

THERMO-ECONOMIC HEAT EXCHANGER OPTIMIZATION FOR ELECTRO-THERMAL ENERGY STORAGE BASED ON TRANSCRITICAL CO₂ CYCLES

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ABSTRACT

The Electro Thermal Energy Storage (ETES) based on thermodynamic cycles using transcritical CO₂ was first developed by ABB corporate research [1]. The target being to set up a large-scale site-independent electricity storage system. The system has been reviewed and developed further by MAN-ES to an energy management system capable of storing not only electricity, but heat and cold as well.

The paper first presents the basic thermodynamic cycle layout, characterized by its heat pump and heat engine concept. Special attention is paid to the high-pressure (i.e. high-temperature) side of the system, where the rapid changes of sCO₂ in heat capacity result in additional modeling complexity. The major challenge lays in the design of heat exchanger (HEX) that provides the best trade-off between cost and performance. On top of that, the novelty of using the same heat exchanger(s) for both charging and discharging cycles on high and low pressure sides of the cycles represents an interesting solution to improve the performance to cost ratio of the entire system. However, this comes at the expense of potential oversizing of the heat exchanger(s). The paper presents the steady state model developed to simulate both thermodynamic cycles. Using a constrained optimization approach, the solver is guided towards solutions where the required heat exchanger surface for both cycles is similar. The consequences of using the same heat exchanger for both cycles are presented for different ETES configurations, allowing identifying significant investment cost reductions and in some cases even improving the entire system's performance. A significant reduction of up to 50% heat exchanger cost can be reached. Additionally, 14% of plant cost for a given round-trip efficiency can be obtained for certain system configurations.

INTRODUCTION

The need to meet the high expectations governments have set regarding reduction of greenhouse gases (GHG) has

substantially increased the interests in large-scale energy storage systems capable of coping with the fluctuating nature and unpredictability of renewable energy [2-4]. The current established technologies based on rotating turbomachinery to face this growing need are Pumped Hydroelectric Storage (PHS) and Compressed Air Energy Storage (CAES). These technologies are however constrained by the topology of the site in which they are installed. The Electro Thermal Energy Storage (ETES) developed in its first stages by ABB corporate research [1] allowed to store large amounts of electricity without relying on any geological constraint. The main thermodynamic performance indicator for electrical storage systems is the so-called roundtrip efficiency (η_{RT}) given by Equation 1.

$$\eta_{RT} = \frac{\text{electric energy provided during discharging}}{\text{electric energy consumed during charging}} = \frac{E_{out}}{E_{in}} \quad (1)$$

Going one step further, the system could also be conceived for heating and cooling applications. The temperature ranges (0°C – 180°C) of the ETES system allows to consider it as a tri-generation energy management system, providing heat, cold and electricity on demand to various process industries or large scale utilities. A schematic representation of a typical ETES plant is presented in Figure 1.



Figure 1. Schematic representation of the tri-generation energy management system: MAN ETES

Performance indicators defined to characterize the efficiency of cooling and heating applications are hot and cold coefficients of performance (COP) defined by Equations 2 and 3 respectively.

$$COP_{hot} = \frac{Hot_{DH}}{E_{in}} \quad (2) \quad COP_{cold} = \frac{Cold_{DC}}{E_{in}} \quad (3)$$

Previous studies of ETES based on transcritical CO2 cycles

The basic layout of the ETES process and component design as well as the preliminary economic study was firstly presented in [1]. CO2 (R744 in refrigerant nomenclature) was selected working fluid due to its suitable thermo-physical properties, high energy density and low critical point ($T_c = 31.1 \text{ }^\circ\text{C}$, $P_c = 73.8 \text{ bar}$). A heat pump (HP) in which the CO2 is compressed by a compressor (COMP) beyond the critical point characterizes the charging cycle. Making use of HEXs, the sCO2 is cooled down and afterwards expanded to a sub-critical state using a work-recovery expander (EXP) instead of a throttling valve to improve the COP of the heat pump (HP). An isothermal evaporation closes the cycle. The discharging cycle on the hand has a similar configuration as a heat engine (HE). A pump (PUMP) will pressurize the fully-condensed subcritical CO2 up to a supercritical state. Making use of the stored heat during the charging cycle, the sCO2 will be heated up and afterwards expanded through power turbines (TURBs) that will drive the generator to provide electricity to the grid. An isothermal condensation of the subcritical CO2 will close the cycle. Typical state point diagrams of both cycles are found in Figure 2.

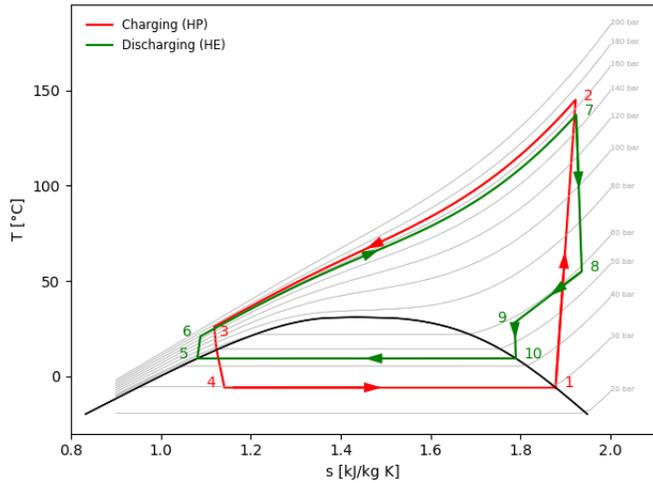


Figure 2. Temperature-entropy diagram of the ETES transcritical CO2 cycles

The possible storage technologies were also studied in [1], where it became obvious that the cold side of the cycle should use a phase changing material in order to make use of an isothermal storage. Ice was selected as cold storage material due to its wide use in the industry as well as its low price. On the hot side of the system, the gliding nature of the temperature profile of sCO2

(see Figure 2) led to select sensible heat storage in order to reduce the exergy losses between charging and discharging cycles. The excellent heat transfer properties of water combined with its low cost and temperatures in which the ETES system operates made it the ideal medium. The number of hot tanks as well as different cycle configurations was further studied in [5-6], where a maximum $\eta_{RT} = 62.3\%$ was obtained using a Pinch Analysis approach.

