Techno-economic analysis of an innovative purely solar driven combined cycle system based on packed bed TES technology

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Techno-Economic Analysis of an Innovative Purely Solar Driven Combined Cycle System based on Packed Bed TES Technology

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Abstract. The present work performs a techno-economic analysis of a purely solar driven combined cycle composed of a solar air receiver directly connected to a topping gas turbine coupled to a bottoming packed bed thermal energy storage at the gas turbine exhaust, which runs in parallel to a bottoming steam cycle. Capacity factor, capital expenditure, and Levelised Cost of Electricity have been considered to assess the plant performance. A sensitivity analysis has been performed in order to understand the influence of solar multiple, energy storage capacity and gas turbine expansion ratio over the plant key performance indicators. The results show that the studied solar driven combined cycle is more costly than conventional ones, and therefore it leads to higher Levelized Cost of Electricity. However, it enables a complete reduction of CO₂ emissions and increased flexibility in the system with the help of the introduction of an intermediate packed bed thermal energy storage. Furthermore, large solar multiple, medium storage capacity and complete expansion ratio through the gas turbine enable enhanced system performance. Finally, further works including optimized dispatch algorithms could enable a proper evaluation of the economic profit given by the flexibility offered by the storage unit and by a potential control of the Brayton cycle recuperation level in the modified plant layouts.

INTRODUCTION

A worldwide growing demand for electricity together with worsening global environmental problems has led to the development of innovative and more sustainable power generation solutions. Among these, concentrating solar power (CSP) has begun to achieve a growing share of the global energy market. Recently several research projects have been focused on the improvement of CSP technologies. Of particular interest in the context of this work, high-temperature pressurized air receiver have been developed [1, 2], hybrid solar gas turbine system have been proposed [2,3] and solar combined cycles have been studied [4]. In this work, a CSP driven combined cycle system has been introduced, sketched in Fig. 1, and its techno-economic performance has been assessed. In doing so, a transient thermodynamic model of the CSP plant has been developed in the Trnsys software®. The study has been extended to the economic side analyzing the capital cost (CAPEX) of all plant components and the Levelised Cost of Electricity (LCoE). A sensitivity analysis has been performed in order to evaluate the influence of the main design choices over the final CSP plant key performance indicators (KPI). Once identified the main weaknesses of the studied CSP plant, two different system modifications have been proposed and their influence on the main KPIs of the plant have been investigated. As expected, it is shown that the analyzed CSP plant layout is more costly than conventional CCGTs. However, thanks to the high level of flexibility, which is granted by the two power cycles and the intermediate thermal energy storage (TES) unit, and the complete cut of CO₂ emissions, a solar driven combined cycle could still be a viable possibility. Modified plants schemes, including recuperated GT cycles could be worth further investigation if included in hybrid solar system, where the turbine inlet temperature can be raised over the receiver’s technical limitations by dint of the use of a combustor. Furthermore, in similar non–hybrid CSP, the relatively low temperature achieved in the bottoming cycle might be used by integrating low temperature Organic Rankine Cycles as bottoming unit.
Ultimately, further researches on optimized control and dispatch algorithms, taking into account variable electricity price scheme, load and weather fluctuations, could highlight the value of decoupling the topping and bottoming cycles through an intermediate TES.

