

Article

Thermodynamic Analysis of an Ideal Compressed Air Energy Storage (CAES) Cycle Integrated with a Solar Booster

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Abstract

This study presents an ideal-cycle thermodynamic analysis of an advanced compressed air energy storage (A-CAES) system with single thermal energy storage (TES) and an external heat boost. The additional heat is represented by a solar heat source, although the analysis is equally applicable to other forms of externally supplied thermal energy. Following the classical thermodynamic approach used for ideal cycles such as the Brayton, Otto and Diesel cycles, the objective is to establish analytical relationships and performance bounds for the integrated system rather than to model a specific engineering configuration. Three principal performance measures are examined: the electrical round-trip coefficient of performance (CoP), the marginal thermal coefficient of performance associated with external heat addition, and the overall second-law efficiency. Closed-form analytical expressions are derived for these quantities under idealised but still practically relevant assumptions. The analysis identifies distinct operating regimes governed by the level of external heat input and establishes analytical transition conditions between them. It is shown that external heat addition can substantially increase the round-trip coefficient of performance and lead to high marginal heat-utilisation effectiveness. A rigorous upper bound on the second-law efficiency is also obtained from a complete-cycle exergy analysis, demonstrating consistency with the laws of thermodynamics. The results provide analytical insight into the fundamental thermodynamic structure of solar-assisted A-CAES systems and establish performance bounds that are independent of any particular engineering implementation.

Keywords: advance compressed air energy storage; thermal energy storage; ideal thermodynamic cycle; round-trip coefficient of performance; marginal thermal coefficient of performance; second-law efficiency



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1. Introduction

Energy storage is a critical enabler of the energy transition, allowing renewable generation to be decoupled from demand and mitigating the inherent intermittency of wind and solar resources [1–3]. Among large-scale storage technologies, advanced compressed air energy storage (A-CAES) provides a promising pathway by storing surplus electrical energy through air compression in underground caverns while simultaneously capturing

the compression heat in thermal energy storage (TES) as illustrated in Figure 1. The long-term stability of underground storage are also an important consideration for large-scale compressed air storage, as rock behaviour, fracture development, and gas migration can influence storage reliability [4,5]. During periods of high electricity demand, the stored thermal and mechanical energy is released through expansion to generate electricity [6,7]. Typical electrical round-trip coefficients of performance for such systems lie in the range of 60–70%, and in some configurations may exceed 70% [8]. To further enhance system performance, external heat integration can be introduced, for example through a solar heat booster, although the thermodynamic framework developed here remains applicable to other sources of external heat [9].

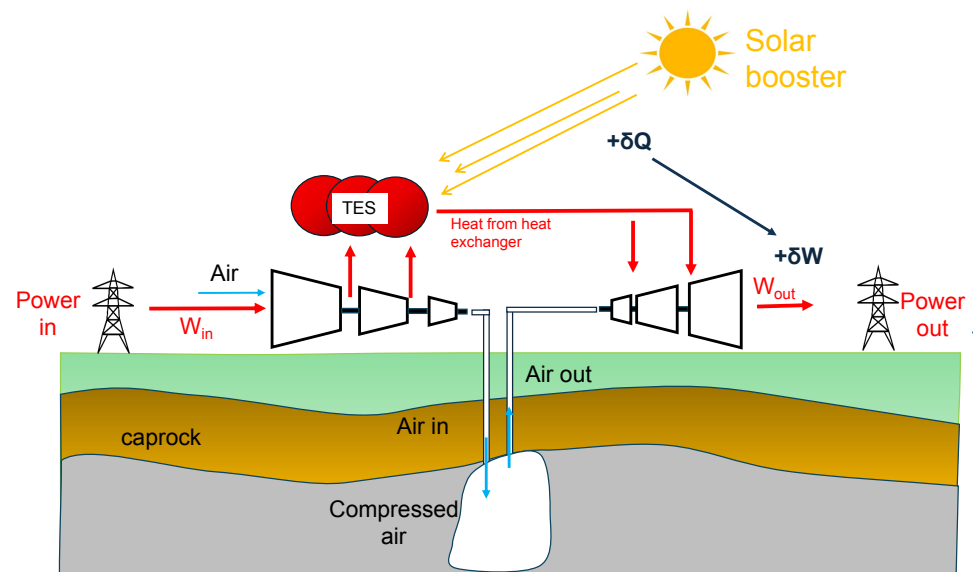


Figure 1. Integrated advanced CAES system with solar booster.

