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Alternating operation of a supercritical CO_2 heat pump / gas turbine for electrical energy storage

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Abstract

To ensure a reliable and efficient energy sector with a growing share of intermittent renewable energy, energy storage is essential. One of the promising solutions to store excess energy (charging mode) and deliver deficit energy (discharging mode) is Thermo-Electrical Energy Storage (TEES). During the charging mode, excess electrical energy is converted into heat at high temperature using a heat pump and during the discharging mode, the stored heat is converted into electrical energy using a heat engine. Hereby, an attractive working fluid for the heat pump and the heat engine cycles is CO_2, due to its non-flammability, non-toxicity, high density, and near-ambient critical temperature.
This paper proposes an innovative TEES based system consisting of a supercritical CO_2 heat pump in the charging mode and a supercritical CO_2 gas turbine in the discharging mode, with a heat storage in between. Hereby, an advantage of the proposed system is the ability to use the same components in the two alternating operational phases. Next to the energy conversion systems, a suitable heat storage medium is required, which allows heat transfer at low temperature differences and is compact. For the proposed system, schematic diagrams, pressure-enthalpy and temperature-entropy diagrams are given. Furthermore, thermodynamic models of the supercritical CO_2 gas turbine and the supercritical CO_2 heat pump are developed. To calculate the COP of the proposed heat pump and the thermal efficiency of the proposed gas turbine a thermodynamic analysis is performed. Hereby, results for the influence of the pressure and the temperature conditions on the round-trip efficiency are presented. The results show that the system can achieve high thermodynamic performance with a round-trip efficiency between 55% and 65%. This indicates that the supercritical CO_2 heat pump / gas turbine system is a competitive energy storage technology for decarbonisation of the energy sector.

Keywords: Thermo-electrical energy storage; supercritical CO_2 gas turbine; supercritical CO_2 heat pump; heat storage; round-trip efficiency;

1. Introduction

As the share of renewable energy grows, the requirement for energy storage grows as well. Renewable energy sources, such as solar and wind, are intermittent, making them unreliable for a consistent, predictable energy supply. To address the intermittency challenge, efficient energy storage and conversion systems are essential. These systems store excess energy generated during periods of high renewable energy production and release it when production is low. Hereby, they need to be able to store energy also for long periods. In this context, a decarbonized energy sector requires two classes of long duration energy storage for electricity,

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one lasting up to 20 h to manage daily cycles and another lasting weeks or months to manage seasonal cycles [1]. A promising energy storage technology for long duration energy storage is Thermo-Electrical Energy Storage (TEES). The operation principle of the TEES is such that during charging mode, excess electrical energy is converted into high temperature heat using a heat pump, whereas in discharging mode, the stored heat is converted into electrical energy using a heat engine. This energy storage system is also referred to as Pumped Thermal Energy Storage (PTES) or Carnot Battery.

In the heat pump and the heat engine cycle, different working fluids can be implemented. Also, in the heat storage, various storage media can be implemented. TEES with nitrogen as the working fluid is proposed in [2]. An attractive working fluid for the heat pump and heat engine cycles is CO_2, due to its non-flammability, non-toxicity, high density, and near-ambient critical temperature. In Morandin et al. [3] TEES has been investigated based on transcritical CO_2 cycles. Different storage media have been investigated for the use in a TEES, including molten salts [4], particle media [2] and water [3]. Morandin et al. [3] proposed water as the hot storage medium because it has a high thermal capacity, is environmentally friendly, and is cheap. Also, cold energy storage is included by incorporating ice as the storage medium. The choice of the working fluid and the storage media still has to be further investigated.

In the literature, CO_2 heat pumps and power cycles have been integrated into energy storage solutions, including pumped thermal energy storage [5], compressed fluid energy storage [6] and hydrogen energy storage [7]. In [8] a transcritical CO_2 system has been applied in a pumped thermal energy storage system, achieving a round-trip efficiency of 54.6% for a considered example with a charging power of 50 MW and storage duration of 10 h. Furthermore, in [9] a compressed supercritical carbon dioxide system with a compression heat recovery is proposed as a novel category of energy storage solutions. An alternative storage solution, preferable for longer durations than TEES, is the production of renewable fuels. Gao et al. [10] proposed an integrated energy storage system consisting of an electrolyser and a hydrogen-oxygen combined cycle, in which pure hydrogen and oxygen are supplied to a combustion chamber to obtain high temperature and high-pressure superheated steam at 1500 °C. All these studies confirm that there is a considerable research interest in the application of CO_2 heat pumps and power cycles within energy storage systems.

To minimize exergy destruction in the heat transfer processes, a low temperature difference is required throughout the heat exchangers. Since there is no phase change in a supercritical cycle configuration, the working fluid can follow the thermal profile of the sensible heat storage. This is a benefit of the supercritical CO_2 heat pump cycle. However, during a temperature change, the thermodynamic properties of CO_2 vary, particularly the specific heat capacity along the supercritical isobars. This variation is the most significant in the supercritical region. To obtain a match in the thermal profiles of the working fluid and the storage medium, and hence a good thermal integration of the thermodynamic cycle with a sensible heat storage, multiple thermal storage tanks have been considered in [3]. Furthermore, multiple cycle configurations were explored for the TEES. Tafur-Escanta [11] highlighted the use of a recuperator in both the gas turbine cycle and the heat pump cycle for the TEES. This simple regenerative configuration was adopted by Marchionni et al. [12], by conducting part-load and transient analyses of the cycles and comparing the results with those from a 50 kWe experimental test facility. A transcritical CO_2 cycle configuration with integrated condensation of the working fluid is implemented in [13]. However, the matching of the thermal profiles in the heat exchangers and the cycle configuration needs to be further explored.

