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Preliminary study of S-CO_2 cycle control logic for part load operation

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1. Introduction

As a part of the Sodium-cooled Fast Reactor development in Korea, the supercritical CO_2 Brayton cycle has been investigated as an alternative power conversion system to the steam Rankine cycle. The benefits of S-CO_2 cycle are relatively high efficiency under the mild turbine inlet temperature region, simple layout configuration and small foot-print. In addition, the safety of the SFR system can be inherently enhanced as the violent sodium-water reaction can be substituted with the mild sodium-CO_2 reaction.

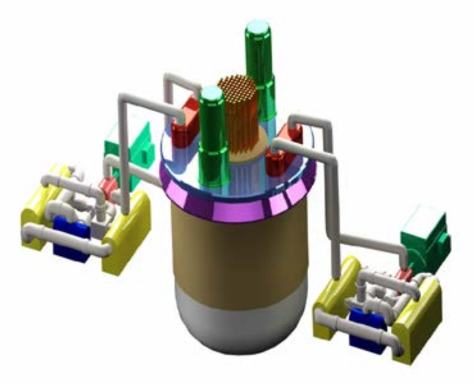

Fig. 1. S-CO_2 power conversion system for SFR application

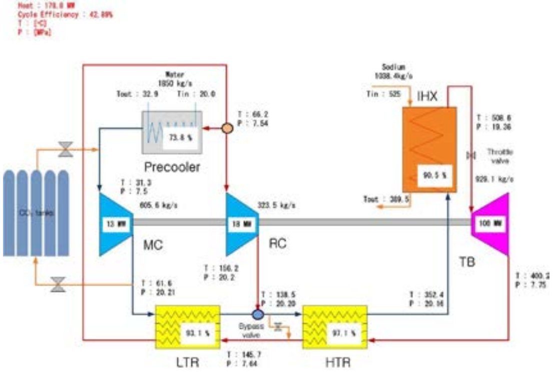

Fig. 2. S-CO_2 cycle condition and component performance for full power operation

The reference SFR system is composed of two 75MWe S-CO_2 power conversion systems as shown in Fig. 1. To assess the cycle performance, the corresponding heat exchanger and turboamachinery performances must be defined. Therefore, radial type turbomachineries and Printed Circuit Heat Exchanger (PCHE) for the SFR system are designed by using the in-house design code [1]. The optimized cycle conditions are shown in Fig. 2.

2. S-CO_2 cycle control analysis for part load operation

The off-design performance of S-CO_2 turbomachineries are utilized for the quasi-static system analysis code evaluation. Based on the design parameters of S-CO_2 turbomachinery, the off-design performance of S-CO_2 compressor and turbine is estimated as shown in Fig. 3. To utilize the off-design performance map for the system analysis, the equivalent mass flow indicates the off-design performance for various operating conditions. The equivalent mass flow ($\dot{m}$) is defined as below.

$$\dot{m}_{eq} = \dot{m}\frac{\sqrt{\theta}}{\delta}\varepsilon \tag{1}$$

$$\theta = \left[\frac{\frac{\gamma_a}{\gamma_a+1}T_a}{\frac{\gamma_{ref}}{\gamma_{ref}+1}T_{ref}}\right]^2 \tag{2}$$

$$\delta = \frac{P_a}{P_{ref}} \tag{3}$$

$$\varepsilon = \frac{\gamma_{ref}\left(\frac{2}{\gamma_{ref}+1}\right)^{\frac{\gamma_{ref}}{\gamma_{ref}-1}}}{\gamma_a\left(\frac{2}{\gamma_a+1}\right)^{\frac{\gamma_a}{\gamma_a-1}}} \tag{4}$$

where $\dot{m}$, γ , P, T are mass flow rate, specific heat ratio, pressure and temperature, respectively.

To analyze the heat exchanger performance, the heat transfer and pressure drop correlations as shown in Table I are adopted in the quasi-static cycle analysis code [2-3]. The preliminary S-CO_2 heat exchanger design parameters are shown in Table II.

