Fundamental Thermo-Economic Approach to Selecting sCO₂ Power Cycles for CSP Applications

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Abstract

The interest in Supercritical Carbon Dioxide (sCO₂) power cycles has grown exponentially in the last decade, thanks to distinctive features like the possibility to achieve high thermal efficiencies at intermediate temperature levels, small footprint and adaptability to a wide variety of energy sources. In the present work, the potential of this technology is studied for Concentrated Solar Power applications, in particular Solar Tower systems with Thermal Energy Storage. A thorough sensitivity analysis based on turbine inlet temperature and pressure ratio is done for twelve sCO₂ cycles, considering their effects on thermal efficiency and specific work, along with solar share and temperature rise across the solar receiver. The most important conclusions of this section are that: a) the peak values of these thermodynamic figures of merit are obtained at different pressure ratios; b) specific work and temperature rise across the receiver seem to follow parallel trends whilst this is not the case for thermal efficiency; c) for a given turbine inlet temperature, higher pressure ratios always increase the receiver temperature rise strongly, but the effect on thermal efficiency is uncertain as this can either increase or decrease, depending on the cycle considered. A deeper analysis of thermal efficiency and receiver temperature rise is therefore mandatory, given that these parameters strongly affect the capital cost of CSP power plants. On one hand, a higher thermal efficiency implies a smaller solar field, the largest contributor to the plant capital cost; on the other, the temperature rise across the receiver is inversely proportional to the size of the thermal energy storage systems, as it is also the case for state of the art steam turbine based CSP plants. An economic analysis is developed using an in-house code and the open-source software System Advisor Model to evaluate the trade-offs between these two effects. The results obtained for the two most representative sCO₂ cycles show somewhat unexpected results.

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Peer-review under responsibility of the scientific committee of the IV International Seminar on ORC Power Systems.

Keywords: sCO₂ power cycle, Thermo-economic analysis, CSP power plant, Thermal Energy Storage,
1. Introduction

The roots of supercritical Carbon Dioxide (sCO$_2$) technology can be tracked back to the initial work by Sulzer [1] and Escher-Wyss even if the theoretical fundamentals are generally attributed to Angelino and Feher [2–4]. The technical challenges posed by closed cycle gas turbines, in particular when fluids other than air were considered, along with the early take-off of combustion turbines hindered the further development of closed cycle engines which, for the following forty years, were hardly considered for large scale applications. In the early 2000’s, the assessment of sCO$_2$ power cycles for Gen IV nuclear reactors carried out at the Massachusetts Institute of Technology, in particular the work by Dostal [5], drew attention to the technology again and triggered a very intense research and industrial activity to explore the potential of these systems in diverse applications: nuclear [6], fossil fuel combustion in stand-alone oxyfuel [7,8] or combined cycle [9] systems, waste heat recovery [10] and Concentrated Solar Power plants. For the latter case, several studies have already suggested that sCO$_2$ systems have the potential to yield lower costs of electricity than state-of-the-art power plants based on steam turbine technology [11–15].

The present paper employs an approach already presented by the authors to analyse the potential of sCO$_2$ power cycles in CSP applications. This methodology relies on the fundamental thermodynamic features of the cycles, considered from a global standpoint for the entire design space, as opposed to the partial analysis of certain cycle layouts in reference cases that is usually found in literature. The concept is set forth in [2,16], showing the need to revisit the original works by Angelino and Feher [3,4] in order to establish the principles and a common framework upon which the technology can be developed further.