The importance of investment costs is another crucial aspect to consider in detail when it comes to building such a system when selecting the system's intensive and extensive parameters. Using the most promising cycle configurations from [5-6], the optimal trade-off between thermodynamic performance and investment costs of the entire system were analyzed in [7]. In order to estimate equipment costs, sizing factors of each component (HEX surface, shaft power, etc) were assumed as the main cost drivers based on interpolation data from selected equipment suppliers.

Use of same heat exchanger for charging and discharging cycles

Taking advantage of the fact that charging and discharging cycles never operate at the same time, the cost-reduction potential of using the same components for both cycles should be considered. As mentioned in [7], using the HEXs between CO2 and storage media for both cycles becomes particularly interesting and has significant impact on total plant cost (CAPEX). This solution requires the flow to be reversed on both sides of the HEX with the target and main challenge to match the required area (A_{HEX}) for the charging case with that of the discharging. Considering that A_{HEX} is calculated as shown in Equation 4, the only component-specific parameter is the overall heat transfer coefficient (k), since Q_{HEX} and MTD (Equation 5) are provided by the process streams boundary conditions. Please note the use of MTD throughout this study instead of the traditional $LMTD$ in order to consider heat exchanges with non-constant thermal capacities process streams. Assuming a total HEX heat duty Q_{HEX} , a constant k through the entire HEX and discretizing the component into n segments of the same duty Q_i , the MTD is calculated as presented in Equation 5.

$$A_{HEX} = \frac{\dot{Q}_{HEX}}{k \cdot MTD} \quad (4) \quad \frac{\dot{Q}_{HEX}}{MTD} = \sum_{i=1}^n \left(\frac{\dot{Q}_i}{LMTD_i} \right) \quad (5)$$

In [7], all of the HEX between CO2 and storage medium were dimensioned assuming $k = 1600 \text{ W m}^{-2} \text{ K}^{-1}$, considered to be an average value for sCO2-water HEX [8]. Assuming the same k represents a modelling simplification advantage. However, the deltas found between predicted and real A_{HEX} result in strong over/under dimensioned HEXs that will not provide the expected

outlet temperatures. Indeed, k depends on a variety of parameters of different nature as shown in Equation 6.

$$\frac{1}{k} = \frac{1}{h_{hot}} + \frac{l_{wall}}{\lambda} + \frac{1}{h_{cold}} \quad (6)$$

The nature of these parameters is diverse:

- HEX: geometry, material, wall thickness, etc.
- Streams properties: thermal conductivity, viscosity, heat capacity, temperature, pressure, etc.
- Pressure drop through the channels.

The relevance of each of these parameters will vary as discussed later on in this paper.

The fact that k depends on so many parameters makes it necessary to calculate this value for each specific HEX both on charging and discharging cycles. Equation 7 summarizes the challenge of using the same HEX for both cycles.

$$A_{ch} = A_{dch} \Rightarrow \frac{\dot{Q}_{ch}}{k_{ch} \cdot MTD_{ch}} = \frac{\dot{Q}_{dch}}{k_{dch} \cdot MTD_{dch}} \quad (7)$$

In other words, the model's goal will be to approximate as much as possible both HEX surfaces by a combination of \dot{Q}_i , MTD_i and k_i for each cycle. The heat duty ratio ($\dot{Q}_{dch}/\dot{Q}_{ch}$) between charging and discharging depends mainly on the process intensive and extensive parameters, namely the charging and discharging times (t_{ch} and t_{dch} respectively) as well as charging and discharging power.

Additionally, the ratio between both pressure drops through the HEX ($\vartheta = \Delta P_{dch} / \Delta P_{ch}$) must also be estimated. For a given geometry, ϑ depends mainly on the different friction coefficients and velocities of the streams. Parameters such as mass flow ratio (δ), viscosity and density of the streams play therefore a crucial role in the value of ϑ .

This paper first defines realistic models for the prediction of k and ϑ . By means of constrained optimization the different HEXs surfaces are matched for both charging and discharging operations. Finally, the results allow to see the advantages in terms of cost to performance ratio of using the same HEX for both cycles for different cycle configurations and ETES boundary conditions. A comparison using different HEX is performed as well as benchmark configuration.

HEAT EXCHANGERS OPERATING CONDITIONS

There are two main sorts of HEX used for the ETES cycles, namely the low and high-pressure HEX. The type of technology used for each group of HEX is different since they operate at different operating conditions.

