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Preliminary Design of S-CO2 Turbomachinery and its Influence on the Performance of an Integrated Solar Combined Cycle

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Abstract. Supercritical CO2 power cycles emerge as the next-generation standard for Concentrated Solar Power technology mainly due to their high efficiency and reduced footprint. This study investigates the role of the S-CO2 turbomachinery in an Integrated Solar Combined Cycle, and performs the design and optimization of the geometries to maximize their efficiency using mean-line models. Centrifugal compressors, axial-flow and radial-inflow turbines have been considered. Various geometries are identified to achieve similar high efficiencies, highlighting the need for additional criteria in selecting the optimal design. For this particular combined cycle, the results also reveal that variations in turbomachinery efficiency have a relatively minor impact on overall cycle efficiency, with turbine efficiency exerting a more decisive influence than compressor efficiency.

Keywords: S-CO2 Power Cycles, Concentrated Solar Plants, S-CO2 Compressors, S-CO2 Turbines, Decarbonization, Integrated Combined Power Cycles

1. Introduction

Concentrated Solar Power (CSP) technology is gaining attention for its potential benefits in future electricity networks due to its characteristics as a dispatchable renewable energy source, capable of storing large amounts of thermal energy and contributing to grid frequency stability.

However, its high levelized cost of electricity (LCOE) remains a significant barrier to widespread adoption [1], mainly due to the large solar fields required for storage. Although the LCOE has significantly decreased over the last decade, due to the increase in global installed CSP capacity, now reaching around 182 $/MWh [2], this cost is still far from the $60/MWh target set by the U.S. Department of Energy's (DOE) SunShot Initiative [3].

Efforts to reduce costs have primarily focused on enhancing system efficiency by raising the turbine inlet temperatures. The DOE's Gen3 CSP Roadmap has identified central receiver systems (CRS) with supercritical CO2 (S-CO2) power cycles as the next-generation standard for CSP plants, aiming for turbine inlet temperatures above 700°C [4].

S-CO2 power cycles potentially present significant benefits with respect to Rankine cycles [5] within the typical CSP temperature range, including higher efficiency, a simpler configuration, and a more reduced footprint [6]. This increased efficiency is due to the S-CO2's ability to recuperate heat from the turbine exhaust and the proximity of the compressor suction conditions to the CO2 critical point, which significantly increases its density, reducing compressor consumption. This last particular aspect has been thoroughly discussed in recent literature [7].

Additionally, hybrid cycles that integrate CSP with fossil fuels have been investigated to improve efficiency and flexibility in solar energy use [8]. Following this trend, a power plant was proposed in [9] combining a regenerative gas turbine with a CRS to serve as the thermal source for an S-CO2 cycle, enabling the turbine's waste heat to compensate for fluctuations in solar radiation and ensuring nearly constant power production.

Turbomachinery plays a crucial role in the implementation of S-CO2 power cycles [10]. Nevertheless, the combination of high values of pressure, temperature, and density, along with the compressor's operation near the critical point, presents significant design challenges. Therefore, substantial engineering efforts are required to displace conventional air and steam compressors and turbines, which have been developed and refined over many years [11]. In spite of recent developments, such as [12], these machines are still far from the commercial state, causing uncertainty about their maximum real achievable efficiency.

In this work, mean-line compressor and turbine designs from the Integrated Solar Combined Cycle (ISCC) described in [9] are performed to give some insights into the design process to optimize efficiency. Moreover, due to the uncertainty in the realistically achievable turbomachinery efficiency, a parametric analysis is conducted to assess the sensitivity of the overall combined cycle performance to the S-CO2 compressor and turbine efficiencies.

2. Turbomachinery mean-line design

Efficiencies of 0.87 for the S-CO2 compressor and 0.92 for the S-CO2 turbine were assumed in [9] to assess the cycle performance. Those values were selected based on what could possibly be achieved in the medium term because, as already mentioned, there are no commercial S-CO2 turbomachines yet and very scarce experimental results can be found in the literature. Therefore, in this work, the efficiencies of the S-CO2 cycle turbomachines have been estimated using mean-line models of centrifugal compressors, axial-flow, and radial-inflow turbines. Additionally, different possible optimal geometries have been evaluated.

