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SUPERCRITICAL CO₂ BRAYTON POWER CYCLE FOR CSP WITH PACKED BED TES INTEGRATION AND COST BENCHMARK EVALUATION

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ABSTRACT

The present work introduces an indirect supercritical CO₂ – air driven concentrated solar plant with a packed bed thermal energy storage. The proposed plant design enables a supercritical CO₂ turbine inlet temperature of 800°C, overcoming the temperature limits imposed by the use of solar molten salts as primary heat transfer fluid. Furthermore, the packed bed thermal energy storage permits the decoupling between thermal power collection from the sun and electricity generation. Besides, the thermal energy storage unit grants operational flexibility and enlarges the plant capacity factor, making it as available as a conventional coal facility. A transient thermodynamic model of the integrated concentrating solar plant, including receiver, thermal energy storage, intermediate heat exchangers and supercritical CO₂ power cycle has been developed. This same model has been used to evaluate the thermodynamic performance of the proposed plant design over a complete year. A similar model has been implemented to simulate a supercritical CO₂ plant driven by a more traditional solar molten salt loop. A comparison of the thermodynamic performance of the two plant designs has been performed. A complete economic model has been developed in order to evaluate the economic viability of the proposed plant. Furthermore, a multi-objective optimization have been executed in order to assess the influence of the thermal energy storage size, supercritical CO₂ turbine inlet temperature and plant solar multiple on the key performance indicators. Results show that the proposed indirect supercritical CO₂ – air driven with a packed bed thermal energy storage concentrated solar plant leads to improved thermo-economic performance with respect to the molten salts driven design. Enhancements in the power cycle efficiency and in the overall electricity production can be achieved, with a consequent reduction in the levelized cost of electricity. Particularly, for a design net electrical power production of 10MWe a minimum levelized cost of electricity has been calculated at 89.4 \$/MWh for a thermal energy storage capacity of 13.9 hours at full load and a plant solar multiple of 2.47 corresponding to a capital investment of about 73.4 M\$.

INTRODUCTION

In light of increasing electricity needs paired with climatic changes, solar energy has become an appealing renewable alternative in the energy market. In this field, a growing share of power production is related to concentrated solar power (CSP) plants. In order to further

enhance the economic viability of CSP plants, different alternatives have been proposed.

Firstly, the coupling of the solar field and receiver unit with a supercritical carbon dioxide power block has recently gained particular attention in the CSP community [1-4]. Indeed, closed loop supercritical CO₂ Brayton cycles allow higher thermal efficiency than supercritical or superheated traditional steam cycles. The main advantages of sCO₂ Brayton cycles, with specific attention to their application in CSP plants, can be summarized as follow [4-6]:

- high thermal and cycle efficiencies;
- high power density and compactness (of both turbomachinery and heat exchangers) thanks to the thermo-physical properties of CO₂ in the supercritical region and particularly to its high density close to the critical point;
- large power scalability (from 1MW to multi-100MW);
- lower corrosiveness of CO₂ than of water/steam, this enables higher working temperature and besides reduces the requirements for the materials used in the different components;
- improved performances in dry-cooling, which makes the implementation of this power cycle easier in CSP plants which are often located in arid area;
- part load operation is more easily manageable especially for short duration due to the thermal inertia in the system and piping.

Secondly, the integration of thermal energy storage (TES) in the system is known to be a key component to enhance the CSP market potential. Notwithstanding the efficiency of the power cycle, CSP cannot simply compete in price with direct photovoltaic plants, which are much cheaper in terms of capital cost and levelized cost of electricity. However, not only TES enables decoupling of the solar concentration and the electricity generation processes, but it also grants operational flexibility to the power plant. Therefore, the TES installation can enlarge the CSP plant capacity factor, making it as available as a conventional coal facility. In this way, when integrated with TES, largescale CSP plants do not compete directly with other fluctuating solar technologies and their market share and potential can be wider.

Traditional CSP plants rely on molten salts as main heat transfer fluid (HTF) in the solar loop to transfer energy from the receiver to the power block. Despite the recent research effort focused on the improvement of molten salts, their properties degrade at temperature above 580 °C limiting the thermal efficiency of the cycle. To overcome

Figure 1: Proposed air driven CSP plant with packed bed TES and integrated with a sCO₂ recompression brayton cycle.