The aim of this paper is to present an original energy storage system consisting of a supercritical CO_2 heat pump that converts excess electrical energy into high temperature heat, and a supercritical CO_2 gas turbine that converts the high temperature heat back to electrical energy. The two operation modes are integrated using high temperature storage and low temperature storage. Heat is stored in the high temperature storage where CO_2 is used as a storage medium. Alternatively, for safety reasons, molten salts could be used as high temperature storage medium. Additionally, a low temperature storage is implemented, which is used to store the heat after recuperation in the supercritical CO_2 gas turbine cycle and to deliver it to the CO_2 heat pump cycle. To analyse the round-trip efficiency of the system under different design and operation conditions, thermodynamic models of the supercritical CO_2 gas turbine and the supercritical CO_2 heat pump are developed in MATLAB/CoolProp. The results show that the system achieves high thermodynamic performance. The novel contributions of this work include:

- Development of an original energy storage system consisting of a supercritical CO_2 heat pump and a supercritical CO_2 gas turbine, integrated with both high temperature storage and low temperature storage with CO_2 or molten salts as the storage medium.
- Development of thermodynamic models of the supercritical CO_2 heat pump and the supercritical CO_2 gas turbine.
- Comprehensive evaluation of the achievable round-trip efficiency depending on the temperatures of the high temperature storage and the low temperature storage, pressure ratio, turbomachinery isentropic efficiency, and temperature difference in heat exchangers.

2. Methodology

The proposed system operates as a supercritical CO_2 heat pump in the charging mode and a supercritical CO_2 gas turbine in the discharging mode. Hereby, the schematic of the system is given in Fig. 1 and the alternating operation modes are presented in Fig. 2 and Fig. 3. For this system, the temperature-entropy and pressure-enthalpy diagrams for a defined set of operation conditions are shown in Fig. 4 and Fig. 5. The design of the system is based on a thermodynamic cycles analysis. Hereby, it is observed that CO_2 as a working fluid for the heat pump and the gas turbine cycles is advantageous because of its non-flammability, non-toxicity, low GWP, high density, and near-ambient critical temperature. Furthermore, with respect to achievable round-trip efficiency, a supercritical configuration with compressor inlet and compressor outlet pressures above the critical pressure, of both the heat pump cycle and the gas turbine cycle, is beneficial. Hence, the expander in the heat pump cycle and the compressor in the gas turbine cycle operate in the supercritical region. Firstly, a supercritical gas turbine based on a regenerative Brayton cycle provides high thermal efficiency due to the low compression work in the critical region, effective heat recuperation, and operation at high temperature and pressure conditions [14], [15]. Secondly, a supercritical CO_2 heat pump based on a regenerative Brayton cycle provides high temperature heat with reasonable COP [16]. Finally, based on the high achievable efficiency of the heat pump and the gas turbine, this configuration leads to a system with high round-trip efficiency.

During the charging mode the system operates as a heat pump, utilizing excess electrical energy to upgrade the heat from low temperature storage to high temperature storage (Fig. 2). The temperature-entropy diagram of the supercritical CO_2 heat pump is depicted with a blue line in Fig. 4. Also, a pressure-enthalpy diagram of the supercritical CO_2 heat pump is depicted with a blue line in Fig. 5. The working fluid is compressed (TC) from state 1 to state 2 and cooled in heat exchanger 1 (HEX1) to state 3, where the heat is transferred to the high temperature storage (H1). Then, the working fluid is further cooled in the recuperator (R) to state 4 and expanded (TE) to state 5. Heat is transferred to the working fluid firstly in heat exchanger 2 (HE2) from the low temperature storage (H2), reaching state 6, and secondly from the working fluid in the recuperator (R), closing the cycle at state 1.

During the discharging mode, the system operates as a gas turbine, utilizing the heat at high temperature, which has been generated during the charging mode, to generate electrical energy. The temperature-entropy diagram of the supercritical CO_2 gas turbine is depicted with a red line in Fig. 4. Also, a pressure-enthalpy