Low-pressure side heat exchanger

This HEX is the so-called *COLD HEX*. The main purpose of this heat exchanger in the ETES process is:

- Produce ice during charging cycle by evaporating CO₂ in subcritical state (4 → 1 in Figures 2)
- Melt ice during discharging cycle by condensing CO₂ in subcritical state (10 → 5 in Figures 2)

The heat exchange between CO₂ and ice storage is done by means of an ethylene glycol-water loop. Throughout this paper, a 25% ethylene glycol in water by mass brine ($\phi_{glyc} = 25\%$) is considered. During charging, the brine would cool down by evaporating the CO₂ and afterwards heated up by freezing the water inside the ice storage tank. On the other hand, during discharging, the brine would be heated up by the condensing CO₂ and afterwards cooled down by melting the ice formerly stored during charging cycle. Figure 3 shows a typical enthalpy – temperature of an ETES COLD HEX. Figure 3 is a clear example of the philosophy behind this paper, where $\dot{Q}_{dch} > \dot{Q}_{ch}$ is compensated by smaller temperature differences in the charging case with respect to the discharging.

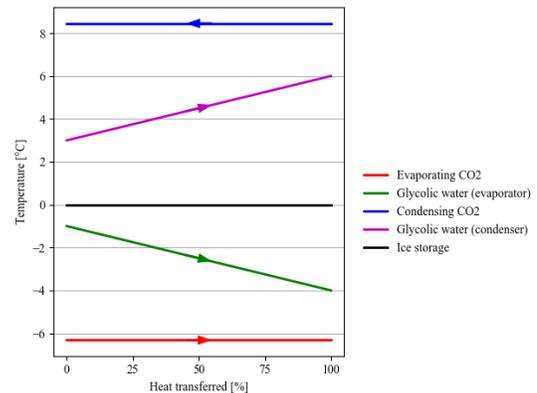


Figure 3. COLD HEX thermal profiles for charging and discharging cycles

The operating conditions of this HEX are bounded by maximum condensing pressure of CO₂. Assuming a maximum saturating pressure $p_{sat} = 50$ bar ($T_{sat} = 14.28$ °C), semi-welded HEX were considered to provide the best performance-to-cost ratio.

The duty of the HEX is determined by both the saturation temperature (i.e. pressure) and the vapor quality x at which the CO₂ enters and exits the HEX. Limitations are however imposed in order to guarantee the integrity and performance the turbomachinery upstream and downstream of the HEX. For instance, in the discharging (i.e. condensing) cycle the CO₂ enters at saturated vapor ($x=1$) and will exit at saturated liquid ($x=0$). In the charging cycle (i.e. evaporating), the quality at the outlet of the evaporator is fixed at saturated vapor ($x = 1$) to avoid

droplets at the COMP inlet. On the other hand, the inlet of the evaporator is a degree of freedom of the process depending solely on the EXP characteristics (see Figures 1, 2). An upper bound value of $x = 0.4$ was imposed to be realistic and this will be assumed throughout this study.

High-pressure side heat exchanger(s)

These HEXs are the so-called *HOT HEX*. The main purpose of these heat exchangers is:

- Heat up water in the hot storage tanks by cooling sCO₂ downstream the COMP (2 → 3 in Figures 1,2)
- Use the hot water to heat up sCO₂ upstream of the TURB (6 → 7 in Figures 1,2)

The first major challenge when it comes to defining the boundary conditions and design of these HEXs relies on the fact that the working fluid through these HEXs is CO₂ at supercritical conditions. As a consequence, the variation in heat capacity c_p during isobaric cooling (2 → 3 Figure 2) or heating (6 → 7 Figure 2) of sCO₂ requires special attention in order to reduce as much as possible the exergetic irreversibilities between charging and discharging cycles. Although early studies of the system suggested the use of only 2 tanks with different temperatures [1], later studies pointed out the performance enhancement opportunity of adding intermediate tanks that would follow the curves for both charging and discharging cycles [5-7]. A schematic representation of the multiple storage tank principle is shown in Figure 4. The ideal water thermal profile would have the same temperature difference between charging and discharging profiles. The only way of achieving this would be with an infinite number of tanks that would be able to adapt to the change in c_p of the sCO₂ by changing the water mass flow from one tank to the subsequent one. A realistic water thermal profile will count with a finite number of tanks n_{tanks} , would be divided into $n_{tanks} - 1$ segments, each of them with a different thermal capacity ($C_w = c_{p,w} \cdot m_w$) of the water. The optimum n_{tanks}

is a compromise between exergy losses and CAPEX of the ETES system. Figure 2 shows how the change in “shape” of the isobars becomes more significant approaching the triple point. The lower the HOT HEX pressure level (i.e. closer to triple point) the more sensitive c_{CO_2} and thus more penalizing since a higher number of tanks is required to follow the sCO₂ thermal profile compared to the subcritical CO₂ exchange, where the pinch point ΔT_{pp} was found at the extremities of the HEX, the change in c_p during isobaric heat exchange can lead to ΔT_{pp} in the middle of the HEX (see Figure 5).