this restriction, previous researches [7-8] have shown that air could be a suitable HTF for CSP by enabling higher working temperature and reducing corrosion and material degradation issues. Furthermore, air driven CSP cycles can be paired with packed bed TES, which allows for several advantages [1] in comparison with traditional solar salt based plants and relative two tanks storage: use of cheap and abundant storing material; wide temperature range and higher maximum temperature limited by the rocks' melting temperature; direct heat transfer between working heat transfer fluid and storing material avoiding the need of an intermediate heat exchanger; no chemical instability, degradation, and corrosion; low safety concerns. The present work introduces an indirect supercritical CO₂ – air driven concentrated solar plant integrated with a packed bed TES, its layout is sketched in Figure 1. The use of air as primary heat transfer fluid, through the solar receiver and the packed bed TES, enable to achieve sCO₂ turbine inlet temperature (TIT) of about 800°C. Therefore, the temperature limit, around 580°C, imposed by the use of solar molten salts as primary heat transfer fluid is overcome. Moreover, the packed bed TES allows storing the energy required to run the sCO₂ power cycle during the majority of the night or during cloudy periods. The HTF flows through the packed bed TES from the bottom to the top during discharge phases, while the flow is reverted during charge periods. A modeling study of a packed bed working with sCO₂ as HTF has already been performed, showing that an exergetic efficiency of above 80% can be achieved [9]. However, as neither cost evaluations nor reliable data are available yet, in this work an indirect cycle has been preferred and therefore the storage unit exploits air as HTF. A transient thermodynamic model of the integrated CSP plant, including receiver, thermal energy storage, intermediate heat exchangers and sCO₂ power cycle has been developed. This same model has been used to evaluate the thermodynamic performances of the

proposed plant design over a period of a whole year. Similarly, a model has been implemented to simulate a supercritical CO₂ plant driven by a solar molten salt loop, whose plant layout is sketched in Figure 3. A comparison of the thermodynamic performances of the two plant designs has been performed. The study has been extended to the economic side considering the capital costs of all the main components and the plant operational costs. Therefore, economic performance indicators as the CAPEX and LCOE have been considered. Finally, a multi-objective optimization, considering a time period of seven consecutive days for the four seasons (from 18th to 24th March, from 18th to 24th June, from 18th to 24th September and from 18th to 24th December) have been executed in order to assess the influence of the thermal energy storage size, sCO₂ TIT, and plant solar multiple on the key performance indicators.

SYSTEM CONFIGURATION

As previously mentioned, the supercritical CO₂ Brayton cycle can be integrated into a CSP plant via a direct or indirect scheme. In the direct approach, the supercritical CO₂ is circulated also through the receiver as the main HTF. This plant design enables to eliminate the intermediate HTF (molten salt or air), the associated heat exchangers (MH, RH) and the relative energy losses and pressure drops. Nevertheless, the packed bed TES solution required for a direct sCO₂ CSP plant, as proposed in [9], is cost prohibitive [3] due to the need of thickly insulated pressure vessels to contain the high pressure and temperature required to maintain the CO₂ in supercritical conditions. For these reasons in this paper, an indirect plant design has been modeled and studied. The proposed 10MWe supercritical CO₂ CSP Brayton plant scheme is shown in Figure 1. The two principal loops can be identified: the air loop (black lines) and the power cycle (sCO₂ loop, green lines). Two heat exchanger units (MH and RH) enable the heat exchange between the two HTFs. In the solar loop, air is circulated as the HTF through the receiver where it absorbs the solar energy. The air mass flow rate is controlled in order to maintain a constant receiver outlet temperature defined on the basis of the TIT. After being heated up in the receiver the air flow is blown towards the heat exchanger units (MH and RH) and/or the TES unit depending on the operating mode. In particular, determined by the DNI and the state of charge (SOC) of the TES five different operation modes are available: TES charge (OM1), TES discharge at nominal air mass flow (OM2), TES discharge at partial air mass flow (OM3), only power production with defocusing (OM4) and stop of the plant without power production (OM5). Further details about this control logic will be explained in the following paragraphs.

Figure 2: T-s diagram of the recompression sCO₂

Figure 3: Molten salts CSP plant integrated with a sCO₂ recompression Brayton cycle.

As the air and sCO₂ mass flow through the loops change according to the working conditions inventory vessels have been introduced in the plant scheme to compensate for flow variations. The CO₂ power loop consists of a supercritical Brayton cycle with recompression and reheating. This cycle layout, whose T-s diagram is shown in Figure 2, introduces an additional compressor and a re-compressor (RC), and subdivide the recuperator into two different sections, high temperature recuperator (HTR) and low temperature recuperator (LTR). At the outlet of the hot side of the LTR the sCO₂ mass flow is subdivided according to the split ratio (SR). The main mass flow component goes through the condenser, main compressor and cold side of the LTR; while the second part of the mass flow goes through the RC. The two streams are mixed at point 9 in Figure 2. Thanks to this design the large difference between the specific heat of the hot/high-pressure and cold/low-pressure sCO₂ streams in the LTR is compensated by a lower mass flow rate in the hot /high pressure side of the heat exchanger reducing the pinch point temperature difference. Further details about the sCO₂ power cycle and its modeling will be given in the following paragraphs.

