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SIMULATION AND PERFORMANCE ASSESSMENT OF LOAD-FOLLOWING CSP PLANTS

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ABSTRACT

The paper is focused on the modeling of Concentrated Solar Power (CSP) plants based on a steam Rankine cycle combined with two different solar field configurations: Parabolic Trough Collectors (PTC) and Heliostats with Central Receiver (HCR). The system is designed to operate as a load following power plant: a Thermal Energy Storage (TES) system allows to compensate fluctuations in solar energy and in power demand, and to operate also during nighttime hours.

Commercial software and in-house developed computer codes are combined together to predict CSP plant performance under real operating conditions. The power block was modeled by Thermoflex® whereas Trnsys® was used to model the solar field operation all over the year.

An optimization procedure interacting with Trnsys® model was used to size the two considered solutions for the solar fields. On the base of annual Trnsys® simulations, the optimization algorithm determined the minimum aperture area of the solar field assuring the required Heat Transfer Fluid (HTF) flow rate from TES. Charging and discharging cycles of TES are ruled by the HTF flow rate required for each hour of the year so as to match the electrical demand.

Results of annual plant operation on a one hour basis are presented and discussed for Upington (RSA). Then the global results are compared with similar plants based in Sevilla (ES).

NOMENCLATURE

CSP		Concentrated Solar Power
DNI	[W/m^2,kWh/m^2]	Direct Normal Irradiation
E	[MWh]	Energy
HCR		Heliostats with Central Receiver
HTF		Heat Transfer Fluid
HX		Heat exchanger
p	[bar]	Pressure
PTC		Parabolic Trough Collector
Q	[MW]	Thermal power
SF		Solar Fraction
T	[°C]	Temperature
TES		Thermal Energy Storage

Subscripts

amb	Ambient
aux	Auxiliary
coll	Collected
cond	Condenser
dem	Demand
el	Electric
rad	Intercepted solar radiation

INTRODUCTION

CSP is an attractive option in climate change mitigation scenarios because electricity is generated by means of solar radiation, an almost infinite energy source, with no direct emissions of CO_2 [1]. Moreover in CSP plants heat can be stored for many hours in a day, allowing overnight power production. By using Thermal Energy Storage (TES) systems, CSP stations operation can be extended to meet base-load and peak load as well [2]. In addition, CSP plant technology is very scalable and it can be employed to generate power in sunny sites from a few megawatts up to hundreds. Also, reliability and design flexibility make CSP plants ideal for remote locations, where they are meant to supplement or substitute other forms of power generation, such as gensets burning fossil fuels. Alternatively, power should be imported via expensive transmission infrastructures.

Recently, an agreement was signed by Aramco to develop plants generating up to 300 MW of solar power distributed in remote areas in Saudi Arabia; the goal is to reduce the burning of diesel for power generation in those areas [3]. The Australian Renewable Energy Agency will fund a 20 MW CSP plant based on solar power tower and thermal energy storage. The project would supply electricity to mining operations and to the rural community of Perenjori (Western Australia); one of the benefits is to eliminate the need for grid expansion [4]. Whatever the goal is to reduce fuel consumption, to increase energy independence, or to minimize environmental impact, solar generated power offers a promising and viable solution.

In the present paper a load-following CSP plant was conceived to generate electricity for remote or weakly interconnected grids. Among the available CSP technologies (namely parabolic troughs, linear Fresnel reflectors, solar towers and dish/engine systems solutions), parabolic troughs and towers have been taken into account since they have now reached a commercial status [5].

In spite of a large number of paper investigating plant performance of a single CSP technology, only a few works present a comparison between solar fields based on PTCs and solar tower. Solar collector efficiency is strongly related to site latitude and meteorological conditions (DNI, ambient temperature) [6]. Generally speaking, PTCs can intercept a larger amount of incident radiation than heliostats in summer months, but their efficiency tends to dramatically decay in winter [7]. Comparing the results for parabolic troughs and tower plants, the latter typically provide a higher uniformity in the electricity production, due to a more constant collection capability of the resource all over the year. However, because of the larger spacing needed by the heliostats, the energy density is lower than for the PTC plants [8].