**SYSTEM CONFIGURATION**

The present study introduces a purely solar driven combined cycle CSP plant with an integrated packed bed thermal energy storage system. The proposed 55 MWe CSP plant scheme is shown in Figure 1. When the direct solar irradiation (DNI) is higher than a minimum value, the air at ambient conditions is compressed up to 15 bar by the compressor of the Brayton cycle, then the air is heated up to 950°C at the outlet of the receiver by means of the concentrated solar irradiation. Then the high-temperature compressed air is expanded in the turbine for power generation. Once at the gas turbine outlet, the exhaust gases still carry a large amount of energy, mainly due to their relatively high temperature. Therefore, in order to recover this energy, a bottoming steam cycle is coupled with the Brayton cycle. Moreover, a packed bed TES system has been integrated into the system in order to store energy during the day and enable power production via the steam cycle during the night. Specifically, at the GT outlet, the air flow is divided into two different streams. One goes to the heat recovery steam generator (HRSG) of the Rankine cycle; while the other flows to the TES unit. The share between the two mass flow rates depends on the actual flow thermodynamic conditions and TES state of charge (SoC). Later on, when no solar power is available, such as during night or cloudy periods, the TES unit is discharged, extending the electricity production well after sunset and increasing the plant capacity factor. The TES discharge is limited by two threshold values for SoC and outflow temperature. This specific arrangement leads to improvements in the system flexibility and thus increases the CSP plant market viability. Indeed, to further profit from the combined cycle and TES integration, specific control strategies can be applied in order to fulfill different load requirements and maximize the economic income according to the electricity market. In this work, a deterministic plant control logic has been developed, based on a constant electricity price. Figure 2 shows a flow chart for the control logic, while Table 1 reports the main implication of each operational mode (OM) on the plant behavior. Two threshold DNI levels are considered: \( DNI_{\text{min}} \) corresponds to the minimum incoming solar energy, which enables to operate the ST at its minimum off-design load; while, \( DNI_{\text{min2}} \) corresponds to the incoming solar energy that permits to operate the ST at its design power output. Besides, TES SoC has been allowed to vary between \( \text{SoC}_{\text{min}} \) and \( \text{SoC}_{\text{max}} \), 0.10 and 0.90 respectively, for stability reasons. During day time, when the DNI is higher than \( DNI_{\text{min2}} \), a variable air mass flow, evaluated in order to obtain a constant receiver outlet temperature, is compressed and flows through the receiver and the GT. Therefore, the GT needs to operate under flexible working conditions enabling wide off-design ranges. Then, at the GT outlet a constant air flow is sent to the HRSG enabling ST power production under design conditions. While the excess of thermal energy, inheriting the DNI fluctuations, circulates to the TES unit, as long as the TES SoC is within the limits (OM 1). Otherwise, if the SoC reaches \( \text{SoC}_{\text{max}} \) all GT outflow is sent to ST, which therefore works under off-design conditions (OM 2). Similarly, the plant undergoes OM 2 also when the DNI level is between the two minimum thresholds. Whenever the incoming solar irradiation is below \( DNI_{\text{min1}} \), the ST is disconnected and only the GT is operated at low partial load. The GT exhaust gases are then directed towards the TES unit, as long as the TES SoC remains within the limits (OM 3). If SoC reaches its maximum threshold, then the energy of the GT exhaust gas is wasted. Finally, during nighttime, for  

![FIGURE 1. CSP driven combined cycle plant layout](image-url)
as long as the TES SoC and its outflow temperature \(T_{\text{topTES}}\) remain above \(SoC_{\text{min}}\) and \(T_{\text{min}}\), the TES unit is discharged and the Rankine cycle is operated with a gradually decreasing power output due to the gradual TES thermocline discharge (OM 5). Once the thermal energy in the TES is depleted the plant is stopped and undergoes OM 6.

![Control logic decision diagram](image)

**FIGURE 2.** Control logic decision diagram

<table>
<thead>
<tr>
<th>OM</th>
<th>GT</th>
<th>ST</th>
<th>TES</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>ON</td>
<td>ON</td>
<td>Design Charge</td>
</tr>
<tr>
<td>2</td>
<td>ON</td>
<td>ON</td>
<td>Off-Design OFF (Full)</td>
</tr>
<tr>
<td>3</td>
<td>ON</td>
<td>OFF</td>
<td>Charge</td>
</tr>
<tr>
<td>4</td>
<td>ON</td>
<td>OFF</td>
<td>OFF (Full)</td>
</tr>
<tr>
<td>5</td>
<td>OFF</td>
<td>ON</td>
<td>Off-Design Discharge</td>
</tr>
<tr>
<td>6</td>
<td>OFF</td>
<td>OFF</td>
<td>OFF (Empty)</td>
</tr>
</tbody>
</table>

**TABLE 1.** Main implications of the different OMs on the CSP plant

Modified Plant Configurations

In order to enhance the performance of the CSP driven combined cycle, two different system modifications have been proposed. The modified plant schemes are reported in Fig. 3a and 3b. Specifically, Fig. 3a shows the modified plant 1 (MP 1), where a recuperator has been integrated before the receiver air inlet, having part of the GT exhaust gas flowing through the hot side and the flow from the compressor to the receiver at the cold side. Therefore, at the GT exhaust, in this specific system configuration, a fixed percentage of the overall air flow is sent to the hot side inlet of the recuperator. This would lead to higher receiver inlet temperature and so higher air mass flow rates, to maintain the receiver power balance. The amount of GT exhaust gas sent to the recuperator, \(f_{\text{GTtoRecap}}\), is a critical value in the overall management. Indeed, at each time step it will affect the receiver air mass flow rate and further influence the GT power production, the air mass flow sent to the TES unit, and the GT exhausts mass flow sent to the recuperator.

