Heat Balance of Supercritical Carbon Dioxide Power Cycles for Thermal Energy Storage System

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1. Introduction

Globally, the environmental pollution such as global warming and fine dust have led to energy conversion to new renewable energy. As the proportion of the new renewable energy increases, the intermittency and volatility of power generation increases, and in order to flexibly respond to such fluctuations, the necessity for the large-capacity energy storage capability of storing and supplying power are increasing. There are various options for large-capacity energy storage devices such as pumped water power generation, compressed air energy storage systems, liquid air energy storage systems, lithium-ion batteries, and thermal energy storage systems. Among these options, the thermal energy storage systems are considered a promising alternative due to their strength such as relatively few installation restrictions, eco-friendly, long-them energy storage, long life, and economical efficiency. In particular, the thermal energy storage systems can be used not only as a power generation source, but also as a heat supply source for industry or heating, and when using heat directly, there is an advantage that the roundtrip efficiency of the system becomes very high because there is no energy conversion loss.

Thermal energy storage systems are usually classified as sensible heat storage, latent heat phase-change materials, and thermochemical storage. Among them, the two tank thermal energy storage system using alkali metal as a storage material is considered an economical storage technology that can be commercialized. In particular, the sodium as a working fluid has a wide operating temperature range, so it is highly usable. Also, the heat transfer coefficient is very large, therefore, the size of the heat exchange device can be minimized. In addition, there is an advantage of increasing the energy storage density by operating a large temperature difference between the hot tank and cold tank.

The supercritical CO2 brayton cycle can be applied as a power cycle that converts stored thermal energy into electrical energy. The supercritical CO2 brayton cycle offers a more efficient, significantly simpler and more compact alternative to the superheated steam cycle.[1] There are many options for the supercritical CO2 Brayton cycle, including a simple recuperated cycle, a recompression cycle, and a cascade cycle. In order to increase the energy storage density of the thermal energy storage system, the temperature
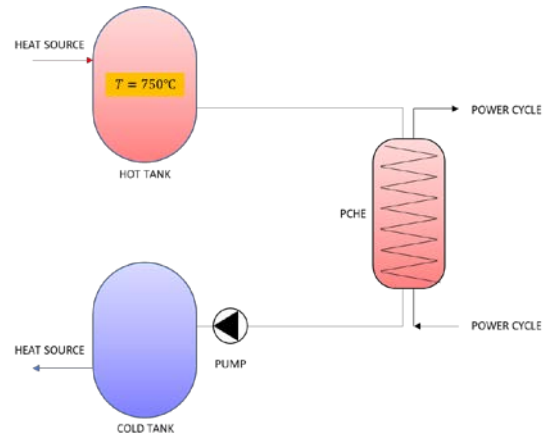

Fig. 1. The schematic diagram of two tank thermal energy storage system.

difference between the hot and cold tanks should be large. It is very challenging to several cycle options such as the recompression cycle, since the operating temperature of heater of the recompression cycle is limited by the outlet temperature of the recuperators.

In this study, the efficiency of the cascade cycle and the partial heating cycle was compared to find an option of a supercritical CO2 brayton cycle suitable for application to a large temperature range of a thermal energy storage device.

2. Methods and Results

2.1 Cycle Design Conditions and Constraints

A schematic diagram of a two tank thermal energy storage system is shown in Figure 1. The heat storage capacity of the thermal energy storage system is 1 GWht and the rated output is 100 MW_{th}, and the capacity is set to supply energy for 10 hours during rated power operation. The temperature of the hot tank was set to 750℃ by maximizing the wide operating temperature range of sodium.

The design constraints on the supercritical CO2 brayton cycle are summarized in Table 1. The compressor inlet condition was assumed to be cooled to the critical point of 32℃ and 7.6 MPa in the cooler. The operating range of the pressure at the outlet of the compressor of supercritical CO2 is generally 20 to 30 MPa, so the maximum pressure was set to 30 MPa or less. According to the preceding study [2], if the effectiveness of the heat exchanger is more than 95%,

the cost of the heat exchanger increases rapidly, so it was limited to less than 95%. Compressor and turbine efficiencies were assumed to be 88% and 92%, respectively, as typical efficiencies of commercial products. In thermal equilibrium, the effect of pressure change in other equipment other than the turbine and compressor was neglected.

Table I: Design constraints for supercritical CO2 brayton cycles

Compressor inlet condition	32 ℃, 7.6 MPa
Maximum pressure	30 MPa
Heat exchanger effectiveness	< 95%
Compressor efficiency	88%
Turbine efficiency	92%

2.2 Cycle Modeling

Cycle modeling is performed by mathematical models of primary model elements for design points. The primary model elements of the supercritical CO2 brayton cycle consist of compressors, recuperator, heating/cooling heat exchangers, turbines, and mixing/splitting junctions, and the cycle layout is composed of different numbers and arrangements of these elements. The mathematical model for each element is as follows.