Amongst the forty-two sCO$_2$ power cycles reviewed in [2], twelve layouts are selected in [16]. For these, the impact of pressure and temperature ratios on $\eta_{th}$ and $W_s$ is studied in order to assess their theoretic potential from a thermodynamic standpoint. The analysis is developed without direct limits to the value of peak pressure, other than those coming from the feasibility of internal heat recovery (recuperator performance). This yields a comprehensive, global approach to cycle performance even if sometimes based on ridiculously high peak pressure values. The approach also provides an insight into remaining efficiency gains in future systems with higher grade materials. Fig. 1(a) shows an example of the results provided by the methodology described above, when applied to the twelve cycles considered. It is observed that each line is made up of two branches with full (black) and empty (white) markers. These correspond to operating conditions involving peak pressures lower or higher than 40 MPa respectively, this threshold set arbitrarily as a representative value for present technology [16]. As said, the combination of both sets (affordable and unaffordable pressures) in one single plot enables an assessment of the remaining gains in performance, whether efficiency or footprint.

The present paper applies the cited methodology to Concentrated Solar Power plants, focusing on the two cycle layouts shown in Fig. 1(b): Transcritical Simple Recuperated [3] and Supercritical Partial Cooling [22]. These
layouts are selected from previous analysis based on their potential for CSP applications [2,16]. The original features of the analysis are presented in the following section.

2. Sensitivity Analysis

The analysis in [16] is now applied to Concentrated Solar Power (CSP) plants incorporating Thermal Energy Storage (TES) systems operating on high temperature molten salts. To this aim, two additional figures of merit are considered: temperature rise of the molten salts across the solar receiver (\(\Delta T_{Solar}\)) and solar share (SS). The former parameter is of interest for its strong effect on the size and cost of the storage system. The Solar Share (SS), defined as the ratio from the solar heat input to the total heat input to the power block, stands as the best index to measure the degree of renewable energy integration when hybrid (solar-fossil fuel) power plants are taken into account.

2.1. General Assumptions

Three turbine inlet temperatures temperature levels are considered in this work: 800, 900 and 1000 °C. The first value\(^1\) (800 °C) sets the peak temperature expected from next-generation power blocks integrated in commercial CSP plants with molten salts TES, and therefore represents the case of a solar-only system. At higher TIT, a complementary energy source (usually fossil fuel through combustion with either air or oxygen) is needed. For the sake of simplicity, the current analysis does not include specific models of this complementary heat addition but, rather, TIT is simply set to 900 or 1000°C arbitrarily depending on the case. This assumption does not affect the validity of the analysis, whose focus is on the thermodynamic potential of the layouts considered and not on the technical feasibility or other aspects such as the technology readiness level.

The high temperatures taken into account are unattainable by the molten salts used in contemporary CSP plants, whose degradation temperature is in the order of 600 °C [17]. Therefore, a FLiNaK molten salt is adopted after a thorough literature review, given that its operating range is compatible with the sCO\(_2\) power blocks considered. A comparison between these two salts, along with their thermodynamic properties, is provided in Table 1. In order to model the heat transfer between FLiNaK and sCO\(_2\), a constant temperature difference of 20°C between the hot (salt) and cold (carbon dioxide) fluids is adopted (note that CO\(_2\) behaves close to ideally in this temperature and pressure region far from the critical point).

\(^1\) Private communication with Abengoa Solar.
but not for the (empty markers) whereas the maximum \[\Delta \] is attainable at lower pressures (except for T\_in\_solar = 1000 °C). On the contrary, the Supercritical Partial Cooling cycle can virtually achieve the highest efficiency at turbine inlet temperatures above 800°C. More specifically, the difference is \[\Delta \] = 5 percentage points in favour of the cited layout even if this is at substantially higher operating pressures.

The first conclusion drawn from these figures is that, for given turbine inlet temperature, the maximum values of \[\eta_{\text{th}}, W_s \text{ and } \Delta T_{\text{solar}}\] are obtained for different PRs in each cycle, Figs. 2 and 4. Moreover, peak performance (highest \[\eta_{\text{th}}\] and \[W_s\]) for the Transcritical Simple Recuperated cycle is always achieved above the technological limit of 40 MPa (empty markers) whereas the maximum \[\Delta T_{\text{solar}}\] is attainable at lower pressures (except for T\_in\_solar = 1000 °C). On the contrary, the Supercritical Partial Cooling cycle can virtually achieve the highest efficiency within the technological limits.

