A CFD Investigation of Using an Isothermal Compressor in a Supercritical CO_2 Cycle for SMART Applications

Jin Young Heo [a], Seong Gu Kim[a], Jeong Ik Lee [a]
[a]*Department of Nuclear and Quantum Engineering, KAIST, Daejeon, South Korea*
Email: jyh9090@gmail.com, skim07@kaist.ac.kr, jeongiklee@kaist.ac.kr

1. Introduction

After the MOU between Saudi Arabia and the Republic of Korea in 2015 [1], the attention towards the deployment and development of the SMART (System-integrated Modular Advanced Reactor) developed by KAERI (Korea Atomic Energy Institute) has escalated. The major advantage of the small modular reactor (SMR) type technology such as SMART points toward modularized components, which allow significant reduction of construction cost as well as the overall size.

The supercritical carbon dioxide (s-CO_2) power cycle further maximizes such benefits of modularization. The s-CO_2 power cycle utilizes CO_2 in the supercritical state as a working fluid to operate the cycle at a higher density than steam. Hence, the system enjoys the reduction of compression work, and therefore, achieves high efficiency, by operating near the critical point, a region where the working fluid exists at a higher density. Furthermore, the power system also benefits from a lower volume due to its simplified layout and size reduction of turbomachinery compared to the conventional steam Rankine cycle. Due to these advantages, the s-CO_2 power cycle can be adopted in SMART for further improvement [2].

In order to further enhance the s-CO_2 power cycle for the SMART application, adopting an isothermal compressor to the cycle layout can be advantageous especially in terms of cycle efficiency. In the previous analysis conducted by the authors [3-4], the advantage of using an isothermal compressor in the reference s-CO_2 cycle layouts has been investigated. For the recompression Brayton cycle, which can avoid the pinch point from occurring inside the recuperators, adopting the isothermal compressor can improve the efficiency by 1-2% points.

Although the potential of using an isothermal compressor is understood, whether the turbomachine can operate realistically is a main issue to tackle. One important aspect of the isothermal compressor is the internal cooling that will remove the heat of compression during the compression process. Previously, the authors have adopted the infinitesimal approach to calculate the performance of the turbomachine, and the amount of heat removal required has been calculated [4]. However, it is needed to examine whether heat exchange area can be properly held within the system needs to be verified.

To study deeper about the aforementioned issues of interest for the s-CO_2 isothermal compressor, a study using a CFD software needs to be conducted. In this research, a CFD analysis using Star-CCM+ software is performed to observe the major effects when internal cooling is applied to the isothermal compressor. Similar analyses have been undergone to investigate the improvement of cooling a centrifugal compressor [5]. Moosania et al. sought after the most effective cooling place to increase the performance of the compressor. A similar methodology can be adopted to analyze the effectiveness of the isothermal compressor for s-CO_2 power cycle applications.

2. Methods and Results

Before designing in specifics related to the s-CO_2 isothermal compressor for SMART application, a preliminary study on the concept is studied. In order to do so, the geometry and the design variables are taken from the s-CO_2 compressor from the SCIEL loop facility in KAERI [6]. This is to draw a comparison with previously conducted results and to create the geometry model from a preexisting compressor.

Table 1: Design variables from the SCIEL compressor [6]

Design Variables	SCIEL Compressor
Total inlet temperature	33°C
Total inlet pressure	7.8MPa
Pressure ratio	1.8
Mass flow rate	3.2kg/s (each side)
Total to total efficiency	65%
Number of vanes	16
Shaft speed	70,000rpm

In this study, a commercial CFD code STAR-CCM+ V11.06 is used, under k-ω SST (Shear Stress Transport) model with wall function. Also, the CSV table of s-CO_2 thermodynamic properties read from NIST REFPROP database [7] is adopted for calculations. For the mesh scheme, it consists of polyhedral elements with 10 prism layers.

