

1 Trigenerative micro compressed air energy storage:
2 concept and thermodynamic assessment

3 Andrea L. Facci^{a,*}, David Sanchez^b, Elio Jannelli^c, Stefano Ubertini^a

4 ^a*School of Engineering, University of "Tuscia", 01100 Viterbo, Italia.*

5 ^b*School of Engineering, University of Seville, 41092 Seville, Spain.*

6 ^c*Department of Engineering, University of Napoli "Parthenope", 80143 Napoli, Italia.*

7 **Abstract**

8 Energy storage is a cutting edge front for renewable and sustainable energy
9 research. In fact, a massive exploitation of intermittent renewable sources, such
10 as wind and sun, requires the introduction of effective mechanical energy storage
11 systems.

12 In this paper we introduce the concept of a trigenerative energy storage
13 based on a compressed air system. The plant in study is a simplified design of
14 the adiabatic compressed air energy storage and accumulates mechanical and
15 thermal (both hot and cold) energy at the same time. We envisage the possibility
16 to realize a relatively small size trigenerative compressed air energy storage to
17 be placed close to the energy demand, according to the distributed generation
18 paradigm. Here, we describe the plant concept and we identify all the relevant
19 parameters influencing its thermodynamic behavior. Their effects are dissected
20 through an accurate thermodynamic model. The most relevant technological
21 issues, such as the guidelines for a proper choice of the compressor, expander and
22 heat exchangers are also addressed. Our results show that T-CAES may have
23 an interesting potential as a distributed system that combines electricity storage
24 with heat and cooling energy production. We also show that the performances
25 are significantly influenced by some operating and design parameters, whose
26 feasibility in real applications must be considered.

27 *Keywords:* Compressed Air Energy Storage, Trigeneration, Distributed
28 generation, CHP, Electricity storage

29 1. Introduction

30 Balancing supply and demand of electricity is nowadays a key issue for many
31 countries, due to the increasing penetration of intermittent renewable energy
32 sources (RES) and of distributed generation (DG)[1, 2]. Different approaches
33 are possible to cope with this problem including, updating the power regulation
34 strategy for DG plants, utilizing electrical boilers and heat pumps, modifying the
35 consumer demand of energy, switching to electricity based mobility, fast ramping
36 conventional plant, and updating the electricity distribution infrastructure [1–
37 3]. In this scenario, electricity storage certainly plays a key role especially for a
38 large scale exploitation of intermittent RES, such as wind or solar power [2, 4].

39 Despite only two working applications of Compressed Air Energy Storage
40 (CAES) exist [3, 5, 6], these storage systems claims the greater economical fea-
41 sibility [1, 2], among all the technological alternatives for large scale electricity
42 storage (e.g. pumped hydro and batteries), thanks to their relatively low invest-
43 ment cost per unit capacity [2]. Nevertheless, the assessment of the economic
44 potential of such a technology is still an open research issue [1, 2, 4, 7, 8]. In a
45 CAES the mechanical energy is stored by compressing air from the environment
46 into a high pressure reservoir, through a compressor driven by an electrical mo-
47 tor. When electricity is required from the CAES the air is expanded through
48 a turbine. In case of Diabatic-CAES, air is heated through the combustion
49 of a fossil fuel before expansion [6]. On the other hand, in Adiabatic-CAES
50 (A-CAES), air heating is performed by recovering the heat in the inter-cooled
51 compression and no additional fuel is required [9–13].

52 Most of the literature related to CAES is focused on utility-scale applica-
53 tions [1, 4, 7, 14], that are very large plants located close to the energy produc-
54 tion. Nevertheless, the development of small scale A-CAES (micro-CAES) to be
55 placed close to the energy demand is of great interest [15–17]. Such a solution
56 allows a more tight coupling between the energy storage systems and the energy
57 demand. Moreover, micro-CAES may act also as trigenerative systems, which
58 is to say combined heat, cooling, and power plants (CHCP) by recovering heat

59 after air compression and cooling energy during or after the expansion phase
 60 [15, 16]. Note that trigeneration is possible only placing the CAES close to the
 61 energy demand, given the technical difficulty to transfer thermal and chilling
 62 energy at large distances. Such a plant allows to conjugate the widely recognized
 63 advantages of distributed trigeneration (see for instance [18–20]) with those of
 64 electricity storage. The energy and exergy analyses performed in [16] for dif-
 65 ferent micro-CAES configurations show the effectiveness of combining energy
 66 storage and trigeneration in a single plant, in particular for DG applications.
 67 The ability of such a plant to satisfy the electrical thermal and chilling demands
 68 is also demonstrated in [15]. However, a rigorous analysis of the potential of
 69 such a system in an energy management environment is lacking.

70 In this paper we study a simple plant configuration for a trigenerative CAES
 71 (T-CAES) in order to assess its thermodynamic feasibility and to define its de-
 72 sign guidelines. Moreover, we complement the analysis introducing some tech-
 73 nological considerations on the most relevant aspects, such as the compressors
 74 and expanders technologies, and the heat exchangers design. The plant configu-
 75 ration and operating principle are described in section 2. Therein, the proper
 76 parameters to evaluate the T-CAES performances are also defined. In section 3
 77 we describe the thermodynamic model utilized for the assessment of the plant
 78 performances. A sensitivity analysis on all the relevant parameter influencing
 79 the T-CAES operations is then performed in section 4. First, a default param-
 80 eter set is defined and utilized to determine the reference performances of the
 81 plant. Then, each parameter is systematically varied to assess its influence on
 82 the T-CAES operation and performances. Considering that the design of the
 83 T-CAES rises non trivial technological issues, such as the selection of the most
 84 appropriate fluid machinery as well as the choice and the design of the heat ex-
 85 changers, a discussion of these aspects is reported in section 5, before drawing
 86 conclusions.

87 2. Trigeneration through CAES

88 The concept of trigenerative compressed air storage (T-CAES) [15, 16] de-
89 rives from the adiabatic one [9–13]. The T-CAES simplifies the design of the
90 A-CAES, by eliminating the regenerative air re-heating. The heat removed dur-
91 ing the compression phase is stored and, rather than being utilized to reheat the
92 air at the turbines inlet (see for example [9]), may be used to fulfill a thermal
93 energy demand [15]. As a consequence, air enters the expansion train at low
94 temperature and chilling energy is obtained at the expanders outlet, without the
95 need an inverse cycle. Placing such a plant close to the energy user facilitates
96 the effective utilization of all the energy streams.

97 A schematic of the T-CAES system is represented in Fig. 1. Therein, all the
98 relevant components of the energy storage and release processes are evidenced.
99 When mechanical or electrical energy is available at low cost (i.e. during off peak
100 hours), or for free (i.e. from RES), the compressor train pumps environment
101 air into the high pressure reservoir. In this respect, it is worth noting that the
102 compression technology of choice has a strong influence on delivery temperature
103 to the storage tank and on the efficiency and cost of the system, as noted in
104 section 5.1. The impact of intercooling on delivery temperature deserves also
105 attention considering that a high temperature at the inlet of the high pressure
106 tank (T_{ri}) is not expected to generate significant thermodynamic advantages
107 (see section 4.4), while it could pose significant technological concerns for the
108 mechanical resistance of the high pressure reservoir [10]. As a consequence, the
109 choice of the proper compressor design should result from an accurate optimiza-
110 tion process, accounting for investment as well as current costs and for plant
111 performances. The result of such an optimization is a function of the plant size
112 and utilization factor. Considering the preliminary nature of the present study,
113 here we are assuming a train of generic dynamic compressors. The compression
114 is inter-cooled to reduce the specific compression work and to limit T_{ri} . We
115 further comment that the model proposed here and described in section 3.3 is
116 general, and could be easily extended to any kind of cooled compression vary-

ing the exponent of equation (16). Nevertheless, some hints on the effects of switching to reciprocating compressors are given in section 5.

The heat is removed from the air stream through pressurized liquid water and stored or directly used to meet the thermal energy demand. Since thermal energy is stored and utilized on a daily basis, the heat storage does not pose any particular technological problems, as it may be simply realized through a properly insulated tank.

When mechanical energy is required from the T-CAES, the air stored in the high pressure reservoir is expanded through a re-heated multi-stage turbine train, without usage of additional fuel. As for the compression, the optimal choice for the expansion technology depends on the mass flow rate and expansion ratio, and it is commonly assumed that one could switch to small centripetal turbine whilst still having to use volumetric compressor to obtain high efficiency. Technological aspects related to the expanders technology are addressed in section 5.1. Given that the very high expansion ratios are likely to bring about an important drop in gas temperature, re-heating is performed using low temperature heat from the environment, which is possible only when the air temperature is lower than the environmental temperature (T_{env}). Water, initially assumed at T_{env} , is utilized to reheat the air stream thus producing a flow of chilling energy at lower temperature compared to T_{env} . As for heat, chilling energy can be stored or directly used for space conditioning or other technological usages.