The present study demonstrates that a solar-driven Rankine cycle with thermal storage and auxiliary heater can be flexibly operated to match electric demand over a one-year period. The load-following power system was assumed to be isolated and conceived to satisfy electric energy demand for a mid-size community (roughly around 50,000 inhabitants). Daily patterns of power demand were defined for an entire one-year period: Figure 1 shows the power load for a typical summer and winter day. The peak power results to be about 51 MW in summer, and only 28 MW in winter. A typical large variation in the power request can be noticed between day and night time.

CSP PLANT DESCRIPTION

The power plant configuration assumed for the present analysis is shown in Figure 2. It is based on a solar-driven Rankine cycle integrated with an auxiliary natural gas heater.

Power block

The power block is a single reheat, regenerative Rankine cycle with 6 water pre-heaters: three LP feed-water heaters, a de-aerator and two HP feed-water heaters. Primary thermodynamic design parameters are from [10]. Molten salt was chosen to transfer heat to the water loop in the Rankine cycle so to increase the cycle efficiency by about 2-3% as compared to the oil case [9].

Steam pressure and temperature at turbine inlet were set at 100 bar and 540°C, respectively (Table 1). Steam is then reheated up to 500°C. Constant molten salt temperature values of 300°C and 550°C were assumed at the solar field inlet and outlet respectively. An air cooled condenser with a design condensing pressure of 0.06 bar at ISO conditions was considered. It is worth noting that the heat exchanger HX1 includes an economizer, an evaporator and a superheater while HX2 is a superheater.

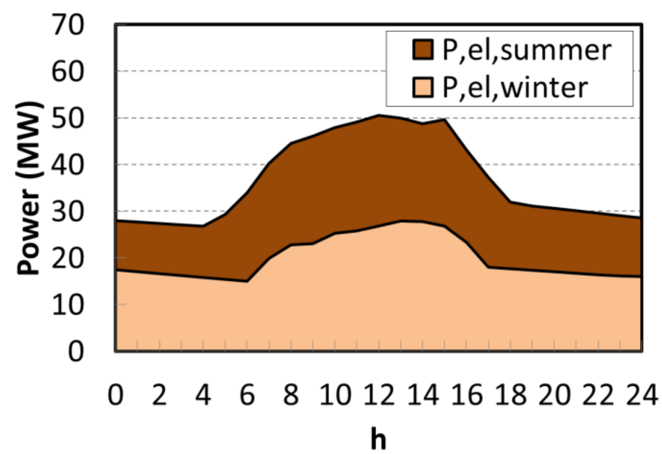

Figure 1 Day electric load profiles

The solarized Rankine cycle was designed to produce 62.2 MW at ISO condition in order to match the summer peak of the electric demand, i.e. 51 MW (Figure 1). The net electric efficiency at design point results to be 41.4%. Sliding pressure working conditions were assumed for all turbine sections except for steam admission to the HP turbine, which was regulated by a 4-sector multi valve, with a minimum pressure drop of 2%. The auxiliary boiler was included in the plant to provide backup capability.

Table 1 Rankine cycle design parameters (ISO cond.)

Turbine inlet temperature (°C)	540
Turbine inlet pressure (bar)	100
Steam mass flow at turbine inlet (kg/s)	53.1
Reheat temperature (°C)	500
Average turbine efficiency (%)	88.1
Condenser pressure (bar)	0.06
HX1 thermal power (MW)	131.0
HX2 thermal power (MW)	19.4
Net electric power (MW)	62.2
Thermal efficiency (%)	41.4

Solar field

Two different configurations have been considered for the solar field: i) Parabolic Trough Collectors (PTC) and ii) Heliostat field with Central Receiver (HCR). The goal is to investigate which solar configuration is the most appropriate to meet a variable heat demand, according to the instantaneous electric load on the grid. It has to be reminded that the CSP plant is designed to operate in "*island mode*": hence TES and solar field must cover hour by hour the heat demand required by the Rankine cycle all over the year.

For each solar configuration a two-tank molten salt direct storage system is considered. HTF coming from solar field fills the hot tank; then it is withdrawn to transfer heat to the steam generator. A cold tank finally collects molten salt exiting boiler heat exchangers and acts as a buffer.

When the hot storage tank level is reduced to a minimum, the auxiliary heater is switched on to heat HTF mass flow rate required by the power block.