To investigate the changes in compressor performance, 4 different sets of thermal specification conditions are provided to the reference compressor. The combination of thermal boundary conditions is the following: adiabatic (case 1), constant temperature for hub surface (case 2), constant temperature for shroud surface (case 3), and constant temperature for hub and shroud surfaces (case 4). The comparison of these four cases is analyzed for temperature, velocity, total pressure, and enthalpy.

Figures 1.1 and 1.2 show the total pressure and temperature distributions in the SCIEL s-CO_2 compressor impeller when all surface boundary

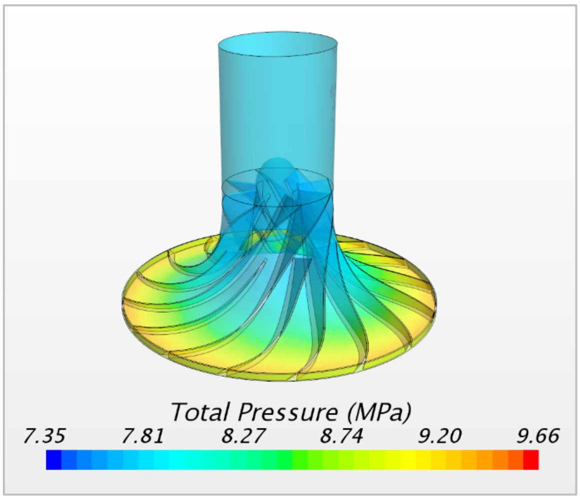

Fig. 1.1 – Total pressure of the SCIEL s-CO_2 compressor impeller (adiabatic)

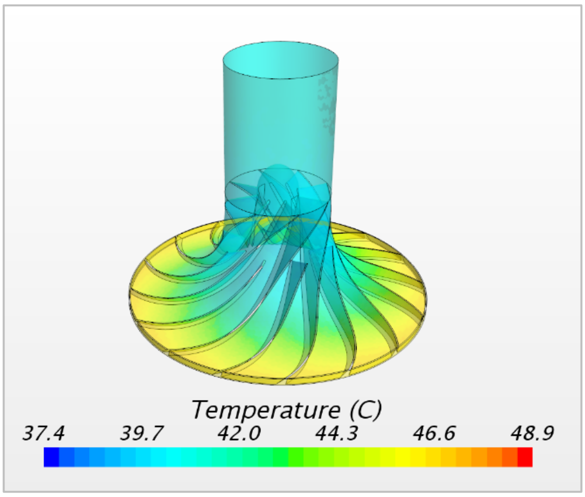

Fig. 1.2 – Temperature of the SCIEL s-CO_2 compressor impeller (adiabatic)

conditions are given as adiabatic. When the thermal specification condition on the impeller hub surface is changed to constant temperature at 35°C, it generates result figures as shown in Figs. 2.1 and 2.2.

In the comparison, one noticeable difference is in the total pressure distributions between case 1 and 2. At the impeller exit surface, the total pressure is about 5-10% greater for case 2. This can be explained by the increased density due to the lowered temperature near the flow path. As the temperature is maintained at a low value throughout the hub surface, the compressed flow undergoes heat transfer with the cooled surface to have increase in density. This leads to the overall increase in total pressure as same work is added to a denser working fluid.

However, it is difficult to identify that the exit temperature of the impeller has been sufficiently lowered so that it equals the inlet temperature. This implies that not enough heat exchange for cooling has occurred to cool the entire volume of the compressed air passing

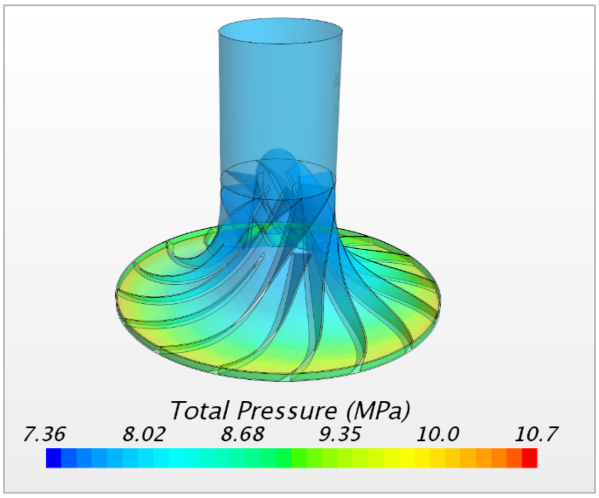

Fig. 2.1 – Total pressure of the SCIEL s-CO_2 compressor impeller (constant temperature on the hub surface)

