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Preliminary Approach of 1-Dimensional Streamline Method for S-CO_2 Axial Compressor

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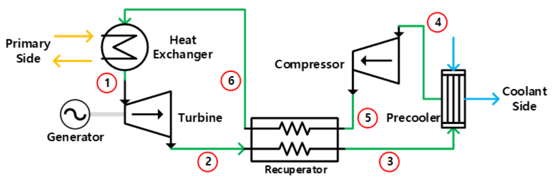
1. Introduction

In recent years, the development of the supercritical carbon dioxide (S-CO_2) power cycle has emerged as a significant advancement in enhancing the efficiency of nuclear power generation [1]. These cycles operate by utilizing carbon dioxide in a supercritical state as the working fluid, promising higher thermal efficiency compared to traditional steam turbine systems. Designing turbomachinery, such as compressors and turbines, plays a crucial role in determining the overall thermal efficiency of the power cycle.

The design of the axial compressor becomes imperative as the system capacity exceeds 10 Mwe. Thus, the challenges associated with compressor changes from radial compressor to axial compressor. Figure 1 shows the 10 MWe S-CO_2 simple recuperated cycle for the Molten Salt Reactor Experiment (MSRE). Molten salt reactors can provide a higher turbine inlet temperature for the power cycle. The cycle mass flow rate is 222 kg/s, and due to the large mass flow rate, the blade velocities are designed to have a Mach number in excess of unity when it is designed as a radial compressor.



Channel ID	Channel T(℃)	Channel P(MPa)
1	630.0	14.7
2	585.0	10.53
3	53.08	10.38
4	35.0	10.23
5	43.75	15.0
6	511.1	14.85

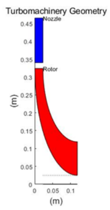


Fig. 1. 10MWe S-CO_2 Simple recuperate cycle and component design

A notable recent research has successfully designed a S-CO_2 axial compressor, utilizing a genetic algorithm with machine learning method [2]. The designed compressor was validated by Computational Fluid Dynamics (CFD) and experiments with gas condition CO_2. This achievement highlights the potential for significant advancements in the efficiency of large-capacity S-CO_2 power cycles.

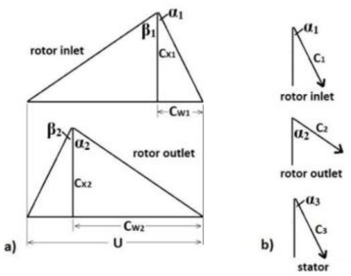
Building on this work, this paper employs the 1-Dimensional mean streamline method to evaluate how accurately this method can predict the compressor's experimental results previously disclosed. This method has previously been validated in the analysis of S-CO_2 radial compressors [3]. However, no studies have been conducted to determine if this method can be applied to the experimental results of CO_2 axial compressors. By utilizing this approach, the research aims to reduce the complex process often encountered in the compressor design, enhancing the accuracy and reliability of design methodologies.

2. Methods and Results

2.1 1-Dimensional Mean Streamline Analysis Method

The Euler work equation and velocity triangle govern the operation and performance of an axial compressor as shown in Equation 1 and Figure 2 [4]. In an axial compressor, mean stream line is where the radius is the root mean square of tip radius and hub radius as shown in Equation 2. In the first stage of an axial compressor, working fluid flows through the Inlet Guide Vain (IGV), rotor 1, and stator 1 as shown in Figure 3.

$$\text{Euler work} = \Delta h_o = U * (C_{w2} - C_{w1}) \qquad \text{(Equation 1)}$$



[5]

Fig. 2. Velocity triangle of compressor

$$r_{mean} = \sqrt{(r_{tip}^2 + r_{hub}^2)/2} \qquad \text{(Equation 2)}$$

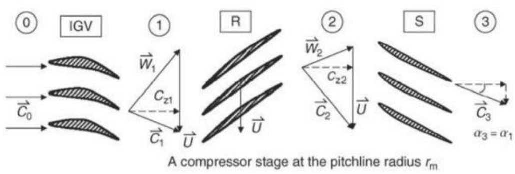


Fig. 3. Flow stream of the beginning stages in an axial compressor

The thermal stations of each stage are determined by the Euler work equation, the relationship between static enthalpy and stagnation enthalpy, as shown in Figure 4, where subscription 'o' means stagnation state. The determination of the thermal state of each blade reflects the entropy generation model shown in Equation 3 [6].