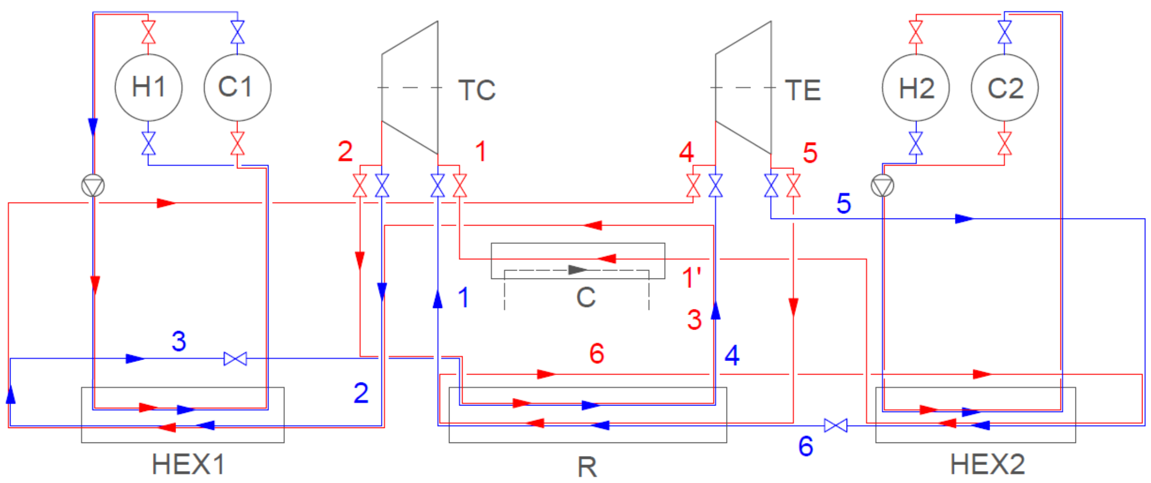

Figure 1. Schematic of the supercritical CO_2 heat pump / gas turbine system

diagram of the supercritical CO_2 gas turbine is depicted with a red line in Fig. 5. The working fluid is compressed (TC) from state 1 to state 2. Then, the working fluid is heated in the recuperator (R) to state 3 and in heat exchanger 1 (HEX1) to state 4, using the heat from the high temperature heat storage. The working fluid is expanded (TE) to state 5 and cooled firstly in the recuperator (R), reaching state 6 and then in heat exchanger 2 (HEX2) using the low temperature storage. In the cooler (C) the working fluid is cooled from state 1' to state 1, allowing a lower temperature at the compressor inlet and generating additional heat. The cooler C has two purposes. Firstly, it reduces the compressor inlet temperature, thereby reducing compression work in the gas turbine cycle. Secondly, it enables combined heat and power generation within the system by effectively utilizing the additional heat generated.

Since a smaller temperature lift is preferred in the heat pump cycle, the temperature of the low temperature heat storage should be higher. Therefore, the rejected heat from the gas turbine cycle is stored and delivered in the heat pump cycle. Also, in the gas turbine cycle, a larger temperature difference is preferred. Therefore, additional cooling is envisioned in the gas turbine cycle in the cooler (C) to reduce the compressor inlet temperature and consequently the compression work. The additional heat generated in C (Eq. (8)) could be used for heating water (up to 95 °C), depending on the end user's needs. Alternatively, this cooling could be realised using the ambient air.

For the supercritical CO_2 gas turbine / heat pump, thermodynamic models are developed to calculate the COP of the proposed heat pump and the thermal efficiency of the proposed gas turbine. Input parameters in the model are the temperature of the high temperature storage at the cold side, the temperature of the low temperature storage at the hot side, the compressor inlet pressure, the pressure ratio, the compressor isentropic efficiency, the expander isentropic efficiency, and the minimal temperature difference in heat exchangers.

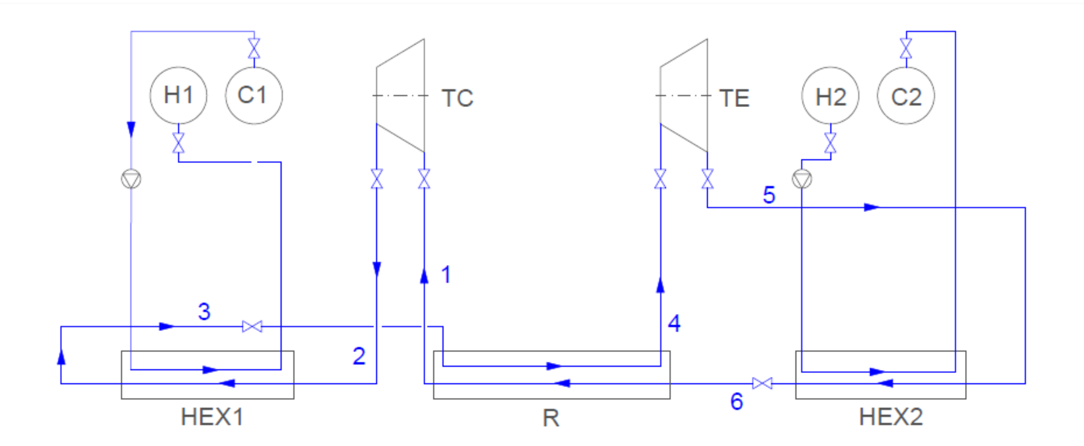

Figure 2. Schematic of the supercritical CO_2 heat pump – charging mode of the system

