Thermodynamic Study of Compressed CO_2 Energy Storage System Integrated to a Conventional PWR

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1. Introduction

Recently, the ratio of renewable energy in the grid has increased globally due to climate change caused by greenhouse gas emissions. In Korea, renewable energy will account for 30~45% of the nation's power generation by 2040. [1] However, renewable energy has unpredictable intermittency in power generation. This issue can be alleviated by load-following operation of a nuclear power plant (NPP). However, it is not economical to control power output of the reactor in an NPP and can have a problem in nuclear fuel integrity. Energy Storage System (ESS) attached to the power cycle can solve this issue. Various ESS types (e.g., Thermal Energy Storage (TES), Li battery) can be considered and they are shown in Figure 1.

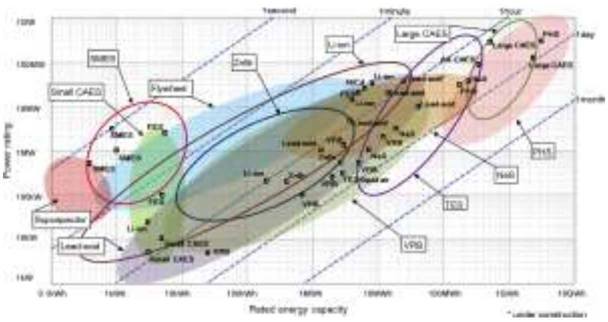

Figure 1. Comparison of power rating and rated energy capacity with discharge time duration at power rating [2]

Among them, the compressed air energy storage (CAES) system has high efficiency, technical feasibility, great power rating and capacity. Supercritical-CAES (SC-CAES) has quite high round-trip efficiency, but the critical point of air cannot be reached easily. If air is substituted with CO_2, its critical point can be easily reached with the current technology. A concept of supercritical compressed CO_2 energy storage (CCES) system was developed previously and showed good expected performance [3].

Therefore, in this paper, a thermodynamic modeling and analysis of a supercritical compressed CO_2 energy storage (CCES) integrated to a conventional PWR thermally and mechanically are studied. The thermodynamic analysis of PWR steam cycle integrated with CCES was studied [4] and its results and modeling are used in this paper. Thus, the performance of CCES in terms of round-trip efficiency and power density are presented in this paper.

2. Thermodynamic modeling

2.1 Steam cycle modeling

In order to store energy in CCES, it is necessary to branch steam from the steam cycle of an NPP and determine which section to bypass in the steam cycle before it merges back. Figure 2 is the layout of PWR steam cycle with CCES. CCES has two energy storage methods. First, there is a thermal energy storage (TES). The next is a mechanical storage using a steam turbine that drives a CO_2 compressor in CCES. Part of the branched steam passes through a heat exchanger, and the rest of it passes through the steam turbine. Since the mass flow rate of LP turbine inlet and inlet of feedwater heater (FWH) are changed from the nominal mass flowrate, off-design models for the LPT and FWH are applied to evaluate the lost work due to energy storage. Steam turbine off-design model and cycle evaluation are explained in this paper [4]. ε-NTU is used for the off-design model of FWH.

2.2 Thermodynamic modeling of CCES

Assumptions used for the modeling are as follows:

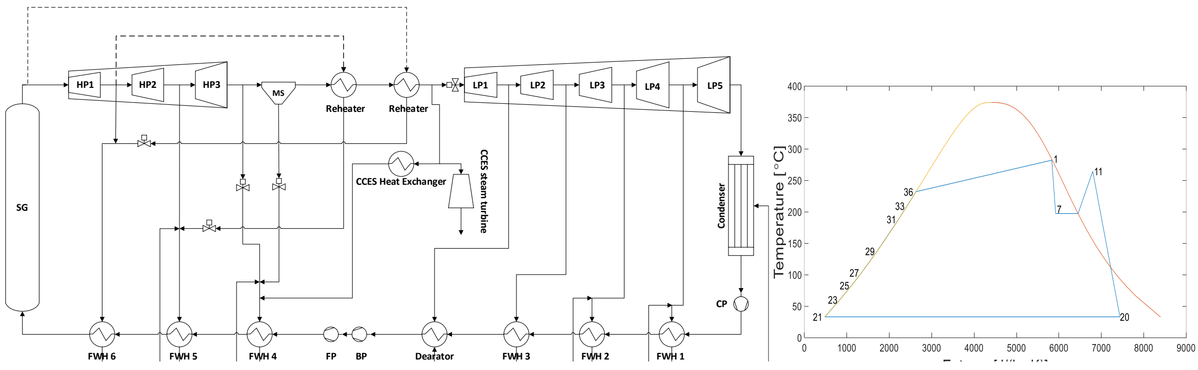

Figure 2. Layout and T-s diagram of Steam Cycle integrated with CCES

1) CO_2 tanks and the TES tanks have the same temperature, pressure, and therefore, thermophysical properties at the inlet and outlet, respectively.
2) There is no pressure drop in the pipelines.
3) Turbines and compressor have constant isentropic efficiencies, respectively.
4) The ratio of charging time to discharging time is unity.
5) There are no changes in potential and kinetic energies