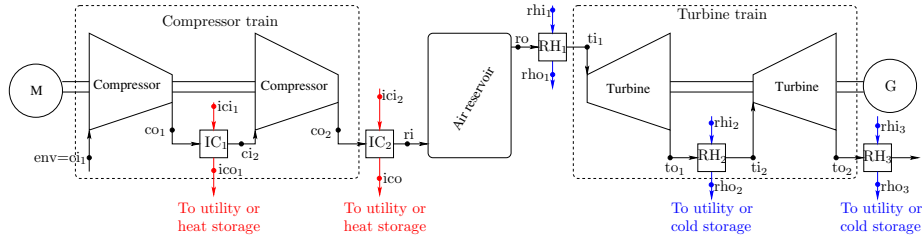


Figure 1: Schematic of the compressed air storage plant in study. Note that, for simplicity, only two compressors and two expanders are represented, though the number of stages could be higher for both trains. M stands for motor, G for electrical generator, IC for inter-cooler, and RH for re-heater.

138 Since the first aim of the T-CAES is to store electrical energy for deferred
 139 usage, the main performance parameter is the electrical efficiency

$$140 \quad \eta_{el} = \frac{W_d}{W_c}, \quad (1)$$

141 which represents the ratio between the electrical energy obtained from the air
 142 expansion, W_d , and the electrical energy spent to pump the air into the reservoir,
 143 W_c . Given the compression and expansion trains power or, equivalently, their
 144 mass flow rate, the storage duration $\mathcal{T}$ is determined by the volume V of the
 145 reservoir, and by the maximum ($p_{\max}$) and minimum ($p_{\min}$) pressures allowed
 146 inside the air tank. To account for the production of thermal and chilling energy,
 147 the exergy efficiency (i.e. the second law efficiency) is defined as follows,

$$148 \quad \eta_{ex} = \frac{W_d + B_t + B_c}{W_c}. \quad (2)$$

149 Note that the exergy associated to electrical work is equal to W_c and W_d for
 150 the charge and the discharge phases, respectively. The exergy associated to the
 151 thermal energy is estimated as [21]

$$152 \quad B_t = \sum_{i=1}^{N_{IC}} Q_i \left(1 - \frac{T_{\text{env}}}{T_{\text{ico}_i}} \right), \quad (3)$$

153 where Q_i is the thermal energy exchanged in the i -th inter-cooler, T_{ico_i} is the
 154 water temperature at the outlet of the i -th inter-cooler, and N_{IC} is the number
 155 of inter-coolers. The exergy of the chilling energy flux is calculated as follows
 156 [21]

$$157 \quad B_c = \sum_{j=1}^{N_{RH}} C_j \left(\frac{T_{\text{env}} - T_{\text{rho}_j}}{T_{\text{rho}_j}} \right), \quad (4)$$

158 where C_j is the chilling energy exchanged in the j -th re-heater, T_{rho_j} is the
 159 temperature at the outlet of the j -th re-heater, and N_{RH} is the number of
 160 re-heaters.

161 3. Thermodynamic model of the T-CAES

162 3.1. Properties of the working fluids

163 In the following thermodynamic analysis air is considered to behave as an
 164 ideal gas.

The specific heats at constant pressure (c_{p_a}) and volume (c_{v_a}) of air are calculated as functions of gas temperature T [22],

$$c_{p_a} = a + bT \quad (5a)$$

$$c_{v_a} = c_{p_a} - R = a^* + bT, \quad (5b)$$

where $a = 959 \text{ J/(kgK)}$, $b = 1.54 \times 10^{-1} \text{ J/(kgK}^2\text{)}$, $R = 288 \text{ J/(kgK)}$ is the gas constant, and $a^* = 671 \text{ J/(kgK)}$.

Water used in the inter-coolers and re-heaters is assumed to be subcooled with a constant specific heat capacity $c_w = 4186 \text{ J/(kgK)}$.

The other thermo-physical properties of water and air such as viscosity, thermal conductivity, and Prandtl number are evaluated using data tabulated in [23] as a function of the fluid temperature.

3.2. High pressure reservoir modeling

The T-CAES operation is divided into three separate phases: (i) charging; (ii) storage; (iii) discharging. In each phase, the time dependent thermodynamic state of air inside the tank is determined by mass and energy conservation.

During the charging phase, a constant mass flow $\dot{m}$ of air is pumped by the compressor train into the high pressure reservoir. Thus, mass conservation reads

$$\frac{d}{dt}M(t) = \dot{m}, \quad (6)$$

where $M(t)$ is the mass of air contained within the high pressure reservoir at time t . Generally speaking, $\dot{m}$ is a function of the pressure ratio and of the rotating speed of the compressor. Under real operating conditions, $\dot{m}$ is ultimately determined by the choice of the compressor control strategy, which, in turn, should be selected to optimally meet the T-CAES energy demand profile. Since the interaction between the demand and the CAES is not addressed in this analysis, $\dot{m}$ is assumed to be constant in time. We further comment that, removing this assumption does not invalidate the analysis reported in section 4.

188 The variation of air temperature in the reservoir T_R is estimated with the
 189 following energy conservation equation

$$190 \quad \frac{d}{dt}T_R = \frac{\dot{m} [h_{ri} - (a^*T_R + \frac{1}{2}bT_R^2)] - \dot{Q}_R}{M(a^* + bT_R)}, \quad (7)$$

191 where $h_{ri} = aT_{ri} + 0.5bT_{ri}^2$ is the enthalpy of the air entering the tank (note
 192 that the reference state for enthalpy is 0 K: $h_{ri}(0\text{ K}) = 0\text{ kJ}/(\text{kgK})$) and $\dot{Q}_R$ is
 193 the thermal power lost through the reservoir walls. T_{ri} is determined by the
 194 control strategy adopted for the inter-coolers and is assumed to be constant
 195 along the entire charging process. The cooling water mass flow rate is adjusted
 196 accordingly to this assumption as described in section 3.4. We comment that
 197 this assumption allows a systematic study of the effects of T_{ri} on the T-CAES
 198 performances and efficiency (see section 4.4) that would be hindered by a time-
 199 varying temperature.

200 Finally, the pressure in the tank is calculated from the ideal gas equation of
 201 state

$$202 \quad p_R = \frac{MRT_R}{V}, \quad (8)$$

203 The system of eqs. (6), (7), and (8) is solved using the second order Runge-
 204 Kutta method, starting from $t = 0$ until $p_R = p_{\max}$. The initial conditions for
 205 eqs. (6) and (7) are: $T(0) = T_{\text{env}}$ and $M(0) = p_{\min}V/(RT_{\text{env}})$.

206 During the storage phase the mass of air constrained within the tank is
 207 constant, and its temperature is reduced due to heat losses through the reservoir
 208 walls, as evidenced by the following equation

$$209 \quad \frac{d}{dt}T_R = \frac{-\dot{Q}_R}{M(a^* + bT_R)}. \quad (9)$$

210 The initial conditions for eq. (9) are determined by the solution of eqs. (6) and
 211 (7).

212 For the discharge phase, mass and energy conservation read

$$213 \quad \frac{d}{dt}M(t) = -\dot{m}, \quad (10)$$

$$214 \quad \frac{d}{dt}T_R = \frac{-\dot{m}R - \dot{Q}_R}{M(a^* + bT_R)}. \quad (11)$$

216 In general, the mass flow rate through the expander is determined by the
 217 instantaneous electrical power demand. Since this paper concentrate on the
 218 T-CAES thermodynamics, rather than on its control strategy, we considered
 219 that the expander mass flow rate equals the compressor one, without loosing
 220 generality. Thus, the same time spent to charge the tank ($\mathcal{T}$), is needed for
 221 its discharge, having imposed that the final mass is equal to $M(0)$. We also
 222 note that the present model and the analysis reported in section 4 apply also
 223 for different values of $\dot{m}$, if one considers different charging and discharging
 224 durations.

225 The heat exchange through the tank walls is modeled through a quasi-steady
 226 approach assuming a spherical geometry of the reservoir. The tank structure
 227 consists of an iron shell externally covered with a rock wool insulation to mini-
 mize thermal losses (see. Fig 2).

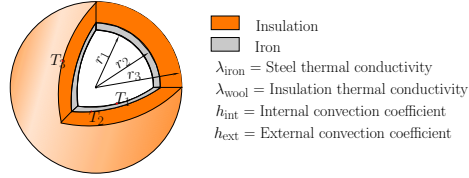


Figure 2: Sketch of the structure of the high pressure air reservoir. Note that, for clarity, dimensions are not in scale.

228
 229 The heat power flowing from the inside of the tank to the environment is
 230 calculated as

$$231 \quad \dot{Q}_R = \frac{T_R - T_{\text{env}}}{\frac{1}{h_{\text{int}} 4\pi r_1^2} + \frac{r_2 - r_1}{4\pi r_1 r_2 k_{\text{iron}}} + \frac{r_3 - r_2}{4\pi r_2 r_3 k_{\text{wool}}} + \frac{1}{h_{\text{ext}} 4\pi r_1^2}} \quad (12)$$

232 with reference to the notation of Fig. 2. Natural convection within a spherical
 233 cavity follows the non-dimensional equation [23]

$$234 \quad \text{Nu}_{\text{int}} = \begin{cases} 0.13 \text{Ra}^{1/3} & \text{for } \text{Ra} > 10^9 \\ 0.59 \text{Ra}^{1/4} & \text{for } \text{Ra} < 10^9 \end{cases}, \quad (13)$$

235 where $\text{Nu}_{\text{int}} = h_{\text{int}} 2r_1 / k_{\text{air}}$ is the Nusselt number and Ra is the Rayleigh

number. The Nusselt number for external convection, Nu_{ext} , is expressed as

$$\text{Nu}_{\text{ext}} = \frac{h_{\text{ext}} 2r_4}{k_{\text{air}}} = 2 + 0.45 \text{Ra}^{1/4}. \quad (14)$$

The thermo-physical properties of air (including conductivity k_{air}) are estimated at film temperature using data from [23] whilst the conductivity of the structural shell and thermal insulation are set to $k_{\text{iron}} = 44 \text{W}/(\text{mK})$ and $k_{\text{wool}} = 44 \times 10^{-2} \text{W}/(\text{mK})$, respectively [23].

