Design Study of Surface-Cooled Centrifugal Compressor in a Supercritical CO_2 (S-CO_2) Brayton Cycle

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1. Introduction

Nuclear power has been playing a significant role in generating electricity for industries and society. Until recently, nuclear power plants using water as a working fluid are dominant. However, the demand for safer and more efficient reactor has driven a wide range of research for the next generation nuclear reactors. The S-CO_2 Brayton cycle is identified of having high thermal efficiency, simple layout and compactness, which all traits can enhance performance of the next generation nuclear reactor greatly.

The advantages of S-CO_2 Brayton cycle mainly come from reduced compression work than conventional Brayton cycle [1]. To further exploit this advantage, the previous research was performed to use an isothermal compressor for the S-CO_2 power cycle [2]. In Fig 1, blue color process 4-5 is the conventional adiabatic compression process. On the contrary, red line 4-5 process is the isothermal compression process. The isothermal compression is elaborated in Fig 2. This paper is the follow up research to further the design of isothermal compressor. The study focuses on designing the compressor with one-dimensional infinitesimal cooling and compression model under constant cooling flux conditions. KAIST_TMD, an in-house 1D mean stream line code, was modified for this purpose [3].

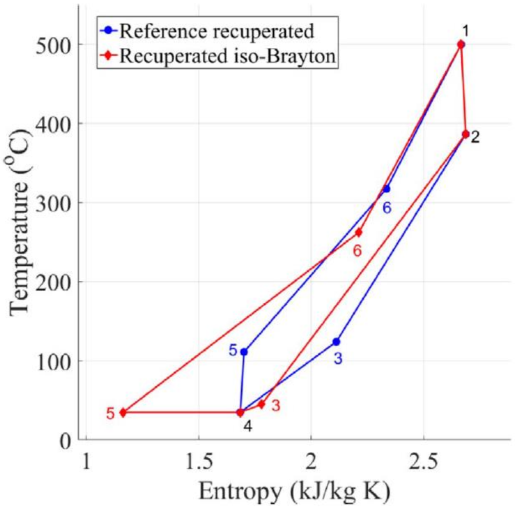

Fig 1. Variation in thermodynamic cycle with isothermal compression [2]

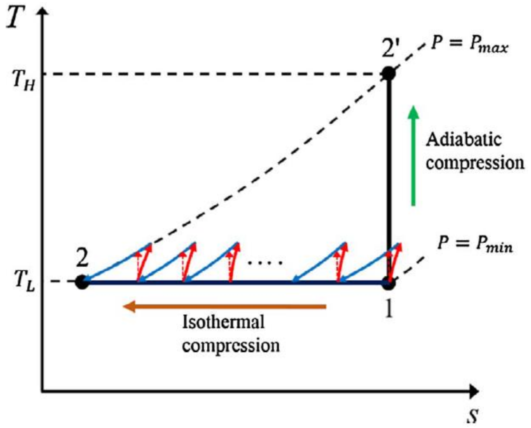

Fig 2. Infinitesimal isothermal compression process [2]

2. Methods and Results

2.1 General compressor model

Since it is very difficult for a designer to consider three-dimensional compressor geometry at first, 1D meanline design method is widely applied to first calculate basic geometries and performance for turbomachinery. In this study, KAIST_TMD, a MATLAB code developed by KAIST research team, was modified. However, complexities including loss models and slip factor were neglected for simplicity and reducing uncertainty. A simplified calculation flowchart of the modified code is shown in Fig 3.

