World Academy of Science, Engineering and Technology
International Journal of Mechanical and Mechatronics Engineering
Vol:8, No:8, 2014

Performance Variation of the TEES According to the Changes in Cold-Side Storage Temperature

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***Abstract*—**Surplus electricity can be converted into potential energy via pumped hydroelectric storage for future usage. Similarly, thermo-electric energy storage (TEES) uses heat pumps equipped with thermal storage to convert electrical energy into thermal energy; the stored energy is then converted back into electrical energy when necessary using a heat engine. The greatest advantage of this method is that, unlike pumped hydroelectric storage and compressed air energy storage, TEES is not restricted by geographical constraints. In this study, performance variation of the TEES according to the changes in cold-side storage temperature was investigated by simulation method.

***Keywords*—**Energy Storage System, Heat Pump.

I. Introduction

NEW and renewable energy sources such as wind and photovoltaic has been developed. However, because of the intrinsic characteristics of these energy sources, a time discrepancy between supply and demand exists. In order to overcome this problem, electrical energy storage technologies have been developed, including pumped hydroelectric storage, batteries, flywheels, capacitors, and compressed air energy storage. Several researchers have recently proposed thermo-electric energy storage (TEES). Surplus electricity can be converted into potential energy via pumped hydroelectric storage for future usage. Similarly, TEES uses heat pumps equipped with thermal storage to convert electrical energy into thermal energy; the stored energy is then converted back into electrical energy when necessary using a heat engine. The greatest advantage of this method is that, unlike pumped hydroelectric storage and compressed air energy storage, TEES is not restricted by geographical constraints.

Desrues et al. [1] proposed a thermal energy storage process based on the Brayton cycle with argon as the working fluid. Peterson [2] proposed a scheme that uses latent heat storage at sub-ambient temperatures and demonstrated that his charging and discharging processes are favorable in the nighttime and daytime, respectively, owing to the differences in atmospheric temperature. Henchoz et al. [3] also noted the effectiveness of sub-ambient temperature TEES. Some studies have recently attempted to use CO_2 as the working fluid for TEES. Mercangöz et al. [4] proposed an energy storage scheme called electrothermal energy storage that uses a transcritical CO_2 cycle. Morandin et al. [5], [6] used pinch analysis tools to optimize various forms of transcritical-CO_2-cycle-based TEES with multiple water tanks. More recently, Kim et al. [7] proposed a unique scheme in which an isothermal process was added to the conventional transcritical-CO_2-cycle-based scheme for increased efficiency.

In this study, performance variation of the TEES according to the changes in cold-side storage temperature was investigated by simulation method.

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II. Modeling

Figs. 1 and 2 show schematics of the charging and discharging processes, respectively, of the TEES. This can be expressed as a combination of the charging process based on heat pump and Rankine power cycle. The operation principle of each process is as follows.

During charging, CO_2 working fluid in a low-temperature, low-pressure state (state point 1 (SP1) in Fig. 1) passes through the compressor, to which power is supplied from outside of the system, and goes into a high-temperature, high-pressure state (SP2). Then, the fluid passes through the high-temperature-side heat exchanger and cools while transferring heat to water to go into a low-temperature, high-pressure state (SP3). The water from hot storage tank 1 passes through the high-temperature-side heat exchanger; here, its temperature increases owing to heat received from the working fluid, following which it is stored in hot storage tank 2. The low-temperature, high-pressure working fluid (SP3) goes into a low-temperature, low-pressure state (SP4) as it passes through the expansion valve. The working fluid in a low-temperature, low-pressure state (SP4) is introduced into the low-temperature-side heat exchanger receives heat from water, becomes vapor (SP1), and re-enters the compressor. The water from the cold storage tank 1transfers heat to the working fluid as it passes through the low-temperature-side heat exchanger, cools, and is stored in the cold storage tank 2. After charging, discharging can take place if necessary. Discharging can be explained as the reverse process of charging. First, CO_2 working fluid from the low-temperature-side heat exchanger in a low-temperature, low-pressure state (SP5) passes through the pump and goes into a low-temperature, high-pressure state (SP6). The working fluid then passes through the high-temperature-side heat exchanger to receive heat from water and goes into a high-temperature, high-pressure state (SP7). The water from hot storage tank 2 transfers heat to the working fluid as it passes through the high-temperature-side heat exchanger, cools, and is stored in hot storage tank 1. The working fluid in a high-temperature, high-pressure state (SP7) generates power as it passes through the expander (turbine) and goes into a low-pressure state (SP8). The low-pressure working fluid is introduced into the low-temperature-side heat exchanger to transfer heat to water, becomes liquid (SP5), and re-enters the pump. The water from the cold storage tank 2 receives heat from the working fluid as it passes through the low-temperature-side heat exchanger such that its temperature increases and is released.