3.3. Compressor and expander modeling

As already mentioned in section 2, the modeling of the compression and expansion transformations is derived hypothesizing to utilize dynamic and adiabatic machineries. Nevertheless, the resulting model is general and applies also to volumetric components, with only minor modifications. A quasi-steady model of performance is considered, since the time constant of the compressor and expander are at least one order of magnitude smaller than that of the boundary conditions (seconds vs. minutes).

The compression is divided into N_c stages. Each stage has the same small stage efficiency η_c and compression ratio

$$\beta_c(t) = [(1 + \delta) p_R(t) / p_{\text{env}}]^{1/N_c}, \quad (15)$$

where δ represents the pressure loss in the heat exchangers and connecting pipes. Equation (15) is derived minimizing the compression work, under the hypothesis that T_{ci} the same for all the stages. Despite this hypothesis may not be completely fulfilled, dividing the compression into equal parts is a generally accepted compromise between efficiency and ease of implementation. The air temperature at the outlet of the n -th compressor is given by

$$T_{\text{co}_n}(t) = T_{\text{ci}_n} \beta_c(t)^{\frac{\varepsilon}{\eta_c}}, \quad (16)$$

where T_{ci_n} is the inlet temperature to each compression stage (note that $T_{\text{ci}_1} = T_{\text{env}}$) and $\varepsilon = R/c_{p_a}$ for adiabatic (i.e dynamic) machinery. Equation

(16) also applies to cooled compression, which is more representative of reciprocating machinery, having care to assume $\varepsilon = (\xi - 1)/\xi$, where ξ is the polytropic exponent of the thermodynamic transformation ($1 < \xi < R/c_{p_a}$). The mechanical work consumed by the compressor train to fill the air reservoir is thus

$$W_c = \dot{m} \sum_{n=1}^{N_c} \left\{ \int_{t=0}^{\mathcal{T}} c_{p_a}(T) [T_{\text{co}_n}(t) - T_{\text{ci}_n}] dt \right\}. \quad (17)$$

The expansion is divided into N_e stages. Each stage has the same small stage efficiency η_e and expansion ratio

$$\beta_e(t) = [(1 - \delta) p_R(t) / p_{\text{env}}]^{1/N_e}. \quad (18)$$

Equation (18) comes from the maximization of the expansion work assuming that T_{ti} is the same for all the stages. Note that, the latter hypothesis is not satisfied in general and Equation (18), is applied as a compromise between optimality and ease of practical implementation. In fact, T_{ti} varies as a function of the operating conditions whilst the subdivision of the expansion between the stages is selected and fixed during the design of the CAES. The air temperature at the outlet of the l -th expander is given by

$$T_{\text{to}_l}(t) = \frac{T_{\text{ti}_l}}{\beta_e(t)^{\varepsilon \eta_e}}, \quad (19)$$

where T_{ti_l} is the inlet temperature. The mechanical work produced by the expander during the discharge of air from the reservoir is calculated as

$$W_d = \dot{m} \sum_{l=1}^{N_e} \left\{ \int_{t=0}^{\mathcal{T}} c_{p_a}(T) [T_{\text{ti}_l}(t) - T_{\text{to}_l}(t)] dt \right\}. \quad (20)$$

3.4. Heat exchangers modeling

The heat that is co-generated in the i -th air inter-cooler is

$$Q_i = \int_0^{\mathcal{T}} \dot{Q}_i dt = \int_0^{\mathcal{T}} \dot{m} c_{p_a} (T_{\text{co}_i} - T_{\text{ci}_{i+1}}) dt, \quad (21)$$

where $\dot{Q}_i$ is the thermal power exchanged at the i 'th inter-cooler. For a counter-current heat exchanger, the hot water temperature (T_{ico_i}) is estimated through

the following relationship

$$\frac{\dot{Q}_i}{UA} = \frac{T_{co_i} - T_{ico_i} - (T_{ci_{i+1}} - T_{ici_i})}{\ln \left(\frac{T_{co_i} - T_{ico_i}}{T_{ci_{i+1}} - T_{ici_i}} \right)}, \quad (22)$$

where $T_{ici_i} = T_{env}$ and $T_{ci_{i+1}} = T_{ri}$ are fixed, U is the global heat exchange coefficient, and A is the heat exchange surface. Under the hypothesis that the convective resistance on the water side of the heat exchanger is much lower compared to the air side, we may write

$$\frac{(UA)}{(UA)^*} = \left(\frac{Re}{Re^*} \right)^{0.8} \left(\frac{Pr}{Pr^*} \right)^{1/3} \left(\frac{k_{air}^*}{k_{air}} \right), \quad (23)$$

where $Re = \dot{m}d/(\rho\nu S)$ is the Reynolds number, d being the hydraulic diameter of the heat exchangers ducts and S their flow area. The apex $*$ indicates the values calculated at the design conditions.

Since constant $\dot{m}$ is assumed and the constructive parameters (i.e. d and S) do not change from design conditions, the following functional dependence holds true

$$\frac{Re}{Re^*} = \left(\frac{\nu^*}{\nu} \right) \left(\frac{\rho^*}{\rho} \right) \quad (24)$$

Finally,

$$(UA)^* = \frac{\dot{m}c_{p_a} (T_{co_i}^* - T_{ci_{i+1}}) \ln \left(\frac{T_{co_i}^* - T_{ico_i}^*}{T_{ci_{i+1}} - T_{ici_i}} \right)}{T_{co_i}^* - T_{ico_i}^* - (T_{ci_{i+1}} - T_{ici_i})}, \quad (25)$$

with $T_{co_i}^* = 0.5 \{ \max [T_{co_i}(t)] + \min [T_{co_i}(t)] \}$. The design temperature for the hot water, $T_{ico_i}^*$, is fixed a-priori as it must fulfill the constraints imposed by the energy user.

The same procedure, with proper change in notation, is utilized to model the design operation and off-design of the re-heaters. Each air re-heater recovers the following chilling power

$$C_j = \int_0^T \dot{C}_j dt = \int_0^T \dot{m}c_{p_a} (T_{ti_{j+1}} - T_{to_j}) dt. \quad (26)$$

Figure 3 reports the temperature and pressure variation of the T-CAES during each phase, according to the model described in this section. The default parameters reported in Table 1 are considered, except for the duration of the

storage phase, which is set to $\Theta = 6$ h, to highlight the reservoir pressure and temperature variations within this phase.

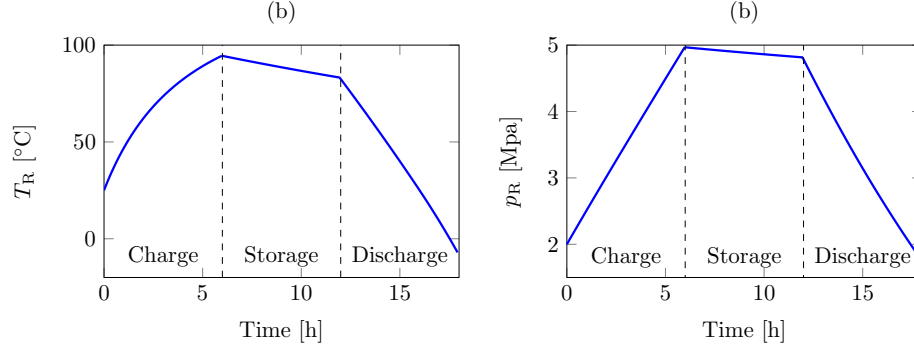


Figure 3: Air temperature (a) and pressure (b) in the T-CAES reservoir as a function of time during the charge, storage, and discharge phases.

4. Sensitivity analysis

Here, we dissect the effects of the most relevant parameters on the performances of the T-CAES. Specifically, N_c , N_e , the small stage efficiencies of the turbine and of the compressor, T_{ri} , $\beta = p_{\max}/p_{\text{env}}$, $\gamma = p_{\max}/p_{\min}$, and the time interval, Θ , between the charging and discharging phases are systematically varied. These parameters are varied one at a time in order to identify the key drivers and dissect their effects on the performances of the T-CAES. The results of such an analysis are the fundamental informations for the design of a T-CAES prototype. Given the preliminary nature and the general objective of this study the mutual effects between the parameters can be neglected. We comment that the evaluation of non-linear interactions between the components would require some knowledge of their off-design performances, that could be achieved after the choice of the proper technology for each plant section. Nevertheless, any arbitrary combination of the parameters reported in Table 1 can be studied through the mathematical model described in section 3. The default values for all the relevant parameters are reported in Table 1, together with

330 their range of variation. When not explicitly specified, each parameter assumes
331 the default value reported in Table 1.