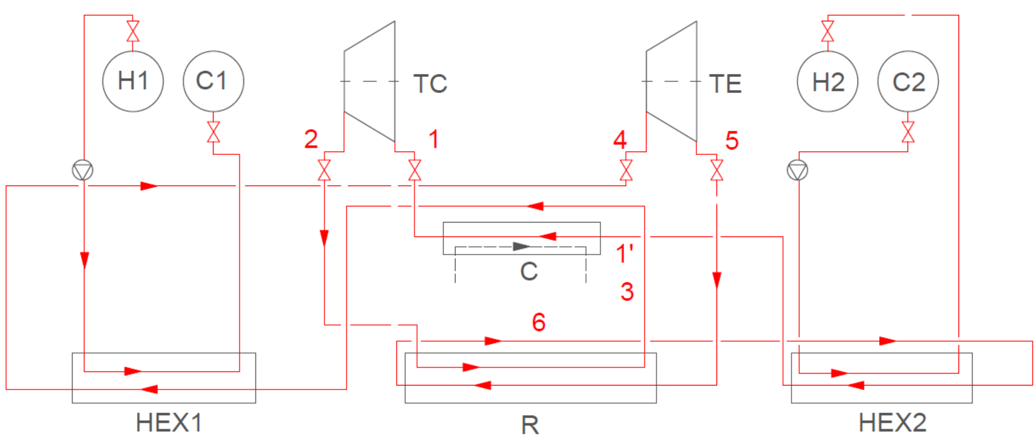

Figure 3. Schematic of the supercritical CO_2 gas turbine – discharging mode of the system

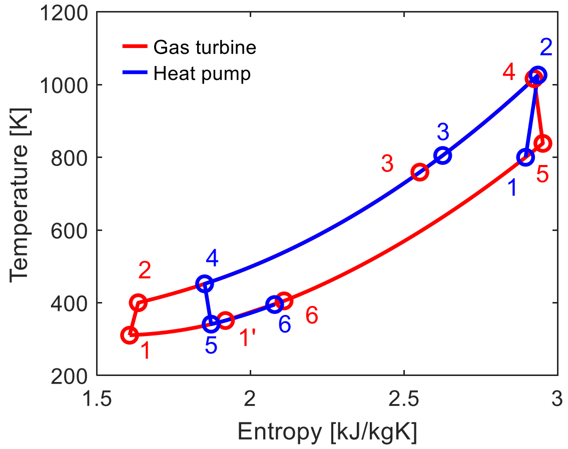

Figure 4. Temperature-entropy diagram of the supercritical CO_2 heat pump / gas turbine system

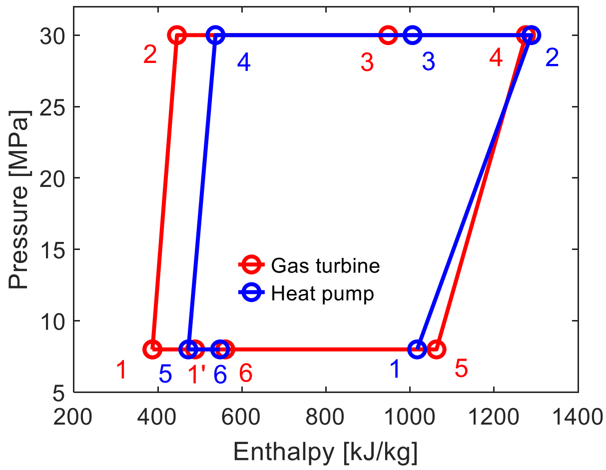

Figure 5. Pressure-enthalpy diagram of the supercritical CO_2 heat pump / gas turbine system

The thermodynamic analysis of the system starts with the calculation of the thermodynamic states of the heat pump cycle. As an output of this model, the net work input, heat output, heat input, and COP are obtained, as well as the temperature of the high temperature storage at the hot side and the temperature of the low temperature storage at the cold side. The analysis then proceeds with the calculation of the thermodynamic states of the gas turbine cycle. As an output of this model, the net work output, heat output, heat input, and thermal efficiency are obtained. Based on this, the round-trip efficiency is calculated. The following modelling assumptions are made:

- Steady state conditions in the heat pump and gas turbine cycle components.
- Adiabatic compression and expansion processes.
- Pressure drops in the heat exchangers are neglected.
- Kinetic and potential energy effects are negligible.

The thermodynamic modelling of the heat pump cycle and gas turbine cycle considers the real gas thermodynamic properties of CO_2. Hereby, the variations of the thermodynamic properties, such as specific heat capacity and enthalpy, especially in the supercritical region, need to be addressed. In this context, CoolProp is used for the thermodynamic properties of CO_2. The thermodynamic models also address the real thermodynamic processes of compression and expansion by including turbomachinery isentropic efficiency in Eq. (1). Furthermore, the models account for heat transfer across a finite temperature difference. In this context, a positive temperature difference is ensured in all heat exchangers (recuperator, high temperature storage heat exchanger HEX1, low temperature storage heat exchanger HEX2), at both the cold side and the hot side. It is noted that heat recuperation is very important for the supercritical CO_2 heat pump and gas turbine cycles because it enables high efficiency. In the heat pump cycle, heat is transferred from the high pressure CO_2 at the HEX1 outlet to the low pressure CO_2 at the HEX2 outlet. In the gas turbine cycle, heat is transferred from the low pressure CO_2 at the expander outlet to the high pressure CO_2 at the compressor outlet. The energy balance in the recuperator considers the enthalpy change of the working fluid on both, the high pressure side and, the low pressure side for the heat pump and the gas turbine.

$$\eta_c = \frac{h_{2s}-h_1}{h_2-h_1}; \quad \eta_e = \frac{h_4-h_5}{h_4-h_{5s}}; \tag{1}$$

$$h_{HP3} - h_{HP4} = h_{HP6} - h_{HP1}; \quad h_{GT3} - h_{GT2} = h_{GT5} - h_{GT6}; \tag{2}$$

Thermodynamic states of the cycle are defined:

- Thermodynamic states 3 and 6 of the heat pump cycle are specified by pressure (considering the compressor inlet pressure and the pressure ratio) and temperature (considering the temperature difference on the HEX1 cold side and the temperature difference on the HEX2 hot side).
- Thermodynamic state 1 is defined with pressure and temperature (considering the temperature difference in R hot side). Thermodynamic state 2 is obtained using Eq. (1).
- Thermodynamic state 4 is obtained using Eq. (2).
- Thermodynamic state 5 is obtained using Eq. (1).