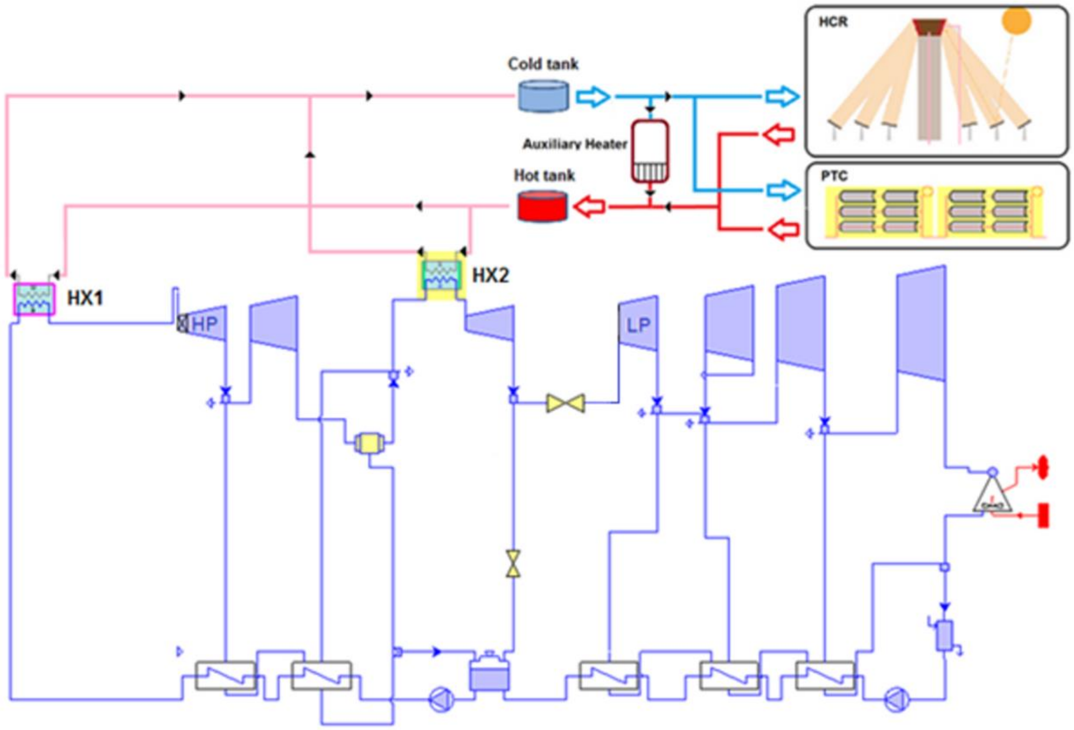

Figure 2 Schematic of the investigated CSP plant: screenshot of Thermoflex®

SIMULATION METHOD AND ASSUMPTIONS

Simulations have been carried out by means of an integrated procedure based on two different codes: Trnsys® with the model libraries STEC and TESS, for the solar field and thermal storage section, while Thermoflex® for the power block. This procedure allowed for taking the best features of both codes, so to achieve accurate off design simulations for an entire year. Meteonorm database from the Trnsys® weather library provides annual meteorological data for the selected site climatic conditions and latitude. In the present study CSP plant was supposed to be located in Upington (RSA). Full details on solar devices modeling are given by reference [7,10].

An iterative procedure within Thermoflex® provided hour-by-hour HTF flow rates ensuring that the Rankine cycle power output matches the electric demand. Then, Trnsys® took those HTF flow rates as a mandatory request to be fulfilled by the solar field (PTC or HCR) coupled with TES and auxiliary heater. Therefore, the solar block provides the required molten salt flow rate just to feed the power cycle through HX1 and HX2.

Rankine cycle design conditions may differ significantly from actual operation. In fact the steam flow rate entering the HP turbine varies strongly with electricity demand throughout the day. In order to carefully simulate the steam turbine off-design behavior, admission control valves and exhaust losses were included in the modeling.

An optimization procedure interacting with Trnsys® model was used to size the two considered solar field solutions. On the base of annual Trnsys® simulations, the optimization algorithm determines the minimum aperture area of the solar field assuring the required HTF flow rate from TES and a minimum annual solar fraction of 90%. Charging and discharging cycles of TES were ruled by the HTF flow rate required for each hour of the year. When molten salt level in the hot storage tank falls below 1% of the total capacity, the gas-fired auxiliary heater switches on and integrates the solar collected heat to guarantee the required flow rate.