Variable	Default value	Range
V	25 m ³	fixed
β	50	[10, 100]
γ	2.5	[1.1, 10]
$\dot{m}$	2.75×10^{-2} kg/s	fixed
$\eta_c = \eta_e$	0.7	[0.4, 0.9]
T_{ri}	50°C	[30, 100]°C
T_{env}	25°C	fixed
T_{rho}^*	0°C	fixed
T_{ico}^*	75°C	fixed
δ	2.5 %	[0, 20] %
N_c	3	[1, 5]
N_e	3	[1, 5]
Θ	0 h	[0, 24]h
$r_2 - r_1$	2.5×10^{-2} m	fixed
$r_3 - r_2$	25×10^{-2} m	fixed

Table 1: Default values and range of variation for the parameters influencing the T-CAES thermodynamic behavior.

332 When all the default values are assumed, the T-CAES is characterized by a
333 maximum mechanical power of about 18 kW, and by $\mathcal{T} = 6$ h. In details, 97 kWh
334 of mechanical energy are required to completely charge the volume, producing
335 also 100 kWh of thermal energy, or, equivalently, 14 kWh of thermal exergy,
336 from the inter-refrigerators. When the T-CAES is completely discharged, the
337 turbine produces 29 kWh of mechanical energy, while the re-heaters, recover
338 15 kWh of chilling energy, that correspond to 2.3 kWh of chilling exergy. The
339 resulting electrical and exergy efficiencies are $\eta_{el} = 30.2\%$ and $\eta_{ex} = 46.1\%$,
340 respectively. The above characteristics for the T-CAES are compatible for a

small to medium size civil application. In fact, T_{ico} and T_{rho} are coherent with space heating, domestic hot water, and air conditioning.

4.1. Number of stages of compression and expansion

The efficiency of the CAES is increased by utilizing a multi-stage inter-cooled compression, as depicted in Figure 4(a).

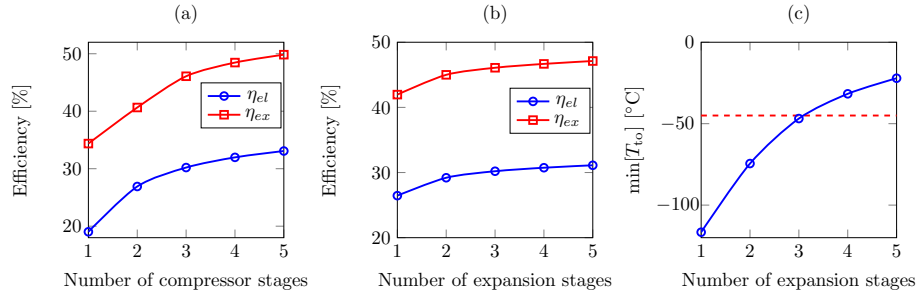


Figure 4: T-CAES performances as a function of the number of stages of the compressor and of the expansion: (a) Efficiency as function of the number of compressor stages; (b) Efficiency as function of the number of expansion stages; (c) $\min[T_{to}]$ as function of the number of expansion stages.

The inter-cooling reduces the compression work, without affecting the expansion one, as the enthalpy of the air at the inlet of the high pressure reservoir is not varied, having fixed T_{ri} . As a consequence, η_{el} is increased by 74% and η_{ex} is increased by 45% using a 5 stages compressor, with respect to single stage compression. Dividing the compression into 2 or 3 stages could be considered as a good compromise between efficiency and plant complexity. In fact, the introduction of one inter-cooler yields more than 40% increase in η_{el} and about 20% increase in η_{ex} . Additional 12% improvement for η_{el} and 13% for η_{ex} are obtained through a 3 stages compression. A further increase of the number of stages generates only minor marginal improvements of the T-CAES efficiency.

Increasing the number of expansion stages has a moderate effect on the T-CAES efficiency. With reference to the single stage expansion, electrical and exergy efficiencies are increased by 18% and 12% respectively, through a 5 stage expansion (see Figure 4(b)). Nevertheless, air re-heating during expansion is

360 mandatory to limit the minimum temperature at the expanders outlet. A single
 361 stage expansion would lead to $\min(T_{to}) < -100^\circ\text{C}$, hampering the technological
 362 feasibility of the plant. Therefore, the expansion should be divided at least into
 363 3 stages to obtain $T_{to} \geq -45^\circ\text{C}$, as shown in Figure 4(c). For more conservative
 364 designs aimed at preventing freezing of working gas or lube oil ($T_{to} \geq -25^\circ\text{C}$),
 365 five stages or more could be needed though.

366 4.2. Small stage efficiency effects

367 Here, we take the small stage efficiency as an indicator of the fluid machinery
 368 technological level. For sake of simplicity and without losing generality we
 369 assume that $\eta_e = \eta_c = \eta_p$, which entails that compressor and turbine has the
 370 same technological level.

371 The small stage efficiency significantly influences the electrical efficiency and
 372 the exergy efficiency of the T-CAES, as depicted in Figure 5. Specifically, η_{el} is
 373 increased by a factor 6, varying η_p from $\eta_p = 40\%$ to $\eta_p = 90\%$ (Figure 5). The
 374 exergy efficiency is increased of a factor 3.5 by increasing η_p in the same range.

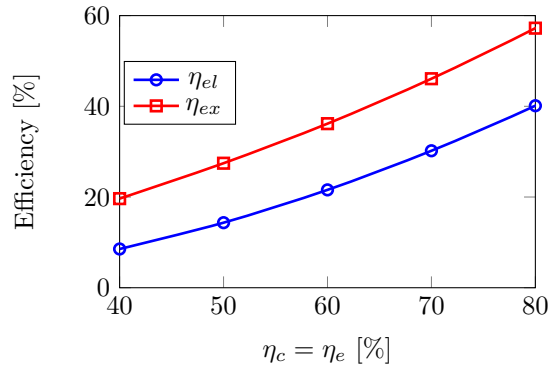


Figure 5: T-CAES efficiency as a function of the compressor and expander small stage efficiency.

375 The target for the compressor and expanders efficiency should be $\eta_p > 60\%$
 376 in order to have $\eta_{el} > 20\%$ and $\eta_{ex} > 30\%$. Note that, having assumed that η_p
 377 is constant, this target should be interpreted as an average efficiency over the

whole functioning interval of the fluid machinery, since both compressor and expander need to work with a variable pressure ratio.

4.3. Pressure losses effects

Friction and pressure losses within the heat exchangers and the connecting ducts dissipate a part of the mechanical energy of the working fluid, as shown in Eqs. 15 and 18

The electrical and the exergy efficiencies are linearly reduced by 17% and 12% respectively, increasing δ from 0 % to 20 % (see Figure 6). Thus, pressure losses needs to be considered for a precise evaluation of the plant performances though they are not determinant for the evaluation of the thermodynamic feasibility of the plant in study.

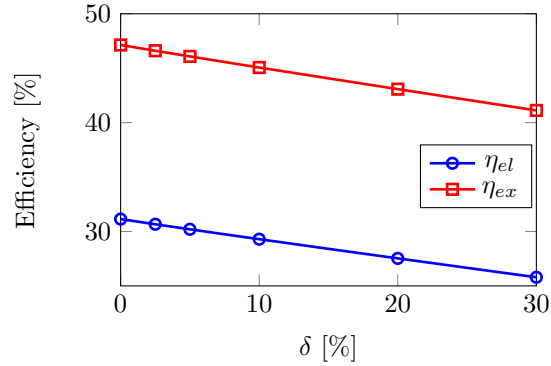


Figure 6: T-CAES efficiency as a function of the pressure losses.

4.4. Effects of the air temperature at the inlet of the reservoir

We study the effect of T_{ri} under the assumption that all the inter-coolers are identical, which means that $T_{ri} = T_{ci} \forall i \in [2, N_c]$.

The capacity of the T-CAES is reduced by 18% increasing T_{ri} from 30°C to 100°C (see Figure 7(a)). In fact, the pressure in the tank is utilized as the indicator of the state of charge of the T-CAES, and charging is completed when $p_R = p_{max}$. This means that, the mass that can be stored in the reservoir is reduced increasing the enthalpy of the air introduced in the volume. Moreover,

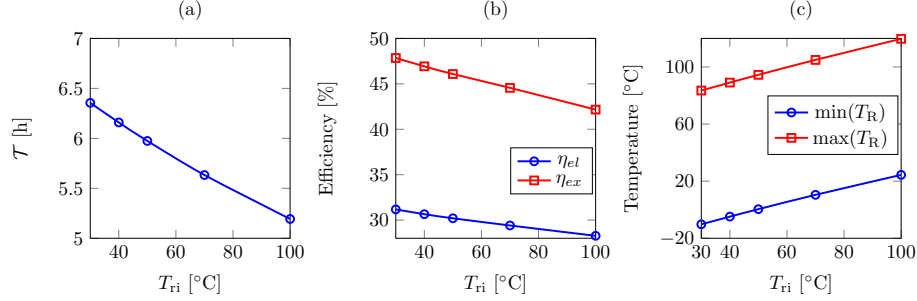


Figure 7: T-CAES performances as a function of T_{ri} : (a) Capacity; (b) Efficiency; (c) Reservoir temperatures.

