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(coordinadores)

Proceedings of the 7th International Seminar on ORC Power Systems

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OPERATIONAL OPTIMISATION OF THE MAIN HEAT REJECTION UNIT OF CSP PLANTS BASED ON CARBON DIOXIDE MIXTURES

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ABSTRACT

This research, developed in the framework of the SCARABEUS project, studies the off-design performance of transcritical power cycles running on CO_2-SO_2 mixtures in Concentrated Solar Power applications. The objective of this work is to identify optimum operational strategies that maximise net energy production when exposed to variable ambient temperature, with special focus on the operation of the Heat Rejection Unit (Air-Cooled Condenser). The power cycle is simulated in Thermoflex, modified with user-defined scripts to account for the specific off-design performance characteristics of key components. The Air-Cooled Condenser is modelled by means of an in-house Matlab tool, already validated in previous publications, able to accurately simulate the heat transfer process between working fluid and cooling medium (air) and to calculate auxiliary power consumption.

Four different strategies are identified, depending on ambient temperature: variable or constant condensation pressure for ambient temperatures lower than the design value, and constant turbine inlet temperature or constant return temperature of the heat transfer fluid for ambient temperature higher than design value. The results show that a combination of variable and constant minimum cycle pressure is the best alternative for low ambient temperatures, enabling net system efficiencies higher than 41%. On the other hand, constant turbine inlet temperature enables higher net performance than constant return temperature of the heat transfer fluid, even if at the expense of a reduction in energy storage capacity for the same inventory of molten salts.

1 INTRODUCTION

Supercritical Carbon Dioxide (sCO_2) power cycles are widely considered as one of the most promising alternatives to enhance the performance of next-generation (higher-temperature) Concentrated Solar Power (CSP) plants, with the final aim of increasing the cost-effectiveness of this technology, thus making it more competitive in the energy market. This is due to some outstanding features of sCO_2 power cycles such as their high thermal efficiency and low footprint, which are in turn enabled by the fact that the compression process is undertaken at high density in the proximity of the critical point (31ºC). Unfortunately, this operating temperature is difficult to achieve in CSP sites, with high Direct Normal Irradiance and ambient temperatures (T_{amb}), resulting in a clear drop in cycle thermal efficiency (Crespi *et al.*, 2022b). This is aggravated by the need to adopt dry-cooling systems in order to reduce water consumption of the power plant, what brings about two harmful effects on cycle efficiency: higher compressor work (lower thermal efficiency) and higher auxiliary power consumption (lower net efficiency). Several authors have investigated different cooling options (Moisseytsev *et al.*, 2017; Pidaparti *et al.*, 2020) and operational strategies to optimise the off-design performance of sCO_2 power cycles (Alfani *et al.*, 2020; Neises, 2020). Nevertheless, none of these solutions is really able to effectively tackle the detrimental effect of increased T_{amb} on cycle performance.

An alternative solution to this problem is proposed by the SCARABEUS project: increasing the critical temperature of the working fluid (WF) thanks to the addition of certain dopants to the raw CO_2, thus allowing condensation of the working fluid at high T_{amb}. This restores thermal efficiency to the very high values that are achieved at lower ambient temperature when pure CO_2 is used. The thermodynamic potential of this concept has already been demonstrated extensively in previous publications (Crespi *et al.*, 2022a; Crespi *et al.*, 2022b). Building upon that, and in order to take a further step towards the commercialisation of this technology, it is necessary to evaluate possible operational strategies in off-design conditions, especially taking into account the different nature of the SCARABEUS cycles (transcritical instead of supercritical). In this regard, the behaviour of the working fluid during condensation becomes very relevant (more information in Section 3).

Bearing all this in mind, this paper presents the principles of the operational optimisation of CSP plants based on CO_2 mixtures. Firstly, the computational environment and the reference power plant are presented. Secondly, the possible operating strategies are discussed, together with the specific constraints posed by the use of a condensing cycle; this calls for a focus on the operation of the Air Cooled Condenser unit (ACC). Finally, the results of a sensitivity analysis to ambient conditions are presented and discussed, identifying the operation strategy yielding the best off-design performance of the SCARABEUS system.