2.1 Compressor design

Two different tools have been employed to pre-design the compressor and estimate its total-to-total efficiency, both following a mean-line approach. First, an in-house model based on loss correlations proposed by Aungier [13], and second, the commercial software COMPAL® from Concepts NREC.

The in-house model has been used to estimate the losses in the impeller, the vaneless diffuser, the volute, and the parasitic losses and has been validated by comparing the estimated performance with that published by Sandia [14]. In this case, however, a vaneless diffuser has been considered. Five parameters were selected to carry out an optimization procedure to maximize the compressor efficiency: the specific speed (n_s), the impeller blade outlet backsweep angle (β_{2bl}), the impeller blade inlet angle at the middle height (β_{1M}), the hub/tip radius ratio at the impeller inlet (r_{1h}/r_{1t}), and the variation of the meridian velocity through the impeller (c_{m2}/c_{m1}). The optimization analysis has been performed with the genetic algorithm

Matlab® function varying the parameters in the following ranges: 0.65< n_s <0.71; 54<β_{2bl} <71; 50<β_{1M} <65; 0.57<c_{m2}/c_{m1}<0.7; 0.25<r_{1h}/r_{1t}<0.3.

Table 1 presents the results corresponding to the highest efficiencies obtained, showing that there are different geometries leading to very similar performances. In all cases, the optimal specific speed is roughly 0.7, which implies a rotational speed of 18,000 rpm. Together with the total-to-total efficiency, several geometric magnitudes and parameters are presented, which allows the comparison of the different designs. It is found that the values obtained are within the margins that are generally recommended in the literature in relation to centrifugal compressors (0.02<b_2/D_2<0.12; 0.5<D_{1shr}/D_2<0.8; $D_{eq}=w_{max}/w_2$<1.7).

Table 1. *Summary of the compressor geometries leading to the highest performance values*

Input parameters				Efficiency	Parameters selected for design evaluation					
β_{2bl} (º)	$\frac{cm_2}{cm_1}$	β_{1M} (º)	$\frac{r_{1h}}{r_{1t}}$	η_c	D_2 (cm)	$\frac{b_2}{D_2}$	$\frac{D_{1shr}}{D_2}$	D_{eq}	Head coef.	D_s
54	0.57	50	0.25	0.859	23.92	0.091	0.53	1.656	0.68	3.7
63	0.6	55	0.3	0.861	24.92	0.09	0.53	1.47	0.62	3.9
71	0.587	57.7	0.25	0.862	26.28	0.09	0.51	1.286	0.56	4.1

In the case of COMPAL®, from the different available options, an efficiency optimization based on both the diameter (D_2) and the height at the impeller exit (b_2) was selected. Additionally, the rotational speed was estimated as 18,000 rpm, derived from the assumption of n_s=0.7, in the range of high-performance centrifugal compressors. The impeller blade outlet backsweep angle (β_{2bl}) was iterated, finding a value of β_{2bl}=65º as optimum. The maximum efficiency (η_c) achieved was 0.862 with D_2=26.71 cm, b_2/D_2=0.071, and a head coefficient of 0.486.

As can be observed, various geometries are possible for achieving a high value of efficiency, which suggests the need for further evaluation using additional criteria to select the best option. In the case of the in-house model, it is observed that as β_{2bl} increases, the maximum peripheral speed of the impeller also increases, which in turn enlarges the impeller's diameter, leading to a consequent growth in mechanical stresses. However, as the influence of the centrifugal force in the compression process increases, the contribution of diffusion decreases, a behavior that is shown in the reduction of D_{eq}. A lower flow deceleration implies greater stability due to a lower tendency for boundary layer detachment. Considering the pros and cons, the design corresponding to the intermediate value of β_{2bl} =63º could be the most suitable of those shown in Table 1. Furthermore, this design aligns more closely with the COMPAL® design results.

2.2 Turbine design

Two different geometries have been studied, in both cases using the rotational speed selected as optimal for the compressor (18,000 rpm).