Once all the thermodynamic states are defined, the COP is calculated Eq. (3), as well as the temperature difference in R cold side Eq. (4). Based on the temperatures of state 2 and 5 of the heat pump, considering the temperature differences, the temperature of the high temperature storage at the hot side and the temperature of the low temperature storage at the cold side are calculated.

$$COP = \frac{q_{HPout}}{l_{HPc}-l_{HPe}} = \frac{h_{HP2}-h_{HP3}}{(h_{HP2}-h_{HP1})-(h_{HP4}-h_{HP5})} \tag{3}$$

$$\Delta T_{RHPcs} = T_{HP4} - T_{HP6} \tag{4}$$

The thermodynamic states 1', 3, 4, and 6 of the gas turbine cycle are defined with pressure (considering the compressor inlet pressure and the pressure ratio) and temperature (considering the temperature difference in HEX1 cold side and hot side, and the temperature difference in HEX2 hot side and cold side). Thermodynamic state 2 is calculated based on the energy balance in R using Eq. (2). Thermodynamic states 1 and 5 are obtained by Eq. (1). Once all the thermodynamic states are defined, the thermal efficiency is calculated by Eq. (5), as well as the temperature difference in R cold side by Eq. (6) and the R hot side by Eq. (7). The additional heat generated in the gas turbine is calculated using Eq. (8).

$$\eta_t = \frac{l_{GTe}-l_{GTc}}{q_{GTin}} = \frac{(h_{GT4}-h_{GT5})-((h_{GT2}-h_{GT1})}{h_{GT4}-h_{GT3}} \tag{5}$$

$$\Delta T_{RGTcs} = T_{GT6} - T_{GT2} \tag{6}$$

$$\Delta T_{RGThs} = T_{GT5} - T_{GT3} \tag{7}$$

$$\Delta q_{CGT} = h_{GT1'} - h_{GT1} \tag{8}$$

A positive temperature difference needs to be ensured in all heat exchangers at both the hot side and the cold side. If the obtained temperature difference in R is lower than the defined minimum, a different procedure with an implied temperature difference in the R cold side is used to calculate the thermodynamic states. Hereby, the thermodynamic states 1', 4, 6, and 2 of the gas turbine cycle are defined with pressure (considering the compressor inlet pressure and the pressure ratio) and the temperature (considering the temperature difference in HEX1 hot side, the temperature difference in the HEX2 hot side and cold side, and a temperature difference in the R cold side). Thermodynamic states 1 and 5 are obtained using Eq. (1). Thermodynamic state 3 is calculated based on the energy balance in R by Eq. (2). Once all the thermodynamic states are defined, the thermal efficiency is calculated by Eq. (5), as well as the temperature difference in the R hot side by Eq. (7) and the temperature difference in the HEX1 cold side by Eq. (9). The additional heat generated in the gas turbine is calculated using Eq. (8).

$$\Delta T_{HEX1GTcs} = T_{GT5} - T_{GT3} \tag{9}$$

After calculation of the thermodynamic states of the heat pump and the gas turbine cycle, the round-trip efficiency is calculated as the ratio of net mechanical work output of the gas turbine and net mechanical work input in the heat pump, considering the different mass flows in the cycles using Eq. (10).

$$\eta_{round-trip} = \frac{l_{GTe}-l_{GTc}}{l_{HPc}-l_{HPe}} \frac{q_{HPout}}{q_{GTin}} = \eta_t COP \tag{10}$$

3. Results and Discussion

The thermodynamic models of the supercritical CO_2 heat pump cycle and the supercritical CO_2 gas turbine cycle are used in this section to study the effect of various design and operating conditions on the round-trip efficiency of the system. The round-trip efficiency of the system strongly depends on the temperature of the high temperature storage and the temperature of the low temperature storage. For example, when the temperature of the high temperature storage cold side is 800 K and the temperature of the low temperature storage hot side is 400 K, the net work input in the heat pump cycle is 207.1 kJ/kg, whereas the net work output in the gas turbine cycle is 154.4 kJ/kg. The resulting round-trip efficiency of the system is 64.31%. For the gas turbine cycle, a higher temperature difference between the high temperature storage and the low temperature storage results in higher thermal efficiency. However, for the heat pump cycle, a lower temperature difference results in a higher COP. Additionally, the effects of the pressure ratio, turbomachinery isentropic efficiency, and temperature difference in the heat exchangers are assessed.