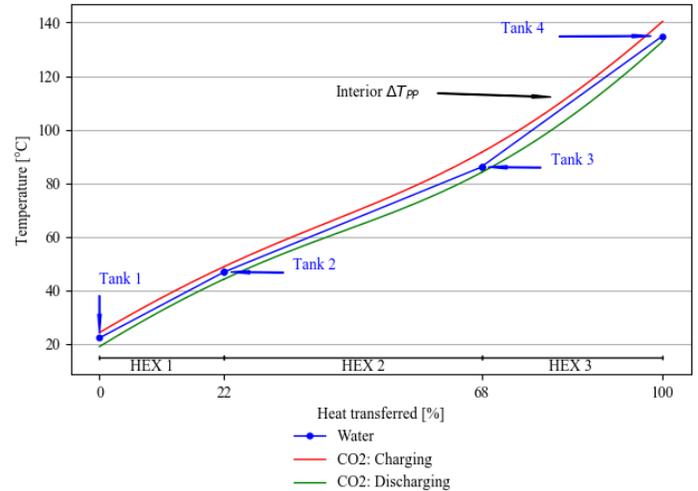


Figure 5. HOT HEX thermal profiles for charging and discharging cycles in a 4-tank ETES configuration

The main challenge of the high pressure side is then to determine the temperature of the tanks as well as the heat duty assigned to each of the HEX. This duty is determined by the volume flow leaving and entering each of the tanks. To guarantee the tanks filling and emptying at the same rate, the same ratio of water volume flow δ_{H_2O} through each of the HEX should be kept as shown in Equation 8 below. Since the water side of the HEX is

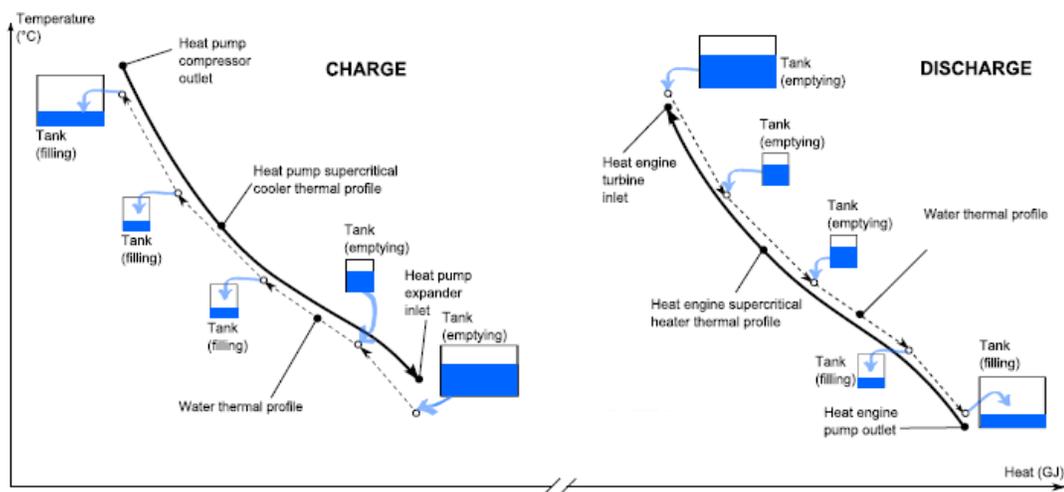


Figure 4. Multiple hot storage tanks principle for a 5-tank ETES configuration [7]

at the same operating conditions in both cycles, the ratio of volume flows is equivalent to δ .

$$\delta_{H2O} = \left(\frac{\dot{v}_{ch}}{\dot{v}_{ach}} \right)_i = \left(\frac{\dot{m}_{ch}}{\dot{m}_{ach}} \right)_i \quad \forall i = 1, \dots, n_{HEX} \quad (8)$$

The type of HEX was selected both on the operating conditions of the sCO₂ (80-270 bar, 10 – 200 °C) and the requirement of having small ΔT_{pp} between both streams to reduce exergetic losses. Among the potential technologies that could cope with this criteria, printed circuit heat exchangers (PCHE) was considered to provide the best compromise between performance and footprint compared to other traditional solutions such as shell and tube HEX.

ETES CYCLE OPTIMIZATION MODEL

The complexity of selecting the numerous parameters defining the ETES requires the development of models capable of simulating and predicting the sizing and performance of all key components constituting the system. The retained ETES configuration is usually selected after an optimization of performance against cost. Therefore, the prediction of cost and performance integration of each key components is fundamental to allow for the best cycle definition. Moreover, the models shall permit the required sensitivity to modification of thermodynamic cycle parameters such as mass flows, temperatures and pressures. To address this challenge, a steady-state ETES model has been developed in python [9] capable of carrying out single/multi-objective and multi-variable optimization. It is based on a sequential calculation solver, so the design is achieved starting from a fixed point and proceeds solving mass and energy balances sequentially component by component following the flow and defining for each of them boundary conditions and applying design rules and component limitation. NIST REFPROP [10] is used to calculate the thermodynamic and transport properties of the different fluids involved in the system. The nature of the software makes the choice of degrees of freedom a crucial step to achieve a feasible and meaningful design. It is characterized by an objective function subject to a set of constraints (linear and non-linear) that the optimizer should satisfy to come to feasible ETES configurations. The general formulation of a constrained optimization problem is presented in Equation 9:

$$\begin{aligned} \min_x & \left(f(x) + \sum_i Pen_i(x) \right) \\ \text{subject to} & \quad b_j \leq g_j(x) \leq c_j \quad \forall j = 1, \dots, n_{constr} \quad (9) \\ & \quad l_b \leq x \leq u_b \\ & \quad x_k \in \mathbb{R} \quad x_l \in \mathbb{Z} \end{aligned}$$

Under this formulation, for a set of decision variables x the optimizer will try to minimize the objective function $f(x)$. The decision variables can be either real (x_k) or integers (x_l) and are limited within an upper and lower bound (u_p and l_b respectively). By including the sum of penalties $Pen(x)$ in the minimization

problem, the optimizer is guided towards feasible solutions by penalizing any violations on constraints. The value of the penalty is proportional to the violation. A weighting factor is also imposed in order to address differences in the magnitudes of each penalty as well as to address the severity of this penalty (more critical constraint violations will have greater weighting factors). Two are the main types of penalties applied to the ETES optimization problem:

- Cycle unfeasibility: one or both cycles are not closed.
- Component limitation: the calculated solution leads to an unfeasible component (either rotating machinery or HEX)

Thermodynamic modelling

The independence of each individual component of the system allows to prioritize the complexity of the components based on their sensibility to modification of input boundary conditions.