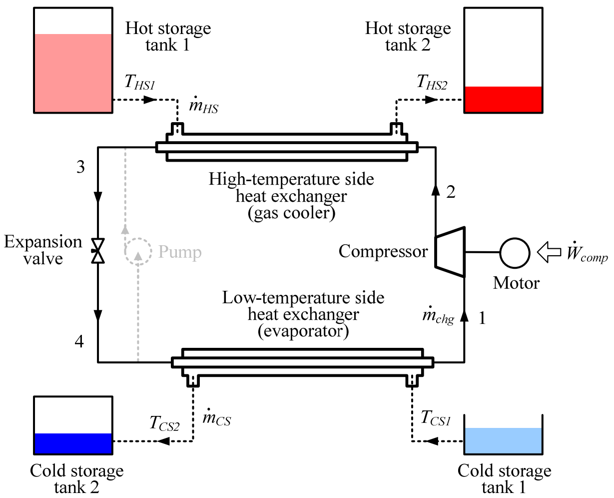

Fig. 1 Schematic diagram of charging process

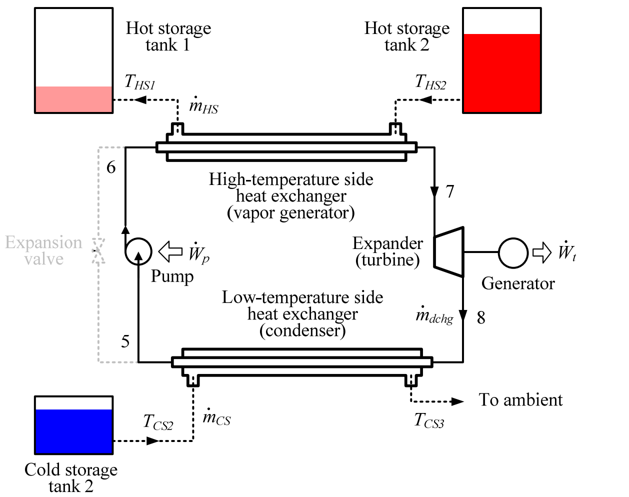

Fig. 2 Schematic diagram of discharging process

The number of unknown variables for the charging process becomes seven. A set of seven simultaneous equations is necessary. An isentropic efficiency for the compressor can be calculated as

$$\eta_{comp} = (i_{2s} - i_1)/(i_2 - i_1). \tag{1}$$

For the discharging process, there are eight unknown variables. A set of eight simultaneous equations is necessary. An isentropic efficiency for the turbine and the pump can be calculated as

$$\eta_t = (i_7 - i_8)/(i_7 - i_{8s}) \tag{2}$$

$$\eta_p = (i_{6s} - i_5)/(i_6 - i_5). \tag{3}$$

The high- and the low-temperature-side heat exchangers were regarded as a concentric double-pipe heat exchanger. Because it is recommended that a counter-current heat exchanger be used in transcritical-CO_2-cycle-based TEES to utilize the temperature variation of CO_2 during heat transfer, a simple double-pipe heat exchanger was chosen for convenience. A working fluid flows through the inner tube, and water flows counter-currently through the annulus. To consider the longitudinal property variations of the working fluid and the water, the entire length of the heat exchanger is divided into 10 discrete segments of equal length, each of which is treated as a counter-flow double-pipe heat exchanger. Heat transfer in the i^{th} segment of the heat exchanger is expressed as