Substantial progress has been made in the modelling and optimisation of integrated A-CAES systems, with most existing studies focusing on numerical simulation, system-specific design assumptions, or techno-economic optimisation frameworks [8,10,11]. These investigations have considered various aspects of A-CAES development, including advanced thermal energy storage integration, multi-parameter system optimisation, alternative thermodynamic storage concepts, and different approaches for improving storage capacity and efficiency [12–19]. Previous studies mainly focus on system-level performance and practical design considerations, analysing the interactions between work, heat transfer, and storage through specific modelling frameworks. Within these frameworks, exergy-based approaches have also been used to evaluate irreversibilities and identify performance limitations. While these studies provide valuable insights into system performance and optimisation, closed-form thermodynamic relationships and basic performance limits are still not fully established and remain an open area of research.

To overcome this, it is essential to return to an idealised thermodynamic representation of the system. Such idealisations are not simplifications in a restrictive sense but rather a fundamental approach for understanding the main relationships between the processes. The development of classical thermodynamics has historically relied on this approach, as demonstrated by ideal cycles such as the Carnot, Brayton, Otto, and Diesel cycles [20,21]. Idealised cycles are deeply embedded in the foundations of thermodynamics, providing basic and transparent descriptions of energy conversion processes. By neglecting secondary engineering complexities, these models provide clear analytical relationships and performance limits. We need to follow this fundamental tradition by extending the

ideal-cycle approach to CAES, which is an important energy storage technology in the current energy transition.

Within this context, the present work develops an idealised thermodynamic framework for an integrated solar booster with a single TES A-CAES system, extending the classical role of ideal-cycle analysis to integrated storage systems. Within this framework, analytical expressions are derived for three principal performance measures: the electrical round-trip coefficient of performance, the marginal thermal coefficient of performance associated with external heat addition, and the overall second-law efficiency. Distinct operating regimes are identified, and a rigorous upper bound on the second-law efficiency is established based on a complete-cycle exergy balance.

The proposed framework provides a clear thermodynamic description of the interactions between mechanical work, thermal storage, and external heat input, while establishing analytical bounds within the idealised representation of the system.

2. Integrated A-CAES with a Solar Booster

As illustrated in Figure 2, the thermal energy generated during air compression is captured and stored in the thermal energy storage (TES), and is subsequently utilised for reheating the compressed air prior to the expansion stage. During the charging process, electrical energy is converted into the pressure energy of compressed air, while the heat generated by compression is recovered and stored rather than rejected to the environment. The compressed air is then stored in the cavern until energy is required (see Table 1 for notation).

During the discharge process, the stored compressed air is released from the cavern and reheated using thermal energy supplied by the TES before entering the expansion stages. This reheating increases the air temperature prior to expansion and thereby enhances the work produced by the turbines. The TES therefore acts as an intermediary thermal reservoir, enabling a portion of the energy recovered during compression to be returned to the cycle during discharge.

To further improve the performance of the system, a solar heat booster is integrated with the TES. The solar booster supplies additional thermal energy to the storage system, supplementing the heat recovered during compression. Consequently, the TES can provide a greater amount of thermal energy during the discharge process, increasing the temperature of the air entering the expansion stages and enhancing the potential for power generation. In this way, the integrated system combines electrical energy storage with the utilisation of renewable thermal energy, allowing both stored compression heat and externally supplied solar heat to contribute to the recovery of useful work.

