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Development of Dynamic Model of Supercritical Carbon Dioxide Power Cycle with 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 CO_2 brayton cycle can be applied as a power cycle that converts stored thermal energy into electrical energy. The supercritical CO_2 brayton cycle offers a more efficient, significantly simpler and more compact alternative to the superheated steam cycle [1]. In a previous study, we compared the heat balance and efficiency of the cascade cycle and the partial heating cycle to find the suitable sCO_2 brayton cycle for the large temperature difference between the hot and cold tank of the thermal energy storage system. Finally, a

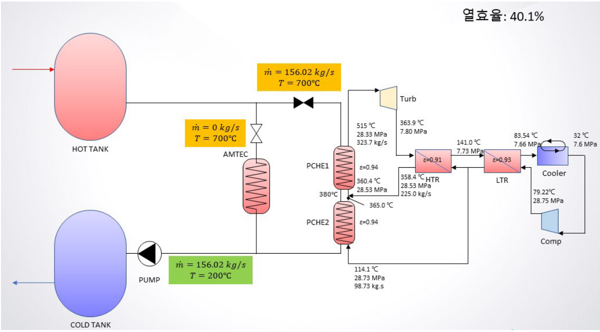

Fig. 1. Heat balance of Partial heating cycle for thermal energy storage system.

partial heating cycle with higher efficiency than the cascade cycle was selected [2].

The thermal energy storage system with the power conversion system aims to operate according to the electric grid demand to reduce the intermittent and volatility of the new renewable energy. The target ramp rate performance of this thermal energy storage system is 5%/min. In order to design control systems for such a load following operation, a dynamic model that can analyze the dynamic characteristics of thermal energy storage and power conversion system is required. In this study, a dynamic model for the selected partial heating cycle with thermal energy storage system was developed. The accuracy of the dynamic model was confirmed by performing steady state calculation and comparing with the heat balance.

2. Methods and Results

2.1 Heat Balance of Partial Heating Cycle

Fig. 1 shows the heat balance of a partial heating brayton cycle using one compressor and turbine, two recuperators, and two heating heat exchangers. The heat storage capacity of the thermal energy storage system is 1 GWht and the rated output is 100 MWth, and the capacity is set to supply energy for 10 hours during rated power operation. The temperature of the hot tank was set to 700℃ by maximizing the wide operating temperature range of sodium. Compressor and turbine efficiencies were assumed to be 88% and 92%, respectively, as typical efficiencies of commercial products. In the heat balance, the effect of pressure change in other equipment other than the turbine and compressor was neglected.

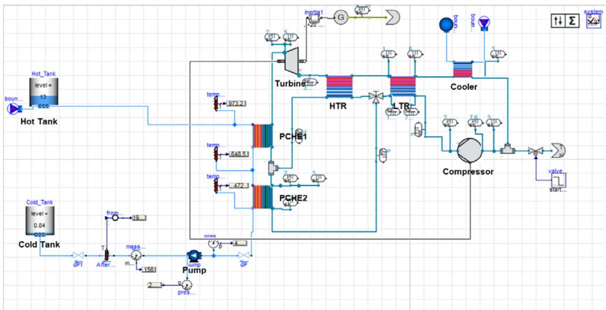

Fig. 2. Integrated analysis model of the thermal energy storage system and partial heating sCO_2 brayton cycle.

2.2 Dynamic Model

Using Dymola software, a dynamic model of the thermal energy storage and utilization system was built. Dymola is a simulation software based on the open Modelica language developed by Dassault systems [3]. For carbon dioxide properties and turbo machinery, the model provided by ClaRaPlus Library v1.3.0 was used [4, 5]. For sodium properties, the properties provided by SolarTherm Library v0.2, which are basically provided by Modelica, were used. The heat exchangers of PCHE, recuperator, and cooler were manually modeled by connecting pipes and heat transfer walls [6, 7].

Table 1 shows the calculation results and accuracies of the steady-state calculation of the integrated analysis model of the thermal energy storage and utilization system by comparing the heat balance of Fig. 1. The main process variables such as temperature and pressure were compared at the inlet point of each device. As a result of comparison, it was confirmed that the maximum error of temperature and pressure at each point in the thermal energy utilization system coincided very well with the heat balance within 4°C and 1.55 MPa, respectively. On the thermal energy storage system side, the maximum error with respect to temperature occurred at 5.2℃. Converting the sodium-side inlet and outlet temperature difference of PCHE1 and PCHE2 in heat transfer rate was good agreement with the heat balance and the error was about 1%. The heat transfer rate of PCHE1 and PCHE2 was 63.88 MW and 36.2 MW.

3. Conclusions

In this study, a dynamic model for the partial heating cycle with thermal energy storage system was developed. To this end, a thermal energy storage and utilization system model was developed by modeling of individual devices developed in references from 4 to 7. In modeling, the Dymola, ClaRaPlus, and SolarTherm libraries were used. The developed model confirmed its accuracy by comparing with the heat balance, and it was

Table I: Comparison of the heat balance and the calculation results from the integrated analysis model of the thermal energy storage and utilization system.

Measured point	Heat balance	Steady Calculation	Error
Comp. inlet	32℃	32.626℃	0.626℃
	7.6 MPa	7.647 MPa	0.047MPa
LTR lower temp. inlet	79.22℃	76.2℃	3.02℃
	28.75MPa	27.2MPa	1.55MPa
HTR lower temp. inlet	114.1℃	115.8℃	1.7℃
	28.73MPa	-	-
PCHE2 lower temp. inlet	114.1℃	115.8℃	1.7℃
	28.73MPa	28.51MPa	0.22MPa
PCHE1 lower temp. inlet	360.4℃	-	-
	28.53MPa	-	-
Turb. inlet	515℃	512.7℃	2.3℃
	28.33MPa	28.4MPa	0.07MPa
LTR higher temp. inlet	363.9℃	362.5℃	1.4℃
	7.8MPa	7.83MPa	0.03MPa
HTR higher temp. inlet	141℃	145℃	4℃
	7.73MPa	-	-
Cooler inlet	83.54℃	83℃	0.54℃
	7.66MPa	-	-
PCHE1 higher temp. outlet	380℃	374.75℃	5.25℃
PCHE2 higher temp. outlet	200℃	198.35℃	1.65℃
Comp. flowrate	323.7 kg/s	325.1 kg/s	1.4 kg/s

confirmed that the maximum error of temperature and pressure of the thermal energy utilization system was well matched within 4℃ and 1.55 MPa, respectively. The heat transfer from the thermal energy storage system was also well matched with a maximum error of 1%. The model developed in this study will be used for the development of control and operation logic in the next step.

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