$$Q^i = (UA)^i (\mathrm{LMTD})^i. \tag{4}$$

The overall heat transfer coefficient for the i^{th} segment is

$$\frac{1}{(UA)^i} = \frac{1}{(h_i)^i \cdot \pi d_i \Delta L} + \frac{\ln(d_o/d_i)}{2\pi \Delta L k} + \frac{1}{(h_o)^i \cdot \pi D_i \Delta L}. \tag{5}$$

For calculating the charging process, the gas cooler model obtains gas cooler exit states with inlet states as input variables. In the gas cooler, the working fluid flows into the inner tube in a supercritical state and is cooled by the water from hot storage tank 1. To calculate the heat transfer coefficient of supercritical working fluid, the Pitla correlation [8] was employed. In the evaporator, the working fluid in a subcritical state undergoes two- and single-phase heat transfer processes. The single-phase heat transfer coefficient was evaluated using Gnielinski's correlation. The two-phase heat transfer coefficient can be calculated by the correlation proposed by Gungor and Winterton.

For calculating the discharging process, the vapor generator model obtains vapor generator exit states with inlet states as input variables. In the vapor generator, the working fluid flows into the inner tube in a supercritical state and is heated by the water from hot storage tank 2. To calculate the heat transfer coefficient of the working fluid, the Krasnoshchekov-Protopopov correlation [9] was employed together with the Petukhov-Kirillov correlation [10]. The pseudocritical temperature of CO_2 required for the calculation of the Krasnoshchekov-Protopopov correlation was obtained from the equation proposed by Baik et al. [11]. The condenser model obtains condenser exit states with inlet states as input variables. As in the evaporator during charging, the working fluid undergoes two-and single-phase heat transfer processes in a subcritical state in the condenser. The correlation proposed by Shah was used in calculating the two-phase heat transfer coefficient. The working fluid's single-phase heat transfer coefficient in the condenser was obtained using Gnielinski's correlation, which was also used to obtain the water-side heat transfer coefficient.

The working fluid-side frictional pressure drop was considered by using Blasius's equation and Müller-Steinhagen and Heck's correlation [12].

III. Numerical Procedure

The system is assumed to be in the steady state for the simulations, and the heat loss in each component is neglected. The thermodynamic properties of the working fluids are calculated by using REFPROP 9.0. The following conditions are given:

(1) Hot storage tank 1's temperature= 40°C, Hot storage tank 2's temperature= 120°C, and the mass flow rate = 0.015 kg/s.
(2) Cold storage tank 1's temperature = 5~20°C, and the mass flow rate = 1.5 kg/s
(3) The isentropic efficiency for the compressor, turbine, and pump is 0.85.
(4) The compressor suction superheat is 5°C.
(5) The condenser exit is in a saturated liquid state.
(6) Compressor exit pressure = 20 Mpa.
(7) Turbine inlet temperature = 115°C.

The length of the high- and the low-temperature-side heat exchanger L is fixed at 20 m and 10 m, respectively. The inner tube is assumed to be a standard stainless steel tube with an outside diameter of 15 mm and a thickness of 2.2 mm to manage high-pressure operation of up to ~400 bar. The thermal conductivity of the inner tube is 15 W/mK. The inside diameter of the outer tube is 15.62 and 20.40 mm for the high- and the low-temperature-side heat exchangers, respectively, with the water-side mass flux G fixed at 1,000 kg/m^{2}s.