THERMO-ECONOMIC MODEL

The model of the cycles has been developed in a MatLab environment coupled with the thermodynamic properties database REFPROP [10]. In order to assess a proper time step for the transient model a sensitivity analysis has been performed considering the following time steps: 5, 10, 20, 30 and 60 minutes. The time step choice affects the plant control logic and the TES SOC while the transient and inertial effects in the turbomachinery have been neglected. Figure 4 shows the absolute error on the final average plant efficiency $\bar{\eta}_{sCO_2}$ and the computational time for the different time step choices. Considering

Figure 4: Sensitivity study for time step choice

the 5 minutes case as a base reference the 10 minutes choice enable to strongly reduce the computational time, around 50% lower, while keeping an absolute error on $\bar{\eta}_{sCO_2}$ lower than 0.05. Furthermore, looking at the operating hours in the different OM, the time step of 10 minutes enables to contain the difference with respect to the 5 minutes case below 2%. Larger time steps lead to higher deviations from the reference case and to limited reductions in the computational time. For these reasons a time step of 10 minutes has been chosen. A further reduction of the time step, 1 minute, has been applied in the TES unit in order to evaluate its transient behavior and the thermocline spread.

Thermodynamic supercritical CO₂ Cycle Model

In the present work, a recompression cycle with reheating has been chosen as the sCO₂ power loop scheme. The recompression cycle has been originally introduced in 1968 by Feher [11]; therefore it is also called “Feher cycle”. Nowadays, several studies have focused on this power cycle [1-6, 12-19], performing comparative studies among different sCO₂ power cycle layouts and confirming its role as the most promising CO₂ cycle candidate. Although the simple regeneration cycle can couple with different working conditions, it is simple and permits to recover a large part of the wasted heat, all authors agree that the recompression cycle improves the thermal performance particularly at higher working temperature. Indeed, the recompression cycle guarantees a lower temperature pitch point in the recuperator. A high temperature pitch point is due to the large difference between the heat capacity of the high-pressure and low-pressure sCO₂ streams.

The expansion processes in both the high and low pressure turbines (HPT and LPT) have been assumed to be adiabatic non-isentropic. In order to take into account worse performances of the turbomachinery away from the design point, the turbine efficiency has been scaled based on the actual load (W_{act}) and inlet temperature (T_1) resulting in:

$$h_2 = h_1 - \left(\frac{W_{act}}{W_{des}} \right) \left(\frac{T_1}{TIT_{des}} \right) \eta_T (h_1 - h_{2s}) \quad (1)$$

And similarly,

$$h_4 = h_3 - \left(\frac{W_{act}}{W_{des}} \right) \left(\frac{T_3}{TIT_{des}} \right) \eta_T (h_3 - h_{4s}) \quad (2)$$

Where, W_{des} is the design thermal power, TIT_{des} is the design turbine inlet temperature and η_T is the turbine iso-entropic efficiency.

Both HTR and LTR have been modeled by means of their effectiveness (ϵ_{HTR} and ϵ_{LTR}) as in Eqs. (3) to (6).

$$\epsilon_{HTR} = \frac{h_4 - h_5}{h_4 - h(p_5, T_9)} \quad (3)$$

$$h_4 - h_5 = h_{10} - h_9 \quad (4)$$

$$\varepsilon_{LTR} = \frac{h_5 - h_6}{h_5 - h(p_6, T_8)} \quad (5)$$

$$h_5 - h_6 = SR(h_{9'} - h_8) \quad (6)$$

The compression processes in both the main compressor and recompressor (MC and RC) have been assumed to be adiabatic non-isentropic, resulting in Eqs. (7) and (8). Where, η_c is the compressor iso-entropic efficiency. Finally, the sCO₂ stream exiting from the hot side of the LTH (state 9') and the one exiting from the RC (state 9'') are mixed (state 9) before entering the hot/high-pressure side of the HTR. The mixing process has been expressed through the energy balance in Eq. (9). The SR has been kept constant to the value that minimizes the difference between state 9' and state 9'' in the design steady state. The thermodynamic model of the recompression with reheating sCO₂ Brayton cycle has been cross-validated at steady state by comparing the results calculated by the present study with those published by Dostal et al. [19]. The main parameters used for the validation of recompression sCO₂ Brayton cycle with reheating are shown in Table 1. The comparison of the cycle thermal efficiency is illustrated in Figure 5. The relative errors are all within 2%; thus the present model is in a good agreement with Dostal's study [19] and it can be used for further studies.

$$h_8 = h_7 + \eta_c(h_{8s} - h_7) \quad (7)$$

$$h_{9''} = h_6 + \eta_c(h_{9''s} - h_6) \quad (8)$$

$$h_9 = SR h_{9'} + (1 - SR) h_{9''} \quad (9)$$

Table 1: Values of the main parameters used for the validation of recompression sCO₂ Brayton cycle with reheating

High pressure, p_h [MPa]	25
Intermediate pressure, p_i [MPa]	16.19
Low pressure, p_l [MPa]	7.38
Turbine Inlet Temperature, TIT [°C]	500-850
Temperature inlet Main Compressor, $TIMC$ [°C]	32
Efficiency Turbine, η_{HPT} and η_{LPT} [%]	93
Efficiency Compressor, η_{MC} and η_{RC} [%]	89
Effectiveness Heat Exchangers, ε_{HTR} and ε_{LTR} [%]	95

