DESIGN AND ANALYSIS OF RADIAL AND AXIAL TURBOMACHINERY OF SUPERCRITICAL CO_2 POWER CYCLES

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ABSTRACT

The most common application of the supercritical CO_2 (S-CO_2) power cycle is a Brayton cycle operating between two supercritical pressures. The key idea of this cycle is that the compression process occurs in the vicinity of the critical point, which results in a consumption power lower than in a classical Brayton cycle, which uses ideal gas as working fluid. This reduction in the compression work enables the S-CO_2 power cycle to work with a medium temperature range in the thermal source, achieving good efficiencies in the range of 300ºC/500ºC. The design of the turbomachinery plays an important role in S-CO_2 power cycles and it is outside the experience of the existing turbomachines. The size of the machines is very small compared to the size of turbines and compressors of classical cycles because of the high density of S-CO_2. In this study, a compressor and a turbine for a high-power S-CO_2 cycle from a fusion power plant are designed and different alternatives are explored. Radial versus axial configuration, number of stages and other relevant characteristics are investigated in order to maximize the efficiency of the machines. Two analysis tools, AXIAL™ and COMPAL™ from Concepts NREC, for multistage axial and radial turbomachinery, respectively, have been used to carry out the mean-line designs. In addition, NIST property database is used in the calculations, so real CO_2 properties have been considered.

INTRODUCTION

The selection and design of the turbomachinery plays an important role in the definition of an S-CO_2 power cycle. The characteristics of S-CO_2 involve the operation of the compressor close to the critical point and the turbine at high pressure and temperature entailing to high values of fluid density. Consequently, compressors and turbines of S-CO_2 power cycles are very compact machines compared to those of classical power cycles (Brayton, Rankine...) [1]. At the turbine exit, the CO_2 density is about 10,000 times greater than for a condensing steam turbine and over 100 times that of a combustion gas turbine [2]. The main advantage of this compactness is the cost reduction, but some issues arise due to the lack of experience in the design of this kind of machines as they lie out of the ranges of pressure and temperature for the existing turbomachinery. So, the design of S-CO_2 turbines and compressors is a challenge that must be overcome for the success of these promising cycles.

The design of the turbomachinery (compressor and turbine) of a S-CO_2 power cycle is carried out in two steps. First, a conceptual design using dimensional analysis is performed, leading to the selection of the machine type and the number of stages. Then a meanline design is completed using AXIAL™ and COMPAL™, from Concepts NREC. The design parameters for both machines are based in the values of DEMO project power plant [3].

NOMENCLATURE

Ds	[-]	Specific diameter
g	[m/s^2]	Acceleration of gravity
H	[J]	Enthalpy drop
Ns	[-]	Specific speed
V	[m^3/s]	Volumetric flow rate at the compressor inlet or at the turbine exit
D	[m]	Diameter of the rotor
p_r	[-]	Pressure ratio

Special characters

ω	[rad/s]	Rotational speed

Subscripts

ad		Adiabatic

CONCEPTUAL DESIGN OF THE TURBOMACHINERY

The selection and conceptual design of the turbomachinery is accomplished using the method developed by Baljé [4] through the nondimensional variables Ns (specific speed) and Ds (specific diameter). The specific speed is calculated using the following equation:

$$Ns = \frac{\omega \cdot \sqrt{V}}{\left(g \cdot H_{ad}\right)^{3/4}} \tag{1}$$

The specific diameter (Ds) is defined as:

$$Ds = \frac{D \cdot \left(g \cdot H_{ad}\right)^{1/4}}{\sqrt{V}} \qquad (2)$$

Baljé showed that well-designed machines, historically, share common values of those nondimensional parameters and plotted them in several Ns-Ds diagrams that include the efficiency contours (Figures 1, 2).

