Preliminary design of S-CO2 turbomachinery and its influence on the performance of an integrated solar combined cycle.

SolarPACES

Arenas, E.[1,2], Muñoz, M.[3], Cantizano, A.[1,2], Linares, J.I.[1], Montes, M.J.[3], Rovira, A.[3]

[1] Rafael Mariño Chair in New Energy Technologies, Comillas Pontifical University, Spain

[2] Institute for Research in Technology, Comillas Pontifical University, Spain

[3] Department of Energy Engineering, Universidad Nacional de Educación a Distancia (UNED), Spain

*Correspondence: Arenas, E., earenas@comillas.edu

1. Introduction

Concentrated Solar Power (CSP) technology is gaining attention for its potential in future electricity networks due to its dispatchable renewable energy and grid stability benefits. However, its high cost, mainly from large solar fields required for storage, restricts widespread adoption. Hybridization with gas turbines, combining different CSP systems, offers a promising solution.

Brayton supercritical CO2 cycles (S-CO2) are proposed to enhance energy conversion efficiency in CSP systems by replacing steam power cycles [1]. A power plant was proposed in [2] combining a regenerative gas turbine with a Central Receiver System (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.

The S-CO2 turbomachinery, particularly the compressor, is not in a commercial state due to its proximity to the critical point, causing uncertainty about its maximum achievable efficiency. A mean-line compressor and turbine design for the Integrated Solar Combined Cycle (ISCC) described in [2] is performed here. A parametric analysis is also conducted to assess the sensitivity of the overall combined cycle performance to turbomachinery efficiency. This work has been supported by the Spanish Ministry of Science and Innovation through the PID2019-110283RB-C33 and PID2019-110283RB-C31 projects.

2. Turbomachinery mean-line design

Efficiencies of 0.87 for the compressor and 0.92 for the turbine were assumed in [2], based on what possibly could be achieved in the medium term. In this work, mean-line models of centrifugal compressors, axial-flow and radial inflow turbines, have been used.

2.1. Compressor design

Two different tools have been employed: an in-house mean-line model based on Aungier's loss correlations [3] and the commercial software COMPAL® from Concepts NREC. These models require several input parameters. Most of them can be selected within narrow ranges based on previous high-performance designs. With the in-house model, four parameters were selected to carry out an optimization procedure to maximize the compressor efficiency: the specific speed (n_s), the impeller blade exit angle (β_{2bl}), the impeller blade inlet angle at middle height (β_{1M}) and the variation of the meridian velocity through the impeller (cm_2/cm_1). In the case of COMPAL®, the diameter and height at the impeller exit were optimized simultaneously. Additionally, the rotational speed and the angle β_{2bl} must be carefully selected. The maximum efficiency (η_c) achieved was 0.862 with β_{2bl}=-65º and β_{1M}=-69.33º, similar for both in-house and COMPAL® models. However, very similar performances were obtained with several other impeller geometries, indicating the need for further evaluation using additional criteria.

2.2. Turbine design

Two different designs have been studied, with the rotational speed selected as optimal for the compressor. In the first place, an axial-flow turbine, achieving an efficiency (η_t) of 0.91 with 3 stages. The highest diameter of the rotor turbine is 39.7 cm. Secondly, a radial in-flow design has been performed, obtaining a slightly lower efficiency, η_t =0.899, with the utmost diameter, corresponding to the intake volute, of 98.96 cm.

3. Influence on cycle performance

The efficiency for the ISCC is η_{cc} =0.5636. A parametric analysis was conducted by varying efficiencies from 0.6 to 0.95 (turbine) and from 0.6 to 0.9 (compressor). The results are shown in Table 1 (in relative values) and Fig.1 (absolute values; nominal point highlighted).

Table 1. *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

Fig. 1 reveals that variations in turbomachinery efficiency do not strongly impact cycle efficiency. Furthermore, the turbine's influence is much greater than that of the compressor. For instance, when considering η_c= 0.8 and η_t= 0.92 (nominal), the ISCC efficiency is 0.56.

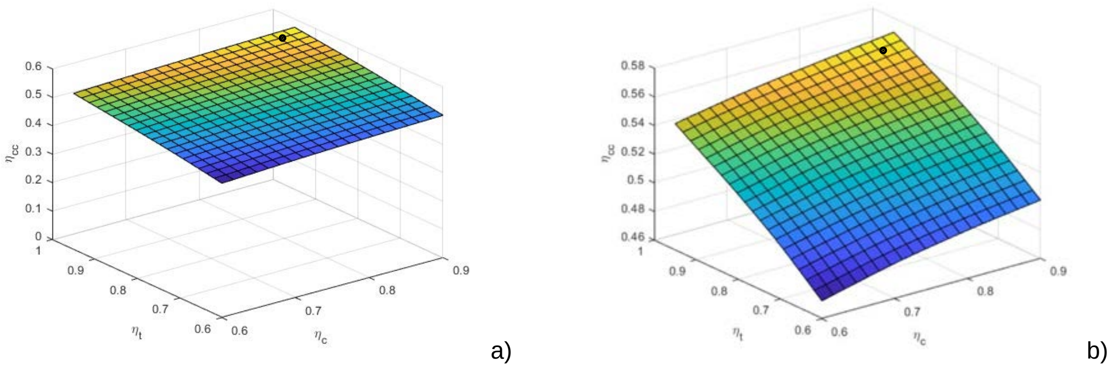

Fig. 1: Combined cycle performance for different values of η_c and η_t. (b) is a larger scale than (a)

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