- Compressor

$$\eta_{isen,comp} = \frac{h_{in} - h_{out,s}}{h_{in} - h_{out}} \qquad (1)$$

- Turbine

$$\eta_{isen,turb} = \frac{h_{in} - h_{out}}{h_{in} - h_{out,s}} \qquad (2)$$

- Heat exchangers and Recuperators

$$\dot{Q}_{hot} = \dot{m}_{hot} \left(h_{hot,in} - h_{hot,out} \right) \qquad (3)$$

$$\dot{Q}_{cold} = \dot{m}_{cold} \left(h_{cold,out} - h_{cold,in} \right) \qquad (4)$$

$$\dot{Q}_{hot} = \dot{Q}_{cold} \qquad (5)$$

$$\varepsilon = \frac{\dot{Q}_{actual}}{\dot{Q}_{max}} \qquad (6)$$

$$\begin{cases} \dot{Q}_{actual} = \dot{Q}_{hot} \text{ or } \dot{Q}_{cold}, \\ \text{For recuperators,} \\ \dot{Q}_{hot,max} = \dot{m}_{hot} \left[h\left(T_{hot,in}, P_{hot,in}\right) - h\left(T_{cold,in}, P_{hot,out}\right) \right] \\ \dot{Q}_{cold,max} = \dot{m}_{cold} \left[h\left(T_{hot,in}, P_{cold,in}\right) - h\left(T_{cold,in}, P_{cold,out}\right) \right] \\ \dot{Q}_{max} = \min\left(\dot{Q}_{hot,max}, \dot{Q}_{cold,max}\right) \end{cases}$$

$$\begin{cases} \text{For heat exchangers,} \\ C_{p,hot} = \dot{Q}_{hot} / \left(T_{hot,in} - T_{hot,out}\right) \\ C_{p,cold} = \dot{Q}_{cold} / \left(T_{cold,out} - T_{cold,in}\right) \\ C_{p,min} = \min\left(C_{p,hot}, C_{p,cold}\right) \\ \dot{Q}_{max} = C_{p,min} \left(T_{hot,in} - T_{cold,in}\right) \end{cases}$$

- Mixing junction

$$\dot{m}_{in,1} + \dot{m}_{in,2} = \dot{m}_{out} \qquad (7)$$

$$\dot{m}_{in,1} h_{in,1} + \dot{m}_{in,2} h_{in,2} = \dot{m}_{out} h_{out} \qquad (8)$$

- Splitting junction

$$\dot{m}_{in} = \dot{m}_{out,1} + \dot{m}_{out,2} = x\dot{m}_{out} + (1-x)\dot{m}_{out} \qquad (9)$$

$$h_{in} = h_{out,1} = h_{out,2} \qquad (10)$$

In compressor and turbine models, η, h_{in}, h_{out}, $h_{out,s}$ mean isentropic efficiency, inlet enthalpy, outlet enthalpy and ideal outlet enthalpy, respectively. Here, ideal outlet enthalpy means enthalpy during isentropic compression and expansion. In the heat exchanger model and recuperator model, $\dot{Q}_{hot}$, $\dot{Q}_{cold}$, $\dot{m}_{hot}$, $\dot{m}_{cold}$, $h_{hot,in}$, $h_{hot,out}$, $h_{cold,in}$, $h_{cold,out}$, ε are the heat transfer rate, mass flow rate, inlet and outlet enthalpy, and effectiveness in the hot and cold areas, respectively. At the mixing junction and splitting junction, $\dot{m}_{in}$, $\dot{m}_{out}$, h_{in}, h_{out} are the mass flow rates and enthalpy at the inlet and outlet, respectively, and x is the flow split ratio. In the above equation, compressor and turbine have one unknown as h_{out} in each equation. In the heat exchanger, there are 4 unknowns as $\dot{Q}_{hot}$, $\dot{Q}_{cold}$, $h_{hot,out}$, $h_{cold,out}$ and 4 equations. In the mixing junction and splitting junction, there are two unknowns, and two equations. Therefore, since the number of equations and unknowns coincide in all models, each unknown can be derived by calculating the solution of the system of equations when various layouts are configured. The CO2 properties were calculated using a lookup table which was created by NIST's RefProp V9.0[3].