The same storage capacity was selected for both solar field configurations. A 48,000 m^3 volume tank was assumed, able to compensate the fluctuations in the heat demand and solar energy availability. Table 2 summarizes main data of TES and solar field for the two investigated configurations resulting from the optimization. It is worth highlighting that solar tower (HCR) requires a significantly lower (-27.4%) aperture area to fulfill the heat demand, thanks to the annual efficiency trend which will be discussed in the following section.

Table 2 Solar Field and TES data

	PTC	HCR
Aperture area (m^2)	612,500	444,960
Tank volume (m^3)	48,000	

RESULTS AND DISCUSSION

Simulations of the CSP plant have been carried out over a one-year period to evaluate the annual performance for both the solar field configurations. Results will focus first on two representative summer and winter days in order to enlighten the plant behavior during the extreme conditions occurring over a year. The Rankine cycle efficiency, evaluated as the ratio of the net power output to the solar heat input through HX1 and HX2, is reported in Figure 3, while Figure 4 shows the condenser pressure variations. Both in winter and summer days, cycle efficiency is in the range between 34% and 38.5%, but with some notable differences. In summer, the highest the efficiency the lowest the condenser pressure, meaning that cycle performance mainly depends on ambient conditions. In winter, the cycle efficiency profile reflects the electric demand, suggesting that ambient conditions, in cold weather, play a minor role with respect to the steam entering the condenser.

The condenser pressure changes during the day (Figure 4) as the result of air temperature variation, but also of the steam flow rate discharged by the turbine. It should be remembered that the steam flowing through the turbine is computed to match the requested electric power hour by hour, and its daily trend closely follows the power demand profile. During the hottest hours of the summer day, the air cooled condenser operates at a pressure of 0.22 bar as a consequence of ambient temperature as high as 38°C. In the winter day condensing pressure gets quite low levels, even lower than the design value of 0.06 bar: this thanks to favorable weather conditions and to reduced steam flow due to the low electric power demand.

Summer and winter day results for the parabolic troughs configuration are shown in Figure 5. Plots report intercepted solar radiation (Q_{rad}), effective collected heat (Q_{coll}), instantaneous heat demand required by the power plant (Q_{dem}) and the fossil auxiliary heat (Q_{aux}). The hot storage tank level is also reported.

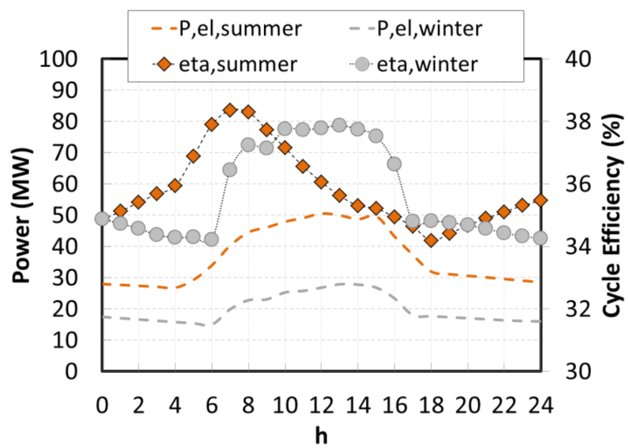

Figure 3 Power and cycle efficiency in summer and winter day

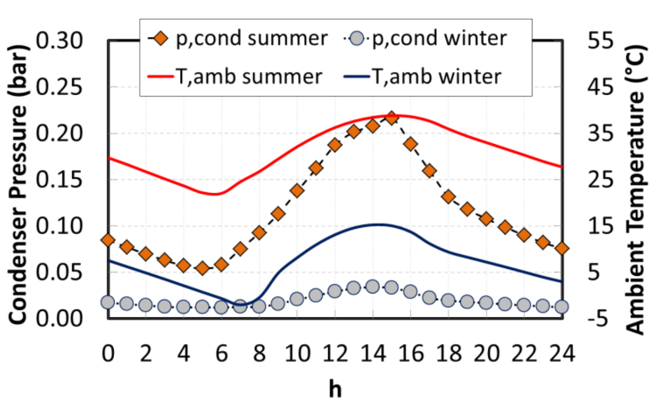

Figure 4 Condenser pressure in summer and winter day

A significant difference in PTC performance between summer and winter days can be noted: in the central hours of a sunny day the collected heat is strongly exceeding the heat input required by the power plant, thus allowing the hot storage