Turbomachinery modelling

For the purposes of this paper, a zero-dimensional model for turbomachinery key components was applied and considered as accurate enough. The assumed isentropic efficiencies (η_{is}) of these machines are shown in Table 1. The generators and motors were assumed to have an efficiency of 99% from electric to mechanical power and vice versa.

Machine	Isentropic efficiency (η_{is})
COMP	81%
EXP	90%
TURB	92%
PUMP	86%
Auxiliary Pumps	85%

Table 1. Assumed turbomachinery isentropic efficiencies

Heat exchanger modelling

The major modelling complexity relies in the challenge of using the same HEX for both cycles, which made it necessary to address the HEX modelling with more care than for the rotating machinery. The use of zero-dimensional models as done in [5-7] can lead to large discrepancies after comparison with realistic supplier data in A_{HEX} (defined by k). A strict technical approach would be to develop a comprehensive geometrical model of each HEX as done in [11]. However, the lack of detailed geometrical information as well as the current necessity of rapid computational time led to choose a mid-way solution. One-dimensional models for both HOT and COLD HEX was considered to be accurate enough to cope with the sensitivity of all relevant input parameters.

For each HEX, a set of parameters is required to calculate k (i.e A_{HEX}). Given the differences in both conditions and type of HEX used, COLD and HOT HEX were treated separately. First of all,

the parameters (both intensive and extensive) having the biggest influence on k and ΔP_{HEX} were identified. On a second step, the impact of each of these parameters on k and ΔP_{HEX} was determined using polynomial regression models based on HEX suppliers' data for different operating conditions. The quality of the approximation of these models is determined by the adjusted Coefficient of Determination (CoD_{adj}) [12]. The higher variety of modeled HEX designs the higher the CoD_{adj} . Additionally, the importance of each variable on the estimation of the output is given by the Coefficient of Importance (CoI) [12]. Table 2 shows the results of the approximation models used to estimate k .

	CO2 state	Variables	CoI	CoD _{adj}
HOT HEX	Supercritical	$\Delta P_{\text{H}_2\text{O}}$	81%	94.9%
		ΔP_{CO_2}	49%	
		λ_{CO_2}	26%	
		μ_{CO_2}	25%	
		$c_p_{\text{CO}_2}$	25%	
		$\lambda_{\text{H}_2\text{O}}$	20%	
		$\mu_{\text{H}_2\text{O}}$	19%	
		MTD	0%	
COLD HEX	Subcritical evaporating	ΔP_{glyc}	77%	97.2%
		MTD	65%	
		T_{CO_2}	47%	
		ΔT_{glyc}	24%	
		x_{in}	3%	
		x_{out}	2%	
		ϕ_{glyc}	0%	
	Subcritical condensing	ΔP_{glyc}	76%	96.3%
		MTD	61%	
		ΔT_{glyc}	42%	
		T_{CO_2}	7%	
		ϕ_{glyc}	3%	

Table 2. Regression models for estimating k

The higher complexity of estimating k between sCO₂ and H₂O is evidenced by the higher number of parameters required for HOT HEX compared to the COLD HEX. Additionally, average values of μ , λ on both sides (sCO₂ and H₂O) are included in the regression model. Furthermore, the significant changes in c_p of sCO₂ are also accounted for by including an average of this parameter throughout the entire HEX in the approximation. ΔP from both sides is also considered given its impact on the turbulence. In the COLD HEX, ΔP_{CO_2} is neglected given the negligible value for a semi-welded plate HEX during evaporation and condensation. The large differences in convective heat transfer h (see Equation 6) between subcritical evaporating and condensing CO₂ led to determine different parameters to determine k_{COND} and k_{EVAP} . Finally, MTD is considered in both types of HEX to account for the engineer's design methodology for each specific HEX. As a general design rule, for smaller MTD , higher ΔP values are necessary to increase

k . The conducted regression studies showed higher values of CoD_{adj} when considering this parameter.

The relevance of ΔP in k estimation becomes clear in Table 2, being the parameter having the greatest impact of all for the three regression studies. The selection of optimal HEX design is closely linked to ΔP : while high values of ΔP results in higher pumping power (i.e. smaller η_{RT}), it also increases drastically k (i.e. smaller A_{HEX} , thus smaller CAPEX). The optimal ΔP is then a compromise between cost and performance and its optimal value is selected based on ETES system boundary conditions such as price of electricity. The approach chosen to estimate ΔP_{HEX} leads to more accurate results for the COLD HEX, since operating conditions do not vary that much between the different operating cases and easily predicted on the HEX supplier side. ΔP_{glycol} is determined as a function of the velocity and density of the brine. Maximum absolute errors up to 5% can be expected for both evaporating and condensing cases. Based on supplier data, the negligible values of ΔP_{CO_2} combined with the complexity associated with phase-changing pressure drops led to impose a constant value for both evaporating and condensing cases.