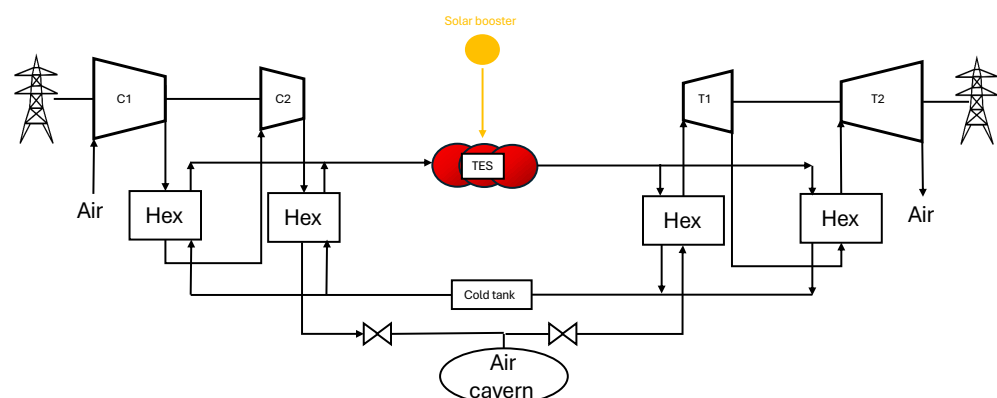


Figure 2. Schematic representation of a two-stage integrated solar-assisted A-CAES system with TES. Although a two-stage configuration is shown for illustration, the thermodynamic framework developed herein is for a general N-stage system.

Table 1. Notation.

Notation	Description
N	Number of compression/expansion stages (assumed equal)
C_p	Specific heat at constant pressure, kJ/(kg · K)
C_v	Specific heat at constant volume, kJ/(kg · K)
n	Adiabatic index (dimensionless)
P_0	Ambient (reference) pressure, kPa
P_s	Air cavern storage pressure, kPa
r_p	Stage temperature-pressure relation factor (dimensionless)
W_T	Total turbine work output, kJ/kg
W_C	Total compressor work input, kJ/kg
W	Net work output, $W = W_T - W_C$, kJ/kg
Q	External heat input to TES (solar booster contribution), kJ/kg
Θ_1	Heat-input threshold separating Regime 1 and Regime 2 (dimensionless)
δQ	Infinitesimal heat input increment, kJ/kg
δW	Corresponding incremental change in net work output, kJ/kg
Q_b	Upper bound of external heat input defining regime transition, kJ/kg
h_c	Heat transferred from compression stages to TES, kJ/kg
h_t	Heat supplied from TES to air before the expansion stage, kJ/kg
T_0	Ambient (reference) temperature, K
T_{TES}	Temperature of thermal energy storage, K
T_1	Outlet temperature of air after the first compression stage, K
T_3	Outlet temperature of air in the consecutive compression stages, K
T_2	inlet temperature of air in the consecutive compression stages, K
T_4	Outlet temperature of air after the expansion, K
$S_1, S_2, \dots, S_N$	Specific Entropy of state, kJ/kg.K
E_r	Exergy recovered from the system, kJ/kg
E_s	Exergy supplied to the system, kJ/kg
x	Heat-quality (Carnot) factor (dimensionless)
y	Dimensionless temperature ratio (dimensionless)
CoP_{rt}	Electrical round-trip coefficient of performance, W_T/W_C (dimensionless)
CoP_{th}	Marginal thermal coefficient of performance, $\delta W/\delta Q$ (dimensionless)
η_{II}	Second-law efficiency (dimensionless)

2.1. Ideal Integrated CAES Cycle with Single TES

As illustrated in the T–S diagram in Figure 3, the introduction of a solar heat boost influences the performance of the integrated A-CAES system. In practice, CAES systems exhibit significant thermodynamic and operational complexity. In the present study, we adopt a set of simplifying assumptions and consider an idealised system configuration.

- Air behaves as an ideal gas with constant thermodynamic properties.
- Compression and expansion processes are internally reversible and adiabatic (isentropic).
- The number of compression and expansion stages is finite and equal.
- Pressure ratios are equal across all compression stages and across all expansion stages.
- Perfect intercooling and reheating are achieved between successive stages.
- The thermal energy storage (TES) is represented as a single thermal reservoir operating at a single temperature T_{TES} .
- No pressure losses, heat losses, leakage losses, or auxiliary power consumptions are present in the system.
- The analysis is carried out on a unit-mass basis; therefore, all work and heat terms are expressed as specific quantities.

- Air is stored in the cavern at atmospheric temperature T_0 .
- The turbine outlet temperature cannot drop below the atmospheric temperature T_0 .
- The heat transfer by the compressor and the heat added by the solar boost to TES is entirely consumed by the turbine.

