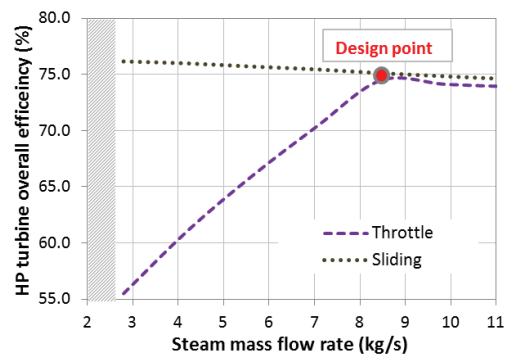


Fig. 8: HP turbine overall efficiency.

In nighttime hours, the nominal fuel consumption in the auxiliary boiler (0.21 kg/s) has been set to generate a steam flow rate equal to 2.8 kg/s, corresponding to about 1.5 MWe. Shifting attention toward the turbine modes, “sliding” profiles of electricity generation indicate superior performance compared to “throttle” mode when the plant is operated far from design point. The benefits of the sliding mode are more significant during all day in winter and summer nighttime, i.e. when the plant works at part load. The reader is remembered that both turbine modes provide identical turbine inlet pressure in case of plant load factor  $\geq 100\%$ . This is why sliding and throttle curves overlap in the central hours of July 20<sup>th</sup>.



The positive influence of the sliding mode on net electric efficiency ( $\eta_{net,el}$ ), with respect to the throttle mode, is evident from Fig.6-right: in night time, at minimum load operation, the efficiency gain is up to 1%; in daylight hours, the incremental efficiency is less than 1% on January 18<sup>th</sup> and becomes negligible on July 20<sup>th</sup>. The lower efficiency of the throttle mode, at part load, is caused by throttling losses in the HP turbine stage. Throttling reduces the HP turbine overall efficiency when the steam flow rate is lower than design point. Conversely, the sliding mode prescribes almost fixed turbine overall efficiency at design value, whatever the steam flow rate (Fig. 8).

Focusing on the sliding mode (Fig. 6-right), it is worth noting that  $\eta_{net,el}$  in the selected days is always lower than the design value (22.3%). The highest  $\eta_{net,el}$  of 21.8% at 10<sup>h</sup> of July is a compromise between high solar radiation and not excessively hot weather. Later that day (until 15<sup>h</sup>), larger amount of solar energy are available at increasing ambient temperature, thus reducing the Rankine cycle efficiency to about 21%. It is well known indeed that heat rejection to air in a dry cooling system is very critical in hot weather. In winter, cold temperatures joined to moderate steam production cause  $\eta_{net,el}$  as high as 21% at 11<sup>h</sup>. In nighttime, the plant efficiency at the minimum load depends only on ambient temperature:  $\eta_{net,el}$  is 2.5% higher in the winter day.

#### 4.2. Annual simulation results

Monthly average net electricity production is shown in Fig. 9 for both turbine modes. The sliding mode ensures higher net power output throughout the year and especially in winter months, when the plant operates at part load all of the time. For the final assessment of the DSG performance, Fig. 10 displays the net electric efficiency for each hour of the year, as a function of DNI. Values of DNI lower than 250 W/m<sup>2</sup> were not considered due to extreme off-design conditions. A second-degree polynomial was found to be the best approximation to  $\eta_{net,el}$  for both turbine modes, in agreement with [13]. The threshold of  $\eta_{net,el} = 22.3\%$  is reached for 176h -15h per year for sliding - throttle mode, respectively.

Results of the annual simulations are summarized in Table 3. Compared to the throttle mode, the sliding option ensures about 5% higher net power production over the whole year, with higher average net efficiency (18.5% vs 17.7%). The average turbine inlet pressure is approximately half the design value, thus confirming that the DSG power plant operates in part load for most of the year.

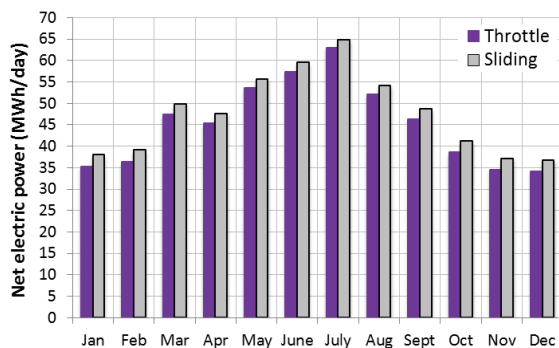


Fig. 9: Monthly average net power production.

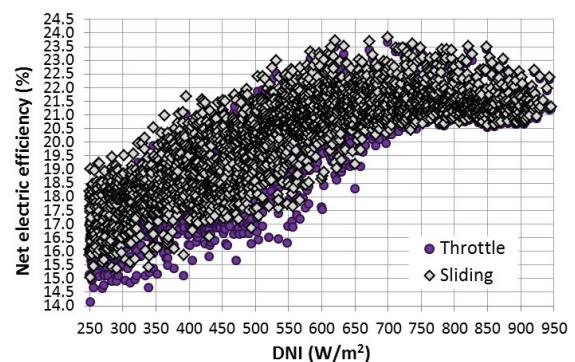


Fig. 10: Net electric efficiency as a function of DNI.

Table 3. Yearly DSG plant performance.

	Throttle mode	Sliding mode
Net electric power (MWh/day)	45.3	47.7
Average net electric efficiency (%)	17.7	18.5
Average turbine inlet pressure (bar)	67.4	32.2
Average fossil fuel consumption (kg/h)	482.85	495.62

The only disadvantage of the sliding mode is slightly higher fossil fuel consumption (+2.6%), due to higher enthalpy of the steam entering the HP turbine (superheated steam has the same temperature of the throttle mode, but with lower pressure levels). This drawback mainly occurs during nighttime.

## 5. Conclusion

A modelling procedure including three different simulation environments, such as TRNSYS®, Thermoflex® and Octave® has been used to predict the performance of a 5MWe DSG solar power plant, designed under the INDITEP project. The coexistence of different codes allowed the user to take the best from each programming language. In particular, TRNSYS is the most suitable interface to deal with transitory periods of radiation; Thermoflex is very effective in predicting the power block behavior at part load whereas Octave can easily manage the details required by the DSG solar field modelling. Yearly simulations with 1-hour time step were run for two different operation modes of the HP steam turbine, i.e. “sliding” vs. “throttle” inlet pressure. The sliding mode, compared to the throttle mode, resulted in 5% higher net electricity production and 0.8% higher net electric efficiency over the year. The only drawback is 2.6% higher fossil fuel consumption.

## Acknowledgements

The authors acknowledge the efficient work carried out by Simone Belotti and Alessandro Pasta.

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