Figure 5: Validation of the recompression sCO₂ Brayton cycle model against Dostal's results [19]

Packed Bed TES Model

A packed bed filled with natural rocks has been assumed as the TES unit integrated into the proposed CSP plant design. As previously said this TES enables several advantages and it is currently under investigation as a viable solution to enhance the storing temperature and energy. Several previous works [20-27] have shown packed bed modeling strategies and thermal behavior. In this work, a thermodynamic model of the axial flow packed bed TES, based on the established Schumann's model [20] has been implemented. The one dimensional, two phases, transient model considers conduction, convection and radiation heat transfer mechanisms. The main assumptions included in the model can be summarized as:

- Mono-dimensional axial Newtonian plug flow
- Uniformly distributed packed bed of spheres (with uniform void fraction)
- Negligible temperature gradient in the radial direction
- Negligible temperature difference within the single solid particles (justified by $Bi < 1$)
- Heat transfer by radiation is accounted by means of a modified conductivity in both fluid and solid phase
- No mass transfer or internal heat generation
- Negligible tank's wall heat capacity

The one dimensional, two phases, transient model can be summarized by a fluid, Eq. (10), and a solid equation, Eq. (11).

$$\frac{\partial T_F}{\partial t} + \frac{G}{\rho_F \varepsilon} \frac{\partial T_F}{\partial x} = \frac{k_{F_{eff}}}{\rho_F c_F \varepsilon} \frac{\partial^2 T_F}{\partial x^2} + \frac{ha_s}{\rho_F c_F \varepsilon} (T_S - T_F) + \frac{U_w D \pi}{\rho_F c_F A \varepsilon} (T_\infty - T_F) \quad (10)$$

$$\frac{\partial T_S}{\partial t} = \frac{k_{s_{eff}}}{(1 - \varepsilon) \rho_s c_s} \frac{\partial^2 T_S}{\partial x^2} + \frac{ha_s}{\rho_s c_s (1 - \varepsilon)} (T_F - T_S) \quad (11)$$

Where, ε is the bed void fraction, G is the mass flow per unit cross-section. While a_s is the bed surface to volume ratio, and U_w is the heat transfer coefficient between the wall and the ambient air, defined in Eqs. (12) and (13), respectively. Where a_i and a_o are the convective heat transfer coefficients on the inside and outside of the TES tank, respectively, while k_j is the conductivity of the different insulation layers. The air thermodynamic properties have been retrieved by the REFPROP database [10] while the thermodynamic properties of the solid filler are reported in Table 2.

$$a_s = \frac{700}{6(1 - \varepsilon)} G^{0.76} d_p^{0.24} \quad (12)$$

$$\frac{1}{U_w} = \frac{1}{a_i D} + \frac{D}{2} \sum_{j=1}^n \frac{1}{k_j} \ln \frac{d_{j+1}}{d_j} + \frac{1}{a_o d_{n+1}} \quad (13)$$

Table 2: Thermodynamic properties of the solid filler material

Density, ρ_s [kg/m ³]	2850
Specific Heat, c_s [J/(kgK)]	1172.5
Conductivity, k_s [W/(mK)]	$(807/(350 + T[°C])) + 0.64$
Emissivity, ε [-]	0.85
Particle mean diameter, d_p [m]	0.035

The convective heat transfer coefficient between the HTF and the filler material, h , has been evaluated by means of the correlation proposed by Wakao [28].

$$h = \frac{k_f}{d_p} \left(2 + 1.1 Re^{0.6} Pr^{\frac{1}{3}} \right) \quad (14)$$

Besides, an effective temperature dependent overall conductivity has been calculated by means of the model proposed by Kunii and Smith [29], Eq. (15), where conduction and radiation within the packed bed and among the single pebbles are considered. The constants β and γ are equal to 0.895 and $2/3$ respectively. K is equal to the temperature dependent ratio between solid and fluid conductivity. The effect of radiation on the effective thermal conductivity of the packed bed is taken into account through k_{rv} and k_{rs} , radiation from void to void and from solid to solid respectively, expressed in Eqs. (16) and (17):

$$\frac{k_{eff}}{k_F} = \varepsilon \left[1 + \frac{\beta k_{rv} d_p}{k_F} \right] + \frac{\beta(1-\varepsilon)}{\left[\frac{1}{\psi_t + k_{rs} d_p / k_F} + \frac{\gamma}{K} \right]} \quad (15)$$

$$k_{rv} = \frac{4\sigma T^3}{1 + \frac{\varepsilon}{2(1-\varepsilon)} \frac{(1-\varepsilon_r)}{\varepsilon_r}} \quad (16)$$

$$k_{rs} = 4\sigma T^3 \left(\frac{\varepsilon_r}{2 - \varepsilon_r} \right) \quad (17)$$