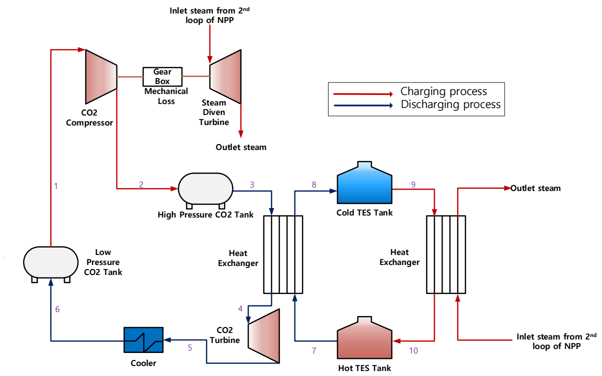

Figure 2. Layout of CCES integrated to PWR steam cycle

As shown in Figure 2, processes 1-3, 10-12 and 15-16 are the energy storage process (Charing operation) and the rest of processes are the energy recovery process (Discharging operation).

2.2.1 Heat exchanger

All heat exchangers in this CCES system have constant pressure drop rate. For given temperature and pressure of inlet of hot side and cold side, the outlet of hot side and cold side can be obtained from using heat exchanger effectiveness and following equations.

$$Q_{max} = \min(\dot{m_{hot}}(h_{hot,in} - h_{hot,out,i}), \dot{m_{cold}}(h_{cold,out,i} - h_{cold,in}))$$

$$\varepsilon_{HX} = \frac{Q_{act}}{Q_{max}}$$

$$Q_{act} = \dot{m_{hot}}(h_{hot,in} - h_{hot,out}) = \dot{m_{cold}}(h_{cold,out} - h_{cold,in})$$

In this paper, pinch in heat exchangers should be larger than 5K. If it has a pinch problem, its HX effectiveness is decreased until satisfying this condition.

2.2.2 TES

HITEC is usually used for the material of TES. This melting temperature is 142.35℃ and specific heat is constant at the molten state. Thus, its enthalpy is the function of its temperature.

$$\mathrm{h_{HITEC}} = 1.555 * 10^3(\mathrm{T_{HITEC}} - 422.039) + 2.56 * 10^5$$

HITEC TES has 2 heat exchangers for steam and HITEC, and for CO_2 and HITEC to store and transfer the heat from steam to CO_2.

2.2.3 Compressor

This compressor is driven by the steam turbine of PWR steam cycle and called steam turbine driven compressor (STDC). Thus, it doesn't need motor and electricity to run the compressor. The outlet pressure, temperature and mass flow rate of a compressor can be obtained from the equation and given compressor work, isentropic efficiency, inlet temperature and pressure ratio.

$$\mathrm{\eta_c} = \frac{\mathrm{h_{out,s}} - \mathrm{h_{in}}}{\mathrm{h_{out}} - \mathrm{h_{in}}}$$

$$\dot{m}, T_{out} = f(\eta_c, T_{in}, P_{in}, P_{out}, W_{comp})$$

2.2.4 Turbine

The pressure ratios of the turbines are determined by the inlet/outlet pressure of the compressor. Then, the outlet pressure and temperature of turbine are obtained from the below equation.

$$\mathrm{\eta_t} = \frac{\mathrm{h_{in}} - \mathrm{h_{out}}}{\mathrm{h_{in}} - \mathrm{h_{out,s}}}$$

$$P_{out}, T_{out} = g(\eta_t, T_{in}, P_{in}, PR)$$

2.2.5 Pipe sizing

Since the flow rate is quite large in certain conditions, the pipe size issue must be addressed. The maximum diameter was referred from the ASME standard [5], and the diameter of each point for S- CO_2 is obtained from an empirical formula suggested by Ronald W. Capps [6].

$$\mathrm{D} = 2\sqrt{\frac{\dot{m}}{\pi f_{pv} \rho^{0.7}}}$$

where D represents diameter of a pipe, f_{pv} represents pipe velocity factor and its optimal value is 29.