2 COMPUTATIONAL ENVIRONMENT

This article presents an in-depth analysis of a central receiver CSP plant with a two-tank molten-salt TES system and a transcritical recompression cycle operating on a mixture of 80%CO_2-20%SO_2 in the power block (PB), see Figure 1 (left). This is an adaption of the recompression cycle using pure CO_2 that has been widely studied in literature for next-generation CSP applications (Neises, 2020; Alfani *et al.*, 2022). The PB has been modelled using the Thermoflex v30 with specific performance models for various of the components added via LUA scripts. The thermophysical properties of the working fluid have been incorporated via look-up tables using the ‘user-defined refrigerant fluid’. This approach has been adopted in order to obtain a robust calculation platform, while overcoming the lack of specific performance models for equipment running on CO_2 mixtures in Thermoflex. The complete set of specifications is provided in Table 1. The reference location is Seville, Spain, for which meteorological data have been obtained from a public database (EnergyPlus). The design temperature is estimated at 36°C, corresponding to the 95th percentile; according to the available data, the annual ambient temperature in Seville ranges from -2°C to 43°C. To assess the performance of the system under extreme conditions, a range of interest from 0°C to 50°C has been assumed (a conservative assumption, considering that temperatures exceeding 47ºC have never been recorded in the area).

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Table 1: Design specifications of the power block

Parameter	Value	Parameter	Value	Parameter	Value
W_{Gross} [MW]	100	TIT [ºC]	550	TOT [ºC]	412.2
T_{min} [ºC]	50	$T_{HTF,in}$ [ºC]	565	$T_{HTF,out}$ [ºC]	387.2
P_{max} [bar]	250	P_{min} [bar]	79.1	η_{th} [%]	44.1

The Heat Rejection Unit of the power block is a modular air-cooled condenser (ACC), capable of condensing the working fluid even in warm operating conditions. The technology employed is a horizontal, direct air-cooled heat exchanger, with axial induced-draft fans (see Figure 1, right), following the indications of the industrial partner of the SCARABEUS consortium. The authors have developed a Matlab-based design and simulation tool of the ACC, built upon preliminary work of Illyés *et al.* (2022) and first presented in (Rodríguez-de Arriba, 2023a). Three different types of bays are studied in (Rodríguez-de Arriba, 2023b): bay with 3 single-speed (i.e., on/off) fans (called *Bay1*), bay with 2 single-speed and 1 Variable Frequency Drive (VFD) fan (*Bay2*) and bay with 3 VFD fans (*Bay3*). These three scenarios provide an opportunity to evaluate tradeoffs between capital and operational expenses of the ACC. Increasing the number of VFD fans reduces the fan power consumption but also raises installation costs (fans equipped with VFD are ~60% costlier than single-speed fans).

The Primary Heat Exchanger (PHX) is a shell-and-tube heat exchanger with the working fluid in the tubes and molten salts on the shell side. Thermoflex is used to produce a design of the PHX whereas off-design performance is evaluated using the Conductance Ratio Method through a Matlab routine (Crespi *et al.*, 2017).

For further information about the assumptions underlying the aforementioned simulation platform, interested readers may refer to (Rodríguez-de Arriba *et al.*, 2023b).

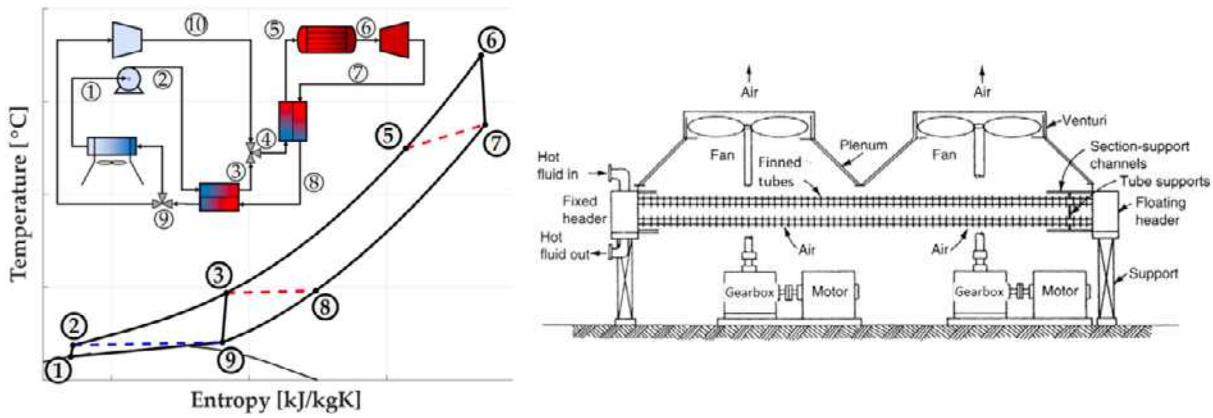

Figure 1: Left: Transcritical Recompression Cycle running on 80%CO_2-20%SO_2 mixture: layout and temperature-entropy diagram – **Right:** General scheme of an induced draft ACHE (Adapted from Serth & Lestina, 2014)

3 OPERATING STRATEGIES AND CONSTRAINTS

This study aims to investigate the operation of the subsystem composed by the PB-ACC-PHX in off-design conditions, a topic already assessed by the authors in (Rodríguez-de Arriba *et al.*, 2023b). The objective is to define an operating strategy that yields the best performance of the subsystem while satisfying the operational constraints. The system is assumed to operate base load, and the objective is maximise the annual yield of the power plant. This means that circulating mass flow rate in the system remains somewhat constant, modified by the need to compensate for the variable ambient conditions.

