Thermodynamic study of SCO2 Recompression Brayton Cycle with Intercooling and Reheating for Light Water Reactor

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1. Introduction

The supercritical CO2(SCO2) Brayton cycle has been considered to replace the steam Rankine cycle for next-generation energy conversion systems. Dostal et al [1] reported the higher efficiency, simpler layout, and compact size of SCO2 Brayton cycle compared to steam cycles. Especially, the advantage of thermodynamic efficiency was maximized around the heat source temperature of 350~550°C.

Not only for next-generation power systems, but the water-cooled reactor of which cycle maximum temperature typically limited under 330°C can be modified by using SCO2 cycle. Yoon et al [2] conducted the thermodynamic analysis for the SCO2 recompressing cycle coupled SMART (System-integrated Modular Advanced ReacTor) and reported that the competitiveness of system was improved with maintaining thermodynamic efficiency of 30.5%.

The concept of reheating is to increase gross turbine power by heating the turbine outlet fluid and then running the turbine again. Meanwhile, intercooling strategy improves net power by cooling the fluid passing through the compression stage to reduce compression work. Although the previously mentioned system of Yoon et al's study did not include intercooling and reheating, the reported efficiency was reasonable compared to the current steam cycle. Therefore, the cycle efficiency could be improved by using intercooling and reheating.

The objective of this study is to investigate intercooling and reheating effects on the SCO2 recompression Brayton cycle for light water reactor in terms of the cycle efficiency. We analyzed the SCO2 cycle including intercooling and reheating and derived an optimized efficiency.

2. Method

2.1 Cycle component analysis methodology

In this study, the cycle consisted of compressors, turbines, and heat exchangers. The performance of compressors and turbines was represented by isentropic efficiency and effectiveness represented the performance of the heat exchangers. The following assumptions were made for the analysis:

- Pressure drop and heat loss terms in all flow paths and heat exchangers are negligible.
- Each compressor has the same isentropic efficiency.
- Each turbine has the same isentropic efficiency.
- All heat exchangers have the maximum effectiveness regardless of the inlet conditions.

The isentropic efficiency of each component was assumed as 0.89 for compressors and 0.90 for turbine based on the study of Dostal et al [1]. The maximum effectiveness of each heat exchanger assumed as 0.90 to avoid pinch point issue.

2.2 Reference water-cooled reactor: SMART

We selected SMART as reference water-cooled reactor. SMART is a small-sized integral type PWR with a rated thermal power of 330 MWt, which is one of the advanced SMRs. Detailed operating conditions are summarized in **Table 1**. Based on these operating conditions, the cycle maximum/minimum temperature was assumed at 310°C and 32°C, and the system maximum pressure was fixed at 25 MPa of which value was selected by previous studies [1-3].

Table 1 Operating conditions of SMART.

Core thermal power	330 MWt
Electric power output	100 MWe
Thermodynamic cycle efficiency	30.3%
Thermodynamic cycle type	Rankine
Core coolant outlet temperature	323°C
Core coolant inlet temperature	295.7°C
HP turbine inlet pressure	5.2 MPa(a)
HP turbine inlet Temperature	296.4°C