2.3 Modeling of parameters

Table1. Design parameters of CCES

Parameters	Value	Unit
Temperature of low-pressure reservoir	308.15	K
Temperature of HITEC cold tank	423.15	K
Mass flow rate of HITEC	9000	kg/sec
Isentropic efficiency of turbines	0.9	
Isentropic efficiency of compressor	0.85	
Effectiveness of heat exchangers	0.9	
Ratio of charging time to discharging time	1	
Pressure drop in HX	1	%
Total steam bypass fraction	20	%
Minimum pinch in HX	5	K
Mechanical loss of gear box	5	%

Table2. Variables of CCES

Parameters	Range of Variation	Unit
Steam bypass fraction to TES of CCES	0.1-0.9	
Pressure of low-pressure reservoir	7.8-8.3	MPa
Pressure of high-pressure reservoir	20-25	MPa

The design parameters are shown in Table1 and the variables and ranges of variation are shown in Table2. For the supercritical compressed CO_2 energy storage system (SC-CCES), the minimum temperature and pressure range of CO_2 are set above the critical point of CO_2 (7.39MPa, 31°C). Total steam to bypass from steam cycle is fixed at 20% of nominal LPT mass flow rate and performance changes are analyzed according to the steam distribution of TES and steam turbine.

3. Thermodynamic evaluation and Results

A round-trip efficiency (RTE) is the ratio of discharge work to charging work in the energy storage system. This is the criteria for cycle optimization. The round-trip efficiency in this system can be calculated using,

$$\eta_{RT} = \frac{W_{turb}}{W_{PWR,loss}}$$

where W_{turb} represents the CO_2 turbine work and $W_{PWR,loss}$ represents the difference of work before and after bypass the steam to CCES.

It is necessary to determine the amount of work that can be produced per unit volume of storage capacity. It is called power density or energy density.

$$\rho_{power} = \frac{W_{turb}}{\dot{m}_{charge/}\rho_{LPT} + \dot{m}_{discharge/}\rho_{HPT}}$$

KAIST CCD code developed by KAIST research team is used for cycle evaluation of round-trip efficiency and power density calculation.

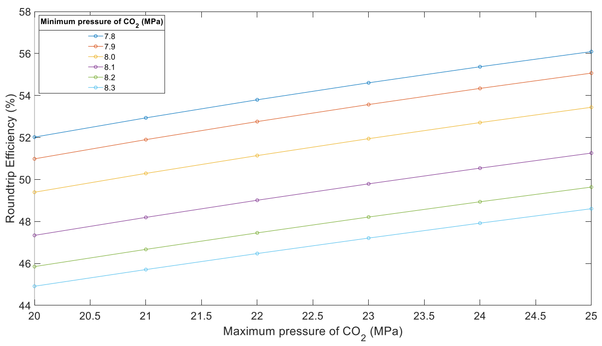

Figure 3. Round-trip efficiency vs Minimum pressure of system (Steam bypass fraction to HX of CCES: 0.5)

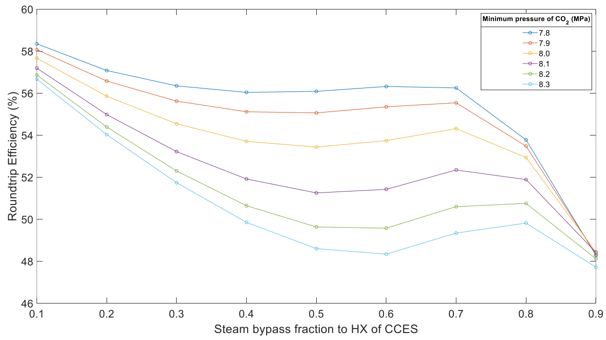

Figure 4. Round-trip efficiency vs Steam bypass fraction to HX of CCES (Maximum pressure of system: 25MPa)

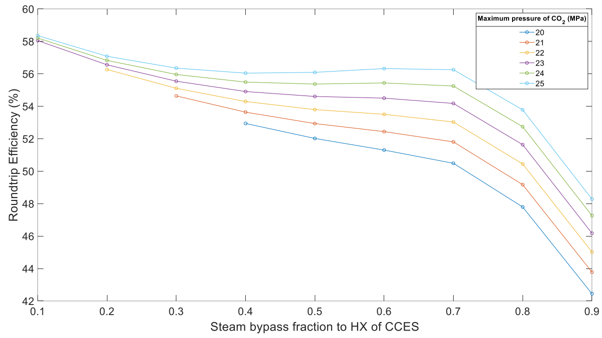

Figure 5. Round-trip efficiency vs Steam bypass fraction to HX of CCES (Minimum pressure of system: 7.8MPa)

As shown in Figure 3, it is observed that the lower the minimum pressure of the system is, the higher the maximum pressure is, and the higher the round-trip efficiency will be. This is because as the work of the compressor is fixed, lower minimum pressure and higher maximum pressure will lead to greater work from turbine.