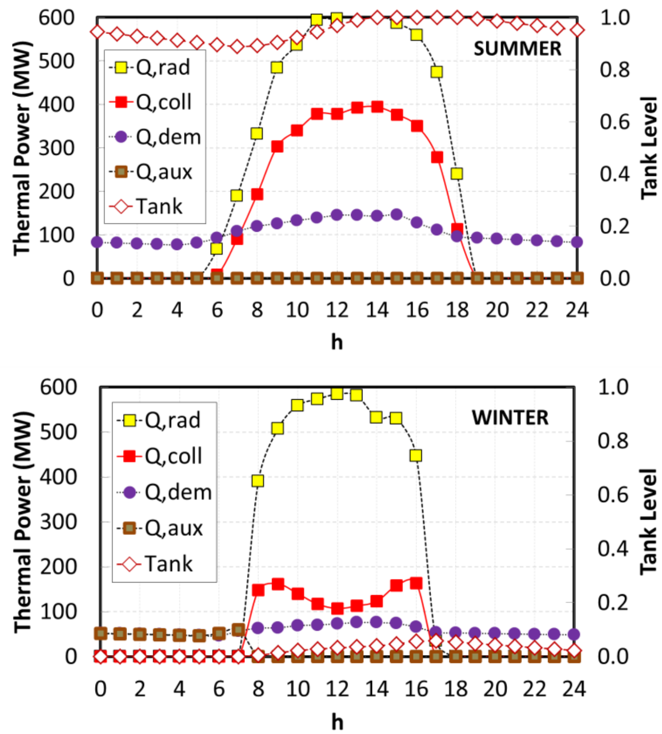

Figure 5 Solar Block day simulation results (PTC case)

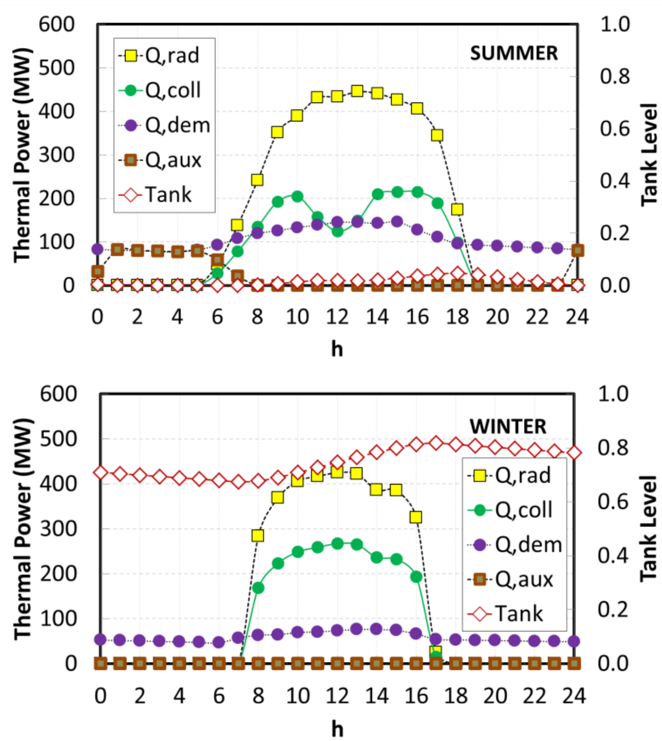

Figure 6 Solar Block day simulation results (HCR case)

tank to charge. In the early afternoon of the summer day, i.e. from 14h to 18h, TES is completely full: this requires defocusing of some parabolic troughs [11]. The storage permits a 24-hour operation without auxiliary heater integration.

In winter day a quite different behaviour takes place. In sunny hours PTC field is able to collect the necessary heat to drive the power block and the slight energy surplus allows only a small TES charging. However the stored heat is not enough to fulfil all the night-time power demand and auxiliary heater integration takes place (from 0^h to 7^h).