Secondly, it worth noting that \[\Delta T_{\text{solar}}\] and \[W_s\] are almost directly proportional until the maximum values of \[\Delta T_{\text{solar}} = 346^\circ\] C is reached. The same applies to the \[\Delta T_{\text{solar}}\] vs. \[\eta_{\text{th}}\] plots for the Transcritical Simple Recuperated cycle in Fig. 3(a) but not for the Supercritical Partial Cooling layout in Fig. 5(a) which shows a different pattern. The horizontal
segment of the $\Delta T_{solar} - W_s$ plots come about because the recuperator outlet temperature increases and eventually achieves the freezing point of FLiNaK when PR is raised. The latter is found at around 454 °C which, considering a safety margin of 20 °C, implies that the minimum salt temperature allowed is in the order of 474 °C.

A final remark is that, for given turbine inlet temperature, increasing PR brings about higher $\Delta T_{solar}$ until FLiNaK achieves the minimum temperature allowed, whereas the effect on $\eta_{th}$ is uncertain and can vary depending on the cycle considered; this is observed in Figs. 2(a) and 4(a). This last conclusion might actually become of uttermost importance for the economic feasibility of a CSP plant. Indeed, high $\eta_{th}$ leads to a smaller solar field, thus decreasing the plant capital cost, whereas high $\Delta T_{solar}$ has a direct impact on the size and cost reduction of the TES system. Unfortunately, $\eta_{th}$ and $\Delta T_{solar}$ achieve their maximum values at different PRs and hence a compromise between both figures of merit is mandatory. A deeper analysis is provided in the following sections.
3. Thermo-economic Analysis

The contributions of solar field and thermal energy storage system to the total cost of a 50 MW CSP plant with 10 hours storage capacity (TES made up of hot/cold tanks for extended operation at full load) is now assessed with a dedicated tool based on the open-source software SAM (solar field) [18] and a complementary in-house code (TES) [23] respectively. To this end, a sensitivity analysis of each cost to variations of $\eta_{th}$ and $\Delta T_{solar}$ is done by varying these parameters in the ranges 30-60 % and 90-350 °C respectively. This means that all the cases in the previous section are covered enabling a complete analysis of the contributions of solar field and thermal energy storage system to the total capital cost of the plant.

The rated performance of the reference power plant (reference case) in as far as the costs of solar field and storage systems are concerned is defined by $\eta_{th} = 45\%$ and $\Delta T_{solar} = 230^\circ C$. For this reference case, Fig. 6(a) shows the cost of the solar field; as expected, increasing $\eta_{th}$ brings about a significant reduction of this cost. The cost of the storage system presented in Fig. 6(b) includes the cost of purchasing the major equipment as well as those related to installation, insulation, foundations and all the auxiliaries equipment required for system operation. These are corrected with the cost indexes published in the International Journal of Production Economics to account for the time value of money [24], and with exponential cost scaling factors taken from literature [25]. Starting from a detailed project budget reported in [19], this approach provides cost estimates of every single component of a real storage project. Additional information, like wages, labour hours and productivity is taken from [26].

The plots in Fig. 6(b) show that the size of the storage system is inversely proportional to $\Delta T_{solar}$. Indeed, when $\Delta T_{solar}$ increases, the specific energy storage capacity (kJ/kg) of the storage material increases, thus reducing the inventory of molten salts for the same specifications (hours of extended operation at full load). On the other hand, $\eta_{th}$ also contributes to reducing the size of the storage system since the heat input required for a given electric output is reduced proportionally to $\eta_{th}$.
These economic assessment becomes more interesting when combined with the thermodynamic analysis presented in the previous section. Let turbine inlet temperature be set to 800°C and turbine inlet pressure be limited to 40 MPa. When PR is varied, the cost of the solar field ranges from 203 to 125 M$ for the Transcritical Simple Recuperated cycle and from 122 to 120 M$ for the Supercritical Partial Cooling layout. Also, the cost of the thermal energy storage system ranges from 120 to 53 M$ for the former cycle and from 68 to 57 M$ for the latter layout.