The following operational constraints set by the safe operation of the system are considered:

- Maximum cycle pressure (P_{max}) cannot exceed the design value (250 bar) by more than 5%;
- The turbine operates in sliding pressure;
- Turbine inlet temperature cannot exceed the design value (550ºC), in order to maintain a 15ºC temperature difference with the hot molten salts ($T_{HTF,in}$ fixed at 565ºC).
- Turbine outlet temperature should not exceed 450ºC, a threshold value set by the industrial partners of the project with the aim to avoid the installation of a specific cooling system in the turbine exhaust duct.
- The power consumption of the fans cannot exceed its rated value by more than 5%.
- Turbomachinery (recompressor and turbine) must operate within the safe and stable limits of their respective performance maps.

Regarding the last bullet point, it is of utmost importance to carefully consider the operation of the recompressor, owing to its close proximity to the two-phase region of the phase diagram. This applies to the operating conditions in the inducer in particular, where cooling arises due to local acceleration. The numerical quantification of this effect can be implemented through a similarity parameter (C) defined by Romei *et al.* (2021), which is a generalisation of the cavitation parameter (see Eq. 8 in the reference). In this work, C^* is defined as the ratio from the actual value of C for a given set of operating conditions to the rated value of such parameter. Bearing this in mind, $C^*>1$ denotes a safer operation than in the design point, $C^*=0$ indicates that the inlet to the recompressor falls inside the condensation dome, and $0<C^*<1$ indicates operating conditions in between the former two (the higher C^*, the safer the operation). Similarly, the flow coefficient of the recompressor and the turbine is evaluated by normalising it to its design value.

Special attention is paid to the operation of the ACC, for which an approach similar to that presented in a previous study by the authors (Rodríguez-deArriba, 2023b) is adopted. In this previous work, component count (i.e., the installation of several units in parallel) was studied as a means to minimise auxiliary power consumption at reduced mass flow rates by independently controlling the mass flow rate flowing through each bay. However, since all bays must operate under the same conditions as long as mass flow rate takes the rated value, this is less relevant in this study. Therefore, the results obtained for one bay are extrapolated to the entire set of bays in this work.

The performance of the ACC is studied at the rated condensation pressure (P_{min}=79.1 bar, corresponding to T_{bub} of 50ºC). Figure 2 illustrates the influence of ambient temperature on the minimum temperature of the working fluid (T_{min}, Figure 2a) and on relative fan power consumption ($W_{ACC,norm}$, Figure 2b) for various combinations of fans in operation and/or reduced fan speeds. Different ACC operation modes are represented (coloured lines) and labelled according to the rotational speed of the VFD fans (ω_i) relative to its rated value: ω_i=1 means fan operating at rated speed, ω_i=0 means fan not in operation, $0<\omega_i<1$ means a VFD fan operating at reduced speed (more information in Rodríguez-de Arriba, 2023b). Any point of the map exhibiting a T_{min} below the bubble temperature (50ºC) represents an operating point where the working fluid is subcooled. The operation of the ACC according to the type of bay under consideration (*Bay1, Bay2* or *Bay3*) can be plotted on this figure. 528

Avoiding subcooling of the working fluid is not possible if all cells use single-speed fans (*Bay1*), and this has a negative impact on cycle efficiency since the best thermal performance is obtained when the condenser yields saturated fluid at pump inlet (T_{min}=T_{bub}); unfortunately, this is possible only if -at least- one fan in each bay incorporates VFD (*Bay2* and/or *Bay3*). It is also observed that, at T_{amb}<27ºC, one of the three fans of *Bay1* can be turned off (ACC operation changes from [1,1,1] to [0,1,1]), hence reducing the fan power consumption to around 67% of the design value. However, it is not possible to shut down a second fan within the temperature range studied (>0ºC), and this yields a higher W_{ACC} for the *Bay1* case than for *Bay2* and *Bay3*. This applies to *Bay2* also but, for this case, the fan with VFD facilitates the transition from the design point (36ºC) to 27ºC, thus reducing fan power consumption. Moreover, a significant reduction in power consumption with respect to *Bay1* is also noticeable below 27ºC, with fan power consumption reaching 35% the rated value. As expected, *Bay3* offers the largest reduction of fan power at part load, reaching 80% when ambient temperature drops from 36ºC to just 30ºC; this is at the expense of a 8% higher capital cost (more info in Rodríguez-de Arriba, 2023b).