The large changes in operating conditions for the HOT HEX led to take another approach to estimate ΔP_{CO_2} and $\Delta P_{\text{H}_2\text{O}}$. In this case, the HEX is assumed to be design for a specific ΔP_{HEX} for each of the two streams (water and CO₂). The charging case is considered as the limiting case. If the same HEX is used for both cycles, the ΔP_{HEX} on both sides for the discharging case will no longer be imposed, but calculated and proportionally based on the ΔP_{HEX} applied on the charging case. A new equation is therefore required to determine ϑ on both sides of the HEX. If different HEX are used for both cycles, ϑ is obviously not calculated and ΔP_{dch} is specified before the calculation. A summary of the approximation of ϑ as well as the importance of the parameters influencing its value is available in Table 3. The dependency of μ on ΔP via Reynolds number is accounted for in both functions. Since $\theta_{\text{H}_2\text{O}} = 1$ for every HOT HEX, the $\mu_{\text{H}_2\text{O}}$ was selected to estimate $\vartheta_{\text{H}_2\text{O}}$ instead of θ as done for ϑ_{CO_2} .

	Variables	CoI	CoD _{adj}
Water	$\delta_{\text{H}_2\text{O}}$	92%	98.8%
	$\mu_{\text{H}_2\text{O}}$	31%	
CO ₂	δ_{CO_2}	96%	97.4%
	θ_{CO_2}	40%	

Table 3. Regression models for estimating ϑ for HOT HEX

The high values of CoD_{adj} obtained for both k and ϑ estimation confirm the one-dimensional model as accurate enough for the implementation into the ETES process simulation and optimization routine.

Economic modelling

In order to reach a meaningful thermo-economic optimization analysis, the associated component costing model of the plant are of equal importance as the performance models. For this paper, the purchasing cost of the different components included in the system are calculated based on characteristic sizing factor (A_{HEX} , shaft power, volume of storage tanks, etc.). To properly address the cost against performance trade-off, it is crucial to count on reliable data regarding equipment costs. The numerous exchanges with suppliers of the various components allows to have consolidated up-to-date data to estimate with a relatively low uncertainty. Additional costs related to engineering, civil work and procurement are assumed to be a fixed percentage of the total equipment cost.

Modelling approach using same heat exchanger

The current steady state model of the ETES system is conceived to provide after each iteration for a given set of decision variables:

- Process state conditions at inlet and outlet of each component
- Cycle performance (η_{RT} , COP_{hot} , COP_{cold})
- Cost and size of equipment

A_{HEX} for charging and discharging are firstly calculated independently from one another. By applying a constraint where a maximum oversizing is allowed between charging and discharging cases, the optimizer should try in the next iteration to reduce that penalty. The addition of this new constraint reduces the number of non-penalized solutions, resulting in longer computational times (i.e. number of evaluations) to obtain optimal ETES systems. Since this penalty is applied to each HEX, increasing the number of tanks (i.e. HOT HEX) reduces the number of non-penalized solutions and ultimately longer computational time is required by the optimizer to find optimal ETES systems. Maximum oversizing of 15% for both HOT and COLD HEX was considered to be an adequate compromise between computational time and accuracy of the solution inside the conventional design margin applied by suppliers. The penalty assigned to an oversizing higher than the abovementioned value is proportional to the committed violation.

RESULTS AND DISCUSSION

The benefits of using the same HEX are evaluated comparing different ETES configurations and system boundary conditions. This evaluation is carried out launching 36 single-objective optimization runs which target being to maximize η_{RT} . ETES designs with thermal export are not evaluated in this study, therefore COP_{hot} and COP_{cold} are not included in the objective function. To evaluate a broad spectrum of ETES configurations, the following four parameters are imposed for each of the runs:

- Using same and different HEX (x2)

- Different number of hot storage tanks: 2 and 5 (x2)
- Different minimum allowed ΔT_{PP} inside HEX: 1, 2 and 4 K (x3)
- Different β : 0.25, 0.5, 1 (x3)

The considered optimization variables are:

- Evaporation temperature [$^{\circ}C$]
- COMP outlet pressure [bar]
- Condensation temperature [$^{\circ}C$]
- Pressure difference between COMP and PUMP outlet pressures [bar]
- δ_{CO2} [-]
- δ_{H2O} for HOT HEX [-]
- Heat distribution among HOT HEX [-]
- Hot storage tank temperatures [$^{\circ}C$]

A total of 8 optimization variables is considered for the simplest case with only 2 tanks. Two additional optimization variables must be considered (heat distribution and tank temperature) should be considered by adding an additional tank. As extensive variables, all optimization runs are characterized by a charging power $P_{ch} = 8.5$ MW during 8 hours.

Additionally, a summary of the optimization set-up for the different runs is presented in Table 4. To guarantee more reliable optimization results, the smaller the number of non-penalized solutions, higher number of iterations are imposed as stopping criteria. This results in longer computational times for cases using the same HEX as well as cases with higher number of hot storage tanks.

Approach	Number of storage tanks	Number of iterations	Population size
Same HEX	2	3500	50
	5	5000	
Different HEX	2	2000	
	5	3500	

Table 4. Single-objective optimization run set-ups

Figure 6 offers good summary of the cost and performance of the different optimization runs, separating the results into cases using the same and different HEX. The CAPEX of the different solutions is displayed as the cost reduction opportunity of each of the solutions with respect to the most expensive of all. The results of Figure 6 confirm that the covered spectrum of ETES configurations is quite broad, varying both in cost and performance. Looking closely at the solutions using the *different HEX* approach, six types of solutions can be clearly identified, one per each combination of n_{tanks} (x2) and minimum allowed ΔT_{PP} (x3). Since no component is shared between both cycles, the optimum ETES system in terms of performance does not depend on β , which defines the size of the discharging cycle key

components. Therefore, for a given η_{RT} , the CAPEX is reduced as β increases when using different HEX on both cycles.