397 given that we assumed a constant $\dot{m}$, a reduction of the stored mass results into
 398 lower duration of the charging and discharging phases.

399 An increase of the air temperature is detrimental to the efficiency of the
 400 T-CAES (see Figure 7(b)), since it increases the mechanical energy spent for
 401 compression, and only part the additional thermal energy of the working fluid
 402 can be recovered during expansion. As a result, η_{el} is linearly decremented of
 403 about 9% increasing T_{ri} in the range $[30-100]^{\circ}\text{C}$, while η_{ex} is decreased of about
 404 12% in the same T_{ri} range, as reported in Figure 7(b). The exergy efficiency is
 405 more sensitive to the variations of T_{ri} , with respect to η_{el} , because B_t and B_c
 406 both decrease as T_{ri} is increased.

407 The temperatures of the air in the reservoir at the beginning and at the end
 408 of the discharge both linearly increase increasing T_{ri} (see Figure. 7(c)). Thus,
 409 T_{ri} can be varied to control the maximum and minimum air temperature in the
 410 tank, that influence the heat exchange through the reservoir structure.

411 4.5. Effects of the reservoir pressure ratio

412 The capacity of the CAES is strongly influenced by the reservoir pressure
 413 ratio $\gamma = p_{\max}/p_{\min}$ (see Figure 8(a)). Specifically, for a fixed mass flow, the
 414 duration of the energy storage is incremented by a factor 11 when the pressure
 415 ratio is raised from $\gamma = 1.1$ to $\gamma = 10$. The storage capacity tends to saturate
 416 for $\gamma \geq 5$ as shown in Figure 8(a).

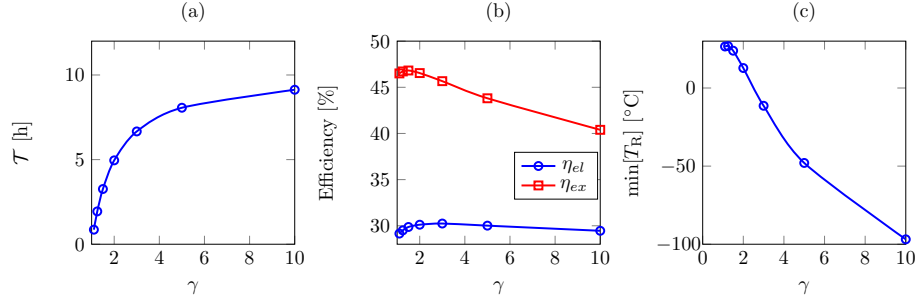


Figure 8: T-CAES performances as a function of γ : (a) Capacity; (b) Efficiency; (c) Reservoir temperature.

Moreover, the reservoir pressure ratio has a minor effect on the electrical efficiency, being η_{el} increased by only 3% varying γ from 1.1 to 5, while a further increasing of γ has a negligible influence on η_{el} (See Figure 8(b)). On the other hand, an increase of γ up to 10 results into a 13% reduction of the the exergy efficiency. Since any variation of γ yields opposing effects on η_{el} and η_{ex} , the determination of its value should not be based on the system efficiency.

The minimum temperature inside the tank is reduced as γ increases, (see Figure 8(c)). Specifically, for $\gamma \geq 3$ the temperature inside the reservoir at the end of the discharge phase falls below 0°C , and reaches -100°C for $\gamma = 10$.

In summary, whilst a relatively large γ should be selected to increase energy density of the T-CAES, the maximum value for γ is on the other hand limited by $\min[T_R]$. In particular, since $\min[T_R] < -50$ is undesirable for technological reasons, one should select $\gamma \leq 4$. Furthermore, the influence of higher γ on compressor and turbine average efficiencies across the entire charging and discharging process will have to carefully considered as very large differences between different types of machinery are expected to be encountered.

4.6. Effects of the turbines expansion ratio

The duration of the T-CAES charge and discharge phases is linearly proportional to β , as shown in Figure 9(a). In fact, the density of the air in the reservoir, and, consequently, the mass that can be stored, are proportional to

437 p_R . Thus, the larger β the smaller will be the volume of the air reservoir for a
 438 given $\mathcal{T}$.

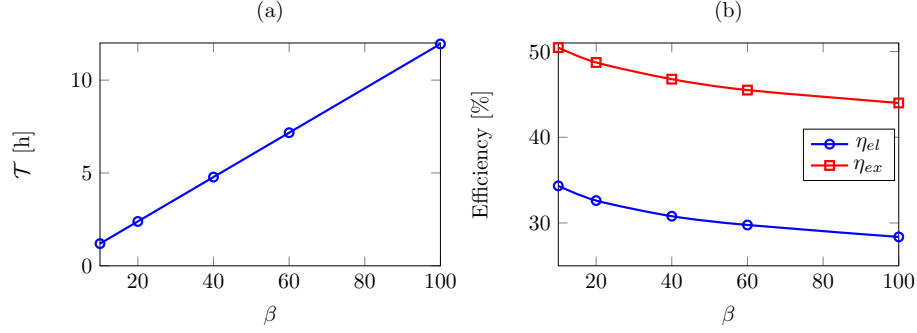


Figure 9: T-CAES performances as a function of β : (a) Capacity; (b) Efficiency.

439 However, both η_{el} and η_{ex} are reduced as β is increased (see Figure 9(b)).
 440 Since the mean temperature of the expansion is lower, than that of compression,
 441 the increment of compression work with β is larger than that of the expansion
 442 work. As a consequence, the electrical efficiency decreases by about 17% with
 443 β increasing from 10 to 100. A slighter detrimental effect is observed for the
 444 exergy efficiency, as B_t and B_c both grow at the same rate of W_c , smearing the
 445 unbalance between compression and expansion work.

446 Summarizing, β should be selected as a compromise between the efficiency
 447 and reservoir capacity, which in turn affects the size of the T-CAES.

448 4.7. Effects of the storage time

449 As p_R grows, during the charging phase, the air temperature inside the tank
 450 is increased above T_{env} . Thereafter, a part of the fluid energy is wasted into the
 451 environment as the heat is lost through the reservoir walls. The influence of the
 452 heat losses through the high pressure reservoir structure on the T-CAES oper-
 453 ation is studied varying the time interval between the charging and discharging
 454 phases.

455 Figure 10(a) shows that heat losses have a minor effect on both the electrical
 456 and the exergy efficiencies of the system. In fact, η_{el} is decreased by about 4.7%

457 in 24 h, while the total efficiency is decreased by 2.3%. The exergy efficiency
 458 is less sensitive to Θ , since more chilling energy is recovered decreasing the
 459 temperature of the air within the tank.

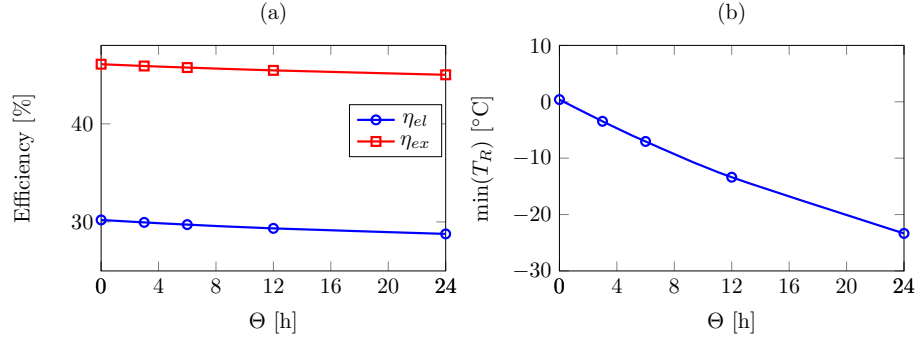


Figure 10: T-CAES performances as a function of the time interval between the charging and discharging phases: (a) Efficiency, (b) Reservoir temperature.

460 Figure 10(b) shows that the temperature within the tank at the end of the
 461 discharge phase is strongly influenced by the duration of the storage (i.e. by the
 462 heat losses). However, when large Θ are expected, $\min[T_R]$ can be controlled by
 463 properly selecting the values of γ and T_{ri} , as highlighted in sections 4.4 and 4.5

464 Thereafter, the control of the heat losses from the high pressure reservoir does
 465 not represent a serious technological issue for the T-CAES. Thermal insulation
 466 can be realized using common construction materials (i.e. rock wool), for daily
 467 energy storage.

468 5. Technological aspects

469 So far, the analysis presented in this paper sets forth a conceptual approach
 470 to the thermodynamic principles of CAES systems. As far as these fundamentals
 471 are concerned, it has already become clear that CAES systems have the potential
 472 to increase the techno-economic performance of the grid providing balance to
 473 the usually unbalanced power generation and demand curves. At the same
 474 time though, it must also be acknowledged that the results shown indicate ideal
 475 performances which will have to be corrected in real applications. The type and

476 extent of these corrections depends mostly on the actual equipment used which,
 477 in turn, depend on the system size (energy storage capacity and output). This
 478 is discussed in the following subsections.

479 5.1. Fluid machinery

480 There are two salient features of a dynamic compression or expansion device
 481 which must be analysed when selecting the most appropriate fluid machinery for
 482 a CAES system. These are the volumetric flow and stage pressure ratio and their
 483 combination results critical when selecting the most suitable turbomachinery.
 484 This is shown in Figure 11 for the case of compressors [24]. It is observed
 485 that axial compressors perform best for high flow capacity applications whereas
 486 multistage reciprocating units achieve outstanding pressure ratios even if they
 487 are limited to lower flows.

