Design of 100MWth Class S-CO_2 Power Conversion System for MSR Application

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1. Introduction

As the global demand for energy increases, the need for innovative thermal power systems has grown. High-temperature reactors, which operate at temperatures between 400°C and 750°C, are now being researched in line with this trend. When coupled with high-temperature reactors, the Supercritical Carbon Dioxide (S-CO_2) Power Conversion System is a promising alternative for its effectiveness. S-CO_2 is the state of carbon dioxide where its temperature and pressure are both above critical temperature and pressure. S-CO_2 is particularly effective for energy conversion process.

This study presents an optimized design and comprehensive analysis of a Supercritical Carbon Dioxide (S-CO_2) Power Conversion System (PCS) integrated to a 100 MW thermal Molten Salt Reactor heat source. The main objective is to present the viability and practicality of adopting S-CO_2 PCS due to its suitability for a 100 MWth heat source. By utilizing in-house codes developed at KAIST: the Closed Cycle Design (CCD), Turbomachinery Design (TMD), a comprehensive turbomachinery design study is performed. Six Brayton cycles – Recompression, Modified Recompression, Recuperated, Reheating, Inter-recuperation, and Pre-compression – are meticulously optimized.

2. Methods

The KAIST-CCD Code is an in-house MATLAB code designed to evaluate the steady-state performance of thermodynamic cycles. The analysis is carried out using enthalpy-based calculations and the REFPROP fluid property database from NIST. KAIST-CCD is utilized to predetermine inlet and outlet values for principal components, such as turbomachinery and heat exchangers. The assumed values for CCD in this study are listed in Table 2.

The minimum temperature and pressure of the cycle should be distinct from the critical point (31.1 ˚C, 7.38 MPa). Due to rapid changes in CO_2 compressibility factor, it is difficult to effectively operate and control the compressor near the critical point [1]. Thus, the precooler's outlet temperature is set to 35 ˚C and the pressure is higher than at least 7.5 MPa. As depicted in Figure 1, the compressibility factor is more stable in this state than it is near the critical point.

In addition, Southwest Research Institute (SwRI) proposed the minimum pressure of 20 MPa for the nuclear application with indirect S-CO_2 Brayton cycle [2]. For safety considerations, the highest cycle pressure in this study is fixed at 20 MPa, despite the fact that this reduces the net efficiency by a significant amount.

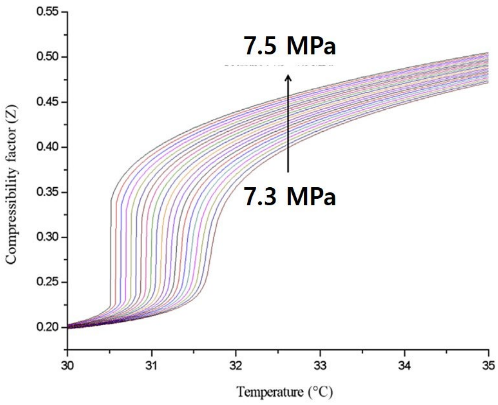

Figure 1. CO_2 Compressibility Factor

Table 1. The S-CO_2 PCS application range for nuclear thermal power (SwRI, 2013)

Application	Nuclear
Cycle Type	Indirect S-CO_2
Motivation	Efficiency, Size, Water Reduction
Size (MWe)	10 - 300
Temperature (˚C)	350 - 700
Pressure (MPa)	20 - 35

Table 2. The assumed values for KAIST-CCD

Turbine Inlet Temp (˚C)	600
Precooler Outlet Temp (˚C)	35
Turbine Efficiency (%)	90
Compressor Efficiency (%)	80
Generator Efficiency (%)	95
HTR Effectiveness (%)	90
LTR Effectiveness (%)	90
HTR Hot Side Pressure Drop (kPa)	60
HTR Cold Side Pressure Drop (kPa)	30
LTR Hot Side Pressure Drop (kPa)	40
LTR Cold Side Pressure Drop (kPa)	20
Precooler Pressure Drop (kPa)	20
Heating Pressure Drop (kPa)	50

On-design performances of axial and radial turbomachinery are calculated by the KAIST-TMD. Calculated designs must match the output data from the

KAIST-CCD, which is part of the design iteration process. The KAIST-TMD code has been developed specifically for analyzing S-CO_2 turbomachinery. Numerous empirical loss models for S-CO_2 have been validated and verified using KAIST's experimental facilities and KAIST-TMD.