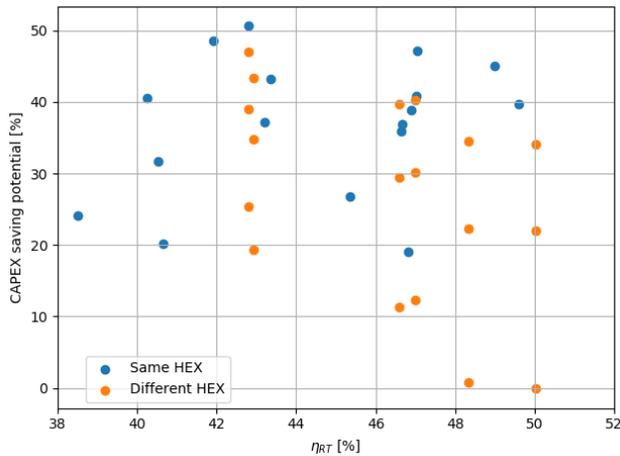


Figure 6. CAPEX saving potential against η_{RT} using same and different HEX for several ETES configurations with $t_{ch} = 8h$ and $P_{ch} = 8.5 MW$

To better understand the benefits derived from using the same HEX approach, solutions with similar configurations (n_{tanks} , ΔT_{PP} and β) should be compared separately since each of them have different costs and performances. CAPEX per discharged electrical energy provides the thermo-economic performance of the ETES system. The difference of this value is compared in Figures 8 and 9 for the different variables defining the considered ETES configurations. Maximum ETES plant cost per discharging electricity of up to 14% are expected as a result of using the same HEX for both cycles. The identification of the most promising cases is carried out by evaluating each of the three parameters individually.

Minimum allowed pinch point (ΔT_{PP})

Unlike n_{tanks} and β , which play an important role in the costing of other key ETES components, ΔT_{PP} only impacts the cost and performance of the HEX. This is evidenced by the greater influence of this parameter on CAPEX saving potential per discharged electricity shown in Figures 7 and 8. Since the impact of HEX cost is directly proportional to the inverse of MTD (see Equation 4), smaller ΔT_{PP} result in more expensive HEX and a higher portion of the total plant cost is assigned to this component. The results of Figure 7 and 8 allow to confirm the higher thermo-economic benefit of the *same HEX* approach for ETES configuration with smaller ΔT_{PP} .

Time ratio (β)

Contrary to *different HEX* approach, if the HEXs are used for both charging and discharging operations, β plays a crucial role in thermodynamic performance of the system. Configurations with small values of β area characterized by high discharging

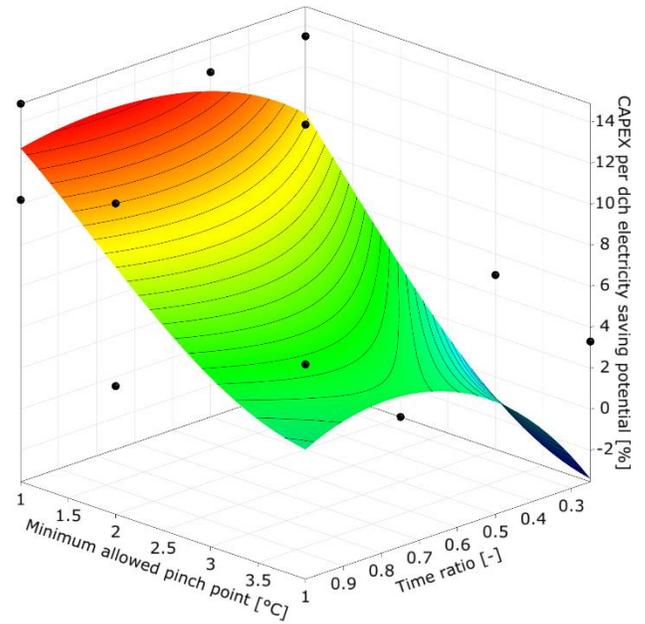


Figure 7. Response surface of CAPEX saving potential per discharged electrical energy as a function of minimum allowed ΔT_{PP} and β

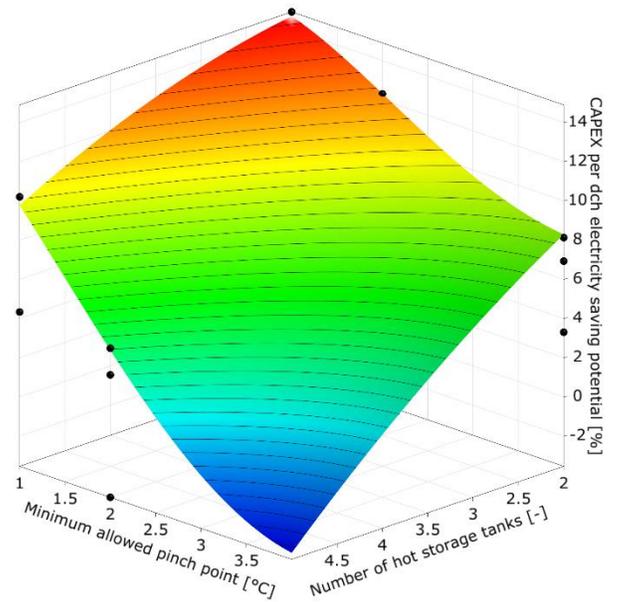


Figure 8. Response surface of CAPEX saving potential per discharged electrical energy as a function of minimum allowed ΔT_{PP} and n_{tanks}

cycle sizes (i.e. high δ) to compensate for the small t_{dch} . Since δ determines Q_{ch}/Q_{dch} , the MTD of the smallest duty is reduced to find solutions with minimum HEX oversizing. A smaller value of β results in:

- Closer charging and discharging thermal profiles (see Figure 5) leading to fewer exergetic losses take place in the HEXs.