The influence of the high temperature storage temperature on the thermal efficiency of the supercritical CO_2 gas turbine, COP of the supercritical CO_2 heat pump, and the round-trip efficiency of the system is shown in Fig. 6. Increasing the high temperature storage temperature results in higher thermal efficiency since the gas turbine inlet temperature is increased. However, the COP of the heat pump decreases, since a larger temperature lift needs to be delivered to store the heat at a higher temperature. The joint effect on the system's round-trip efficiency is shown with the yellow line in Fig. 6. An increase from 36.7% to 62.1% is observed for an increase in the high temperature storage temperature from 450 K to 1000 K. Cycles with high temperatures require suitable materials that can withstand the operating conditions of the gas turbine in the gas turbine cycle, the compressor in the heat pump cycle, and the recuperator in both operation modes. Also, such high cycle temperature exceeds the limitation of the molten salts as heat storage media (730°C [17]). In this paper, CO_2 is proposed as the storage medium in both high temperature storage and low temperature storage. This not only allows higher storage temperature but also positively influences the matching of the thermal profiles in HEX1 and HEX2. The increase in the temperature of the high temperature storage at the cold side above 800 K, causes a slower increase in round-trip efficiency (from 57.2% to 62.1%) and significant requirements for suitable materials of the system's components. Hence, a reasonable limit of the temperature of the high temperature storage at the cold side is 800 K. The results in Fig. 6 are shown for a low temperature storage temperature of 430 K, a pressure ratio of 3.75, a compressor isentropic efficiency of 80%, an expander isentropic efficiency of 85%, and a temperature difference in the heat exchangers of 5 K.

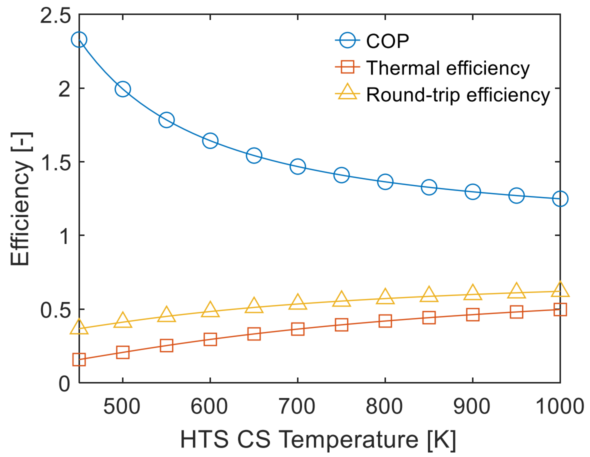

Figure 6 Influence of the temperature at the high temperature storage cold side (C2) on the round-trip efficiency

Similarly to the influence of the high temperature storage temperature, the influence of the low temperature storage temperature on the round-trip efficiency is also assessed (Fig. 7). By increasing the temperature of the low temperature storage at the hot side, the COP of the heat pump increases, whereas the thermal efficiency of the gas turbine decreases. As a result, the round-trip efficiency firstly increases from 57.3% for a temperature of 350 K to 58.5% for a temperature of 400 K. Then, the round-trip efficiency decreases to 43.1% for a temperature of 500 K. The results in Fig. 7 are shown for a high temperature storage temperature of 800 K, a pressure ratio of 3.75, a compressor isentropic efficiency of 80%, an expander isentropic efficiency of 85%, and a temperature difference in the heat exchangers of 5 K.

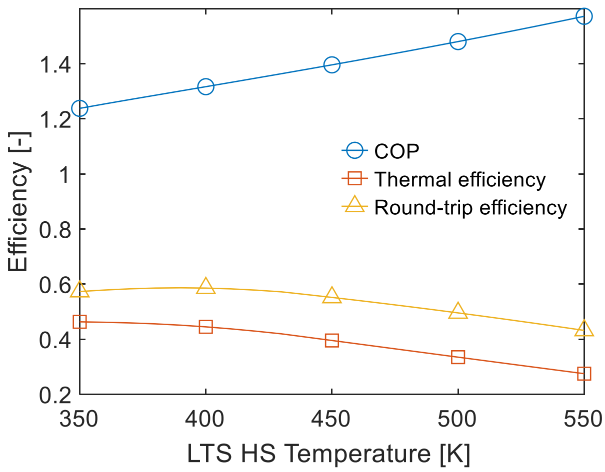

Figure 7 Influence of the temperature of the low temperature storage hot side (H1) on the round-trip efficiency

Increasing the pressure ratio in the cycles results in increase of the round-trip efficiency. The supercritical CO_2 heat pump and gas turbine cycles have a low pressure ratio (2.75 – 3.75). However, both the compressor inlet and outlet pressures exceed the critical pressure of 7.37 MPa. As a result, the system components need to operate under high pressure conditions, which is especially demanding for the recuperator, which must operate at large pressure differentials. Increasing the pressure ratio from 2.75 to 3.25 up to 3.75, for the same temperature conditions at the high temperature storage (800 K) and the low temperature storage (430 K), results in an increase of the round-trip efficiency from 51.8% to 54.9% up to 57.2% (Fig. 8). Additionally, in this analysis the pressure ratio of the heat pump and the gas turbine is equal, however, different pressure ratios can be further analyzed. At a lower temperature of the high temperature storage, the influence of the pressure ratio is higher. The results in Fig. 8 are shown for a low temperature storage temperature of 430 K, a compressor isentropic efficiency of 80%, an expander isentropic efficiency of 85%, and a temperature difference in the heat exchangers of 5 K.