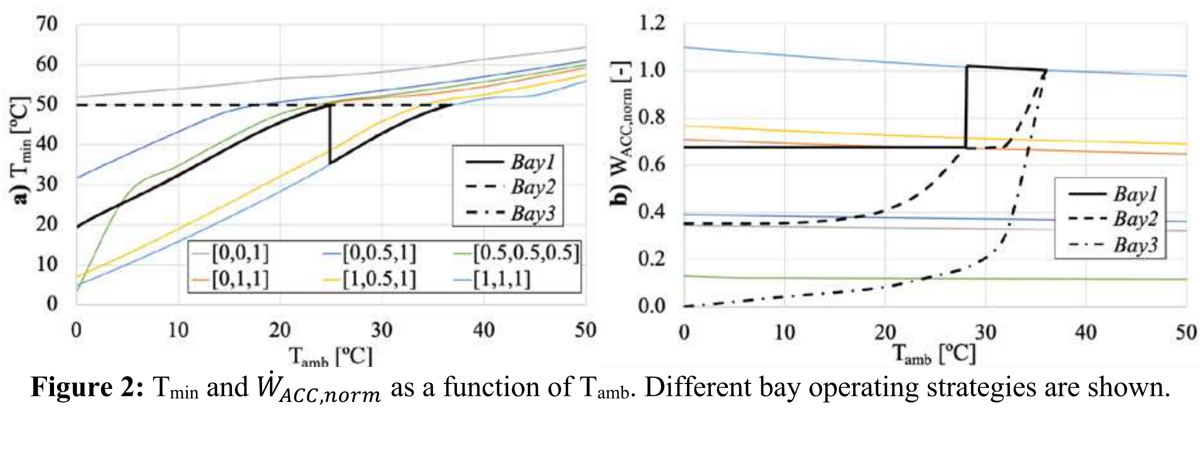

Figure 2: T_{min} and $\dot{W}_{ACC,norm}$ as a function of T_{amb}. Different bay operating strategies are shown.

The previous considerations have been made for constant P_{min}, equal to the design value and corresponding to T_{bub}=50ºC. Nevertheless, the same methodology can be applied to other condensation pressures (hence, different T_{bub}), to create an operational map of the ACC for operation at variable pressure. This is presented in Figure 3, where ACC power consumption is shown as a function of T_{amb} for different T_{bub} and considering T_{min}=T_{bub} (no subcooling at condenser outlet). The blue line represents the running line of the condenser when the fans operate at rated speed (thus, maximum condenser capacity). This operation strategy results in variable P_{min}; also, since subcooling of the working fluid is not allowed (T_{min}=T_{bub}), a lower T_{amb} brings about lower T_{min} (thus, T_{bub}) and P_{min}; in other words, T_{min} always reaches the minimum value achievable by the heat rejection capacity of the ACC. For instance, T_{min} can be reduced from 50ºC to 45ºC when T_{amb} decreases from 36ºC to 30ºC though this is partly offset by the effect of a higher density on power consumption.

The black line in Figure 3 represents operation at constant P_{min}, same case as the dashed line in Figure 2b. It is to note that Figure 3 is based on an ACC with the *Bay2* architecture, a choice aimed at balancing operational and investment costs; a similar study could have been undertaken also considering *Bay1* and *Bay3*, but this is not reported here due to length limitations.

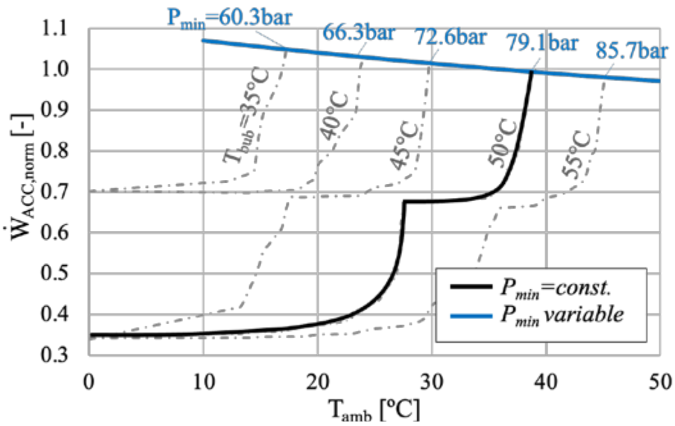

Figure 3: Operational map of the ACC at different T_{amb}

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From the foregoing results, the following two strategies for operation at variable ambient temperature are identified:

1. *Constant condensation pressure*: the condenser is oversized for the required cooling capacity at reduced ambient temperature. Therefore, it is possible to reduce fan speed or even shutdown some of them whilst the power block remains unaffected, hence with constant gross efficiency. Net efficiency, on the contrary, increases due to the lower power consumption of the ACC fans.
2. *Variable condensation pressure*: condensation pressure is reduced as much as allowed by the condenser; by doing so, it is possible to achieve lower minimum cycle temperature, potentially increasing cycle efficiency. The fans run at their nominal speed and, hence, fan power remains at the rated value approximately. As a drawback, this strategy poses several challenges to system control, as discussed in the next section.