- Higher ΔP due to higher δ on both sides of the HEX

The compromise between these two parameters determines greatly the benefits of using the same HEX approach compared to using two separate components. For the assigned boundary conditions of the optimization runs conducted in this paper, high values of β yield higher thermo-economic performances (see Figure 7).

As seen in Figure 7, the impact of β is smaller than ΔT_{PP} on cost savings per discharging electricity since β also impacts the cost of other equipment, mainly TURB and PUMP. This results in smaller CAPEX reduction when one of the HEX is removed from the cycle.

Number of hot storage tanks (n_{tanks})

ETES configurations with small n_{tanks} reduces the number of total HEX and thus the contribution of this component to total plant cost. On the other hand, the exergetic losses due to the changes in c_p of the CO₂ in the supercritical region increase when n_{tanks} decrease since the water thermal profile is unable to follow the CO₂ charging and discharging profiles (see Figure 5).

The thermo-economic impact of n_{tanks} using the *same HEX* approach is shown in Figure 8. The results show more promising results for ETES configurations with smaller number of hot storage tanks. These results are however highly influenced by the complexity of finding global minima by the optimizer and thus number of iterations (see Table 4). Higher potential η_{RT} for similar CAPEX are expected for configurations with higher n_{tanks} when increasing the number of iterations.

CONCLUSIONS

The methodology to use the same HEX for charging and discharging process cycles applied to the ETES system and its various configurations was firstly presented. By means of constrained optimization, maximum A_{HEX} difference of 15% appears to lead to a satisfactory compromise between computational time and the prediction of realistic HEX performance.

The importance of k and ΔP on both cost and performance of the overall system confirms the importance of implementing accurate estimations for both parameters. The dependency of k on ΔP on both sides of the HEX is to be carefully considered as shown in Tables 2.

A series of single-objective optimization runs were conducted to evaluate the thermo-economic advantage of using the same HEX for both charging and discharging cycles in steady-state conditions. From a cost perspective, the higher the contribution of HEX on total ETES plant cost, the more interesting becomes to use the same component for both cycles. On the other hand, from a performance perspective, the increase of cycle

complexity limits the number of possible ETES solutions for given set of system boundary conditions. Among the considered parameters that define the ETES system, ΔT_{PP} proved to be most decisive when evaluating the possibility of using the same HEX for both cycles. Smaller values of ΔT_{PP} represent the most promising cases, given the greater portion of the cost allocated for the HEXs. Additionally, β values between 0.6 and 1 show great potential of applying the same HEX approach in an ETES system. Maximum CAPEX per discharged electricity reductions of 14% are expected for ETES configurations with $n_{\text{tanks}} = 2$, $\beta = 0.7$ and minimum allowed $\Delta T_{PP} = 1\text{K}$. For particular ETES configurations, favorable thermo-economic are reached by using different HEX for both charging and discharging cycles. Nevertheless, the reduction in footprint as a result of using the same component should be considered, making it crucial to use the same HEX for both cycles to obtain the most compact ETES plant possible.

The behavior of the HEXs in transient and off-design operating conditions should be considered in a further step for a more comprehensive assessment and to eventually identify other component limitations still unknown at this stage.

NOMENCLATURE

A_{HEX}	Heat Exchanger Surface
CAPEX	Capital Expenditure
COMP	Compressor
COP_{hot}	Hot Coefficient of Performance
COP_{cold}	Cold Coefficient of Performance
h	Convective heat transfer
ρ	Density
μ	Dynamic Viscosity
θ	Dynamic Viscosity Ratio for a specific HEX (discharging/charging)
ETES	Electro-Thermal Energy Storage
EXP	Hydraulic Expander
HE	Heat Engine
HEX	Heat Exchanger
HP	Heat Pump
k	Overall Heat transfer coefficient
δ	Mass flow ratio (discharging/charging)
$LMTD$	Logarithmic Mean Temperature Difference
\dot{m}	Mass flow
MTD	Mean Temperature Difference
OPEX	Operating Expenses
TURB	Power Turbine
$Pen_i(x)$	Penalty i for a set of input parameters x
ΔT_{PP}	Pinch Point of Heat Exchanger
ΔP_i	Pressure drop inside Heat Exchanger i
ϑ	Pressure drop ratio in heat exchanger (discharging/charging)
PUMP	CO ₂ Pump
φ_{glyc}	Composition of ethylene glycol in water by mass brine

η_{RT}	Round-trip Efficiency
λ	Thermal Conductivity
t_{ch}	Charging time
t_{dch}	Discharging time
β	Ratio between discharging and charging times (t_{dch}/t_{ch})
v	Volumetric flow

ACKNOWLEDGEMENTS

The writers would like to recognize the support and preliminary studies carried out by ABB which have set the baseline for the further development of the ETES system.

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Published in: 3rd European sCO2 Conference 2019

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DOI: 10.17185/duepublico/48917

URN: urn:nbn:de:hbz:464-20191004-160339-4



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