The integrated A-CAES system considered here shares certain thermodynamic features with a Brayton cycle, since both involve compression, heat addition, and expansion processes. However, the present configuration differs from a conventional Brayton cycle in several important respects. The system operates as an energy-storage cycle with thermal recovery and heat augmentation rather than as a conventional continuous power-generation cycle.

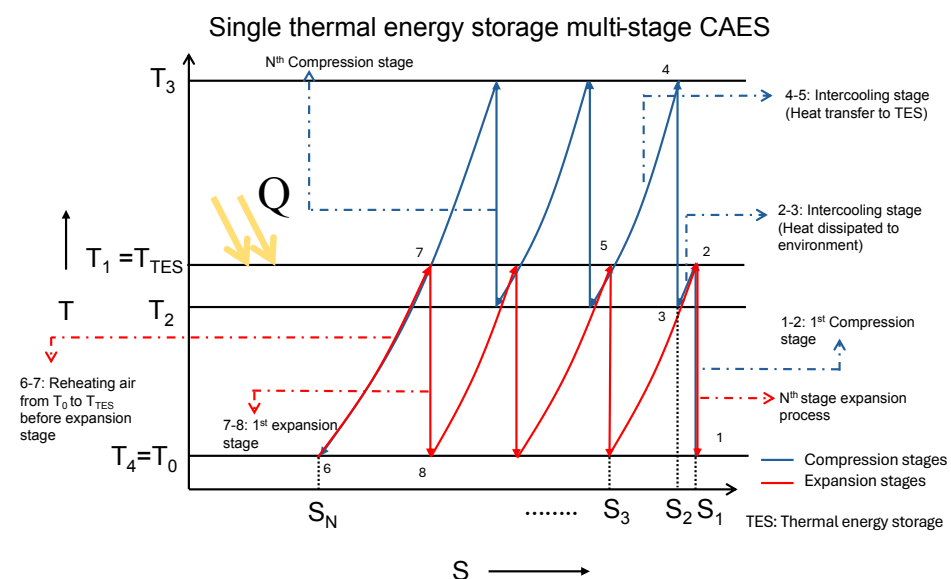


Figure 3. T-S plot of the integrated cycle in regime 1. Blue lines indicate compression stages and red lines indicate turbine stages. The yellow arrow represents the heat addition from the booster to TES.

The blue lines indicate the compressor work, and the red lines indicate the turbine work. To produce turbine work, the TES supplies the thermal energy required for reheating the air. The TES, in turn, is charged by energy extracted from the compressed air during the compression stages. With the addition of a heat booster, two operating regimes need to be considered.

2.1.1. Regime 1: Solar-Assisted Energy Storage

In this regime, a limited amount of heat is supplied by the solar booster, as illustrated in Figure 3, where

$$T_4 = T_0 < T_2 < T_{TES} = T_1 < T_3 \quad (1)$$

Air is compressed through N stages, and during intermediate intercooling a portion of the thermal energy from the compressed air is transferred to the TES operating at T_{TES} , while the remaining heat is rejected to the surroundings. Here, W_C is the cumulative compression work and W_T represents the total turbine work over all stages. Let $h_c = C_p(T_3 - T_{TES})$ denote the heat transfer from the subsequent compression stages. Similarly, $h_t = C_p(T_{TES} - T_0)$ represents the heat supplied by the TES for reheating the air in each expansion stage. The thermal energy stored in the TES is then used to reheat the air prior to expansion. With the introduction of the solar booster, the overall energy balance of the system can be expressed

so that the total heat required for reheating equals the sum of the external heat input Q and the thermal energy accumulated in the TES during compression:

$$Q + (N - 1)h_c = Nh_t \quad (2)$$

This relation represents the total thermal energy necessary for one complete cycle of the integrated A-CAES system. In the absence of any external heat input, $Q = 0$, the entire reheating requirement is met by the thermal energy supplied from the compression process. However, as the solar heat input Q increases, the energy transfer h_c from the compression stages decreases, leading to a reduction in T_3 and T_2 (see Figure 3) and, therefore, a corresponding decrease in h_c . This, in turn, lowers the required compressor work. The operational parameters of the turbine do not change. The upper bound for this regime is $Q \leq Q_b$, where the case $Q = Q_b$ is shown in Figure 4. This limit is reached when T_3 approaches its minimum value, equal to T_1 , so that