Once the temperature of the cold storage tank 1 is given, the round-trip efficiency can be obtained as follows. First, calculation for the charging process is performed. We assume the values of seven unknown variables. The gas cooler exit states are determined by using the gas cooler model; evaporator exit states are evaluated by using the evaporator model; and, then, the compressor efficiency can be calculated by using (1). This process is repeated until a set of seven simultaneous equations is solved. Once the calculation for the charging process is complete, the compressor's power input can be calculated.

Next, the values of the 8 unknown variables in the discharging process are assumed. With assumed values as input variables and using the vapor generator model and condenser model, the exit states of each heat exchanger are obtained. An efficiency of the turbine and the pump can be calculated by using (2) and (3), respectively. This procedure is repeated until a set of eight simultaneous equations is solved. The power generated by the turbine and the power consumed by the pump can be calculated and their difference is the net power output.

Having obtained the compressor power input (charging) and net power output (discharging) for a given temperature at the cold storage tank 1, a round-trip efficiency is determined.

IV. Results

Fig. 3 shows performance variation of the TEES according to the changes in the temperature at the cold storage tank 1. According to Fig. 3, as the cold-side storage temperature increases, the round-trip efficiency decreases.

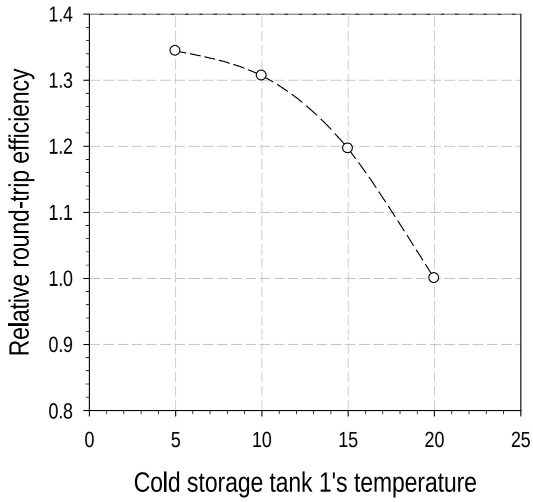

Fig. 3 Performance variation of the TEES according to the changes in the temperature of the cold storage tank

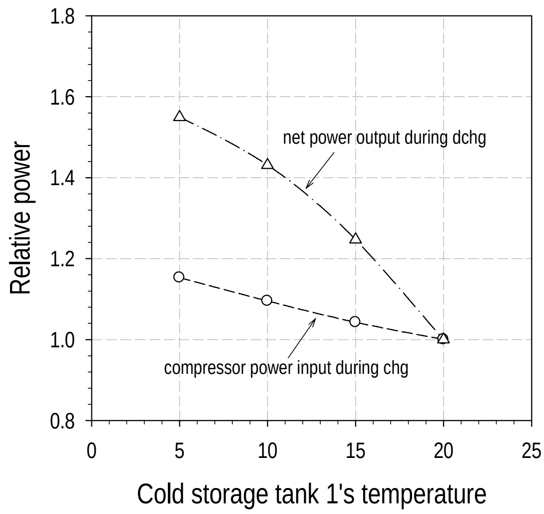

Fig. 4 Relative power variation according to the changes in the temperature of the cold storage tank

The reason is that as the cold-side storage temperature decreases, although the heat pump performance decreases, the heat engine performance increases more than that as shown in Fig. 4. Regarding this issue, Mercangöz et al. [4] also investigated that the lower the ambient temperature the higher the TEES efficiency. They also added that the TEES roundtrip efficiency is more strongly influenced by the heat engine, and this is what the present study confirmed.

V. Conclusion

Performance variation of the TEES according to the changes in cold-side storage temperature was evaluated by the simulation method. We have confirmed that as the cold-side storage temperature increases, the round-trip efficiency decreases. Therefore, in order to increase the round-trip efficiency of the TEES, it is necessary to decrease the temperature of the cold-side storage.

Acknowledgement

This work was conducted under the framework of Research and Development Program of the Korea Institute of Energy Research (KIER).

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