![Modified CSP plant layout 1, MP 1 (a); Modified CSP plant layout 2, MP 2 (b)](image)

**FIGURE 3.** Modified CSP plant layout 1, MP 1 (a); Modified CSP plant layout 2, MP 2 (b)
during the following time step. Fig. 3b shows the modified plant 2 (MP 2) where the recuperator hot side outlet has been further connected to the evaporator hot side inlet. This arrangement enables to further exploit the thermal energy carried by the recuperator hot side outflow, increasing the thermal power input at the HRSG evaporator. Besides the considerations relative to MP 1, in MP 2 the fraction of GT exhausts sent to the recuperator influences also the ST power production, and potentially it could increase the power output during OM 1 and OM 2. In both MP 1 and MP 2, in order to obtain thermodynamic conditions in the Brayton cycle, which could enable a proper recuperation, the GT expansion ratio has been reduced. Thus in the modified plants a maximum GT expansion ratio of 12 has been considered.

**METHODOLOGY**

Several interconnected models have been built in order to study the thermo-economic performance of the proposed solar driven combined cycle CSP plant:

- A steady state thermodynamic model of the CSP plant to assess its design working conditions and the main design input values for the different components of the plant.
- A transient Trnsys® model, with integrated control logic and meteorological data gathered from Meteonorm database, where annual simulations have been performed.
- A System Advisor Model model to evaluate the solar field efficiency matrix.
- An economic model, in MatLab scripts, to calculate all capital and operational costs functions, based on [3, 5, 6], and finally to determine the CSP plant KPIs.

At first, the techno-economic performance of the studied CSP plant have been evaluated for a fixed base scenario. Table 2 summarizes the main plant design parameter assumed for the base case. Secondly, in order to evaluate the influence of the main design choices over the CSP plant KPIs, a sensitivity analysis has been performed. The solar multiple (SM), the size of the TES unit, the expansion ratio of the gas turbine, $\varepsilon_{GT}$, and, as a consequence, the nominal power ratio between gas turbine and steam turbine, $\gamma = \frac{P_{GT}}{P_{ST}}$, have been considered as the input variables of the sensitivity analysis. Such a set of decision variables results to be of particular interest for the overall plant behavior. Specifically, the solar multiple directly affects the solar field size, the collected thermal power, the air mass flow rate and the associated costs. The TES unit is the major responsible for the plant flexibility and for its CF extension during nighttime. Therefore, its size refers to the amount of storable energy, which could be further exploited by the Rankine power unit. The expansion ratio of the gas turbine has been considered as a last input parameter of the sensitivity analysis. Indeed, it is directly connected to the GT outlet thermodynamic flow conditions and therefore to the potential power production in the bottoming steam cycle. As a consequence of varying the $\varepsilon_{GT}$, the ratio between the nominal power in the GT and ST has been varied. Particularly, a decrease of $\varepsilon_{GT}$ leads to a reduction of the nominal power of the GT turbine and an increase of the ST nominal power due to the increase of the steam turbine inlet temperatures. Table 3 reports the ranges and step considered for the sensitivity analysis. The overall net energy production, capacity factor, CAPEX and LCoE have been considered as the plant KPIs. Specifically, the plant CF and LCoE has been defined as in Eqs. (1) and (2).

$$CF = \frac{E_{NET}}{\left[P_{GT} \cdot (24 - TES_{size}) + P_{ST} \cdot 24\right] \cdot 365} \quad (1)$$

$$LCoE = \frac{\alpha \cdot CAPEX + \beta \cdot C_{decom} + OPEX}{E_{NET}} \quad (2)$$

**TABLE 2. Plant design parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Direct Normal Irradiance, $DNI_{DES}$ [W/m²]</td>
<td>850</td>
</tr>
<tr>
<td>Receiver Outlet Temperature, $TIT_{DES}$ [°C]</td>
<td>950</td>
</tr>
<tr>
<td>Net Electrical Power, $W_{DES}$ [MWₜ]</td>
<td>60</td>
</tr>
<tr>
<td>GT to ST Power Ratio, $\gamma$ [-]</td>
<td>10</td>
</tr>
<tr>
<td>GT Expansion Ratio, $\varepsilon_{GT}$ [-]</td>
<td>15</td>
</tr>
<tr>
<td>Solar Multiple, $SM$ [-]</td>
<td>2</td>
</tr>
<tr>
<td>TES size, [h]</td>
<td>10</td>
</tr>
</tbody>
</table>