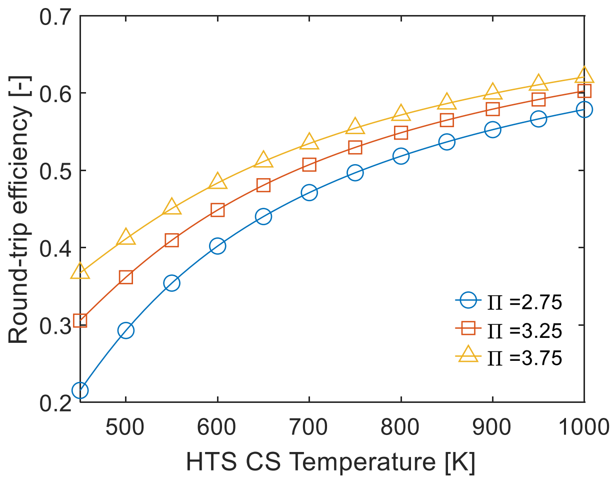

Figure 8 Influence of the pressure ratio on the round-trip efficiency

The turbomachinery isentropic efficiency strongly influences the round-trip efficiency since it directly affects the mechanical work consumed by the heat pump and the mechanical work delivered by the gas turbine. Three configurations were considered, with compressor isentropic efficiency and gas turbine isentropic efficiency of 75% and 80%, 80% and 85%, and 85% and 90%, respectively (Fig. 9). Since the turbomachinery type corresponds to the power capacity, radial and centrifugal turbomachinery are preferred for capacity below around 10 MW, whereas axial turbomachinery is preferred for large-scale systems (> 100 MWe) [18]. Hereby, the lowest turbomachinery isentropic efficiency is considered for small centrifugal compressors and radial-axial gas turbines, and the highest values for large axial turbomachinery. For a temperature of the high temperature storage of 800 K, the round-trip efficiency increases from 49.6% to 57.2% up to 64.2%. In the gas turbine cycle, the influence of the gas turbine isentropic efficiency is greater than that of the compressor isentropic efficiency, since the compression work is small in the supercritical CO_2 gas turbine. In the heat pump cycle, the influence of the compressor isentropic efficiency is greater than that of the gas turbine isentropic efficiency, since the expansion work is small in the supercritical CO_2 heat pump. At a high temperature of the high temperature storage of 1000 K, the highest round-trip efficiency is achieved of 68.5%. The results in Fig. 9 are shown for a low temperature storage temperature of 430 K, a pressure ratio of 3.75, and a temperature difference in the heat exchangers of 5 K.

The influence of the temperature difference in the heat exchanges (recuperator, HEX1 and HEX2) on the thermal efficiency is assessed. The results for different temperature differences (3 K, 5 K, and 10 K) are given (Fig. 10). By decreasing the temperature difference, the recuperation effect increases significantly, resulting in higher thermal efficiency and higher COP. As a result, the round-trip efficiency increases from 54.9% to 57.2%, up to 58.1%, for a temperature of the high temperature storage of 800 K, with the change of temperature difference from 10 K, to 5 K, up to 3 K. However, the minimum temperature difference in the heat exchanger determines the heat transfer area. Lower recuperator temperature difference requires a larger recuperator size.

The design of the recuperator needs to account for the challenging operating conditions at high temperature and high pressure, as well as the requirement for a large heat transfer area, especially when a small temperature difference is needed. Printed Circuit Heat Exchanger (PCHE) has been implemented in the supercritical CO_2 gas turbines, specifically when the gas turbine heat source is nuclear energy or solar energy. Their characteristic manufacturing process, diffusion welding, and material choice (316L SS, Inconel 625/718, Titanium, Hastelloy) enable operation at extremely high temperatures and pressures. Several manufacturers have noted the extreme conditions (up to 1000 °C and 50 MPa) under which PCHEs operate, the extremely small temperature difference (up to 3 K), and the extremely large surface area density (up to 2500 m^2/m^3).

The practical implementation of the proposed supercritical CO_2 heat pump / gas turbine system is conditioned by the development of suitable materials and devices operating at extremely high temperatures and pressures. Turbomachinery research in gas turbine systems has demonstrated that operation at such extreme conditions is possible, reaching gas turbine inlet temperatures of 1600 °C with gas turbine cooling systems [15]. Furthermore, the development of PCHE allows the operation of heat exchangers at extreme pressure differentials. Hereby, the research and development of turbomachinery and heat exchangers provide the foundation for the proposed system.

Another technical challenge is the efficient and safe long-term storage of CO_2 at high temperature and high-pressure conditions. In the literature, several studies show the use of CO_2 storage at high pressure and at high temperature in the order of 800 K, demonstrating the technical feasibility of such operating conditions [19], [20]. This aligns with the thermodynamic results presented in this paper, which indicate that the increase in round-trip efficiency is slower when the temperature in the high temperature storage exceeds 800 K. These results suggest that the use of expensive materials capable of withstanding higher temperatures and high pressures will not lead to substantial performance benefits. Still, to identify the corresponding practical design trade-offs, a technoeconomic analysis is needed, which is left for future work. Although the storage of CO_2 at high pressure and temperature, intended for utility-scale energy storage, is challenging, the experience and available equipment for compressed air energy storage and compressed CO_2 energy storage [20] (which are commercially available as a grid-scale energy storage solution) provide a relevant technological basis for the development of the proposed storage. Also, safety requirements can be addressed from a system level design perspective. Note that, the round trip efficiency demonstrated in this paper emerges from the heat pump / gas turbine cycle and remains relevant regardless of whether the storage medium is molten salts or CO_2. Molten salts reduce the high-pressure storage requirement. Additionally, compared to compressed fuel storage (e.g., compressed hydrogen), the proposed system has comparatively fewer safety issues because no flammable gas is stored. However, further system-level design should involve the integration of the proposed system into hybrid energy storage solutions, whereby chemical energy storage is used to provide long-term storage and is complemented with the supercritical CO_2 system that provides short-term (e.g. weekly), high-efficiency storage. In this case, the storage volume, pressure, and associated safety concerns are reduced.