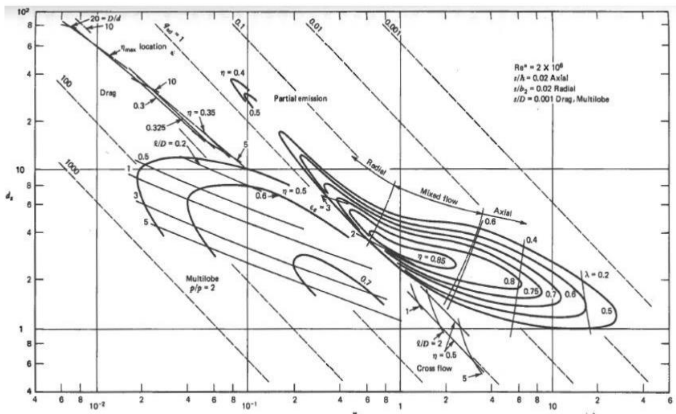

Figure 1 Specific diameter versus specific speed diagram for compressors [4]

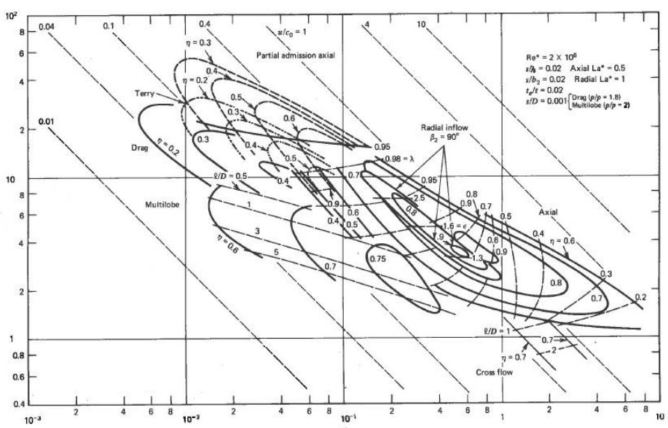

Figure 2 Specific diameter versus specific speed diagram for turbines [4]

So, knowing the nominal operating conditions, Ns can be obtained for the compressor and for the turbine. If the values do not lie in a high-performance region, the total mass flow rate can be split to obtain several parallel lines, or several stages can be considered to obtain the required Ns, as Fuller et al., describe in [5] for S-CO_2 turbomachinery.

Both radial and axial compressors may be considered, but radial compressors are preferred if high efficiency needs to be guaranteed at off-design conditions [6]. The turbine is designed to be axial, as recommended in [1]. A rotational speed of 3000 rpm is assumed for both machines, allowing a single shaft configuration, where the turbine may directly drive the compressor and the synchronous generator.

CONCEPTUAL DESIGN OF THE COMPRESSOR

The design parameters of the CO_2 compressor are summarized in Table 1. These are typical values in a CO_2 fusion power cycle [3] and corresponds to a 300 MW compressor.

Compressor	
Inlet pressure (bar)	85
Inlet temperature (ºC)	35
p_r (-)	4.12
Mass flow rate (kg/s)	7135
Power (MW)	300

Table 1 Design parameters for the CO_2 compressor

An analysis of the specific speed (Ns) of the compressor is performed using the steady state operating conditions and a rotational speed of 3000 rpm, assuming several serial and parallel configurations, as depicted in Figure 3.

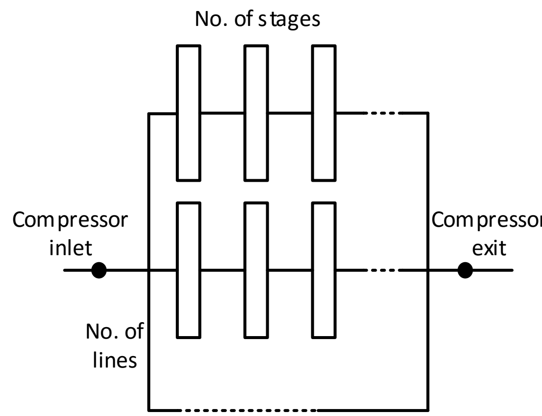

Figure 3 Serial and parallel configurations

Depending on the mass flow rate or the pressure ratio, the Ns of the individual stage will be different, meaning that the geometry required to maximize the efficiency will be different. Different values of Ns mean different geometry (from radial to axial) and different values of maximum efficiency.