3. S-CO_2 Brayton Cycle Layout Analysis

Multiple Shaft Configuration is suitable for power less than 10 MWe according to the Argonne National Laboratory. Single Shaft Configuration (SSC) is suitable for around 10 MWe or more [3]. Based on this, this study optimized the high-efficiency SSC cycle layout. KAIST conducted a quantitative analysis of the efficiency of several typical S-CO_2 system layouts (Fig. 2) [1]. The top six Cycle Efficiency SSC configurations (Recompression, Modified Recompression, Recuperated, Reheating, Inter-recuperation, and Pre-compression) were chosen as candidates for optimization based on Ahn's research. The detailed cycle layouts are shown in Appendix A.

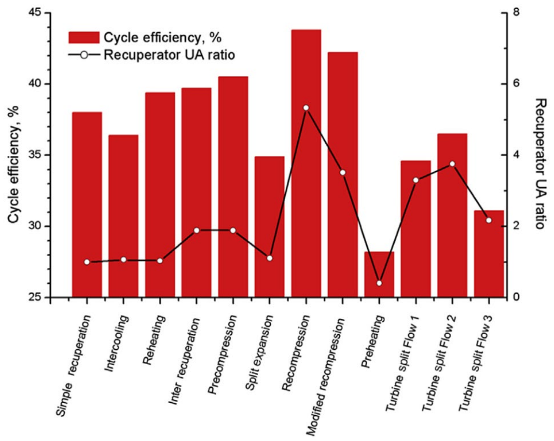

Figure 2. The cycle efficiencies of various layouts

Molten Salt Reactor (MSR) is chosen as the reference system. S-CO_2 PCS is an important topic in the MSR research field **[4]**. SwRI's Supercritical Transformational Electric Power Pilot Plant (STEP) is a demonstration project for 10MWe S-CO_2 PCS, and successful first operation was recently reported **[5]**. In Korea, a research project is ongoing for 100 MWth MSR-powered commercial ships **[6]**. Since the technologies for 10MWe PCS can be mostly shared for 100MWth class PCS, the authors believe that the S-CO_2 PCS system is realizable and practical for Korean MSR-powered ships.

The operating temperature of MSR is 600 ˚C ~ 750 ˚C **[7]**. Considering the intermediate loop, the inlet temperature of the turbine was fixed at 600 ˚C. The optimized points were determined by the point with the highest efficiency. As optimization parameters, the pressure ratio and the flow split ratio are selected. The compressor operates near the critical point, so the low compressor load has a significant operational advantage. Therefore, the design optimization is determined by total compressor load, specific efficiency, net efficiency, mass flow, and layout complexity. Specific efficiency is calculated by dividing net efficiency by mass flow; its unit is %·s/kg. The net efficiency, mass flow, and specific efficiency for each layout are shown in Figures 3, 4, and 5, which are obtained at each optimized point.

In the Modified Recompression layout, the minimum pressure point can exist in either the supercritical or gas regions. Therefore, optimization was accomplished by separating S-CO_2 (supercritical) and T-CO_2 (trans-critical) cycles. Consequently, the minimum pressure of the T-CO_2 layout is below 7.5 MPa.

The Recompression layout has the highest net efficiency. However, the Recompression layout has significantly higher flow rate than other layouts, leading to the lowest specific efficiency.

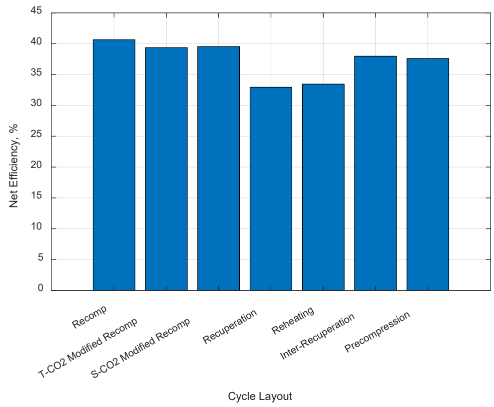

Figure 3. Cycle net efficiency (%)