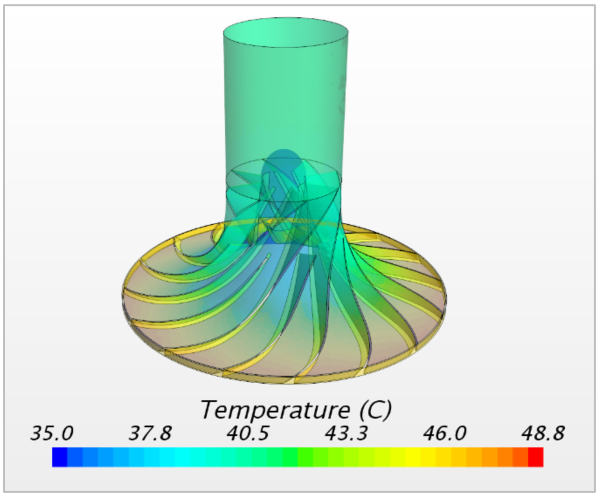

Fig. 2.2 – Temperature of the SCIEL s-CO_2 compressor impeller (constant temperature on the hub surface)

through. In order to increase the heat exchange rate of cooling inside the isothermal compressor, either the heat exchange area or the heat flux can be increased.

Case 3 results shown in Figs 3.1 and 3.2 compare similarly to those of case 2, as there is an increase in total pressure but not enough of temperature reduction. The total pressure at the exit of the impeller is increased about 2-3% compared to that of case 2. It can be explained that since the centrifugal force acts dominantly on the fluid in rotating motion, more fluid is in contact with the shroud surface than the hub surface, and therefore, increasing the amount of cooling done to the working fluid.

Finally, results shown in Figs 4.1 and 4.2 describe the conditions under which both hub and shroud surfaces are providing the cooling of the working fluid passing through. The total pressure of the impeller exit is even greater than that of case 3, and its temperature ranges slightly lower due to extensive cooling. It is still far from

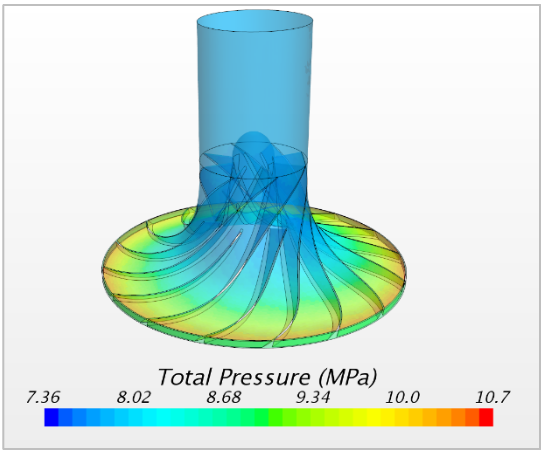

Fig. 3.1 – Total pressure of the SCIEL s-CO_2 compressor impeller (constant temperature on the shroud surface)