$$T_4 = T_0 = T_2 < T_{\text{TES}} = T_1 = T_3 \quad (3)$$

The thermal regimes for the compressor and turbine then become similar, and

$$Q_b = Nh_t = NC_p(T_{\text{TES}} - T_0) \quad (4)$$

In this limiting case, $h_c = C_p(T_1 - T_{\text{TES}})$ is achieved because $T_2 = T_0$.

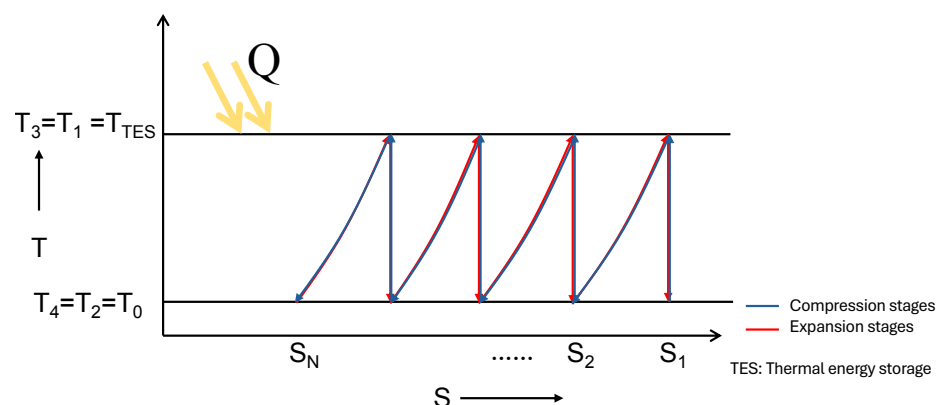


Figure 4. The T-S diagram of the integrated cycle showing the transition from Regime 1 to Regime 2. The temperature notations are consistent with previous figures.

2.1.2. Regime 2: Solar Co-Generation

In the second regime, illustrated in Figure 5, the operating conditions of the A-CAES system are such that $Q > Q_b$, where Q_b is defined in (4), and the solar heat dominates, implying that

$$T_0 = T_2 < T_4 < T_1 = T_3 < T_{\text{TES}} \quad (5)$$

The system increasingly operates as a heat engine. The heat balance for this case can be expressed as

$$Q = Nh_t = NC_p(T_{\text{TES}} - T_4) + C_p(T_4 - T_0) \quad (6)$$

As Q increases further, there is a corresponding increase in T_{TES} and T_4 , leading to an increase in h_t and turbine work output. The compressor work remains unchanged and minimal.

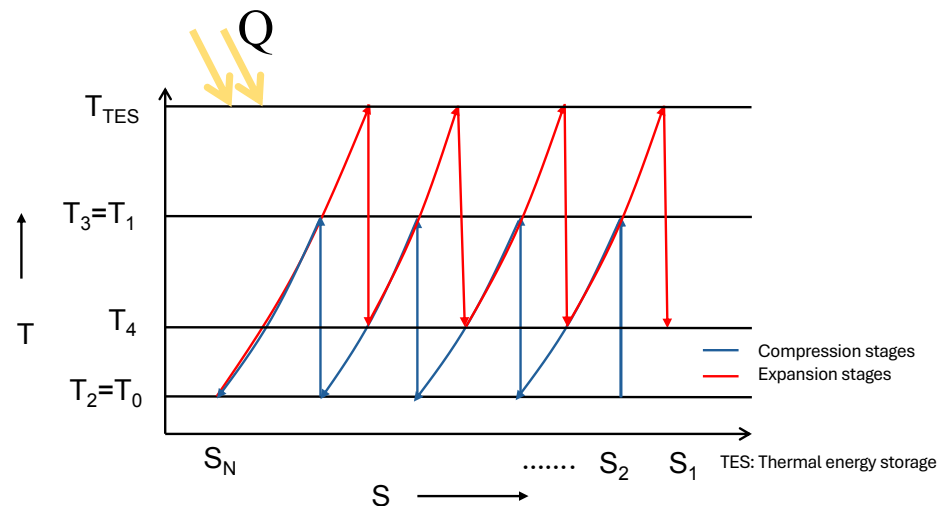


Figure 5. T-S plot of the integrated cycle in regime 2a. Notations are as in the previous figures.