The value of ψ_t depends on the number of contacts among particles, defined as n , coordination flux number. For loose packing $n_1 = 1.5$, while for close packing $n_2 = 4\sqrt{3}$. Then, ψ_t is calculated accordingly to Eq. (18), where $\psi_{1 \text{ or } 2}$ are defined in Eq. (19).

$$\psi_t = \psi_2 + (\psi_1 - \psi_2) \left(\frac{\varepsilon - 0.26}{0.216} \right) \quad (18)$$

$$\psi_{1 \text{ or } 2} = \frac{\frac{1}{2} \left(\frac{K-1}{K} \right)^2 \sin^2 \theta_0}{\ln(K - (K-1) \cos \theta_0 - \left(\frac{K-1}{K} \right) (1 - \cos \theta_0))} - \frac{2}{3K} \quad (19)$$

Then the method explained by Cheng and Hsu [30] has been applied to share out the effective conductivity contributions between the fluid and the solid phase, obtaining $k_{F_{eff}}$ and $k_{S_{eff}}$ respectively. The implemented one dimensional, two phases, transient model of the packed bed TES has been validated by comparing the temperature profiles across the storage calculated in the present study with those published by Hänchen et al. [8]. The main parameters used for the validation of the packed bed TES model are shown in Table 3, while the comparison of the temperature profiles along the TES height is illustrated in Figure 6. The present model is in a good agreement with Hänchen's study [8] and it can be used for further studies.

Table 3: Values of the main parameters used for the validation packed bed TES mode

Inlet temperature, T_{in} [K]	823
Ambient temperature, T_{∞} [K]	293
Specific HTF mass flow rate, G [kg/(m ² s)]	0.225
Packed Bed height, H [m]	1.2
Packed Bed diameter, D [m]	0.148
Void fraction, ε [-]	0.4
Particle diameter, d_p [m]	0.02
Solid Density, ρ_s [kg/m ³]	2680
Solid Specific Heat, c_s [J/(kgK)]	1068
Solid Conductivity, k_s [W/(mK)]	2.5
External Heat Transfer Coefficient, U_w [W/(m ² K)]	0.678

CSP Plant Model

In order to evaluate the solar energy input, the studied CPS plant has been located in Daggett, CA, USA (34.85°N, -116.78°E) and the

Figure 6: Validation of the packed bed TES model against Hänchen's study [8]

DNI data have been gathered from the Meteorism database. The transient model of the complete CSP plant cycle considers a time step of 10 minutes, neglecting transient effects in the turbomachinery. In order to fully understand the dynamic behavior of the plant the simulations have been performed considering a whole year. The heliostat field and the receiver unit have been modeled by means of two effective efficiencies η_{SF} and η_{REC} . The first has been calculated for the whole year as function of the solar azimuth and zenith by means of the commercial software System Advisor Model [31], the latter has been considered constant and equal to 0.85. The supercritical CO₂ power cycle has been modeled considering the parameters shows in Table 1. The TES unit has been sized accordingly to the number of discharge hours at the design thermal power required by the sCO₂ power cycle.

$$E_{TES} = \frac{\dot{Q}_{HEX} t_{disch}}{\eta_0} \quad [kWh] \quad (20)$$

Where $\eta_0 = 0.7$ has been included in order to oversize the storage unit to take into account thermal losses and a decrease in the outlet temperature during the second phase of the discharge. The operation of the plant has been managed by means of a specific control logic. Depending on the actual DNI and on the SOC at the previous time step, five different operation modes (OM) can be identified. The control logic flowchart is reported in Figure 7, while Table 4 defines the main implication of each operation mode in the plant control. Therefore, if the solar input alone is sufficient to cover the power cycle heat load (DNI_{min} , minimum irradiation that enables a thermal input sufficient to power the sCO₂ cycle) the stored energy is not used or the storage is charged if a surplus in the incoming energy is available and if the SOC is lower than SOC_{max} equal to 0.95. On the contrary, when the solar input is not enough to fulfill the requirement from the heat exchangers, the storage unit is discharged as an additional source. Further on, the storage discharge, and consequently the plant power production, is stopped when the SOC is lower than SOC_{min} , which is equal to 0.05.

Table 4: Operational Modes main implications

OM	PB	REC	TES	$\dot{m}_{TES}$
1	On	On	Charge	$\dot{m}_{REC} - \dot{m}_{HEX}$
2	On	Off	Discharge	$\dot{m}_{HEX_{DES}}$
3	On	On	Discharge	$\dot{m}_{HEX} - \dot{m}_{REC}$
4	On	On	Full	0
5	Off	Off	Empty	0

RESULTS AND DISCUSSION

In this section, the thermodynamic performance of the proposed air-driven CSP sCO₂ plant are investigated for a whole year and for seven consecutive days during the four seasons (from 18th to 24th March, from 18th to 24th June, from 18th to 24th September and from 18th to 24th December). As main results, the overall plant efficiency and the operative hours in the different operation modes will be reported. Besides, the trend of the SOC of the TES units is analyzed. Finally, the main economic indicators and the cost share among the different components are enlightened. All the results have been reviewed in comparison with the ones provided by the molten salt sCO₂ CSP plant.