Where $P_{GT}$ and $P_{ST}$ are the nominal gas and steam turbine power, $C_{decom}$ is the cost for plant decommissioning, considered equal to 5% of the cost for all equipment, civil engineering and installation. $\alpha$ and $\beta$ are two factors considered in order to covert the capital expenditure and the decommissioning costs into equivalent annual payments, the factors’ definition have been gathered from [3]. Finally, in order to enhance the CSP plant performance two
different system modifications have been proposed, the relative plant layout are shown in Fig 3a and b. In order to
assess the KPIs for the two changed CSP plants, the whole set of models has been modified to represent the new plants
behaviors. For both MP 1 and MP 2, a further sensitivity analysis has been performed in order to assess the influence
of $\epsilon_{GT}$ and the amount of GT outflow sent to the recuperator, $\epsilon_{StoRecup}$. Indeed, the latter has been identified as a
critical parameter for the overall plant behavior. For this part of the study the variation of the considered parameters
is summarized in Table 4.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>TESsize</th>
<th>SM</th>
<th>$\epsilon_{GT}$</th>
<th>$\gamma$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base case</td>
<td>10 h</td>
<td>2</td>
<td>15</td>
<td>10</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>8 h,12 h, 15h</td>
<td>1.5, 1.75</td>
<td>7.5, 10, 12</td>
<td>2.5, 5, 7.5</td>
</tr>
</tbody>
</table>

**RESULTS AND DISCUSSION**

**Base Case Scenario**

In this section, the thermodynamic performance of the proposed air-driven CSP sCO2 plant are investigated. The
studied CSP plant is located in Seville, Spain (37.39°N, -5.99°E). Figure 4a reports the evolution over a single day
(5th January) of the net power produced by the GT and ST, the DNI with the indication of the two thresholds level
($DNI_{min1}$ and $DNI_{min2}$) and the TES SoC. Furthermore, the specific OM are also highlighted. The plant works as
imposed by the control logic, producing power during daylight mainly with the GT, and extending the generation
during night though at a much lower load, defined by $\gamma$. It is worth noting that the flexibility requirements for the two
turbomachinery units are different. Indeed, the GT has to be able to work under highly fluctuating air mass flow,
determined by the local DNI value. On the contrary, the ST works under design or slightly away from design
conditions for the majority of the time. During the second part of OM5, the thermal input power at the HRSG gradually
decreases due to a gradual decrease of the TES outlet temperature, therefore, only during this period, ST has to work
under partial load conditions. Figure 4a shows also the TES SoC and its variable charge rate. However, due to the
large thermal inertia offered by the TES unit, the sharp variations of the DNI are flattened down, and are just partially
mirrored by a variation in the TES rate of charge.

The share of the capital cost among the different major plant components is summarized in Fig 4b. The whole
solar part, consistent of solar field, tower and air receiver accounts for about half of the overall CAPEX. The two
major components are the solar field, mainly due to the large SM considered for the base case (SM=2), and the

**TABLE 3.** Design choice parameters variation for the sensitivity analysis on the original plant layout.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>TESsize</th>
<th>SM</th>
<th>$\epsilon_{GT}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base case</td>
<td>10 h</td>
<td>2</td>
<td>15</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>8 h,12 h, 15h</td>
<td>1.5, 1.75</td>
<td>7.5, 10, 12</td>
</tr>
</tbody>
</table>

**TABLE 4.** Design choice parameters variation for the sensitivity analysis on MP 1 and MP 2.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>TESsize</th>
<th>SM</th>
<th>$\epsilon_{GT}$</th>
<th>$\gamma$</th>
<th>$\epsilon_{StoRecup}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base case</td>
<td>10 h</td>
<td>2</td>
<td>15</td>
<td>10</td>
<td>0.2</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>-</td>
<td>-</td>
<td>7.5, 10, 12</td>
<td>2.5, 5, 7.5</td>
<td>0.15, 0.25</td>
</tr>
</tbody>
</table>

**FIGURE 4.** Net power, DNI, SoC and OMs for 5th January (a); CAPEX share among all plants components for base case (b).
receiver, which becomes particularly expensive due to the need of high temperature materials to enable the target working conditions. Among the power cycles, the large $\gamma$ leads to costs for the Brayton unit much higher than the one for the Rankine unit. Besides, the considered TES unit results to be an extremely cheap solution.