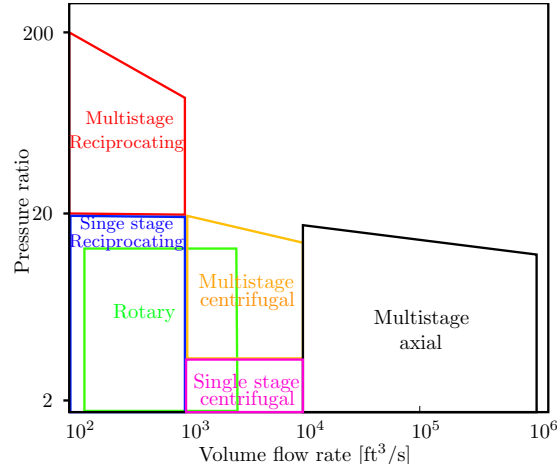


Figure 11: Preliminary compressor selection diagram. Data from [24].

488 In similarity analysis, flow and pressure ratio are expressed by the flow and
 489 load coefficients, ϕ and μ , respectively

$$490 \quad \phi = \frac{\dot{V}}{u_2 d_2^2} \frac{4}{\pi} \quad (27)$$

491

$$\mu = \frac{H_s}{u_2^2} \quad (28)$$

where $\dot{V}$, u_2 and d_2 are the volumetric flow rate, peripheral blade speed and diameter at the rotor/stator interface and H_s stands for the isentropic enthalpy change across the machine. In a first order approach, the following statement holds true

$$\eta_{st} = f(\phi, \mu) \quad (29)$$

whereby the efficiency of a given turbomachine (η_{st}) depends on the combination of flow and load coefficients only. Nonetheless, even if these coefficients are widely used in aerodynamic design of turbomachinery, they are not really practical at the initial system design stage. This is due to two main reasons: (i) their explicit dependence on the rotor characteristics, which are initially not known; and (ii) because these coefficients apply to turbomachinery only and thus they are not useful for small scale applications. For these reasons, specific speed and diameter (N_s and d_s) are more commonly used

$$N_s = \frac{N\dot{V}^{1/2}}{H_s^{3/4}} \quad (30)$$

$$d_s = \frac{DH_s^{1/4}}{\dot{V}^{1/2}} \quad (31)$$

where D is a representative length scale of the equipment: rotor diameter for turbomachinery and cylinder diameter for reciprocating systems.

Given the equivalence between the aforementioned parameters, the statement in Eq. (29) can now be expressed as

$$\eta_{st} = f(N_s, d_s) \quad (32)$$

whose graphical expression was originally developed by Balje [25] and is presented in Figure 12. It is observed that the combination of flow (N_s) and head (d_s) provides a very good estimate of machine type and expected efficiency,

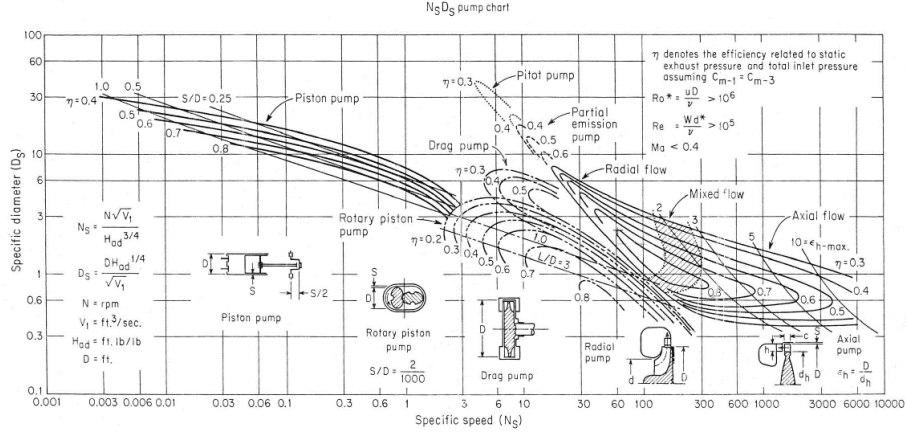


Figure 12: Generalized N_s D_s diagram for pumps and compressors. [25]

even if these are not univocally defined (i.e., for certain flow ranges, two machine types could be appropriate).

A second conclusion drawn from Figure 12 is the existence of an optimum combination of N_s - d_s for which efficiency is highest. This is more easily observed in Figure 13, which is a simplified version of Figure 12 providing the most suitable machinery type and expected peak efficiency for a given specific speed, on the assumption that specific diameter takes the optimum value in Figure 12.

The stage efficiency obtained from Figures 12 and 13 is adiabatic (isentropic) and thus it can be misleading when applied to a cooled compression for which it could happen that $\eta_{ad} > 1$. Such would be the case of low flow reciprocating compressors, which are usually water-cooled, or intercooled multistage turbocompressors. For these cases, the isothermal efficiency η_{iso} comparing the actual compression work with the compression work at constant gas temperature must be used. Typical values for these and for the polytropic efficiency utilized in previous sections are presented in Table 2.

It is thus concluded from this information that the application of the CAES concept put forward in this paper must take into account these technological aspects of the compression and expansion devices in order to yield realistic performance specifications. This optimization of the system with respect to

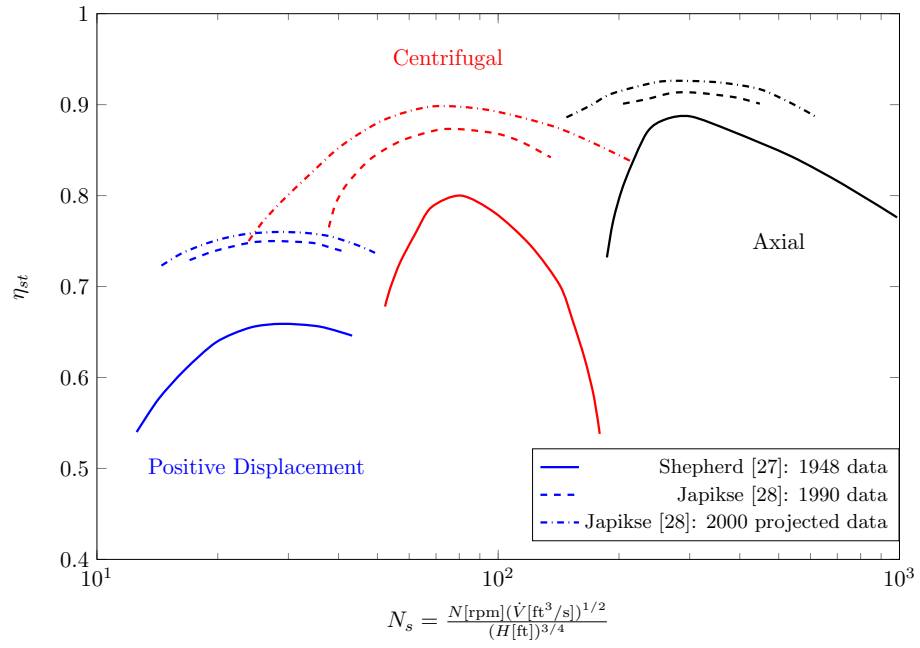


Figure 13: Peak isentropic efficiency (η_{st}) vs. N_s for compressors. Data are retrieved from [26–28].

	η_{ad}	η_{pol}	η_{iso}
Reciprocating	0.83 – 0.94	-	0.55 – 0.75
Rotary	0.70 – 0.80	–	–
Centrifugal SS	0.77 – 0.82	0.80 – 0.86	0.60 – 0.70
Centrifugal MS	–	0.80 – 0.86	0.60 – 0.70
Axial MS	0.85 – 0.90	0.85 – 0.87	–

Table 2: Adiabatic, polytropic and isothermal efficiencies for various compressor types (SS: single stage, MS: multistage). Data are retrieved from [29] and [30].

size and pressure ratio would otherwise be insensitive to the type of rotating machinery used. Specifically, following the design guidelines in Figure 11 the application presented in section 4, would require a multi stage reciprocating compressor or a series of rotary compressors. According to data reported in Table 2, and in Figure 12, the adiabatic efficiency of such fluid machineries lies in the range $[0.7, 0.95]$, with the higher values for cooled reciprocating compressors. We comment that these efficiencies (note that small stage efficiency is slightly larger than the isentropic one with larger differences attaining to higher compression ratios) yield to satisfactory performances of the CAES as evidenced in section 5 and in Figure 5.

All the foregoing discussion about the compression system is applicable to the expander whose type and expected performance must be accurately assessed during the design phase. The following singularities must be borne in mind though:

- Expanders (in particular turbines) can accommodate larger pressure ratios than compressors and thus fewer stages are expected in the expansion section. This brings about significant cost savings.
- The expansion line starts from room temperature and thus cryogenic conditions are met if the turbine pressure ratio is high as highlighted in section 4.6. This is a very interesting feature for a trigeneration facility. On

the other hand, these very low temperatures are expected to potentially trigger icing problems both in the flow path and in the lube oil system. Whilst the former may cause a drop in performance due to erosion, the latter could eventually lead to a fatal failure of the unit.