CSP Plant Performance – Design Case

In the design case, the plant performances have been evaluated considering the design parameters reported in Table 5. Figure 8 shows the TES state of charge over time (seven consecutive days for four seasons) for the two storage solutions and the instantaneous DNI level. The TES starts from a 90% SOC in both designs, then coherently the discharge phases starts after the peak DNI when the solar input itself is not sufficient to fulfill the power cycle heat demand. Furthermore, some particularly cloudy day can be identified, in this case the solar field is not able to charge the TES unit and the SOC remains around its minimum acceptable level. Figure 9 reports the number of working hours of the CSP plants under the five operation modes defined in section 3.3. The results for the yearly simulation are shown. Firstly, neither of the plant designs reaches a full TES, thus OM4, with relative defocusing and wasted energy, is avoided. Furthermore, the hours of TES charge, OM1, are equal for the air and molten salt case, while the air driven CSP plant enables longer discharge periods. Thus, an absolute reduction of 1335 hours of stop of the plant can be achieved thanks to the proposed design. With respect to the molten salt CSP plant a relative reduction of the stop period, OM5, of 27.5% is achieved. As key performance indicators for the thermodynamic behavior of the plant the sCO₂ power cycle efficiency and the net electrical energy produced

Table 5: Plant Design parameters

Direct Normal Irradiance, DNI_{DES} [W/m ²]	900
Tower Height, H_{tower} [m]	117
Turbine Inlet Temperature, TIT_{DES} [°C]	800
Net Electrical Power, W_{DES} [MW _e]	10
Solar Multiple, SM [-]	2
TES size, $t_{disch_{DES}}$ [h]	12

Figure 7: CSP plant control logic scheme

Economic Model

In this study the CAPEX and the LCoE has been chosen as the main economic indicators in order to measure both the investment and the relative profitability of the introduced CSP plant. The capital investment has been calculated by adding the direct cost for all the main specific components (sCO₂ turbomachinery, sCO₂ heat exchangers, heliostat field and land, tower, receiver, TES, balance of plant and contingencies) and indirect cost (engineering, procurement and construction and taxes). The cost of the power block components have been calculated according to the scaling cost function in [32-33]. The heliostat and land cost has been evaluated based on the scaling function in [34]. The receiver and tower scaling cost functions has been taken from [35]. While the remaining cost components have been evaluated thanks to the data and approach presented in [36]. Instead, a more detailed approach has been followed to evaluate the cost of the TES. Indeed the overall TES cost has been calculated by adding the cost of the filler material, insulation, steel and foundation accordingly to [37]. Furthermore, labor and services costs [36] have been accounted in the operational cost, OPEX. Finally, the LCoE is calculated by means of Eq. (21) as a function of the annualized CAPEX, the annual OPEX and the total electricity produced throughout the year. The capital return factor α is calculated by means of Eq. (22) where i is the real debt interest rate, n is the plant life time and k_{ins} is the annual insurance rate, assumed to be equal to 7%, 30 years and 1%, respectively [36-38].

$$LCoE = [\alpha \cdot CAPEX + OPEX] (E_{NET})^{-1} \quad (21)$$

$$\alpha = i(1+i)^n[(1+i)^n - 1]^{-1} + k_{ins} \quad (22)$$

Figure 8: TES SOC for air and molten salts driven CSP plant and DNI, for 7 consecutive days for 4 seasons

Figure 9: Operating hours in the different OMs for air and molten salts driven sCO₂ CSP plant for yearly simulation

during a year have been considered. Table 6 reports both the parameters, the mean sCO₂ power cycle efficiency has been defined in Eq. (23), where h_{op} is the number of operative hours with a positive electricity production (OM1, OM2, OM3 and OM4). Particularly, $\bar{\eta}_{sCO_2}$ considers the net power output of the plant over the thermal power required at the main heater and re-heater, thus it evaluates only the sCO₂ cycle. The air driven design enables a higher sCO₂ cycle efficiency thanks to the higher TIT, otherwise limited by the molten salts stability to 560°C. However, during the packed bed TES discharge the outlet air temperature decreases due to the partial thermocline release. Therefore, a gradually decreasing efficiency during the TES discharge period can be identified. Both the thermodynamic key performance indicator are strongly improved by the proposed design. Particularly, the sCO₂ cycle efficiency is increased by around 49%, while the electricity production increase by more than 34%.

$$\bar{\eta}_{sCO_2} = \left[\sum_{i=0}^{n_i} \frac{W_{HPT}^i + W_{LPT}^i - W_{MC}^i - W_{RC}^i}{Q_{MH}^i + Q_{RH}^i} \right] \cdot (h_{op})^{-1} \quad (23)$$