A final remark regarding operation at higher than rated ambient temperature is also needed. In such conditions, the ACC would not be able to condense the nominal flow rate of working fluid at P_{min} and without violating the restriction not to exceed the rated power of the fans. Thus, the only viable option is to increase condensation pressure, keeping all the fans running at full speed. However, the increase in P_{min} comes with a higher minimum cycle temperature and, thus, with a lower thermal efficiency of the cycle. As a consequence, net system efficiency should be expected to drop significantly; moreover, it has been observed that the return temperature of the HTF in the Primary Heat Exchanger ($T_{HTF,out}$) increases with respect to the design value in these cases, what has a potentially detrimental effect on the energy storage capacity of the TES (Neises, 2020). To overcome this issue, an active control of TIT has already been presented in previous work by the authors (Rodríguez-deArriba, 2023b), and it is investigated in the next section.

4 RESULTS

4.1 Variable condensation pressure strategy: benefits and limitations

This section explores the performance of the power block when condensation pressure is varied through variations of the minimum cycle temperature. The objective of this study is to get a better understanding of how the different operational constraints evolve under different minimum cycle temperature, both below and above the design point, and to estimate the gross performance of the system (η_{th}). The results are presented in Figure 4. Nine figure of merits are reported: (a) η_{th}, (b) P_{max}, (c) TIT, (d) TOT, (e) $T_{HTF,out}$, (f) *C**, (g) m_{HTF}, (h) flow coefficient of the recompressor ($\Phi_{RECOMPR}$) and (i) flow coefficient of the turbine (Φ_{TURB}). The design point is indicated with a black marker and corresponds to the boundary conditions provided in Table 1. It is to note that, if constant condensation pressure operation is adopted, the system always runs at this point.

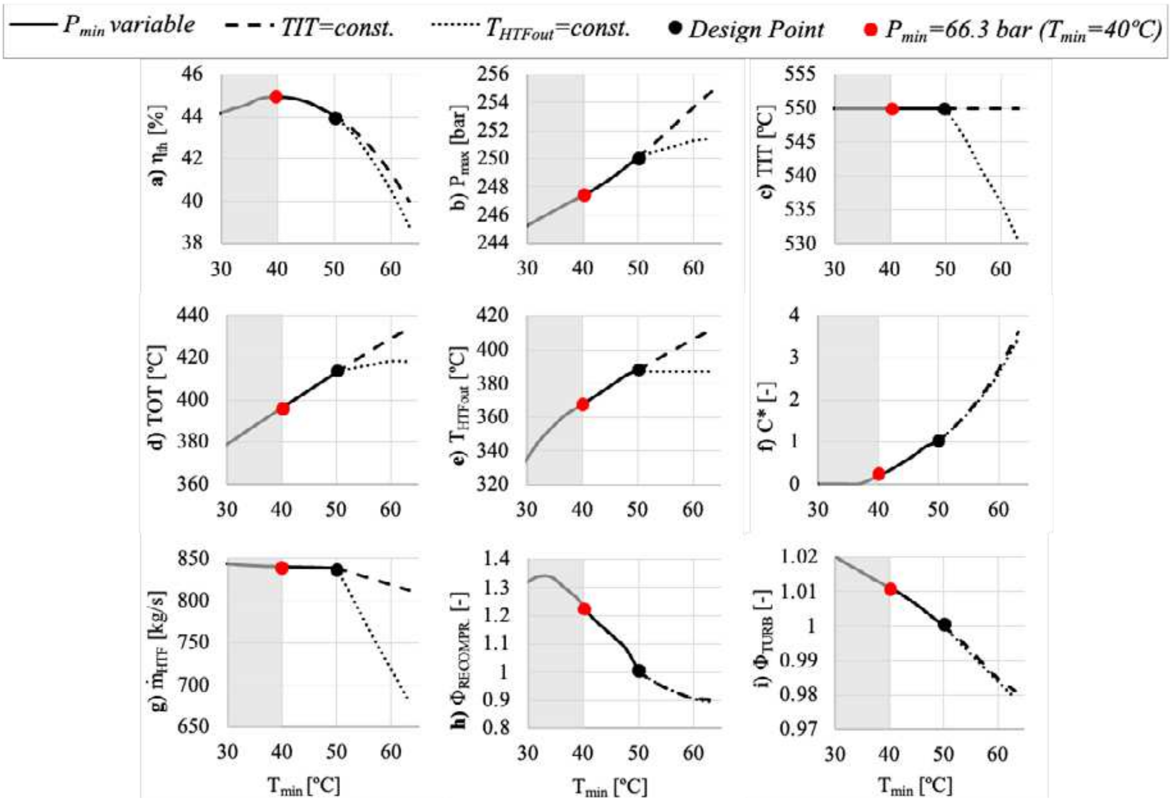

Figure 4: System performance adopting variable P_{min} strategy as a function of T_{min} (various parameters). Sub-figure captions are provided in the y-axis.