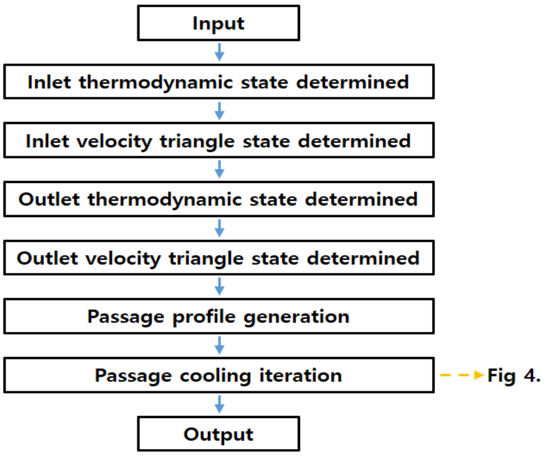

Fig 3. Overview of 1D procedure for compressor rotor

In Fig 3, inlet velocity, RPM, backswept angle and the centerline distance of the outlet are all controlled by a user, which follows the conventional design approach.

Then, firstly, the code determines inlet thermodynamics states and velocity profile. Secondly, according to the target pressure, it calculates for isentropic compression process. The compression work is derived in Euler turbine equation. In common practice, losses and slip factor should be considered because these factors decrease the pressure rise in the outlet. However, in this study, losses and slip factor are neglected. As a consequence, the determined inlet and outlet conditions produce a rotor passage profile. After the calculation process is completed, cooling is considered by adopting the infinitesimal work and cooling concept.

2.2 Infinitesimal compression and cooling model

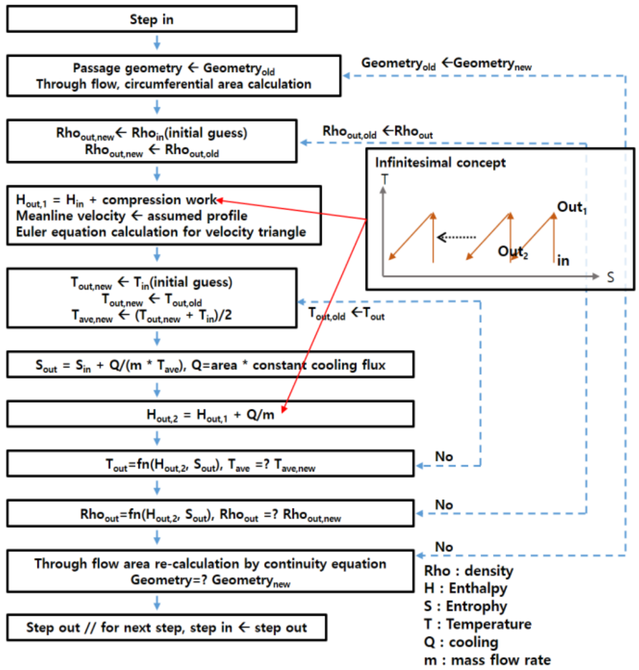

Fig 4. Passage cooling iteration scheme

Impeller outlet conditions are path-independently calculated in most 1D meanline design methods as mentioned in section 2.1. However, as cooling is newly considered in this study, it is necessary to perform the calculation path-dependently for the analysis of isothermal compressor performance, and thus calculations were performed at each step from inlet to outlet. In order to simulate compression and cooling at the same time, infinitesimal concept was introduced. This iteration scheme is shown in Fig 4. For the compression, non-cooling compression work was distributed on each step, and for the cooling, internally reversible heat transfer assumption was applied. This was reflected on $H_{Out,1}$ and $H_{Out,2}$ in Fig 4. In addition, it was assumed that the meanline velocity does not change with respect to the cooling, which induced the change in shape to satisfy continuity equation with density change. As a result, thermodynamic states as well as velocity diagram in the flow-path, and outlet conditions can be all calculated.