Table 6 also shows the main economic results for the two considered plants accordingly to the definitions give in the previous section. While Figure 10 reports the $CAPEX_{Direct}$ share among the different components of the plant. Firstly, the air driven design enables

Figure 10: Direct CAPEX share among all main CSP plant components

large reduction in all the components' costs. Particularly large savings can be identified in the receiver (it should be mentioned that refinements and more precise scaling functions for the air receiver might be introduced) and in the TES unit, mainly due to a cheaper storing material (natural rocks instead of molten salts). Besides, the overall cost of the power block can be sensibly reduced, with lowered costs for each specific unit. The power block cost reduction is mainly related to improvements in the thermodynamics of the cycle, which lead to lower UA of the heat exchangers. Furthermore, the overall CAPEX can be reduced by about 35%: while the resulting LCoE can be cut by more than 50%.

Table 6: Key performance indicators

	Air	Molten Salts
$\bar{\eta}_{sCO_2}$ [-]	0.5201	0.3489
E_{tot} [MW _e]	52426	39073
$CAPEX_{Direct}$ [10 ⁶ \$]	57.91	89.86
$CAPEX$ [10 ⁶ \$]	67.4	104.03
$OPEX$ [10 ³ \$/year]	167.147	168.211
$LCOE$ [\$ /MWh _e]	119.65	245.49

Multi-Objective Optimization

Performed sensitivity analysis indicates the presence of trade-offs between costs and efficiency, depending on both the TES size, the TIT and the SM. Therefore, two complete multi-objective optimizations of the proposed CSP plant design have been implemented. For both the multi-objective optimizations, the plant transient behavior has been studied for seven consecutive days for the four seasons (from 18th to 24th March, from 18th to 24th June, from 18th to 24th September and from 18th to 24th December). As a way to assesses the reliability of the shortened simulations the main results for the design case have been compared showing a relative error lower than 10 % and similar trends regarding the operation modes. In order to evaluate the techno-economic performance of the proposed air driven sCO₂ CSP plant scheme the overall mean sCO₂ cycle efficiency, $\bar{\eta}_{sCO_2}$, the $CAPEX$ and the resultant $LCoE$ have been considered. The multi-objective optimizations have been performed by means of the GA algorithm implemented in a MatLab environment [38-39]. The first multi-objective problem has been theoretically defined as:

- Objective functions: $CAPEX$, $\bar{\eta}_{sCO_2}$, $LCoE$;
- Decision variables: t_{disch} , TIT ;
- Constraints: $8 h \leq t_{disch} \leq 14 h$;
 $600^\circ C \leq TIT \leq 900^\circ C$;

Figure 11 shows the results of this multi-objective optimization for the sCO₂ power cycle. The LCoE has been plotted against the overall CAPEX of the plant. The influence of the TES size and the TIT has been reported and besides the resultant $\bar{\eta}_{sCO_2}$ has been recorded. The LCoE linearly increases with the CAPEX, however it can be clearly seen that the driving variable is the TIT. Indeed, higher TIT leads to higher sCO₂ efficiency, therefore the yearly electricity production increases, enabling a first reduction of the LCoE. Further decrease of the LCoE are related to lower costs of the sCO₂ power block at higher TIT, just partially compensated by an increase in the solar field cost. Particularly, improved thermodynamic working conditions of the sCO₂ flow in the heat exchangers limit the heat transfer area requirements, main driver for the cost of the heat exchangers. Finally, it can be noted that the TES size has a relatively low impact on both the LCoE and the power cycle efficiency. Larger TES lead to slight increase in the plant thermodynamic performances, by enabling longer discharge period.

Figure 11: Multi-objective optimization, with TIT and TES size as decision variables, economic and thermodynamic performance indicators.

As in traditional CSP plant design, the TIT can hardly be modified but it is rather considered as a design parameter, a second optimization problem has been formulated. In this case the TIT has been assumed as the design value (Table 5). Instead, the influence of the TES size and the plant SM has been investigated. Furthermore the number of operating hour of the plant (OM1, OM2, OM3 and OM4) has been considered as an additional objective in order to consider the enhanced flexibility ensured by the TES unit. Thus, the multiobjective optimization problem has been formulated as:

- Objective functions: $CAPEX$, $\bar{\eta}_{sCO_2}$, $LCoE$, h_{op} ;
- Decision variables: TES size, SM ;
- Constraints: $8 h \leq TES \text{ size} \leq 14 h$;
 $1.25 \leq SM \leq 3.25$;