- For small applications, high speed generators are needed if turbines are selected in lieu of volumetric expanders. This technology is, though available, more costly and thus the technical and economic trade offs must be looked into.

With all this in mind, the selection of the best compression and expansion systems call for an integral approach which must comprise technical but also economic and operational considerations.

5.2. Heat exchangers

Heat exchangers also play a fundamental role in CAES systems where they are used to cool down the compression process and reheat the expansion process (both to enhance power generation and avoid potential icing problems). Usually, fundamental analyses of such equipment are based on heat transfer performance only, either by means of the logarithmic mean temperature difference or using the number of transfer units approach, whilst paying little or no attention to the influence of the heat exchanger type on its performance. This can eventually lead to significant deviations.

In the main, heat exchangers can be classified in two groups: standard class and compact. Standard heat exchangers are typically of the shell and tube type and they exhibit satisfactory thermal and hydraulic performances, though at the expense of a large footprint. Compact heat exchangers (CHE) are characterised for their high area to volume ratio which gives place to reduced footprint and fairly high heat transfer rates. Their main drawback is a potentially higher pressure drops due to a smaller hydraulic diameter which can also trigger *fouling* problems if the heat transfer media are not particle-free. For small to intermediate scale CAES systems, CHEs are likely to constitute the best option.

585

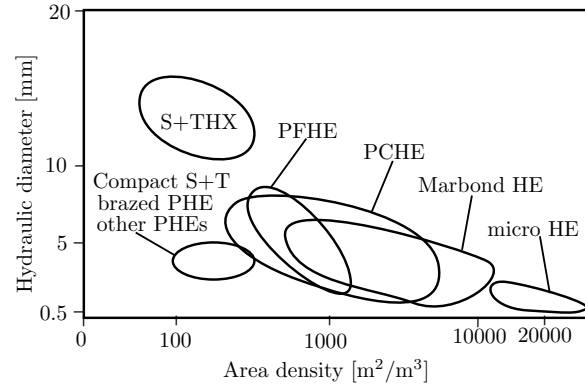


Figure 14: Heat exchanger : STHX, shell and tube; PHE, plate; PFHE, plate-fin; PCHE, printed circuit. Data are retrieved from [24].

586 Figure 14 from [31] evidences the much higher area density (heat transfer
587 area over volume) of CHEs. Reay [31] cites the following advantages of CHEs
588 over shell and tube:

- 589 • Improved effectiveness.
- 590 • Smaller volume and weight for given duty, leading to lower costs an easier
591 retrofit of existing facilities.
- 592 • Lower inventory and thus thermal inertia (easier and faster control).

593 As far as costs are concerned, these are very sensitive to plant size. Thus, for
594 small scale and intermediate applications compact units are reported to be more
595 economical (up to 4-5 times according to [31]) whereas in large facilities shell
596 and tube are a better option for their size scales up at a lower rate. Also, these
597 handle the differential expansion brought about by large temperature gradients
598 much better than the bulky CHEs which might also suffer from sealing problems
599 when their size grow very large. In this respect, the reliability of sealing gaskets,
600 chemical bonds and welding of multiple plates/layers is critical to prevent gas
601 leakages in high pressure operation.

602 In summary, it is of paramount importance to assess the optimum heat ex-
 603 changer technology for a particular CAES system as it affects its performance in
 604 all senses: on/off-design heat transfer rate, thermal inertia, reliability, *fouling*
 605 rate and costs. A very interesting review of these aspects is available in [32].

606 6. Conclusions

607 In this paper we study a trigenerative compressed air storage (T-CAES),
 608 where the heat available at the compressor exhaust could be used for heating
 609 purposes and expanding air can be used to supply a cooling load. We setup
 610 a thermodynamic model of a generic T-CAES, taking into account all the rel-
 611 evant operating and design parameters and we perform a sensitivity analysis.
 612 The proposed study is preliminary with respect to the design of a T-CAES pro-
 613 totype. In particular, the identification of the key drivers, and of the baseline
 614 performances is fundamental since no physical realizations of such a plant exist.

615 The sensitivity analysis is carried out with reference to a T-CAES with a
 616 compressor of about 18 kW and an air reservoir capacity compatible with a
 617 daily energy storage, which results into 97 kWh of mechanical energy needed
 618 to completely charge the volume. The reference system is able to produce 100
 619 kWh of thermal energy, 29 kWh of mechanical energy, and 15 kWh of chilling
 620 energy. The resulting electrical and exergy efficiencies are $\eta_{el} = 30.2\%$ and $\eta_{ex} =$
 621 46.1% , respectively. The above characteristics for the T-CAES are compatible
 622 for a small to medium size civil application. In fact, $T_{ico} \in [53, 89]^\circ\text{C}$ and
 623 $T_{rho} \in [-3, 5]^\circ\text{C}$ are coherent with space heating, domestic hot water, and air
 624 conditioning.

625 The following conclusions could be drawn from the sensitivity analysis:

- 626 • Dividing the compression into 2 or 3 stages could be considered a good
 627 compromise between efficiency and plant complexity. A further increase
 628 of the number of stages generates only minor marginal improvements of
 629 the T-CAES efficiency.

- The number of expansion stages is mainly driven by the need to limit the minimum temperature at the expanders outlet, which requires a minimum of 3 stages. The effect on the T-CAES efficiency is moderate, as a 5-stage expansion would result in an increase of the electrical and the exergy efficiencies of 18% and 12%, respectively, with respect to a single stage expansion.
- An increase of the reservoir air temperature is detrimental to the efficiency of the T-CAES, as its capacity is reduced by 18% increasing T_{ri} from 30°C to 100°C.
- The reservoir pressure ratio $\gamma = p_{\max}/p_{\min}$ should be chosen as a compromise between storage capacity and efficiency, and is bounded below by the minimum reservoir temperature.

Such a fundamental thermodynamic analysis demonstrates that T-CAES may have a potential as a distributed system that combines electricity storage with heat and cooling energy production. In fact, according to the model results, the exergy efficiency of the T-CAES is in the range 40-50% and compares to the expected second low efficiency for traditional trigenerative plants [33]. Moreover, the ratio between peak and off-peak electricity prices on European energy markets can be as high as 3-3.5 (see for instance [34, 35]), demonstrating the interest on a storage system that can reach 30% electrical efficiency while also producing thermal and chilling power.

The selection of suitable compressor and expander technologies, and then their efficiency in real application, depends on the flow and pressure requirements of the specific application. However, the optimization of the system with respect to size and pressure ratio would be insensitive to the type of rotating machinery used. Given the trigenerative nature of such systems, the design of the heat exchangers is also of paramount importance: for small scale and intermediate applications compact units are reported to be more economical whereas in large facilities shell and tube are a better option for their size scales up at a lower rate.

Future work will be devoted to the selection of the most suitable technology for each plant section, leveraging on the results of the present paper. Off-design or dynamic behavior of all the plant components, will be dissected after preliminary design of the T-CAES, together with the influence of the mutual interactions between most relevant parameters.

Nomenclature

$\dot{C}_j$	Chilling power exchanged the j -th re-heater	η_{el}	Electrical efficiency
$\beta_c(t)$	Compression stage pressure ratio	η_{ex}	Second law efficiency
β	Ratio between maximum reservoir pressure and environmental pressure	η_{iso}	Isothermal efficiency
δ	Pressure loss in the heat exchangers and connecting pipes	η_{st}	Generic turbomachine efficiency
$\dot{m}$	Air mass flow	γ	Ratio between maximum and minimum reservoir pressure
$\dot{Q}_i$	Thermal power exchanged in the i -th inter-cooler	$\mathcal{T}$	Storage capacity
$\dot{Q}_R$	Thermal power lost through the reservoir walls	$\mathcal{T}$	Time spent to charge or discharge the reservoir
$\dot{V}$	Volumetric flow rate	Nu_{ext}	Nusselt number relative to the air outside the reservoir
η_c	Compressor small stage efficiency	Nu_{int}	Nusselt number relative to the air inside the reservoir
η_e	Small stage efficiency for the expander	Ra	Rayleigh number
η_p	Small stage efficiency	Re	Reynolds number
η_{ad}	Adiabatic efficiency	μ	Compressor load coefficient
		ν	Air kinematic viscosity
		ϕ	Compressor flow coefficient
		ρ	Air density

707	Θ	Time interval between charge	732	h_{ext}	Convective coefficient outside
708		and discharge of the volume	735		the reservoir
709	ε	Compression transformation	739	h_{int}	Convective coefficient inside
710		exponent	737		the reservoir
711	ξ	Polytropic exponent	738	h_{ri}	Enthalpy of the air entering the
712	A	Heat exchange surface	739		reservoir
713	a	Zero order coefficient of air	740	H_s	isentropic enthalpy change
714		constant pressure specific heat	741		across the compressor
715	a^*	Zero order coefficient of air	742	k_{air}	Thermal conductivity of air
716		constant volume specific heat	743	k_{iron}	Thermal conductivity of struc-
717	b	First order coefficient of the air	744		tural material
718		specific heat	745	k_{wool}	Thermal conductivity of insu-
719	B_c	Exergy relative to a chilling en-	746		lation material
720		ergy flux	747	M	Mass of air contained in the
721	B_t	Exergy relative to thermal en-	748		volume
722		ergy	749	N	Rotating velocity
723	C_i	Thermal energy exchanged at	750	N_{IC}	Number of inter-coolers
724		the j -th re-heater	751	N_{RH}	Number of re-heaters
725	c_w	Water specific heat	752	N_c	Number of compression stages
726	c_{p_a}	Constant pressure specific heat	753	N_e	Number of expansion stages
727		for air	754	N_s	Specific speed
728	c_{v_a}	Constant volume specific heat	755	p_{env}	Environmental pressure
729		for air	756	p_{max}	Maximum reservoir pressure
730	D	Reference length scale	757	p_{min}	Minimum reservoir pressure
731	d	Hydraulic diameter	758	p_R	Pressure of the air in the reser-
732	d_2	Peripheral blade diameter	759		voir
733	d_s	Specific diameter	760	Q_i	Thermal energy exchanged at
			761		the j -th inter-cooler