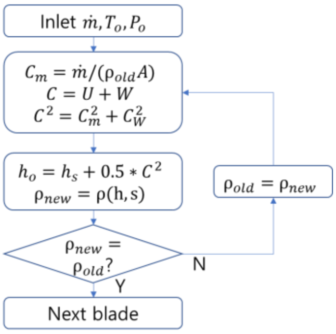


Fig. 4. Algorithm of defining velocity triangle and thermal properties for each blade

$$\zeta = \frac{T * \Delta s}{h_{o1} - h_1} \quad \text{(Equation 3)}$$

2.2 Design Parameters and Validation

The compressor shown in Table 1 was designed by M.G. Turner of NASA [2]. A genetic algorithm was used to determine the number of blades and the leading and trailing angles of each stage. It is the first axial compressor to be built and tested successfully with S-CO_2 as the working fluid. Figures 5 through 7 show pictures of the IGV, rotor, and first stage of the stator. The first stages of the compressor have been successfully performed with high-accuracy to designed values in University of Notre Dame Propulsion and Power Laboratory [7]. Three stages, nine stages, and finally a compressor with a pressure ratio of 10 and an outlet pressure of 30 MPa testing will be conducted. The compressor test section 3D modeling figure is shown in Figure 8. The first stage rotor combined to the test section and efficiency map are shown in Figure 9 [8].

Fig. 5. IGV of the S-CO_2 axial compressor

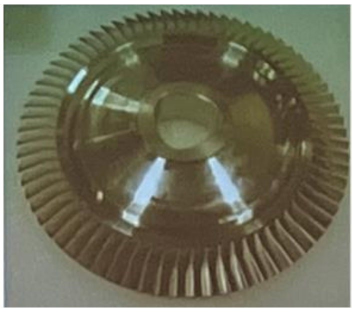


Fig. 6. 1st stage rotor of the S-CO_2 axial compressor

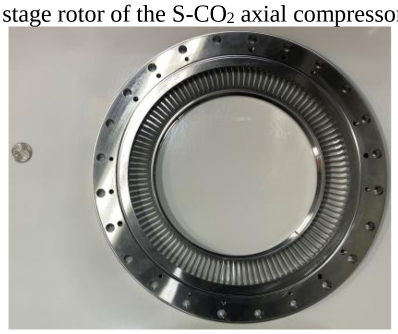


Fig. 7. 1st stage stator of the S-CO_2 axial compressor

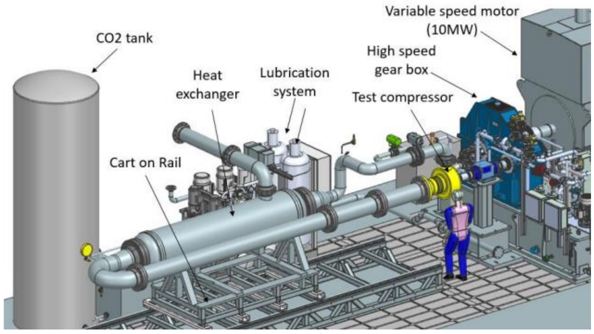


Fig. 8. S-CO_2 Compressor test section in University of Notre Dame Propulsion and Power Laboratory [7]

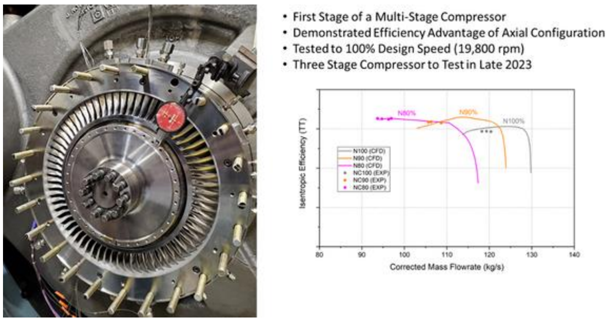


Fig. 9. 1st stage rotor combined to test section and efficiency map [8]

Table 1 lists the S-CO_2 axial compressor design parameters. The entropy generation coefficients are derived from CFD analysis, and the values at each stage are shown in Table 1 [2]. The entropy generation factor can be applied to the 1-D method because the CFD performance predictions are accurate, as shown in Figure 9. The resulting variables for validating the design are shown in Table 2. The flow coefficient is calculated by the mass flow rate and rotational speed, and the loading coefficient is calculated by the pressure ratio, isentropic efficiency, and rotational speed. Power is calculated by the pressure ratio, mass flow rate, and the isentropic efficiency. The pressure ratio data is obtained from a presentation at the S-CO_2 Power Cycle Symposium 2024 by Jeongseek Kang, which has not yet been published. Of these parameters in Table 2, the loading coefficient and the flow coefficient are dimensionless values of the work done by the compressor. These values are considered more important than the others because they are the main parameters used when designing and verifying other compressors. Specific speed and specific diameter of this rotor are 2.00 and 1.75, where axial type is more appropriate than radial type according to the Balje's ns-ds diagram.