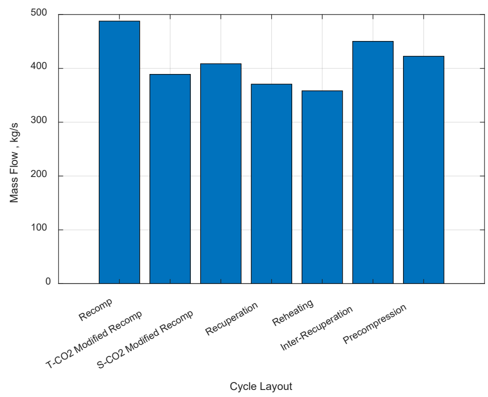

Figure 4. Cycle mass flow (kg/s)

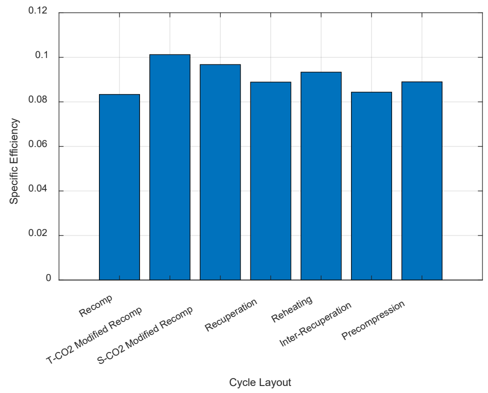

Figure 5. Cycle specific efficiency (%·s/kg)

Compressor load is shown in Figure 6. Due to the compression of the gas phase, the T-CO_2 Modified Recompression has greater compressor load than that of other layouts. Inter-Recuperation layout has a high compressor load due to double compression with no flow split. The compressor load is low in the Recuperated, Reheating, and S-CO_2 Modified Recompression layouts.

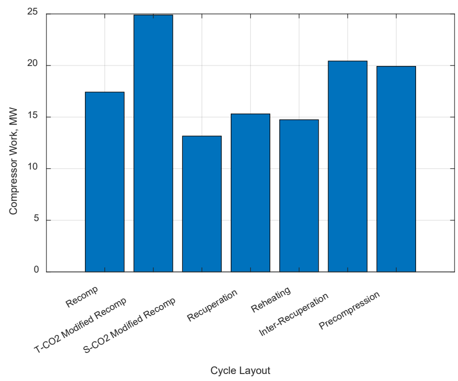

Figure 6. Total compressor work (MW)

Turbine load is shown in Figure 7. T-CO_2 Modified Recompression layout have the highest turbine loads. Turbine load is very low in the Recuperated, Reheating layouts.

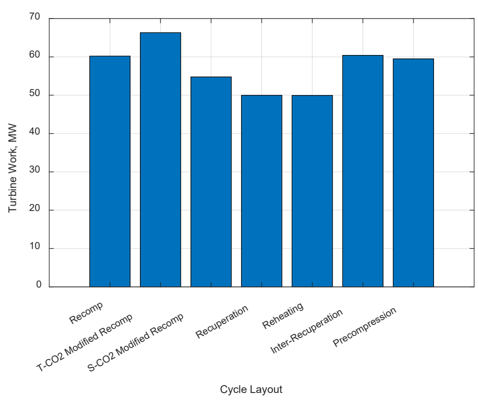

Figure 7. Total Turbine Work (MW)

The KAIST-CCD result is summarized below.

Table 3. The summary of cycle layout optimization

Layout	Pros	Cons
Recompression	Highest efficiency Simple layout	Highest mass flow rate Low specific efficiency
Modified Recompression	T-CO_2 suitable High efficiency	Complex layout High compressor load High turbine load
Recuperated	Moderate specific efficiency Small component load Small mass flow Simple layout	Lowest efficiency
Reheating	Low compressor load Low mass flow Simple layout High specific efficiency	Low efficiency
Inter-Recuperation	High efficiency Simple layout	High compressor load High Mass flow.
Precompression	Low mass flow High efficiency Low compressor load	Complex layout

Both the Modified Recompression layout and the Precompression layout are highly complex than other layouts. The modified recompression layout is less efficient than the recompression layout, but component loads are divided into three compressors and two precoolers with flow split. Despite each component's low load, capital costs and operational difficulties must be carefully considered. Recompression layout has excellent efficiency and a simple layout. However, it was excluded due to its low specific efficiency and highest mass flow. The Inter-Recuperation layout has the greatest compressor load among S-CO_2 cycle layouts except T-CO_2 modified recompression. Simpler layouts are more capital-cost-effective, easier to maintain, and quicker to start up. Complex layouts offer other benefits, like higher efficiency or adaptability. In this study, the authors prioritize simplicity over efficiency, as it directly influences the initial construction costs and the flexibility in operations. Therefore, reheating and recuperated layouts are currently the most feasible and achievable layouts for a 100 MWth heat source. Even though the net efficiency is low, the specific efficiency is practical. Also, mass flow, component load, and layout complexity are moderate.