3. Thermodynamic Analysis

The performance of the energy storage system is characterised by the round-trip coefficient of performance

$$\text{CoP}_{\text{rt}} = \frac{W_T}{W_C} \quad (7)$$

which characterises work returned relative to work expended. Note that this CoP_{rt} does not generally reflect the overall energy balance, since it deals with work only and does not take into account the “free” solar heat, although CoP_{rt} depends on Q .

A small additional amount of solar heat δQ is supplied to the TES induces extra useful work $\delta W = \delta(W_T - W_C)$ produced by the system. This introduces the second important CoP, namely the marginal thermal coefficient of performance

$$\text{CoP}_{\text{th}} = \frac{\delta W}{\delta Q} \quad (8)$$

which characterises the conversion of external heat into additional work.

The ratio factor r_p is given by:

$$(r_p)^N = \left(\frac{P_s}{P_0} \right)^{\frac{n-1}{n}} \quad (9)$$

3.1. Performance Coefficients in Regime 1

In this regime, the electrical round-trip CoP of the storage system is given by

$$\text{CoP}_{\text{rt}} = \frac{NC_p(T_{\text{TES}} - T_0)}{C_p(T_1 - T_0) + (N - 1)C_p(T_3 - T_2)} \quad (10)$$

For regime 1, r_p in terms of temperature relation is given by:

$$r_p = \frac{T_{\text{TES}}}{T_0} = \frac{T_1}{T_0} = \frac{T_3}{T_2} \quad (11)$$

This implies that

$$\frac{1}{\text{CoP}_{\text{rt}}} = \frac{(N - 1)}{N} \frac{T_3}{T_{\text{TES}}} + \frac{1}{N} \frac{T_1}{T_{\text{TES}}} \quad (12)$$

Using Equation (2), the energy balance for the integrated heat-boosted CAES system is

$$Q + (N - 1)C_p(T_3 - T_{TES}) = NC_p(T_{TES} - T_0) \quad (13)$$

Dividing Equation (13) by NC_pT_{TES} gives

$$\frac{Q}{NC_pT_{TES}} + \frac{(N - 1)}{N} \frac{T_3}{T_{TES}} + \frac{1}{N} \frac{T_1}{T_{TES}} = 2 - \frac{1}{r_p} \quad (14)$$

Substituting Equation (12) into Equation (14), we obtain

$$\frac{Q}{NC_pT_{TES}} + \frac{1}{\text{CoP}_{rt}} + \frac{1}{r_p} = 2 \quad (15)$$

resulting in the following value for CoP_{rt} :

$$\text{CoP}_{rt} = \left(2 - \frac{1}{r_p} - \frac{Q}{r_p NC_p T_0} \right)^{-1} \quad (16)$$

In this regime, the useful work $W = W_T - W_C$ increases through a reduction in W_C as Q increases, while W_T remains constant. Therefore,

$$W = W_T - W_C = W_T \left(1 - \frac{1}{\text{CoP}_{rt}(Q)} \right) \quad (17)$$

Hence, using (16),

$$\text{CoP}_{th} = \frac{\delta W}{\delta Q} = \frac{W_T}{NC_pT_{TES}} \quad (18)$$

which, due to

$$W_T = NC_p(T_{TES} - T_0) = NC_pT_{TES} \left(1 - \frac{1}{r_p} \right) \quad (19)$$

is transformed into

$$\text{CoP}_{th} = \frac{\delta W}{\delta Q} = 1 - \frac{1}{r_p} \quad (20)$$

3.2. Performance Coefficients in Regime 2

In this regime, the round-trip CoP_{rt} of the storage system is given by

$$\text{CoP}_{rt} = \frac{T_{TES} - T_4}{T_1 - T_0} = \frac{T_{TES}}{T_0} \frac{1}{r_p} \quad (21)$$

as in regime 2, r_p is given by:

$$r_p = \frac{T_1}{T_0} = \frac{T_3}{T_2} = \frac{T_{TES}}{T_4} \quad (22)$$

The energy balance takes the form

$$Q = NC_p(T_{TES} - T_4) + C_p(T_4 - T_0) \quad (23)$$

Dividing the energy equation by NC_pT_0 results in

$$\frac{T_{TES}}{T_0} = \frac{\frac{Q}{NC_pT_0} + \frac{1}{N}}{1 - \frac{1}{r_p} + \frac{1}{Nr_p}} \quad (24)$$