Table 1. Design Parameters of the S-CO_2 axial compressor [2]

Parameter	**IGV**	**Rotor**	**Stator**
Number of blades (-)	43	69	114
Tip radius (mm)	131	129	128
Hub radius (mm)	102	102	102
Leading edge angle (deg)	0	-58.78	48.53
Trailing edge angle (deg)	10.70	-42.46	14.00
Rotational speed (RPM)	0	19,800	0
Inlet Total Pressure (kPa)	2,770		
Inlet Total Pressure (K)	371.15		
Total to total Pressure ratio (-)	1.41		
Mass flow rate (kg/s)	125		
Entropy generation coefficient (ζ)	0.0085	0.0425	0.0144

Table 2. Validation parameters and the equation

Parameter	**Equation**
Mass flow rate (kg/s)	$\dot{m}$
Isentropic efficiency (%)	$(h^{is}_{o,out} - h_{o,in}) / (h_{o,out} - h_{o,in}) * 100$
Pressure ratio (-)	P_{out} / P_{in}
Loading coefficient (-)	$(h_{o,out} - h_{o,in}) / U^2$
Flow coefficient (-)	$C_{m,rotor,in} / U$
Power (MW)	$\dot{m} * (h_{o,out} - h_{o,in})$

The results and errors of the experiments and the 1-D method are shown in Table 3. Most of the errors are less than 10%, but the loading coefficient error is significantly off by 20%.

Table 3. Validation of 1-dimensional mean streamline method

Parameter	**Experiment**	**1-D method**	**Error (%)**
Mass flow rate (kg/s)	120	120 (Set)	0
Isentropic efficiency (%)	89.5	97.2	8.60
Pressure ratio (-)	1.44	1.52	5.70
Loading coefficient	0.425	0.510	19.8
Flow coefficient	0.567	0.579	2.12
Power (MW)	3.31	3.56	7.62

Table 4. Mach number at each stage

Station	**Mach number**
Before IGV	0.550
Before IGV	0.645
Before R1 (relative)	0.941
After R1	0.723
After R1 (relative)	0.649
After S1	0.417

Using the one-dimensional method, it is found that the Mach number of each stage does not exceed unity, so there is no significant shock wave loss.

2.3 Discussion

There are several reasons for the large discrepancy between experimental results and the 1-D approach. In general, the turning angle is designed to be different at HUB and TIP, which is the difference between the leading edge angle and the trailing edge angle. It is because even if each part is subjected to the same turning angle, the torque at the TIP and the torque at the HUB will be distributed differently. In addition, the blade speed, U, is the radius times angular velocity, so U at the TIP and U at the MEAN are formed differently, and the absolute velocity of the working fluid at the TIP must be designed so that it does not exceed the speed of sound. Since the loading and pressure ratio are determined with the velocity triangle at the meanline, the 1-D method results in larger values than in reality.

While the 1-D method can be useful for predicting compressor operation in off-design conditions, it is expected that the method that will inherently reduce the error is to divide the streamline into several spans radially and analyze each one. In other words, 2-D approach seems to be needed.

3. Conclusions and Future Works

This study evaluated the effectiveness of the 1-Dimensional mean streamline method for predicting the performance of an S-CO_2 axial compressor compared against experimental data. The findings reveal that, while the method generally offers reasonable predictive accuracy, significant discrepancies in loading coefficient prediction indicate limitations in capturing the full complexity of axial compressor dynamics. The primary challenge lies in accurately modeling the varied effects of blade angles and velocities across different radial heights of the compressor, suggesting a need for higher dimensional approaches that consider radial flow variations. The blockage effect and wake effect could not be evaluated due to the lack of blade thickness information. If the blade thickness information is available, the accuracy of the 1-D method is expected to increase.

This study highlights the potential of the one-dimensional mean streamline method as a preliminary tool for compressor design, while also emphasizing the need for more sophisticated models to improve predictive capability. Moving forward, incorporating radial analysis appears to be a promising way to improve the design prediction accuracy of S-CO_2 axial compressors, i.e. 2-D throughflow analysis, which could contribute to the advancement of efficient and sustainable nuclear power generation technologies.

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