First, the region for the radial geometry was analysed. This region comprises values of Ns below 0.7. Up to 3 parallel lines and 4 stages for every line were checked, calculating the specific speed for every configuration and reading from the Baljé diagram (Figure 1) the corresponding value of maximum efficiency. The results are shown in Figure 4.

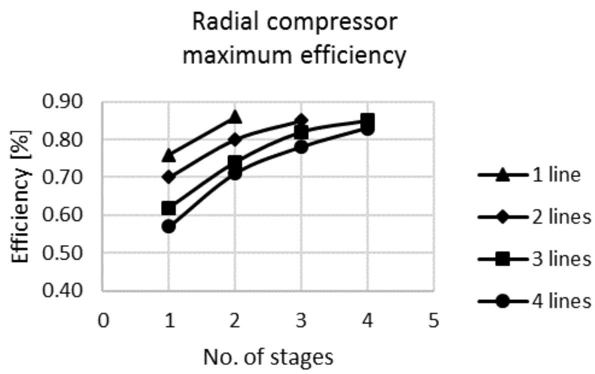

Figure 4. Maximum efficiency of a radial compressor as a function of parallel lines and number of stages.

As can be seen, a compressor based on one line (total mass flow rate through it) and two stages would achieve the best efficiency, being more or less the same as 2 lines and 3 stages or 3 lines and 4 stages. For the sake of simplicity and cost efficiency, the first option would be preferable. The efficiency of such a compressor would be 86%, according to the Baljé diagram.

The axial geometry was then analysed, focusing on the corresponding region of the diagram (Ns greater than 2). In this case, only configurations with high mass flow rates may be considered. Accordingly, just one line is assumed, and the values of Ns and maximum efficiency are calculated for different number of stages. The results are shown in Figure 5, where the maximum efficiency is obtained for 12 stages. The efficiency of this axial compressor would be 83%, which is lower than the expected efficiency for a radial compressor.

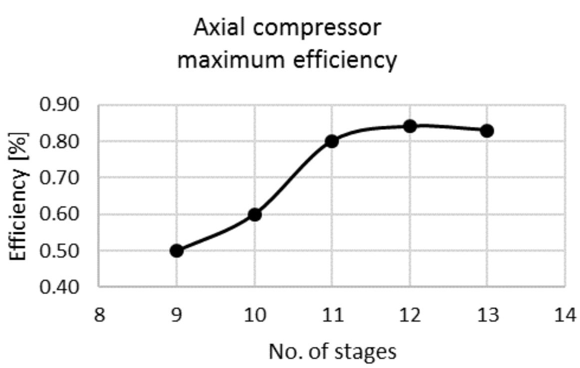

Figure 5. Maximum efficiency of an axial compressor for different number of stages and one line.

CONCEPTUAL DESIGN OF THE TURBINE

The design parameters of the turbine are given in table 2. These values are typical of a CO_2 fusion power cycle [3] and corresponds to a 1200 MW turbine.

Turbine	
Inlet pressure (bar)	350
Inlet temperature (ºC)	485
Exit pressure	85
Mass flow rate (kg/s)	7225
Power (MW)	1200

Table 2 Design parameters for the CO_2 turbine.

The specific speed (Ns) of the turbine is calculated (Equation 1) using the steady state operating conditions, and a value of 0.35 is obtained. The maximum efficiency that corresponds to this value is not high, so several stages must be assumed. Only axial geometry is considered, given the actual trend in the literature for such a high-power turbine [2].

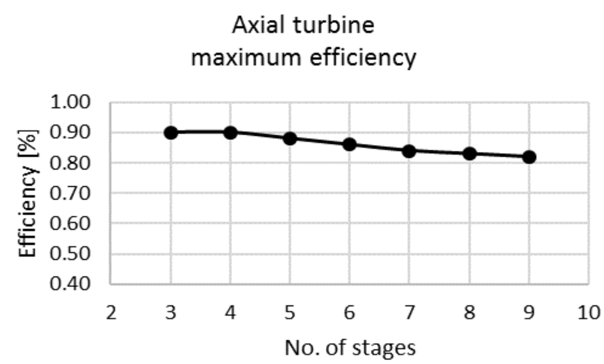

Figure 6. Maximum efficiency of an axial turbine for different number of stages.