**Sensitivity Analysis**

The main results for the sensitivity analysis are summarized in Fig. 5. Figure 5a shows the LCoE and CAPEX behavior with respect to changes in all the considered input variables of the sensitivity study, namely the SM, the size of the TES unit and the expansion ratio of the gas turbine, $\epsilon_{GT}$. The typical trade-off between a decreasing LCoE for increasing CAPEX is identified. As expected, for small solar fields, a decrease in the CAPEX is shown. However, this reduction is not mirrored by the LCoE, which increases. This behavior is mainly attributable to the fact that a decrease of SM leads to lower plant energy production, which hinder any reduction of the LCoE. Furthermore, for increasing $\epsilon_{GT}$, larger capital expenditure are required. Indeed, for higher GT expansion ratio the Brayton unit cost share increases, and this growth is just partially compensated by a reduction of the Rankine unit cost. The Rankine cycle turbomachinery cost decreases but the heat exchangers costs increase due to worse thermodynamic conditions along the cycle. Nonetheless, a reduction of the LCoE is achieved with larger $\epsilon_{GT}$, due to higher amount of energy produced during daylight by the gas turbine. Different behaviors of the CSP plant with respect to the third analyzed parameter, the TES capacity, can be pointed out. Indeed, at high SM and low to medium GT expansion ratio large TES capacities enable clear LCoE reductions, without introducing significant capital cost increases. Reasons for this can be found in the fact that, at large solar field and low $\epsilon_{GT}$, the energy content of the GT exhausts is higher and the plant relies more extensively on the steam cycle power production. Instead, gradually increasing the GT expansion ratio, the TES influence on the LCoE become less and less important. On overall, a minimum LCoE is found at 176.37 $/MWh, for a CAPEX equal to 261.82 M$ and for a TES size, $\epsilon_{GT}$ and SM equal to 10 hours, 15 and 2, respectively. These plant conditions leads to a CF of about 49%. The influence of the considered design parameters over the plant capacity factor is reported in Fig 5b. As expected, the TES size and the SM plays major roles. Specifically, larger TES capacity enables longer discharge phase and therefore higher CF. Similarly, large solar field, particularly if coupled to big TES units, permits to enhance the amount of collected and stored solar energy, leading to improved CF. Differently, high values of the GT expansion ratio cause reductions in the plant capacity factor. Indeed, a decrease of $\epsilon_{GT}$ requires a smaller ratio $\gamma$, therefore the ST nominal power increases leading to higher power production during nighttime. Moreover, for the largest considered TES size, 15 h, a maximum trend of the plant CF for different $\epsilon_{GT}$ can be identified. On overall, a maximum CF is found at 72.34% for a CAPEX equal to 257.14 M$, leading to an LCoE of 186.75 $/MWh. In order to attain these KPIs, a TES size of 15 hours, an $\epsilon_{GT}$ of 12 and a SM equal to 2 are required. The main KPIs achieved by the solar driven CCGT system have been compared against the ones attained by a traditional natural gas fueled combined cycle. The comparative analysis is shown in Table 5. It reveals that the solar driven CCGT is about 2.2 times more expensive than a traditional power unit, at such working conditions and capacity factor. The higher LCoE is principally due to the high initial investment required for the solar part, which, as shown in Fig 4b,
represents about a half of the overall solar driven CCGT CAPEX. However, large savings in the annual operational costs can be obtained, as the major OPEX sources in the conventional system are related to the fuel consumption and CO₂ emissions taxes. Finally, as a pure renewable energy based power unit the analyzed system enables a complete reduction of GHG emissions.

| TABLE 5. KPIs comparison between solar driven and traditional CCGT. |
|-------------------------|------------------|------------------|
| KPIs                    | Solar Driven CCGT | Conventional CCGT |
| LCoE [$/MWh]            | 176.37           | 79.91            |
| CAPEX [MS]              | 261.8            | 100.7            |
| OPEX [MS]               | 1.32             | 9.43             |
| Emissions [kgCO₂/MWh]   | 0                | 383.29           |