4. Turbomachinery Analysis

Compressors and turbines are designed utilizing optimization data from the layouts of recuperated and reheating. Based on conservatism, The axial stage pressure ratio was limited to a maximum of 1.2. KAIST-TMD designed turbomachinery to satisfy the reference efficiency, which is shown in Table 4.

Table 4. Designed Efficiency of Turbomachinery Analysis

Turbo Type	Designed Efficiency
Axial Turbine	~ 96 %
Axial Compressor	~ 90 %
Radial Turbine	~ 90 %
Radial Compressor	~ 80 %

With KAIST S-CO_2 facilities, KAIST-TMD was developed and validated with various radial loss models: volute loss, nozzle loss, vaneless loss, incidence loss, passage loss, clearance loss, friction loss and windage loss. Windage loss, volute loss and vaneless loss are not considered in this study. However, the axial loss model only considers profile loss and endwall loss. Thus, axial types' designed efficiency was set higher than radial types'. It should be noted that the KAIST-TMD does not account for elbow loss, which is crucial for multi-stage radial types. Therefore, the actual efficiency of the radial type is quite lower than the calculations, especially under high mass flow. The design of radial turbomachinery is limited to low stages (≤3) only. As the number of stages increases, the radial type becomes structurally more complex compared to the axial type.

For the design of turbomachinery, the flow coefficient and the number of impeller vanes are considered as the primary parameters. For simple comparison, geometrical parameters like aspect ratio, solidity, blade thickness, etc. are fixed in a specific component design.

Maintaining an appropriate tip speed is crucial to ensure the safe and efficient operation of the turbomachinery. If it's too high, it could result in material degradation, frequent maintenance, and unexpected shutdowns. The maximum tip speed is within the safety range (20 % ~ 40%) of the speed of sound. At the turbine outflow and compressor inflow, the tip velocity is the highest. The maximum tip speed is compared to either the turbine outlet or compressor inlet tip speed limits. [5][6] For example, Oh define the safety limits (400 m/s) of maximum tip speed at radial turbines to 20% (100 m/s) of nominal speed (500 m/s). [6] This approach is effective for the low-stage types but unsuitable for the high-stage types. For high-stage types, the aspect ratio distribution along the stages can lead to excessive tip speed at the compressor outlet or turbine inlet. In this study, the minimal speed of sound of either outlet or inlet is the nominal speed, which is used to determine the tip speed limit of both radial and axial turbomachinery. A design with a large safety margin might reduce performance metrics, but it ensures the machinery's reliability and safety. The turbines are operated at 500 to 600 ˚C, while the compressors are operated at 30 to 100 ˚C. Considering the extremely high temperature at the turbine, the safety range for the turbine's maximum tip speed is 50% and the safety range for the compressor is 30%. If a turbomachinery's maximum tip speed exceeds its tip speed limit (50%, 30%), it is considered to be excluded.

Table 5. Minimum speed of sound at turbomachinery

(m/s)	Recuperated	Reheating
Turbine (Main)	420.20 (x 0.5)	430.62 (x 0.5)
Turbine (Sub)		450.97 (x 0.5)
Compressor	198.68 (x 0.7)	198.49 (x 0.7)

(m/s)	Recuperated	Reheating
Turbine (Main)	210.1	215.31
Turbine (Sub)		225.49
Compressor	139.08	138.94

First, in the Recuperated layout, a radial turbine cannot be employed. This is due to the fact that the radial type is unsuitable for large mass flows. The maximum tip speeds of a multiple-stage radial turbine are higher than the tip speed limit. Recuperated radial compressors are unsuitable for the same reasons as is radial turbines. In conditions with a high mass flow rate, radial turbomachinery is therefore challenging. Under high mass flow circumstances, a radial turbine with more than three stages has few advantages over a low-stage axial turbine. The maximum tip speed of the recuperated axial turbine is lower than the tip speed limit. An optimal choice is a six-stage axial turbine with a stage pressure ratio not exceeding 1.2. Also, 10-stage axial compressors meet tip speed limitations.