In Figures 4 and 5, the round-trip efficiency increases as steam bypass fraction to HX of CCES decreases. It can be seen that in the range of the steam bypass fraction to HX of CCES between 0.3 and 0.7, there is no significant change in RTE. In Figure 5, there are no data for some conditions because the diameter of the pipe exceeds the maximum diameter for ASME standard. When a large amount of steam flows into the steam turbine, the CO_2 mass flow rate increases, resulting in a larger pipe diameter required.

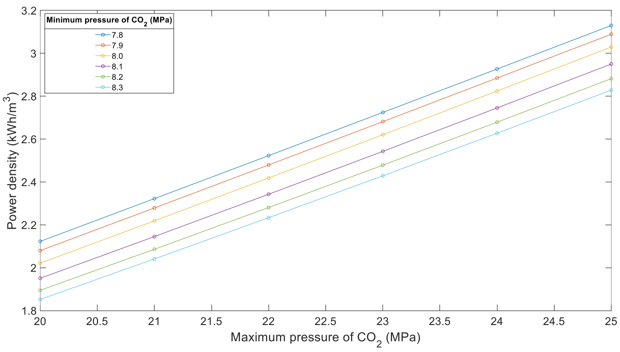

Figure 6. Power density vs Minimum pressure of system (Steam bypass fraction to HX of CCES: 0.5)

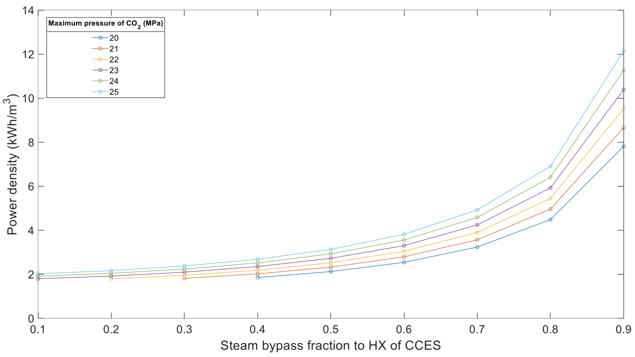

Figure 7. Power density vs Steam bypass fraction to HX of CCES (Minimum pressure of system: 7.8MPa)

Observed from Figure 6, lower minimum pressure and higher maximum pressure lead to higher power density. However, the power density is about 2kWh/m^3, which is not very sensitive to the maximum and minimum pressures.

In Figure 7, the power density increases as steam bypass fraction to HX of CCES increases. The minimum and maximum power densities are 2kWh/m^3 and 12kWh/m^3, respectively. It can be shown that it is quite sensitive to the steam bypass distribution between steam turbine and TES of CCES.

3. Summary and Future works

From the result of the compressed supercritical CO_2 energy storage analysis, it is shown that as the maximum pressure and the steam bypass fraction to steam turbine to drive CO_2 compressor increase and the minimum pressure decreases, the round-trip efficiency increases while power density decreases. The maximum RTE is about 52% and maximum power density is about 12kWh/m^3. Among them, the bypass fraction of steam to HX of CCES is the most effective parameter

In the future, tank modeling will be added to limit the maximum pressure and bypass steam fraction distribution. Further investigation will commence soon regarding optimization of CCES round-trip efficiency and power density by adding off-design model as well.

Acknowledgement

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REFERENCES

[1] 산업통상자원부, 제 9 차 전력수급기본계획(2020~2040), 2020

[2] Luo, Xing, et al. "Overview of current development in electrical energy storage technologies and the application potential in power system operation." Applied energy 137 (2015): 511-536.

[3] Zhang, Yuan, et al. "Thermodynamic analysis of a novel energy storage system with carbon dioxide as working fluid." Renewable Energy 99 (2016): 682-697.

[4] YongJae Chae et al. "Thermodynamic analysis of a Conventional PWR Integrated with Compressed 〖CO〗_2 Energy Storage System" Transactions of the Korea Nuclear Society Spring Meeting (2019).

[5] ASME B36.10M (Welded and Seamless Wrought Steel Pipe), ASME,2010.

[6] Ronald W. Capps, Fluid Handling, textbook of the Gulf Coast Consulting Group, 2013.