762	R	Gas constant relative to air	786	U	Global heat exchange coefficient
763	r_1	Internal radius of the reservoir	787		
764		(See Figure 2)	788	u_2	Peripheral blade speed
765	r_2	External radius of the structural part of reservoir (See Figure 2)	789	V	Storage volume
766			790	W_c	Compression electrical energy
767			791	W_d	Expansion electrical energy
768	r_3	External radius of the reservoir (See Figure 2)	792	A-CAES	Adiabatic-CAES
769			793	CAES	Compressed Air Energy Storage
770	S	Flow Area	794		
771	T	Air temperature	795	CHCP	Combined Heat Cooling and Power plant
772	t	Time	796		
773	T_{ri}	Reservoir inlet temperature	797	CHE	Compact heat exchanger
774	T_{ci}	Compressor inlet temperature	798	DG	Distributed Generation
775	T_{co}	Compressor outlet temperature	799	HE	Heat exchanger
776			800	IC	Inter-cooler
777	T_{env}	Environmental temperature	801	PCHE	Printed circuit heat exchanger
778	T_{ico_i}	Water temperature at the outlet of the i -th inter-cooler	802	PFHE	Plate-fin heat exchanger
779			803	PHE	Plate heat exchanger
780	T_{rho_j}	Temperature at the outlet of the j -th re-heater	804	RES	Renewable Energy Sources
781			805	RH	Re-heater
782	T_R	Air temperature in the reservoir	806	STHX	Shell and tube heat exchanger
783			807	T-CAES	Trigenerative-CAES
784	T_{ti}	Turbine inlet temperature			
785	T_{to}	Turbine outlet temperature			

808 References

- 809 [1] H. Lund, G. Salgi, The role of compressed air energy storage (CAES) in
810 future sustainable energy systems, *Energy Conversion and Management*
811 50 (5) (2009) 1172–1179.
- 812 [2] R. Madlener, J. Latz, Economics of centralized and decentralized com-
813 pressed air energy storage for enhanced grid integration of wind power,
814 *Applied Energy* 101 (2013) 299–309.
- 815 [3] H. Safaei, D. W. Keith, R. J. Hugo, Compressed air energy storage (caes)
816 with compressors distributed at heat loads to enable waste heat utilization,
817 *Applied Energy* 103 (2013) 165–179.
- 818 [4] A. Cavallo, Controllable and affordable utility-scale electricity from inter-
819 mittent wind resources and compressed air energy storage (CAES), *Energy*
820 32 (2) (2007) 120–127.
- 821 [5] C. Xia, Y. Zhou, S. Zhou, P. Zhang, F. Wang, A simplified and unified
822 analytical solution for temperature and pressure variations in compressed
823 air energy storage caverns, *Renewable Energy* 74 (2015) 718–726.
- 824 [6] H. Chen, T. N. Cong, W. Yang, C. Tan, Y. Li, Y. Ding, Progress in electrical
825 energy storage system: A critical review, *Progress in Natural Science* 19 (3)
826 (2009) 291–312.
- 827 [7] D. Swider, Compressed air energy storage in an electricity system with
828 significant wind power generation, *Energy Conversion, IEEE Transactions*
829 on 22 (1) (2007) 95–102.
- 830 [8] W. F. Pickard, N. J. Hansing, A. Q. Shen, Can large-scale advanced-
831 adiabatic compressed air energy storage be justified economically in an
832 age of sustainable energy?, *Journal of Renewable and Sustainable Energy*
833 1 (3) (2009) 033102.

- [9] G. Grazzini, A. Milazzo, Thermodynamic analysis of CAES/TES systems for renewable energy plants, *Renewable Energy* 33 (9) (2008) 1998–2006.
- [10] D. Wolf, M. Budt, LTA-CAES—A low-temperature approach to adiabatic compressed air energy storage, *Applied Energy* 125 (2014) 158–164.
- [11] C. Bullough, C. Gatzen, C. Jakiel, M. Koller, A. Nowi, S. Zunft, Advanced adiabatic compressed air energy storage for the integration of wind energy, in *Proceedings of the European Wind Energy Confer, London*.
- [12] N. Hartmann, O. Vöhringer, C. Kruck, L. Eltrop, Simulation and analysis of different adiabatic compressed air energy storage plant configurations, *Applied Energy* 93 (2012) 541–548.
- [13] S. Zunft, C. Jakiel, M. Koller, C. Bullough, Adiabatic compressed air energy storage for the grid integration of wind power, in: *sixth international workshop on large-scale integration of wind power and transmission networks for offshore windfarms*, 2006, pp. 26–28.
- [14] J. B. Greenblatt, S. Succar, D. C. Denkenberger, R. H. Williams, R. H. Socolow, Baseload wind energy: modeling the competition between gas turbines and compressed air energy storage for supplemental generation, *Energy Policy* 35 (3) (2007) 1474 – 1492.
- [15] Y. Li, X. Wang, D. Li, Y. Ding, A trigeneration system based on compressed air and thermal energy storage, *Applied Energy* 99 (2012) 316–323.
- [16] Y. Kim, D. Favrat, Energy and exergy analysis of a micro-compressed air energy storage and air cycle heating and cooling system, *Energy* 35 (1) (2010) 213 – 220.
- [17] E. Jannelli, M. Minutillo, A. L. Lavadera, G. Falcucci, A small-scale CAES (compressed air energy storage) system for stand-alone renewable energy power plant for a radio base station: A sizing-design methodology, *Energy* 78 (0) (2014) 313 – 322.

- [18] A. L. Facci, L. Andreassi, S. Ubertini, Optimization of chcp (combined heat power and cooling) systems operation strategy using dynamic programming, *Energy* 66 (2014) 387–400.
- [19] A. L. Facci, L. Andreassi, F. Martini, S. Ubertini, Comparing energy and cost optimization in distributed energy systems management, *Journal of Energy Resources Technology* 136 (3) (2014) 032001.
- [20] A. L. Facci, L. Andreassi, S. Ubertini, E. Sciubba, Analysis of the influence of thermal energy storage on the optimal management of a trigeneration plant, *Energy Procedia* 45 (2014) 1295–1304.
- [21] A. Khaliq, Exergy analysis of gas turbine trigeneration system for combined production of power heat and refrigeration, *International Journal of Refrigeration* 32 (3) (2009) 534–545.
- [22] N. A. Gokcen, R. G. Reddy, *Thermodynamics*, Plenum Press, New York and London, 1996.
- [23] F. Kreith, R. Manglik, M. Bohn, *Principles of heat transfer*, Cengage learning, 2010.
- [24] D. Robison, P. J. Beaty, Compressor types, classifications, and applications, *Proceedings of the Turbomachinery Symposium* 21 (1992) 183–188.
- [25] O. Balje, *Turbomachines: A Guide to Design, Selection and Theory*, John Wiley and Sons, 1981.
- [26] S. Sjolander, *Turbomachinery, Supplementary Course Notes*, Carleton University, 2010.
- [27] D. G. Shepherd, *Principles of turbomachinery*, Macmillan, New York, 1956.
- [28] D. Japikse, N. Baines, *Introduction to Turbomachinery*, Oxford University Press, Oxford, 1994.

- 886 [29] M. Khan, Basic practices in compressors selection, in: International Com-
887 pressor Engineering Conference, 1984.
- 888 [30] P. Pichot, Compressor Application Engineering, Volume 1: Compression
889 Equipment, Gulf Publishing Company, Houston, 1986.
- 890 [31] D. Reay, Compact heat exchangers, enhancement and heat pumps, Inter-
891 national Journal of Refrigeration 25 (2002) 460–470.
- 892 [32] K. Manjunath, S. Kaushik, Second law thermodynamic study of heat ex-
893 changers: A review, Renewable and Sustainable Energy Reviews 40 (2014)
894 348–374.
- 895 [33] A. Khaliq, Exergy analysis of gas turbine trigeneration system for com-
896 bined production of power heat and refrigeration, International Journal of
897 Refrigeration 32 (3) (2009) 534 – 545.
- 898 [34] GME, <http://www.mercatoelettrico.org/en/esiti/mgp/esitingp.aspx>, con-
899 sulted on june 29th, 2015 (2015).
- 900 [35] F. Cappa, A. L. Facci, S. Ubertini, PEM fuel cell for cooperating house-
901 holds: a convenient combined heat and power solution for residential ap-
902 plications, Energy (In Press).