Second, the Reheating layout has three main components: the main turbine, sub-turbine, and compressor. The main axial turbine can be designed in four stages for the Reheating layout. The Axial sub-turbine is designed with two stages for a moderate stage pressure ratio. Also, a two-stage radial design is possible for the sub-turbine. However, because of the loss at the elbow of radial turbines, the two-stage axial sub-turbine was chosen for this study instead of the small-stage radial sub-turbine. The axial compressor has 10 stages for tip speed safety.

The 100 MWth turbomachinery charts of recuperated and reheating layouts are summarized in Tables 6, 7 and 8. Due to its high maximum tip speed, radial turbomachinery, except reheating sub-turbines, is not recommendable. Multiple-stage reheating radial sub-turbines can be adopted, but they are not as efficient as small-stage axial turbines. Therefore, radial-type turbomachinery is not suggested for 100 MWth S-CO_2 PCS. However, this result is dependent on how the safety margin of the turbomachinery's maximum tip speed is designed.

Table 6. The summary of recuperated cycle turbomachinery

Recuperated	Turbine	Compressor
Radial (≤3)	High Tip Speed	High Tip Speed
Axial	6 stages	10 stages

Table 7. The summary of reheating cycle turbomachinery

Recuperated	Main Turbine	Sub Turbine	Compressor
Radial (≤3)	High Tip Speed	Uncompetitive	High Tip Speed
Axial	4 stages	2 stages	10 stages

Table 8. The optimized turbomachinery of S-CO_2 PCS coupled with 100 MWth

Layout	Recuperated	Reheating
Main Turbine	Axial 6 stage	Axial 4 stage
Sub Turbine		Axial 2 stage
Compressor	Axial 10 stage	Axial 10 stage

5. Summary and Conclusions

From the design study, the Recuperated and the Reheating cycles are identified as the most applicable layouts for 100MWth MSR application at this point. In addition, a comprehensive comparison between radial and axial turbomachinery configurations is conducted. At a high mass flow rate (~ 400 kg/s), radial turbomachinery with a small number of stages (≤3) is challenging due to high tip speed and high blade loads. A moderate or higher stage is needed to endure the high pressure ratio between the turbomachinery inlet and outlet. Consequently, the authors plan to demonstrate the viability of S-CO_2 PCS for the optimized cycle layouts. Future study will include intermediate heat exchanger with MSR and S-CO2 PCS heat exchangers (precooler, recuperator).

ACKNOWLEDGEMENT

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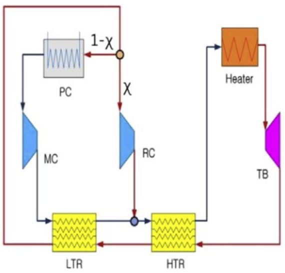
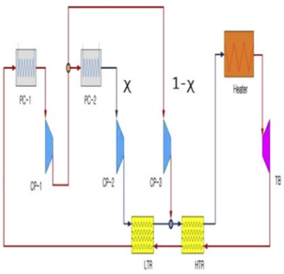
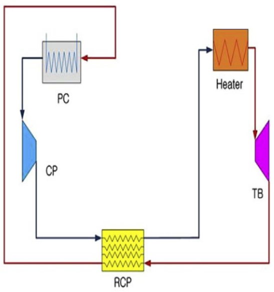
Appendix A : Cycle Layout



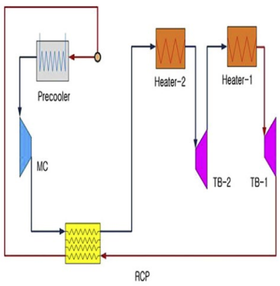
Recompression



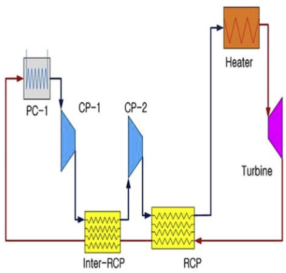
Modified recompression



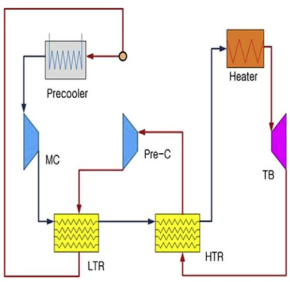
Recuperation



Reheating



Interrecuperation



Precompression