2.3 Cycle layout and cycle analysis methodology

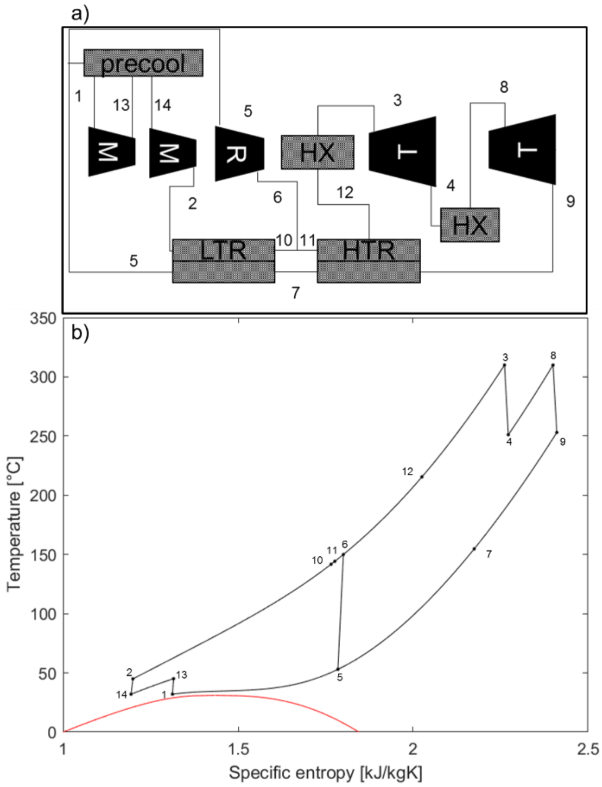

Fig. 1. a) Cycle layout and b) T-s diagaram of SCO2 recompression cycle with MCIC and reheating.

Fig.1 shows the overall layout and the T-s diagram of the SCO2 recompression cycle with the main compressor intercooling and reheating. In this cycle, the working fluid (SCO2) flows to the high-pressure (HP) turbine (points 3-4) and reheated by the heat exchanger (points 4-8), and then goes through the low-pressure (LP) turbine (points 8-9). Due to the high turbine outlet temperature (point 9), SCO2 after the LP turbine passes through the high-temperature recuperator (HTR) and the low-temperature recuperator (LTR) to recover the heat. Some of the SCO2 is compressed by the recompressor (points 5-6) and the rest goes through the precooler to discard the waste heat (points 5-1). Cooled SCO2 flows the progress: compression (points 1-13), intercooling (points 13-14), compression (point 14-2), being heated by the LTR (points 2-10), and then being mixed to the recompressed SCO2 (points 6-10 to 11). The merged SCO2 passes through the HTR (points 11-12) and finally be heated by the main heat exchanger (points 12-3).

To calculate the cycle efficiency, the HTR outlet temperature of the lower pressure side (point 7) was initially assumed as the average value of the LP turbine outlet (point 9) and the compressor outlet temperature (point 2). By using the assumed temperature (point 7), all other points can be calculated including point 7. In each calculation, the properties of SCO2 were calculated through NIST (National Institute of Standards and Technology)'s REFPROP (Version 9.0) program. If the difference between the assumed temperature and the calculated temperature of point 7 was over 0.5K, the temperature of point 7 was newly assumed as the mean value of the previously assumed temperature and the calculated temperature. **Fig. 2** shows the flow chart for analysis the cycle efficiency.