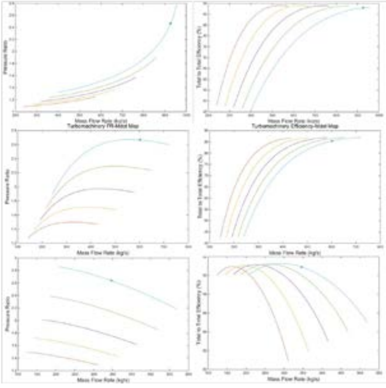

Fig. 3. S-CO_2 turbomachinery performance (turbine, main compressor and recompression compressor)

Fluid	Na	CO_2	Water
Nusselt number	$Nu = 7 + 0.025Pe^{0.8}$	$Nu = 0.1696Re^{0.629}Pr^{0.317}$	
Friction factor	$f = 4(0.0014 + 0.125Re^{-0.32})$	$f = 0.1924Re^{-0.091}$	
N_{Re} range	No limitation	2,500 - 33,000	
Reference	Hejzlar et al., 2007	Ngo et al., 2007	

Table I: Heat transfer and friction factor correlation in PCHE [2, 3]

Heat exchanger	HTR	LTR	PC	IHX
Heat, MW	108.2	268.7	102.1	178.8
Effectiveness, %	97.1	93.1	73.8	90.5
Volume, m^3	6.9	8.5	3.07	2.3
Geometry, m (L x W x H)	1.1x2.5x2.5	1.4x2.5x2.5	1x1.75x1.75	0.6x2.1x2.1

Table II: Heat exchanger design of S-CO_2 cycle

To establish the control logic for the part-load condition, in-house quasi-static cycle analysis code was developed by the KAIST research team. This code requires preliminary component design results. The heat exchanger and turbomachinery geometries are used from the component design results. The code algorithm is shown in Fig. 4.

Based on the component geometry, part load performance was assessed by the quai-static cycle analysis code. For various heat load conditions, three strategies: inventory control, bypass valve control, turbine throttle valve control were compared. The results are shown in Fig. 5.

The turbine work is gradually decreased compared to the compressor works under the part load condition as shown in Fig. 6. The compressor surge is the condition at which the compressor is not capable of providing enough energy to overcome the system resistance or backpressure. Therefore, in the system operation, the compressor surge condition is seriously considered and controlled to avoid the system damage. As shown in Fig. 6, the main compressor surge margin rapidly decreases under the part load operating conditions, especially lower than 50% load condition while the density of part load operating condition changes abruptly. Depending on the system designer, the minimum surge margin varies but 50% of surge margin is usually considered. In this manner, the valve control must be applied during the low load operation (<50%).

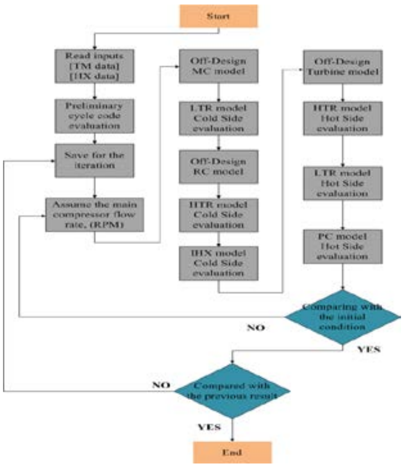

Fig. 4. S-CO_2 cycle quasi-static analysis code algorithm

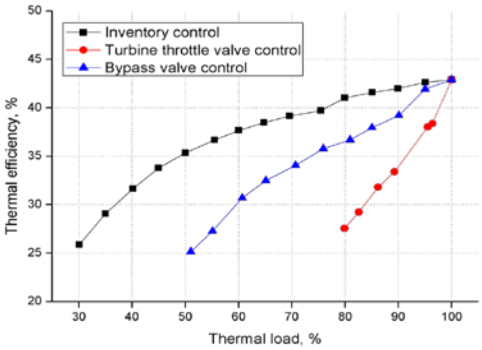

Fig. 5. Comparsion of S-CO_2 cycle performance in part load conditions

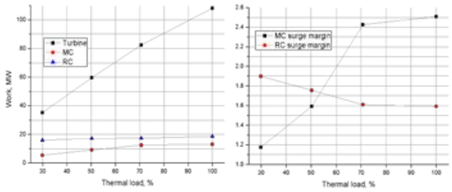

Fig. 6. Comparison of turbomachinery works and compressor surge margin in part load conditions

3. Conclusions

75MWe S-CO_2 recompression cycle with radial type turbomachineries and PCHE was designed. Under various part load conditions (30-100% thermal load), off-design performance of the designed system was assessed, and different control logics were first tested. It was identified that the inventory control strategy is the most efficient logic for the part load operation. However, in low (<50%) part load condition, the surge margin decreases abruptly and relevant valve control is required to maintain the compressor stability.

REFERENCES

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