2.3 Results

Table 1. Design parameters

Inlet pressure	8 MPa
Not cooled target pressure	21 MPa
Mass flow rate	175.69 kg/s
Inlet velocity	30 m/s (No swirl)
RPM	20200 rev/min
Inlet temperature	313.15 K
Backswept angle	-50 º
Cooling flux	10^8 W/m^2
The number of steps	200

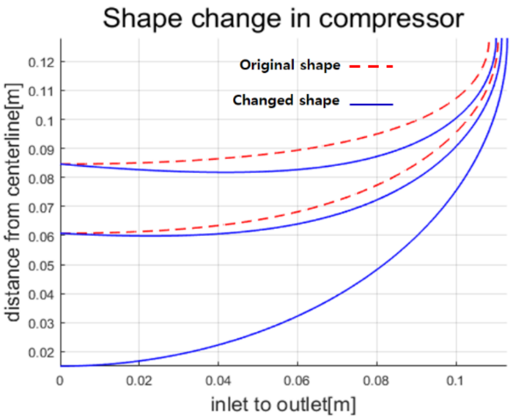

Fig 5. Shape change in compressor

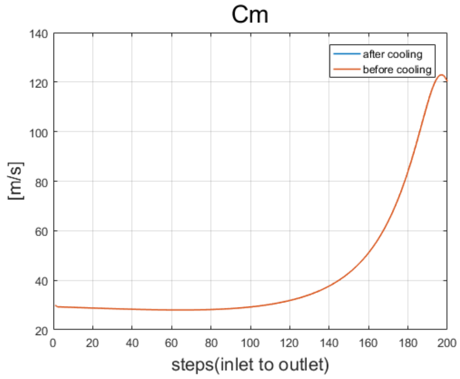

Fig 6. Absolute meanline velocity

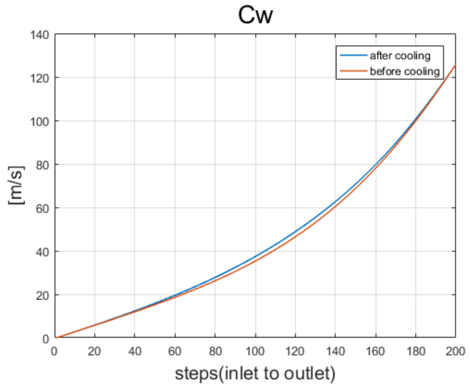

Fig 7. Absolute tangential velocity

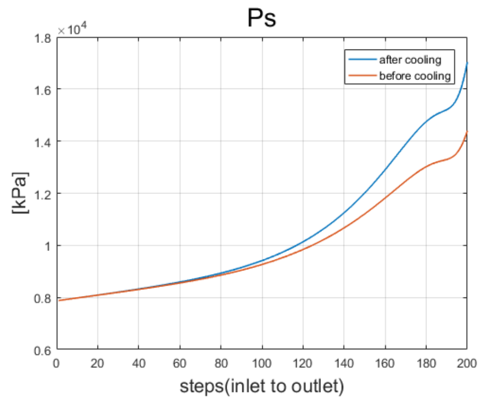

Fig 8. Static pressure

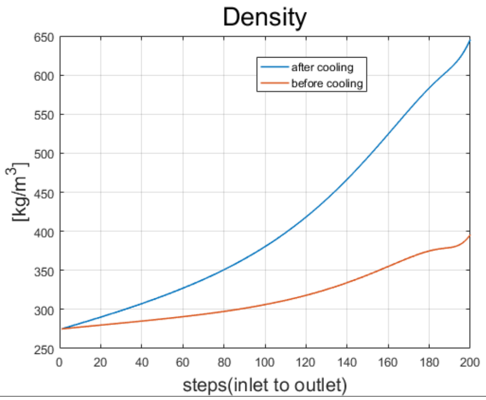

Fig 9. Density

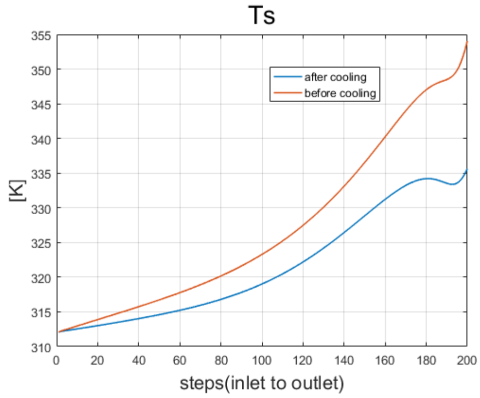

Fig 10. Static temperature

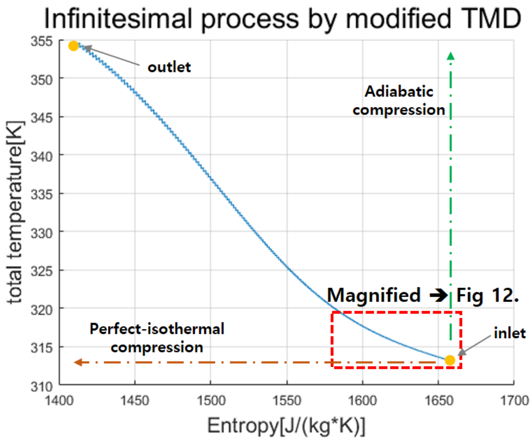

Fig 11. T-S diagram of infinitesimal process

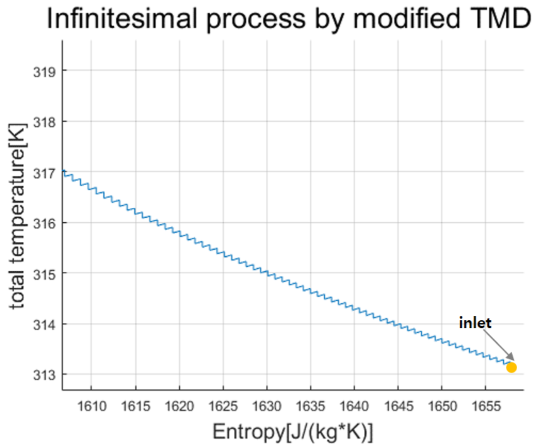

Fig 12. Magnified T-S diagram

Table 1 summarizes design parameters for the infinitesimal compression and cooling impeller model. Figs. 5-12 show the results of this analysis. As mentioned in section 2.2, information along the flow-path could be obtained due to the path-dependent calculation. Since absolute meanline velocity profile is assumed to be unchanged, two profiles are exactly overlapped in Fig. 6. Moreover, the profiles of absolute tangential velocity in Fig. 7 are also very close to each other, which implies they are dynamically similar regardless of cooling. Due to difference in the density along the streamline, one may be able to optimize the shape of passage. In Fig. 5, red dotted lines and blue lines indicate before and after cooling, respectively. It is noteworthy the change of shape in compressor shroud when identical meanline velocity was imposed. In Figs. 8-9, even though the compression