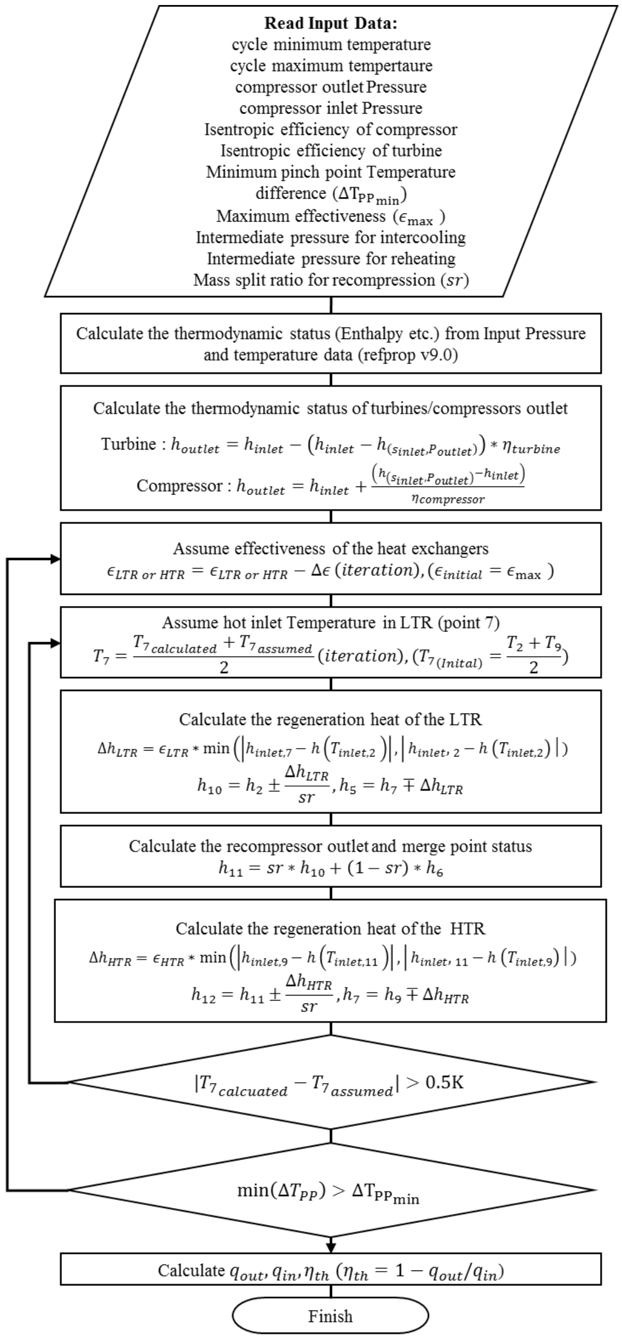

Fig. 2. Flow chart to analysis the cycle efficiency.

If the assumed effectiveness of the heat exchanger is excessively high, a pinch point problem occurs at which the temperature is locally reversed inside the heat exchanger. In order to avoid this problem, we calculated the minimum temperature difference from the temperature profile of heat exchanger. If the calculated

Table 2 Verification of cycle analysis code.

Operating Conditions (Input of the present code)					Cycle efficiency (%)		Error
T_{min}	T_{max}	P_{max}	Split ratio	P_{max} / P_{min}	Reference data [4]	Present code	
32°C	550°C	20MPa	0.666	2.64	41.18	41.92	1.79%
32°C	550°C	30MPa	0.645	3.86	43.32	42.41	2.09%
50°C	550°C	20MPa	0.816	2.40	36.71	37.10	1.07%
50°C	550°C	30MPa	0.746	2.80	38.93	39.81	0.65%

pinch point temperature difference (ΔT_{PP}) is less than the set minimum temperature difference ($\Delta T_{PP_{\min}}$), the effectiveness of the heat exchanger was reduced by $\Delta\varepsilon$ and recalculation was performed. In this study, $\Delta T_{PP_{\min}}$ was assumed as 0K to investigate the maximum improvement of cycle efficiency by intercooling and reheating.

Numerical analysis of cycle efficiency was conducted by MATLAB code. To verify this numerical analysis code, a comparison was performed with the cycle efficiency analysis cases reported by Sarkar and Bhattacharyya [4]. In their study, isentropic efficiency of compressors and turbines was assumed as 85% and 90% respectively. Because mean effectiveness of HTR and LTR was 85%, each effectiveness of the present code was assumed as 85%. Cycle maximum pressure, cycle maximum temperature, and cycle minimum temperature was selected as the operating conditions for the case study but cycle minimum pressure, mass split ratio was the optimization results of each case. The calculated result of present code closely matched the previous data with error of less than 2.1%. This error might be because different property correlations were used, and averaged effectiveness was applied to each recuperator.

2.4 Optimization and Thermodynamic analysis methodology

Since the cycle maximum/minimum temperature is fixed, the cycle efficiency depends on the pressure ratio, the mass split ratio, and the intermediate pressure for intercooling and reheating. To investigate the independent effects of intercooling and reheating, we divided into three cases: 1) both reheating and intercooling applied cycle, 2) only intercooling applied cycle, and 3) only reheating applied cycle.

The optimization of the compressor inlet pressure and the mass split ratio with no intercooling and reheating was conducted to establish the reference efficiency. From the optimization data, the intermediate pressure is optimized by the iteration method. At the obtained intermediate pressure for intercooling and reheating, the results would be verified by re-optimizing the pressure ratio and the mass split ratio.