As Figure 6 shows, according to the Balje´s diagram for turbines (Figure 2), the maximum efficiency is found for 3-4 stages, being around 90%. The solution of 3 stages is discarded as it is quite closed to the radial inflow turbine region in the diagram. Therefore the design with 4 stages is considered to be the best one.

MEANLINE DESIGN OF THE COMPRESSOR

The conceptual design of the compressor is checked using COMPAL™. This software allows to design a compressor stage, to analyse performance, to refine parameters, and model the machine according to a number of performance models. NIST real fluid properties are used in the calculations.

An analysis is accomplished to obtain all the performance parameters for the one line and two stages configuration. An efficiency of 89% was obtained, which is in good agreement with the results of the preliminary design. The improvement of the efficiency in this analysis compared to the conceptual analysis is not relevant and this value can be only considered relative to other configurations. This design is shown in Figure 7.

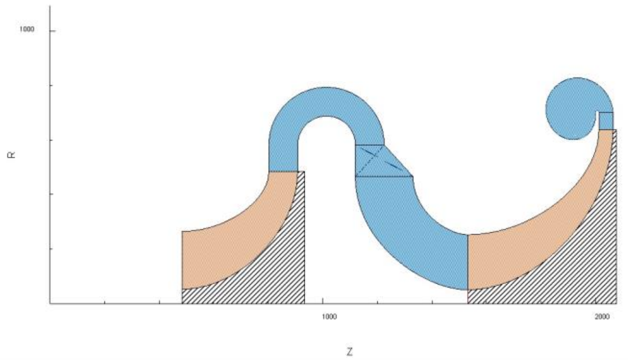

Figure 7. Radial compressor configuration. Units: mm

MEANLINE DESIGN OF THE TURBINE

Once the conceptual design of the turbine is carried out, this design is modelled using AXIAL™. This code supports axial compressors and turbines with design-point and off-design analysis including a state-of-the-art loss modelling system. As in the compressor calculations, NIST real fluid properties are used.

A constant mean radius configuration has been chosen, obtaining the rotor shown in Figure 8. The efficiency achieved with this design is 93%, which is in accordance with the values found in the literature for axial turbines.

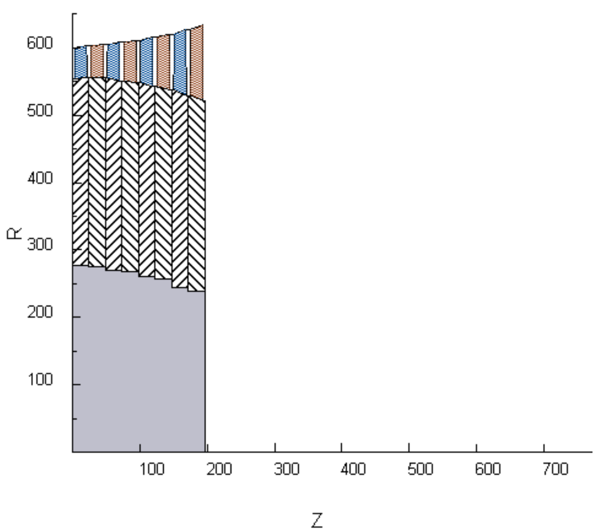

Figure 8. Rotor turbine geometry with 4 stages. Units: mm

CONCLUSION

The design of turbomachinery, compressor and turbine of a S-CO_2 power cycle for a fusion power plant has been carried out in two steps. First, a conceptual design using dimensional analysis based on Baljé's diagram was performed, leading to the selection of the machine type and number of stages. Then a meanline design has been proposed using AXIAL™ and COMPAL™, from Concepts NREC.

The compressor geometry to maximize the efficiency is found to be radial, achieving an efficiency of 89% with a two-stage and one-line configuration. For the turbine, axial turbines have shown the best performance in high power applications; therefore an axial turbine is proposed, with a maximum efficiency of 93% and a 4-stage configuration.

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