In the first place, an axial-flow turbine has been assessed using an in-house code, achieving a total-to-total efficiency (η_t) of 0.88 with 3 stages. The highest diameter of the rotor turbine is 34.66 cm, with a mean diameter of 29.15 cm. To achieve an efficiency of 0.9 the number of stages must be increased to 5. A diagram of the design obtained using the commercial software AXIAL® is shown in Figure 1, where the geometry is quite similar to the one obtained using the in-house code.

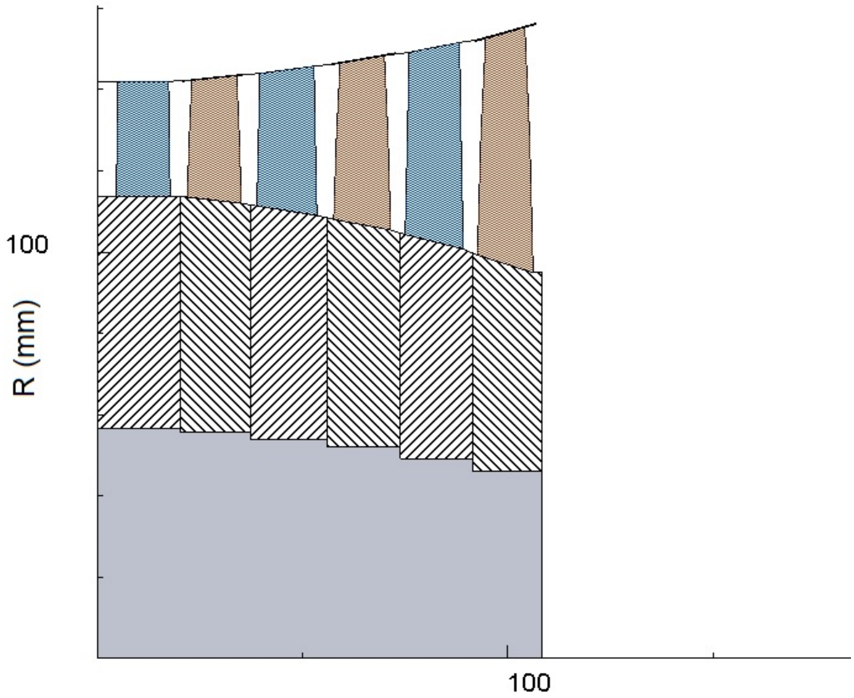


Figure 1. *Axial-flow turbine geometry (mm)*

Secondly, a radial in-flow design has been performed, following the method and loss correlations proposed by Aungier [15]. Also in this case, an optimization study by genetic algorithms was carried out to select the values of some geometrical relations, which are required as input data. The optimized geometry leads to a high efficiency of 0.90, with the utmost diameter, corresponding to the intake volute, of 98.96 cm and an impeller diameter of 48.28 cm. Considering the high value of the average fluid temperature in the turbine (552.6 ºC) and that the maximum peripheral speed is 454.9 m/s, the choice of material can be a relevant aspect in this case.

3. Influence on cycle performance

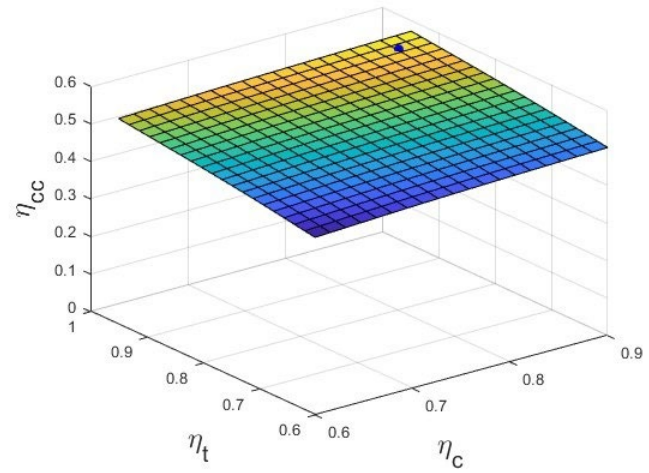
The efficiency obtained in [9] for the nominal point of the ISCC is η_{cc} =0.5636. A parametric analysis has been conducted by varying turbomachinery efficiencies from 0.60 to 0.95 (turbine) and from 0.60 to 0.90 (compressor) to evaluate their impact on the cycle efficiency. The results are shown in Table 2 (in relative values) and in Figure 2 (absolute values; nominal point highlighted).