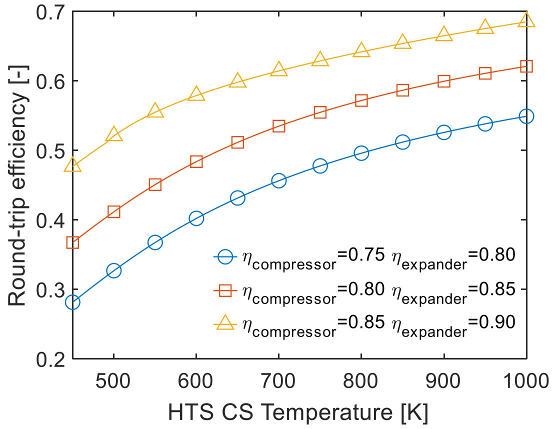

Figure 9 Influence of the turbomachinery isentropic efficiency on the round-trip efficiency

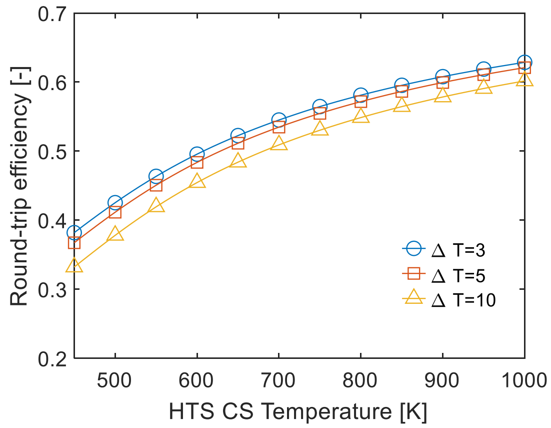

Figure 10 Influence of the temperature difference in heat exchangers on the round-trip efficiency

4. Conclusions

An original energy storage system consisting of a supercritical CO_2 heat pump to convert excess electrical energy into high temperature heat, and a supercritical CO_2 gas turbine to convert the high temperature heat back to electrical energy is presented. Hereby, CO_2 or molten salts are proposed as the storage medium in the low temperature storage and the high temperature storage. The thermodynamic models of the supercritical CO_2 gas turbine and the supercritical CO_2 heat pump are developed in MATLAB/CoolProp. Both the heat pump cycle and the gas turbine cycle, as well as both the low temperature heat storage and the high temperature heat storage, are integrated. A positive temperature difference is maintained in all heat exchangers at both the hot and the cold sides. The results for the assessment of the round-trip efficiency of the system under different design and operation conditions lead to the following conclusions:

- Increasing the high temperature storage temperature at the cold side results in higher thermal efficiency and a lower COP. The round-trip efficiency increases from 36.7% to 62.1% for a temperature increase from 450 K to 1000 K. Due to the limitation of the maximum temperature, a high temperature storage temperature of 800 K is recommended, leading to a round-trip efficiency of 57.2%.

- Increasing the low temperature storage temperature at the hot side results in lower thermal efficiency and a higher COP. The round-trip efficiency slightly increases from 57.3% to 58.5% for a temperature increase from 350 K to 400 K and then decreases towards 43.1% for 500 K.
- The pressure ratios of the proposed cycles are low. An increase in the pressure ratio from 2.75 to 3.75, for the temperature conditions at the high temperature storage of 800 K, causes a round-trip efficiency increase from 51.8% to 57.2%. For lower temperatures, the influence of the pressure ratio is more pronounced.
- The turbomachinery isentropic efficiency strongly influences the round-trip efficiency. For the three considered configurations, with compressor isentropic efficiency and gas turbine isentropic efficiency of 75% and 80%, 80% and 85%, and 85% and 90%, a round-trip efficiency of 49.6%, 57.2% and 64.2% is achieved, respectively.
- A positive temperature difference is maintained in all heat exchangers. A decrease in the temperature difference from 10 K to 3 K, increases the recuperation effect and the round-trip efficiency from 54.9% to 58.1%.

The presented results support the development and further investigation of the supercritical CO_2 heat pump / gas turbine system as a competitive energy storage technology for decarbonisation of the energy sector.

Nomenclature

l_c – compressor work (kJ kg^{-1})
l_t – expander work (kJ kg^{-1})
q_{in} – specific heat input in cycle (kJ kg^{-1})
q_{out} – specific heat output in cycle (kJ kg^{-1})
η_c – compressor isentropic efficiency (-)
η_e – expander isentropic efficiency (-)
η_t – gas turbine system thermal efficiency (-)
COP – coefficient of performance (-)
h – enthalpy (kJ kg^{-1})
p – pressure (MPa)
s – enthalpy (kJ kg^{-1} K^{-1})
T – temperature (K, °C)
ΔT – temperature difference (K)
Δq – additional heat generated in the gas turbine cycle (kJ kg^{-1})

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