These results come to confirm the earlier comments regarding Figs. 3(a) and 5(a). Whilst the former presents significant variations of the figures of merit, and therefore of the correspondent costs, the latter presents an almost vertical slope, and hence the cost of the solar field is very weakly affected by PR. This is further observed in Fig. 7 where the individual and cumulative costs of solar field and storage system are shown for two different turbine inlet pressures (25 and 40 MPa) and temperatures (800 and 1000°C). Thermal efficiency $\eta_{th}$ is also provided on the right vertical axis and the labels in each bar indicate the $\Delta T_{solar}$ across the solar receiver. The first case (800°C and 25 MPa) stands for the current state of technology whereas the last one (1000°C and 40 MPa) shows the performance attainable by these cycles should significant technology improvements be made.

![Image](image_url)

Fig. 7. Capital costs of solar field and TES for different turbine inlet pressure and temperature (bars). Thermal efficiency (dashed line) $\eta_{th}$ and temperature rise across the receiver (labels) $\Delta T_{solar}$ [°C] also shown.

The information in Fig. 7 suggests that raising turbine inlet temperature when pressure is limited to 25 MPa increases efficiency but it does not bring about an economic benefit. This is because despite a slightly lower cost of the solar field, the temperature rise across the receiver drops by more than 100°C due to the higher turbine exhaust temperature. The consequent increase in size and cost of the storage system offsets the aforesaid solar field cost reduction. The situation is somewhat similar for the Supercritical Partial Cooling cycle when peak pressure is set to 40 MPa, even if less acute, whilst the impact of a higher temperature in the Transcritical Simple Recuperated system is virtually null. It must be noted though that these statements refer to an economic analysis only since the thermodynamic benefit when TIT is raised is unquestionable (highest $\eta_{th}$).

Overall, the largest economic benefit seems to come from pursuing higher operating pressures whilst remaining at moderate turbine inlet temperatures (800°C and 40 MPa). This is because such design conditions yield acceptable efficiency but with a very large $\Delta T_{solar}$, the impact of this latter parameter on system cost being dominant for the case considered.

4. Conclusions

The first part of the present paper shows a thermodynamic comparison between the Transcritical Simple Recuperated and the Supercritical Partial Cooling cycles for different turbine inlet temperatures and pressures. When pressure is limited to 25 MPa, the first cycle seems to achieve higher specific work $W_s$ and temperature rise across the receiver $\Delta T_{solar}$, peaking at 350 kJ/kg and 346 °C respectively, whilst the Supercritical Partial Cooling layout achieves higher $\eta_{th}$, 58%. When the pressure limit is raised to 40 MPa and high solar shares are sought, the Transcritical Simple Recuperated cycle seems to perform slightly better: 56% & 70% for the Supercritical Partial Cooling and 54% & 78% for the Transcritical Simple Recuperated (values corresponding to $\eta_{th}$ and SS respectively).
The second part of the paper presents a thermo-economic analysis to assess how $\eta_{th}$ and $\Delta T_{solar}$ affect the costs of solar field and thermal energy storage system. The results show that the overall costs of these systems are fairly similar (within a 5% margin). Nevertheless, the interesting conclusion from the analysis is the different trend followed by thermal efficiency and costs (of solar field and TES) when turbine inlet temperature and pressure are changed, meaning that a purely thermodynamic optimisation is not advisable. Moreover, according to the results, there seems to be a larger benefit coming from higher operating pressures than from increasingly higher temperatures. This conclusion is very positive since higher pressures mean more resistant (bulkier) equipment with essentially the same technology whereas higher temperature would very likely imply new materials and, probably, technologies.

As a final comment, the combined thermodynamic and economic analysis presented in this paper makes use of secondary economic figures to highlight that the pathways towards lower capital costs are not as evident as initially thought. This approach makes more sense in sCO$_2$ applications due to the fairly complex cycle layouts that are used.

References