**Modified CSP Driven Combined Cycle Layouts**

In order to enhance the CSP driven combined cycle performance, two different system modifications (MP 1 and MP 2) have been proposed. The modified plants schemes layouts are shown in Fig. 3a and 3b. The main goals of the introduced changes to the original plant scheme are: to improve the power cycle efficiency, particularly the Brayton unit, to limit the wasted power and consequently to increase the overall energy production. All of these, if achieved, should enable an enhancement in the thermodynamic plant performances. Although, recuperation is not considered for traditional combined cycle, in the context of this work it has been considered has a potential additional source of flexibility for the plant. The suggested modifications require higher initial investments, related to the recuperator unit and to an increased complexity of the whole plant. The main results for the plant KPIs in the two revised plant layouts are shown and compared with the base case in Fig. 6a and 6b. As a base case the results for a variable GT expansion ratio, SM equal to 2 and TES size of 10 hours have been considered. For sake of simplicity, the GT exhausts mass flow rate fraction sent to the recuperator has been called f in both Fig. 6a and 6b. Solid lines represent the results for the MP 1 case and dashed lines report the outcomes for MP 2. In order to obtain the required thermodynamic conditions for a proper recuperation within the Brayton cycle, the GT outlet temperature has to be higher than the air compressor outlet temperature. Therefore, only GT expansion ratios of 12, 10 and 7.5 have been considered. On overall, KPIs trends similar to the base case are recorded, with decreasing LCoE and CF and increasing CAPEX for larger εgt. The MP 1 layout does not lead to any significant improvement, resulting in KPIs similar or slightly worse than the one achieved by the base plant layout. Particularly, CFs slightly higher than the base case are recorded for the minimum fraction of GT exhaust sent to the recuperator (f = 0.15) and low to medium GT expansion ratios as shown in Fig 6b. Besides, higher LCoE are evaluated. Indeed, the energy production of MP 1 increases with respect to the base case only for a low GT exhausts fraction sent to the recuperator (f = 0.15) and for low GT expansion ratio (εgt equal to 7.5 or 10). However, these changes in the energy production do not enable to compensate for the cost increases. Furthermore, higher fraction of GT exhaust gas deflected towards the recuperator hot inlet leads to further plant KPIs worsening. Contrarily, for MP 2 larger deviations from the original base case are shown. Specifically, an improvement of CF of about 3.3% could be achieved for an εgt equal to 7.5. Consequently, notwithstanding the CAPEX increment, LCoE reductions between 2 and 3% would be enabled for the MP 2 layout. Furthermore, especially for high εgt,
larger GT exhaust gas fraction sent to the recuperator enables further KPIs improvements. However, for increasing GT expansion ratios the achieved KPIs enhancements decrease. As a result, the minimum LCoE and the maximum CF achieved by the MP 2 layout are 184 $/MWh and 54.4%, respectively. Although, both results show enhancements with respect to the base case at the correspondent GT expansion ratio, they do not overcome the best KPIs achieved for the base layout working with a full expansion in the Brayton cycle. Nonetheless, the improvements achieved by means of MP 2 could lead to more cost competitive plant solutions when analyzed for larger scale units and including a dynamic and adaptive control logic, involving also \( f_{m_{	ext{ToRecup}}^*} \).  

**CONCLUSIONS**

In this work a techno-economic analysis of a solar driven combined cycle has been performed. The studied plant is composed of a high temperature air receiver, a topping gas turbine, and a bottoming steam cycle connected in parallel to a packed bed thermal energy storage at the gas turbine exhaust. In order to evaluate the proposed plant performances a sensitivity analysis has been performed. The capacity factor, initial investment, and LCoE have been considered as the plant KPIs. Besides, the following decision variables have been analyzed: solar multiple, energy storage capacity, gas turbine expansion ratio and consequently the ratio between nominal gas and steam turbine power. The results show that:

- Purely solar driven CCGT would lead to LCoE about twice the one of comparable traditional, natural gas based, CCGT. However, they enable a completely emission-free power production.
- Large SM and large to medium TES size enable capacity factor enhancement and consequent LCoE reduction.
- Wider heliostat fields and larger TES size lead to increased capital investments, but lower LCoE thanks to higher energy production.
- For a non-recuperated plant layout, a complete expansion through the GT and a ratio between the gas and steam turbine nominal power of about 10 would result as the best design choice, leading to reduced LCoE.
- Higher plant CF are achieved by means of reduced GT expansion ratio and \( \gamma \).
- Modified plant layouts introducing recuperation do not enable sufficient improvements, as the GT expansion ratio result to be limited by the recuperation thermodynamic constraints.

To further enhance the plant performance of solar driven gas turbine system, they might be integrated with low temperature Organic Rankine Cycles as bottoming unit. Such cycles might enable a better exploitation of the relatively low GT outlet temperature. In order to fully investigate the flexibility potential offered by the introduced plant layout, further researches on optimized control and dispatch algorithms, taking into account variable electricity price scheme, load and weather fluctuations, will be developed.

**ACKNOWLEDGMENTS**

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