Table 2. *Relative variation of cycle performance*

$\Delta\eta_c$ (%)	$\Delta\eta_t$ (%)	$\Delta\eta_{cc}$ (%)
+3.4	+3.3	+1.1
-8	-13	-4.5
-20	-24	-9.7
-31	-35	-16

Results reveal that variations in turbomachinery efficiency lead to only minor changes in the ISCC efficiency. Moreover, Figure 2 indicates that the turbine's influence is much greater than that of the compressor. For instance, when considering a η_c= 0.80 (7 percentage points less than the nominal efficiency assumed) alongside the nominal value in the turbine η_t= 0.92, the resulting global cycle efficiency is 0.56, practically the same as the nominal value.

a)



b)

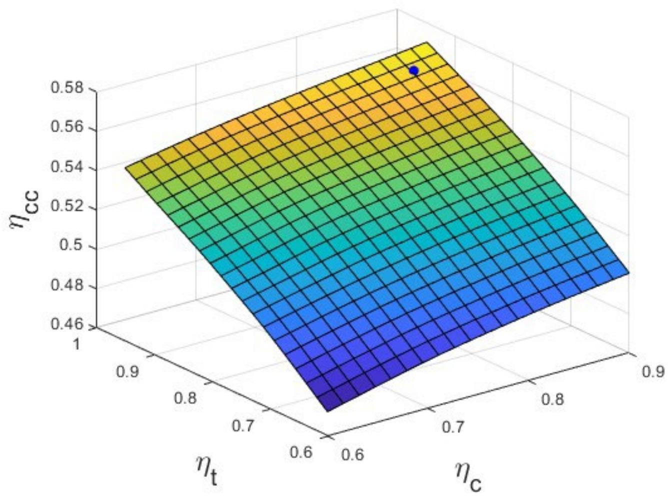


Figure 2. *Combined cycle performance for different values of $\boldsymbol{\eta_c}$ and $\boldsymbol{\eta_t}$. (b) is a larger scale than (a)*

4. Conclusions

A mean-line design procedure has been applied to obtain the geometries of the S-CO2 compressor and turbine of the Integrated Solar Combined Cycle (ISCC) described in [9] to maximize their efficiencies.

The maximum efficiency obtained for the compressor is 0.862, corresponding to a radial compressor, while the assumed efficiency to assess the cycle in [9] was 0.87. Regarding the turbine, a value of 0.88 was achieved for the axial geometry with 3 stages and 0.90 with 5 stages. For the radial geometry, the maximum efficiency was 0.90, all values slightly lower than the value assumed in [9], which was 0.92. It can be concluded that multiple geometries of the turbomachinery can lead to similar high efficiencies, indicating the need for further evaluation with additional criteria to determine the optimal choice.

An analysis of the influence of turbomachinery efficiency on the ISCC performance has also been assessed, showing that variations in turbomachinery efficiency do not significantly impact cycle efficiency. It has also been found that the turbine efficiency has a stronger influence than the compressor efficiency.

Data availability statement

Not applicable

Author contributions

Conceptualization, E.A., M.M., A.C.; Formal Analysis, M.M., A.C., J.I.L..; Funding acquisition, J.I.L., A.R.; Investigation, E.A., M.M., A.C., M.J.M.; Methodology, E.A., M.M., A.C.; Project administration, J.I.L., M.J.M., A.R.; Resources, A.C., M.J.M., A.R.; Software, E.A., M.M., A.C.; Supervision, E.A.; Validation, M.M., A.C., M.J.M.; Visualization, E.A., M.M., A.C.; Writing – original draft, E.A., M.M., A.C.; Writing – review & editing, J.I.L., M.J.M., A.R. All authors have read and agreed to the published version of the manuscript.

Competing interests

The authors declare that they have no competing interests.

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