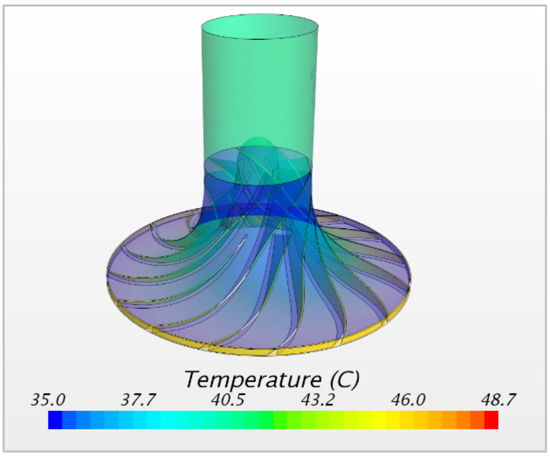

Fig. 3.2 – Temperature of the SCIEL s-CO_2 compressor impeller (constant temperature on the shroud surface)

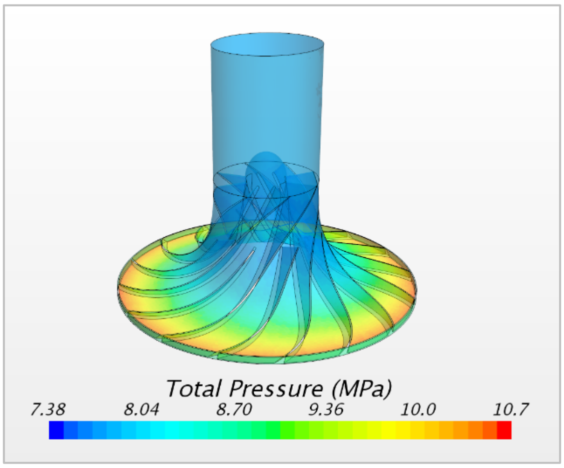

Fig. 4.1 – Total pressure of the SCIEL s-CO_2 compressor impeller (constant temperature on the hub and shroud surfaces)

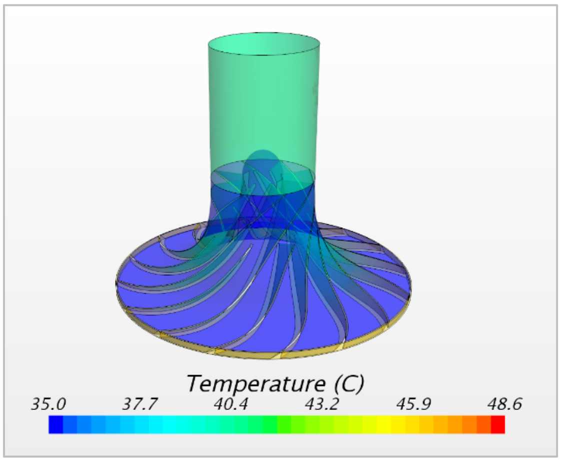

Fig. 4.2 – Temperature of the SCIEL s-CO_2 compressor impeller (constant temperature on the hub and shroud surfaces)

the inlet temperature range, implying that the degree of cooling performed inside the impeller should be much greater than what the constant temperature boundary condition offers.

Several points can be made from the simple CFD analysis of four different cases. The heat transfer surface area inside the impeller of the isothermal compressor remains small considering the available surface for cooling compared to the required amount of heat removal for an isothermal compression process. Further detailed analysis will be conducted to accurately assess the heat flux level for cooling that will closely model the isothermal compression. The required amount of heat removal for isothermal compression can be calculated using the methodology of infinitesimal approach, described in previous works by the authors [4].

3. Conclusions

The isothermal compressor has been shown potential to improve the s-CO_2 Brayton cycle layouts suitable for SMART applications. In achieving closer to the realization of the isothermal compressor for s-CO_2 cycle applications, preliminary CFD investigation has been conducted. Although the results do not yet provide a guideline for a feasible isothermal compression process within the compressor, cooling the hub and shroud surfaces leads to having higher total pressure at the impeller exit. If the rise in pressure converts to the reduction of compression work, it can be implied that the cooling of the surface will bring claimed benefits to improving the efficiency of the s-CO_2 power cycle.

Nonetheless, many obstacles must be overcome in order for the technology to become fully realized. A detailed analysis on the appropriate cooling load for an actual isothermal compression process needs to be conducted.

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