Operating conditions for minimum cycle temperature lower than the design value are represented with a solid black line. Thermal efficiency peaks at T_{min}~40ºC, owing to the lower recuperative potential of the system at lower P_{min}, as a consequence of the lower turbine exhaust temperature (TOT). Regarding turbomachinery, the turbine flow coefficient increases slightly due to changes in the speed of sound caused by the fluctuations of P_{max}. However, the most critical situation arises in the recompressor since its inlet temperature decreases as P_{min} decreases, getting closer to the dew point and reducing the safety margin *C**. When T_{min} falls below 40ºC, the inlet to the recompressor falls inside the condensation dome, meaning that the system cannot be operated at variable P_{min} for T_{min}<40ºC (corresponding to T_{amb}~24ºC, see Figure 3, intersection between variable P_{min} and T_{bub}=40ºC). This condition is graphically represented by the grey areas in Figure 4.

A possible solution to extend the operational range of the system to lower ambient temperatures is to employ a hybrid solution: variable P_{min} down to T_{min}=40ºC followed by constant P_{min}, where P_{min} is kept at a constant value of 66.3 bar (corresponding to T_{min}=40ºC). In this second range, the thermal efficiency of the power block remains unchanged, and this is represented by a red marker in Figure 4. It is worth noting that the selection of P_{min}=66.3 bar (corresponding to T_{min}=40ºC) is an arbitrary choice, driven by the fact that such conditions maximise η_{th} (see Figure 4a). Nevertheless, such value could still be problematic for the safe operation of the recompressor (C^* close to 0). In this regard, future works will investigate different values of P_{min} in order to find the best compromise between thermal performance and operability of the cycle.

It is interesting to observe that the temperature of the Heat Transfer Fluid (molten salts) at the outlet from the Primary Heat Exchanger ($T_{HTF,out}$) decreases for operation at variable P_{min}. One approach to control this temperature would be to reduce the mass flow rate circulating through the system. However, this would also reduce the electric output of the plant and, accordingly, the performance gains in comparison with the operation at constant pressure. Furthermore, as demonstrated by Rodríguez-de Arriba (2023b), reducing the mass flow rate when operating at part load shifts hte inlet conditions to the recompressor towards the dew line, thus reducing C^* even further. Another approach would be increasing turbine inlet temperature above its design value, which has been ruled out in this study in order to maintain a minimum temperature difference of 15ºC between the working fluid and the Heat Transfer Fluid.

The operation at very high minimum cycle temperatures (T_{min}>50ºC) does, on the contrary, not pose any particular challenges to the safe operation of the system. Two operating strategies are investigated for this case, the first one of which relies on operating the system at constant TIT (dashed lines in Figure 4). As mentioned in the previous section, increasing minimum cycle pressure (P_{min}) results in lower gross (thermal) efficiency, around 2 percentage points (p.p) when T_{min} increases from 50ºC to 60ºC. This also translates into maximum cycle pressures (P_{max}) shifting to values slightly above the design value (<5 bar) given that the system operates in sliding pressure; these higher pressures are, nevertheless, not foreseen to incur substantially higher mechanical stresses on the component. 531

Turbine outlet temperature also increases but it always remains below the aforementioned 450ºC threshold, thus avoiding the need for a specific duct cooling system. The flow coefficients of both the recompressor and turbine decrease slightly, well within the limits of the performance map. C^* experiences a large increase, from 1 (design) to 3 at 60ºC, meaning that the recompressor operates with a larger safety margin than in design conditions. Unfortunately, the temperature of the HTF at the outlet from the Primary Heat Exchanger ($T_{HTF,out}$) increases significantly and this reduces the thermal energy storage capacity of the TES for the same inventory of molten salts. In order to avoid this, a new control strategy based on reducing turbine inlet temperature with the aim to operate at constant $T_{HTF,out}$ is considered. This strategy is represented by the dotted lines in Figure 4, where it is observed that TIT is reduced by 15ºC when minimum cycle temperature (T_{min}) increases 10ºC above the design point. This operation brings about lower thermal efficiency than when turbine inlet temperature is kept constant. As a minor benefit though, the increase in turbine inlet pressure when running at high minimum cycle pressure is halved, also with respect to constant TIT operation (1.3 bar *vs.* 3.6 bar). Furthermore, the mass flow rate of HTF is largely reduced, driven by the larger temperature drop of this fluid across the Primary Heat Exchanger. This brings about a lower HTF pumping power (lower auxiliary power consumption) that is beneficial in terms of net cycle efficiency of the plant.