Note that the presence of the auxiliary fossil fuel boiler permits to reduce the required PTC aperture area with respect to a full solar CSP plant. However, the assumed constrain of an annual SF higher than 90% made necessary to install a collector surface of 612,500 m^2. This wide area in summer collects more heat than required, so thermal damping (i.e. PTC defocusing) is needed for long time.

Figure 6 shows the same results for the second plant configuration, based on the solar tower field. In this case, a lower heliostats aperture area and consequently a lower solar intercepted radiation Q_{rad} are needed to cover the heat demand: the peak value is about 450 MW against 600 MW of PTC case. In the central hours of summer days, when the Sun is near to Zenith position, a typical decrease of collected power takes place due to high heliostats to tower reflection angle. In winter this effect does not occur and in sunny hours collected thermal power is more constant. Conversely to PTC case, fossil integration takes place in the summer day, for about 6 hours (from 1^h to 7^h): this is the result of the low HCR optical efficiency when the Sun is close to Zenith. The different behaviour of the two investigated solar fields is compared in more details in Figure 7, where the solar-to-thermal efficiency is reported.

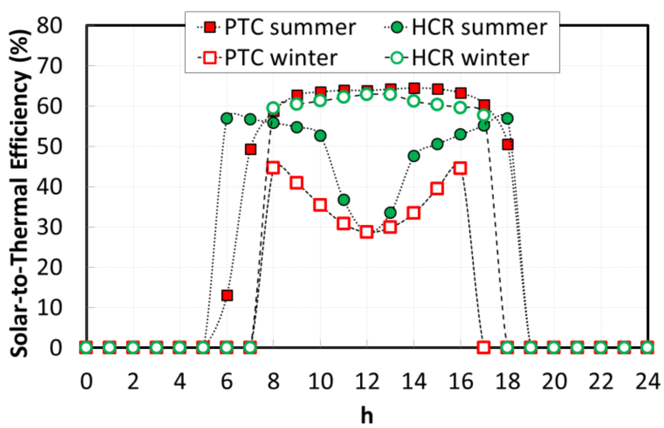

Figure 7 Solar-to-Thermal Efficiency (%)

In summer (full symbols) parabolic troughs exhibit a very high efficiency. Conversely, for the HCR configuration a relevant reduction in the collected solar energy takes place around midday. This is related to geometric limits of heliostats in reflecting the incident radiation toward the top of the tower.

In winter (hollow symbols), when the Sun is farthest from the Zenith, the fraction of solar radiation collected by HCR significantly increases. On the opposite, PTC efficiency dramatically decays because of the cosine effect and lower ambient temperatures.

Global performance from annual simulations over a one-year period have been computed. Figures 8 and 9 report the amount of the available solar energy (E_{rad}), the available collected energy (without defocusing) (E_{coll}), the auxiliary heat (E_{aux}) and the power block thermal energy demand (E_{dem}), on monthly basis.

There are substantial performance differences between the two solar concentration technologies. PTC configuration exhibits a moderate excess (up to 49% in November) in the collected heat from September to February that requires the defocusing of many troughs. This excess is due to the need to cover E_{dem} in winter months when PTC solar-to-thermal efficiency is quite low; so a very large aperture area is required in PTC field to meet the heat demand. Fossil integration occurs between April and August.