For SM equal to [1.5, 2, 2.5, 3] the relative heliostat configuration and solar field efficiency matrix have been evaluated by means of the commercial software System Advisor Model [31]. Then, for each simulation the solar field efficiency has been considered equal to the closest available one. The main results are shown in Figure 12. It should be noted that the non-continuity in the results of the optimization and in the identifiable Pareto front is related to the previously mentioned assumption for the solar field efficiency matrix. In this optimization, an expected trade-off between the CAPEX and the LCoE can be identified. Smaller solar fields (lower SM) lead to lower capital costs but to an increase in the LCoE. Besides, for $SM \leq 2$ larger TES lead to an increase of the LCoE, while for larger SM bigger TES enables longer discharge phases thus a higher overall electricity production, leading to reduction in the LCoE. A larger initial investments enables lower LCoE, however a minimum LCoE can be identified at 89.4 \$/MWh for a TES size of 13.9 hours of discharge operation at full load and a SM equal to 2.47, corresponding to a CAPEX of about 73.4M\$. Furthermore, from Figure 12 the effect of the SM and the TES size on the overall number of operative hours can be seen. As expected, larger heliostat fields and larger TES increase the operative hours of the plant. Particularly, for a SM of 3.25 and 13 hours of TES capacity about 634 operative hours can be reached, thus enabling an almost continuous production (94% of the overall considered 28 days).

Figure 12: Multi-objective optimization, with SM and TES size as decision variables, economic and thermodynamic performance indicators.

Table 7: Main results for the configuration with the minimum LCoE

$LCOE$ [\$/MWh _e]	89.4
$CAPEX$ [10^6 \$]	73.4
$OPEX$ [10^3 \$/year]	168.7
Solar Multiple, SM [-]	2.47
TES size, $t_{disch_{DES}}$ [h]	13.9
Power Cycle Efficiency, $\bar{\eta}_{sCO_2}$ [-]	0.519
Operative hours, h_{op} [h]	585
Turbine Inlet Temperature, TIT [°C]	800
Net Electrical Power, W_{DES} [MW _e]	10

CONCLUSIONS

In this work, an indirect supercritical CO_2 – air driven concentrated solar plant design integrated with a packed bed thermal energy storage has been introduced and studied. In doing so, a transient thermodynamic model of the integrated concentrating solar plant, including receiver, thermal energy storage, intermediate heat exchangers and sCO_2 power cycle has been developed and discussed. In the first stage of the analysis, the thermo-economic performances of the proposed CSP plant design have been compared with the ones evaluated for a sCO_2 CSP plant driven by a solar molten salt loop. It has been shown that the proposed design enables to improve the thermodynamic performances, obtaining a mean sCO_2 overall power cycle efficiency of about 52%. Besides, the introduced sCO_2 plant design lessens the investment costs, particularly leading to a reduction of about 35% in the total CAPEX and of around 50% in the LCoE. Secondly, multi-objective optimizations have been executed in order to assess the influence of the thermal energy storage size, of the supercritical CO_2 turbine inlet temperature and of the plant solar multiple on the thermo-economic key performance indicators. The results shown that the sCO_2 power cycle efficiency linearly depends on the TIT, an increase of the TIT from 600°C to 900°C enable an increment of the power cycle efficiency of about 8%. Furthermore,

considering variable TES size and SM a trade-off between the CAPEX and the LCoE has been identified. A minimum LCoE can be identified at 89.4 \$/MWh for a TES size of 13.9 hours of discharge operation at full load and a SM equal to 2.47, corresponding to a CAPEX of about 73.4M\$. Furthermore, larger heliostat fields and larger TES increase the operative hours of the plant. Particularly, for a SM of 3.25 and 13 hours of TES capacity an almost continuous electricity production at base load can be reached.

This work testifies the potential of air driven CSP cycles coupled with supercritical CO₂ recompressed Brayton power cycles. Future works will be focused on the TES unit considering the inclusion of a cascade PCM – sensible TES that could enable higher and constant temperature during discharge. Furthermore, the direct integration scheme will be assessed considering sCO₂ as the only HTF. Finally, the economic analysis will be improved introducing sensitivity studies to account for the of different specific cost assumptions and looking into different potential markets to assess the influence of different electricity price schemes.

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NOMENCLATURE

Abbreviations

CAPEX	= Capital expenditures
COND	= Condenser
CSP	= Concentrating solar power
DNI	= Direct normal irradiance
HPT	= High pressure turbine
HTF	= Heat transfer fluid
HTR	= High temperature recuperator
LCoE	= Levelized cost of electricity
LPT	= Low pressure turbine
LTR	= Low temperature recuperator
MC	= Main compressor
MH	= Main heater
OM	= Operation mode
PB	= Power block
RC	= Re-compressor
REC	= Receiver
RH	= Re-heater
SOC	= State of charge
SR	= Split ratio
TES	= Thermal energy storage
TIT	= Turbine inlet temperature

Symbols

Bi	= Biot number
d_p	= Packed bed particles mean diameter
$\dot{m}$	= Mass flow rate
Pr	= Prandlt number
Re	= Reynold number
W	= Power

Subscripts

<i>act</i>	= Actual
<i>C</i>	= Compressor
<i>des</i>	= Design
<i>eff</i>	= Effective
<i>F</i>	= Fluid
<i>S</i>	= Solid
<i>T</i>	= Turbine

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