Cycle performance could be represented by efficiency and net power. Due to the fixed heat power of the reactor, the cycle efficiency is directly related to the electric output power. Based on the net power of the reference cycle, the net power of each cycle was normalized to compare.

3. Results and discussions

3.1 The optimization result

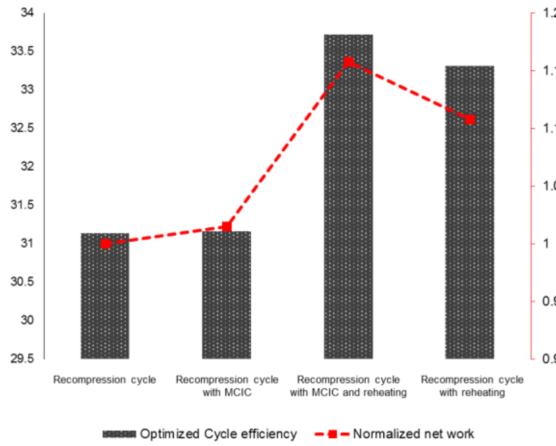

Fig. 3. Cycle efficiency optimization results for the each cycle.

Fig. 3 shows the optimization results. The optimized efficiency of the reference cycle was calculated to 31.13% at the compression ratio of 2.7096 (Compressor inlet pressure: 9.23MPa) and the mass split ratio of 0.7474. The optimization data of the recompression cycle with the main compressor intercooling showed the increased net power of 1.5% but the similar cycle efficiency (31.16%) of the reference with the same mass split ratio. The optimized pressure ratio of the intercooling applied cycle was 2.8135 (Main compressor inlet pressure: 8.89MPa).

On the other hand, in the case of the cycle both intercooling and reheating applied, the maximum efficiency (33.72%) was obtained when the intermediate pressure for the intercooling and reheating was 10.057MPa and 15.475MPa, respectively. The mass split ratio and pressure ratio between LP turbine inlet and main compressor inlet was optimized as 0.6632 and 3.125 (Main compressor inlet pressure: 8.00MPa), respectively.

In the case of the cycle in which only reheating was applied, the maximum efficiency (33.31%) was optimized at the intermediate pressure of 15.657MPa. The optimized mass split ratio and pressure ratio was 0.7053 and 2.8654 (Compressor inlet pressure:8.73MPa)

4. Conclusion

In this study, thermodynamic analysis and optimization of SCO2 recompression cycle for water-cooled reactor was conducted to investigate the effect of intercooling and reheating on the cycle efficiency. The cycle maximum/minimum temperature and the maximum pressure was fixed based on the operating condition of SMART, which is water cooled small modular reactor. In these conditions, the SCO2 cycle efficiency was maximized by using reheating or intercooling strategy.

What was found in this study can be summarized as follows:

- Both the cycle efficiency and the net power were improved at the reheating cycle (with/without intercooling). Reheating, like the general Brayton cycle, could be effective strategy to improve cycle efficiency.
- Cycle efficiency of SCO2 cycle with only intercooling was hardly improved by 0.03% compared to the reference cycle. Even in the case of SCO2 cycle with reheating, the efficiency improvement from intercooling was only 0.4%. Intercooling itself is not an efficient strategy due to the relatively low compression power of SCO2 cycle.
- On the other hand, the net power of both SCO2 intercooling cycle (with/without reheating) increased by 5% and 1.5%, respectively. Relatively high net power could reduce the system scale.
- The largest pressure ratio was recorded at the cycle in which both intercooling and reheating were applied. The low minimum pressure of the cycle has the advantage of lowering the required material conditions to constitute the system.

In this study, the pressure drop of heat exchangers was neglected despite its effect of reducing efficiency. A low ΔT_{PP} could increase the pressure loss due to the expanded scale of heat exchangers. Since the insignificant increase of cycle efficiency was found in this study, despite the lowest ΔT_{PP} of 0K, intercooling and reheating strategy might be meaningless when considering the pressure loss. Therefore, it is necessary to analyze the pressure drop terms of the systems in future studies.

Acknowledgments

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