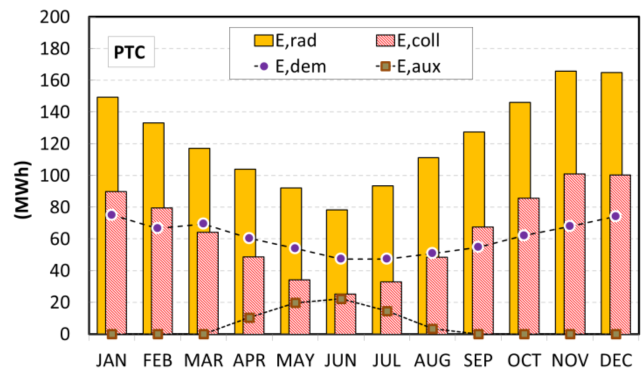

Figure 8 Monthly results for PTC field

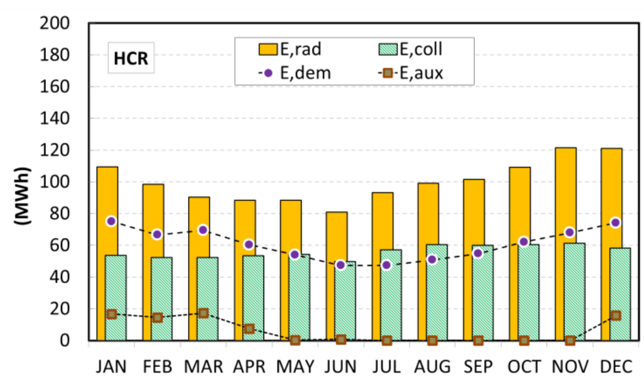

Figure 9 Monthly results for HCR field

Conversely (see Figure 9), the monthly thermal production for the HRC configuration remains fairly stable throughout the year: this behavior is strictly related to solar tower field efficiency that is lower in summer and higher in winter (Figure 7). Only a very small overproduction takes place in July and August, when the power demand is lower. Auxiliary heater needs to be switched on in summer months.

In order to globally evaluate the performance of the two investigated configurations, the annual energy balance reported in Table 3 has been evaluated. One can note that CSP plant based on HCR requires a lower amount of solar energy to drive the power cycle (-18.9%) and the surplus of collected heat is minimized (2.9%). The PTC case, on the opposite, exhibits a 14.6% annual overproduction (requiring collector defocusing) because collector field has been oversized to cover the heat demand in winter.

Table 3 PTC vs. HCR: one-year period performance

	PTC	HCR
Solar energy (w/o defocusing) (GWh)	1482.4	1201.7
Collected heat (w/o defoc.) (GWh)	777.3	671.2
Heat demand (GWh)	729.7	
Heat from solar source (GWh)	659.5	657.0
Heat from fossil fuel (GWh)	70.2	72.8
Energy surplus (%)	14.6%	2.9%
Aver. solar-to-thermal efficiency (%)	52.4%	55.9%

Finally the CSP plant performance for Upington location is compared to that one achievable with the same power block operating under meteorological conditions and latitude of Sevilla (ES). The electricity demand profile was assumed to be the same. Simulations with Thermoflex® have been carried out again to take into account the effect of the different ambient temperatures on the condenser performance. The new profile of the required heat was used as input of the solar field optimization algorithm to determine the new required aperture area for both configurations (PTC and HCR), under the same constrain of a SF higher than 0.9. The results of the optimization and of the annual simulations are summarized in Table 4.

Table 4 Upington vs. Sevilla: one-year period performance

	UPINGTON		SEVILLA	
	PTC	HCR	PTC	HCR
DNI (kWh/m^2)	2700.8		1862.6	
Aperture area (m^2)	612,500	444,960	1,650,000	597,840
Tank volume (m^2)	48,000		48,000	
Energy surplus (%)	14.6%	2.9%	50.4%	0%
Cycle efficiency (%)	36.5%		37.0%	
Solar-to-thermal efficiency (%)	52.4%	55.9%	48.3%	57.6%

The most relevant difference is related to the required aperture area. Because of the lower solar irradiance, moving from Upington to Sevilla makes the collector surface increase by 34.4% for HCR case. The increase is even more dramatic for PTC case: +169.4%. This is due to the PTC low efficiency in winter, when the power demand has to be followed and fossil contribution cannot exceed 10% (on annual basis). This has an impact on the huge energy surplus in summer (more than 50% on annual basis). Looking at the cycle efficiency, the average value is 0.5% higher for the Sevilla case, thanks to lower average ambient temperature. As regards to the solar field efficiency, at Sevilla PTCs show a penalty of 4%, whilst the latitude beneficially affects the HCR performance (+1.7%).

CONCLUSION

A CSP plant including a steam Rankine cycle, with thermal energy storage and auxiliary heater was modeled according with a load following strategy. Two solar field configurations were compared: PTC and HCR. Simulations were performed over a one-year period for Upington. Then the annual simulations were carried out also for Sevilla climatic conditions. For both locations PTC technology requires a larger aperture area than HCR to provide the heat input to the Rankine cycle. HCR appears to perform better than PTCs, and this is particularly evident in the Sevilla case. It can be concluded that solar tower is the best solution for a CSP plant in terms of load-following capability both in Upington and Sevilla.

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