4.2 Comparison between different operational strategies

In this last section, the operational strategies discussed previously are compared in terms of net system efficiency; in the following, net power output (W^*_{net}) and net efficiency (η^*_{net}) account for the contribution of the auxiliary power consumption of the ACC as well as the mechanical losses caused by the electric motors coupled to pump and recompressor, losses in the electric generator and HTF pumping power to enable the flow of molten salts from hot to cold tank through the PHX. A complete description of all the assumptions is provided in Rodríguez-de Arriba (2023b).

The results are presented in Figure 5. Two different strategies are considered at ambient temperature below the design point (T_{amb}<36ºC). First, the operation at constant minimum cycle pressure (P_{min}) (dashed-and-dotted lines) considers that the power block operates at the design point continuously (see marker in Figure 4) and the improvement in W^*_{net} and η^*_{net} is caused by the lower fan power consumption in the ACC, following the map presented in Figure 2b. A smooth transition is observed at ~27ºC, where one single-speed fan of the ACC is shut down.

The operation at variable minimum cycle pressure (P_{min}) is presented in black solid lines in Figure 5. Using this operational strategy, P_{min} is reduced with ambient temperature (T_{amb}) following the blue line in Figure 3 until the minimum cycle temperature (T_{min}=40ºC) is reached, what happens at T_{amb}~24ºC. Beyond this point, minimum cycle temperature cannot be reduced further as, otherwise, the inlet to the recompressor would fall within the two-phase region (see previous section). Consequently, for T_{amb}<24ºC, the system operates at a constant T_{min}=40ºC (red solid lines) with the ACC following the corresponding line in Figure 3; this reduces fan power consumption of the ACC at low T_{amb} (such drop in W_{ACC} being the cause of the smooth increase in W^*_{net} and η^*_{net} when moving from 24ºC to 20ºC). This hybrid strategy achieves an increase in η^*_{net} of ~1 p.p. with respect to constant P_{min} and ~12 MW_e higher electric output (roughly 13% higher W^*_{net}). It is worth noting that this significant gain in W^*_{net} is due not only to the higher η_{th} but also to the fact that more thermal energy is being extracted from the same m_{HTF} (because of the lower $T_{HTF,out}$ enabled in the hybrid strategy than in the constant P_{min} strategy: 360ºC *vs.* 380ºC). This suggests the idea that operating the system at variable minimum cycle pressure (P_{min}) can potentially increase the annual yield of the plant, though it also presents a more complex operation given the potential condensation of the working fluid at the inlet to the recompressor. Further research on this topic is thus required to investigate the actual feasibility of this operational strategy.

For operation at high ambient temperature (higher than the design value), operation at constant TIT or at constant $T_{HTF,out}$ are possible. The resulting trends of η^*_{net} are very similar to those observed in Figure 4a for η_{th}. For operation at constant TIT, a 3 p.p. drop in η^*_{net} is observed when ambient temperature increases from 36ºC to 50ºC. In terms of net power output, this means ~16 MWe. If operation at constant $T_{HTF,out}$ is adopted, the system suffers an additional reduction in η^*_{net} and W^*_{net} of around 0.75 p.p. and 4 MWe, respectively. These results suggest that operating at constant TIT leads to higher net performance of the cycle. Nevertheless, the detrimental effect on thermal energy storage capacity, hence CAPEX (see previous section), must be borne in mind and its impact on the overall economics of the plant must be assessed before a final recommendation can be made. This is currently being explored by the authors in the context of the SCARABEUS project.