2.3 Cascade Cycle

Cascade cycle is designed for waste heat recovery to increase the amount of recovered wasted heat, the cycle has larger temperature difference in heaters. [4] Figure 2 shows the heat balance of a cascade brayton cycle using a compressor, recuperator, and two turbines. The Cascade cycle divides the flow rate discharged from the compressor and supplies some to the high-temperature turbine through an air heat exchanger and some to the low-temperature turbine through a regenerative heat exchanger. Figure 3 shows the T-s diagram of the cascade cycle. In the case of the simple regeneration brayton cycle or the recompression cycle, since the fluid that has passed through the recuperators flows into the
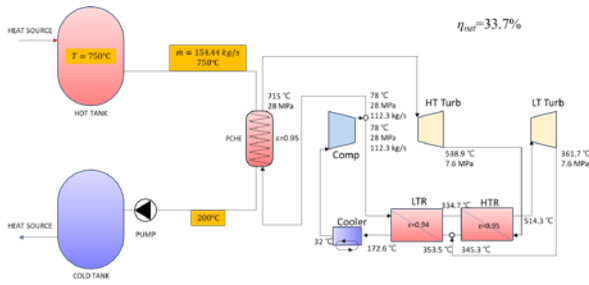
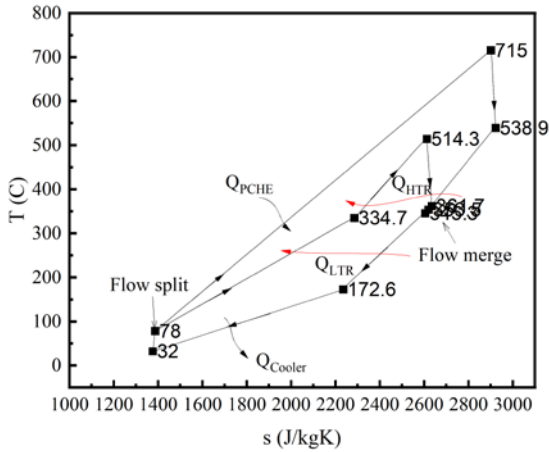Fig. 2. Heat balance of Cascade cycle for thermal energy storage system.

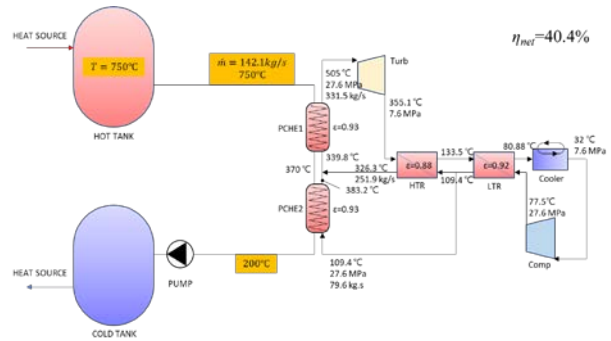Fig. 3. T-s diagram of Cascade cycle for thermal energy storage system.

Fig. 4. Heat balance of Partial heating cycle for thermal energy storage system.
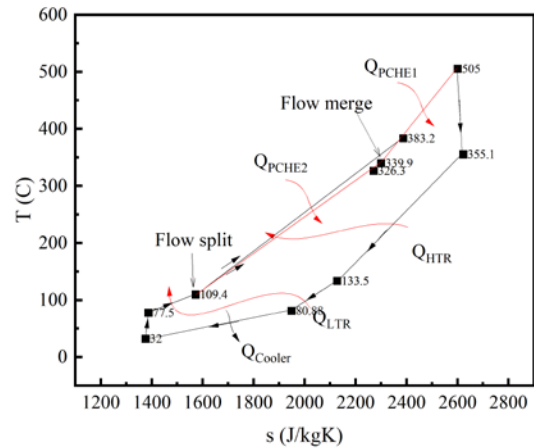
Fig. 5. T-s diagram of Partial heating cycle for thermal energy storage system.

heating heat exchanger, the discharge temperature of the high temperature side fluid of the heating heat exchanger is limited to the discharge temperature of the recuperators. On the other hand, the cascade cycle can cover a wide temperature range because the relatively low temperature fluid discharged from the compressor flows into the heating heat exchanger. When the heat storage system is configured with cascade, the stored heat energy can be utilized from 750°C to 200°C, and the net efficiency was derived as about 33.7%.

2.4 Partial Heating Cycle

Figure 4 shows the thermal equilibrium of a partial heated brayton cycle using a compressor and turbine and two recuperators and two heating heat exchangers. In the partial heating brayton cycle, a relatively low temperature fluid discharged from the low temperature recuperator is branched and a part is supplied to the heating heat exchanger 2 and a part is supplied to the high temperature recuperator. Similar to the cascade cycle, relatively low temperature fluid is supplied to the heating heat exchanger, so it is possible to cope with large temperature differences. Even in the case of the partial heating cycle, the stored heat energy can be utilized from 750°C to 200°C, and the net efficiency of this case was derived to be about 40.4%.

3. Conclusions

In this study, the efficiency of the cascade cycle and the partial heating cycle was compared to find an option of a supercritical CO_2 brayton cycle suitable for application to a large temperature range of a thermal energy storage system. As a result of the thermal equilibrium calculation, the efficiency of the partial heating cycle was 40.4%, which was higher than the efficiency of the cascade cycle, 33.7%. In addition, the cascade cycle was composed of a high-temperature turbine and a low-temperature turbine, so it is expected to be more difficult in terms of control than a partial heating cycle in which only one turbine is applied. Therefore, in order to cope with the large temperature range of the thermal energy storage system, it is concluded that the partial heating cycle is more suitable than the cascade cycle.

ACKNOWLEDGEMENT

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