work was the same, the increase of pressure and density along the path was observed because of cooling effect. Therefore, higher exit pressure was achieved. As shown in Fig. 10, the analyzed case was not a perfect-isothermal process. If the perfect-isothermal process is realized, further improvement in the pressure ratio and more change in geometry can be expected. In addition, it is guessed that small dip near the outlet is due to overcooling of wider available cooling surface there than the inlet, considering constant cooling flux condition was imposed. The comparison among adiabatic, perfect-isothermal, and the studied case is shown in Fig 11 on a T-S diagram. Conventional adiabatic compression process follows the green vertical line. Perfect-isothermal horizontal compression process follows the brown line. In this study, compressor was moderately cooled in the calculation via surface cooling. Therefore, the compression process is in between adiabatic and isothermal processes. In addition, it is confirmed that the modified code simulates infinitesimal process successfully as shown in Fig 12.

3. Summary and further works

As a preliminary study of isothermal compressor design, KAIST_TMD code was modified and constant cooling flux case was first analyzed while idealizing the problems for simplification. By adopting an infinitesimal approach, compression and cooling were performed simultaneously throughout the flow path. The obtained results illustrated thermodynamic states and velocity information at each step. As for the further works, consideration of loss models is now required to predict more realistic performance of the designed compressor.

4. Acknowledgement

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