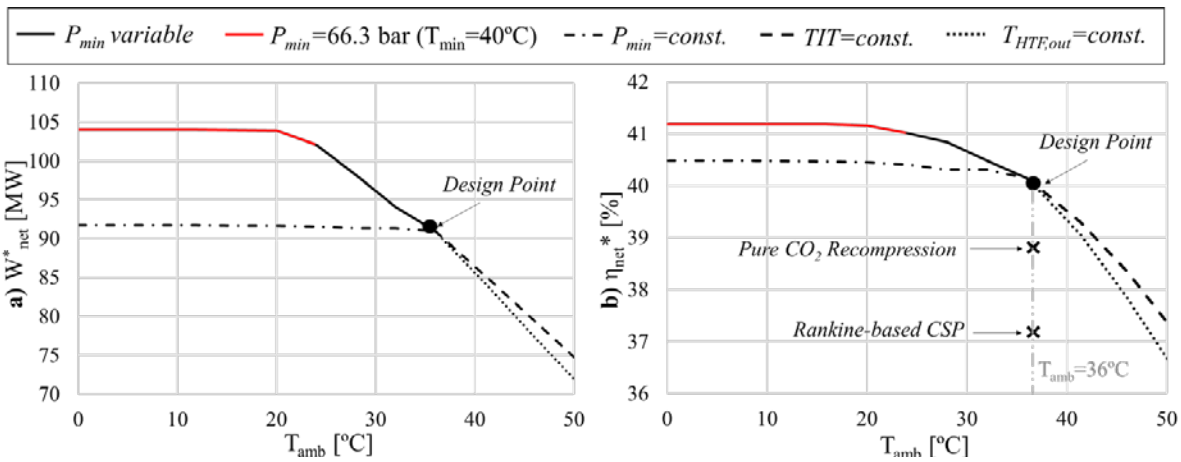

Figure 5: Net power (a) and net efficiency (b) as a function of ambient temperature for various operating strategies

This final part of the paper provides a quick comparison of the SCARABEUS system against the alternative technologies for CSP, either established or under development: steam turbines and pure sCO_2 power systems. To this aim, the net efficiency of the system η^*_{net} is compared for all three cases, considering similar boundary conditions. The results are presented in Figure 5b, where a reference net efficiency of 37.1% is assigned to a state-of-the-art steam-based CSP plant operating at 574ºC (molten salts temperature). This value corresponds to a gross efficiency of 41.2%, corrected with a gross-to-net conversion efficiency of 0.9, according to the recommendations in (NREL, 2022). The net efficiency of a recompression sCO_2 cycle is, on the other hand, calculated with the in-house platform used to estimate the performance of the SCARABEUS system, using the same boundary conditions specified in Table 1. Comparing the rated performance of these three technologies, the large potential of the SCARABEUS concept is confirmed inasmuch as the utilisation of CO_2 mixtures enables 1 p.p. and 2.4 p.p. net efficiency gains with respect to using pure sCO_2 systems and steam turbines, respectively. Future works by the authors will address a detailed techno-economic comparison of these technologies, based on LCoE and annual yield.

5 CONCLUSIONS

This study has discussed different operating strategies of CSP plants based on supercritical power cycles running on CO_2 mixtures in off-design conditions, with special focus on the operation of the Air-Cooled Condenser. The power block is a transcritical (condensing) recompression cycle operating on 80%CO_2-20%SO_2 is considered. Four different strategies are identified, depending on ambient temperature: variable or constant condensation pressure (P_{min}) for ambient temperature lower than the design point, and constant Turbine Inlet Temperature (TIT) or constant HTF return temperature ($T_{HTF,out}$) for ambient temperature higher than the design value.

The following conclusions are drawn:

- Operation at variable minimum cycle pressure (P_{min}) leads to potentially higher net efficiency and net power output than operation at constant P_{min}. As a drawback, this strategy poses the need for a specific control of the inlet conditions to the recompressor, which experience an important excursion towards the dew line.
- In this regard, the plant cannot be operated at variable minimum cycle pressure T_{min}<40ºC (T_{amb}<24ºC), since the recompressor inlet would experience condensation (assuming thermodynamic equilibrium). To overcome this issue, a hybrid strategy combining variable P_{min} down to T_{min}=40ºC and constant P_{min} for T_{min}<40ºC is explored. This hybrid strategy achieves significantly better performance than the operation at constant P_{min}.
- For ambient temperatures higher than the design value, operating at constant TIT brings about better performance than operating at constant $T_{HTF,out}$. Nevertheless, this comes at the expense of a higher CAPEX for the same storage capacity due to the need for a larger inventory of molten salts.
- In order to identify the best operational strategy, maximising the annual yield and minising LCoE, further investigations are needed. Yet, the present study sets the grounds for this further analysis.

NOMENCLATURE

ACC	Air-cooled condenser	T_{bub}	Bubble temperature [ºC]
*C**	Normalised two-phase parameter	$T_{HTF,out}$	HTF return temperature [ºC]
CSP	Concentrated Solar Power	TIT	Turbine Inlet Temperature [ºC]
HTF	Heat Transfer Fluid	T_{min}	Minimum cycle temperature [ºC]
m_{HTF}	HTF mass flow rate [kg/s]	TOT	Turbine Outlet Temperature [ºC]
PB	Power Block	VFD	Variable Frequency Drive
PHX	Primary Heat Exchanger	WF	Working fluid
P_{max}	Maximum cycle pressure [bar]	W	Power [MW]
P_{min}	Minimum cycle pressure [bar]	η	Efficiency [%]
p.p	Percentage points	ϕ	Flow coefficient
T_{amb}	Ambient temperature [ºC]		

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This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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