This determines T_{TES} and, therefore, CoP_{rt} :

$$\text{CoP}_{\text{rt}} = \frac{T_{\text{TES}} - T_4}{T_1 - T_0} = \frac{\frac{Q}{r_p N C_p T_0} + \frac{1}{N r_p}}{1 - \frac{1}{r_p} + \frac{1}{N r_p}} \quad (25)$$

The useful work $W = W_T - W_C$ increases with increasing W_T as Q increases, while $W_C = N C_p (T_1 - T_0) = N C_p T_0 (r_p - 1)$ remains constant. Therefore,

$$W = W_T - W_C = W_C (\text{CoP}_{\text{rt}}(Q) - 1) \quad (26)$$

Differentiating this equation yields

$$\text{CoP}_{\text{th}} = \frac{\delta W}{\delta Q} = W_C \frac{\delta \text{CoP}_{\text{rt}}}{\delta Q} = \frac{(r_p - 1)}{r_p - 1 + 1/N} \quad (27)$$

3.3. Parameters for Transitional Regimes

The transition between regimes 1 and 2 requires the solar boost intensity of

$$\frac{Q}{N C_p T_0} = r_p - 1 = \Theta_1 \quad (28)$$

and corresponds to

$$\text{CoP}_{\text{rt}} = 1 \quad (29)$$

where Θ_1 defines the onset of regime transition due to increasing external heat input.

The operating behaviour can be classified into two distinct regimes: Regime 1, corresponding to low external heat input where the system behaves as a conventional TES-assisted CAES cycle, and Regime 2, corresponding to higher external heat input, where solar assistance becomes dominant and enhanced work recovery is observed.

4. Performance Metrics

4.1. The Round-Trip Coefficient of Performance

The round-trip CoP_{rt} , which is defined as the ratio of work output (W_T) to work input (W_C) in Equation (7), is a principal parameter characterising the performance of energy storage systems. This definition represents a work recovery ratio for an energy storage system and is not a classical thermal efficiency of a heat engine cycle. In such systems, it represents the fraction of input electrical energy that can be recovered after a complete charge–discharge cycle. With the integration of a solar booster, either the required electrical input (W_C) decreases in regime 1 or the work output (W_T) increases in regime 2. The round-trip CoP_{rt} tends to increase substantially with increasing solar boost and is a monotonically increasing function of Q in all regimes. In regime 1, CoP_{rt} remains below 100% until regime 2 is achieved. The baseline CAES work–recovery ratio approaches unity in the transition state between regime 1 and regime 2. However, in Regime 2, where external thermal energy is supplied through the solar heat booster, the round-trip CoP exceeds unity due to the additional energy contribution from the thermal pathway, and therefore should not be interpreted as a closed-cycle thermodynamic efficiency. The major role of the solar boost is to in ideal system is to improve the operational round-trip efficiency. The practical attractiveness of this measure is associated with the relatively small modifications required to provide a solar boost and with the abundant availability of solar energy. It is noted that the round-trip CoP should not be interpreted in terms of net cycle efficiency or heat-engine performance, but rather as a work-based energy recovery metric specific to storage systems. Our findings about CoP_{rt} are now summarised in form of a proposition.

Proposition 1. *The round-trip coefficient of performance CoP_{rt} of solar-boosted CAES with single TES increases monotonically with increasing solar heat input and exceeds 100% when the external heat supplied to the system exceeds $Q = NC_p T_0 \Theta_1$. With solar boosting, the CAES system is progressively transformed from a pure energy storage device into a hybrid solar-thermal generator.*

4.2. Thermal Coefficient of Performance

The thermal performance of the system is assessed by the marginal thermal CoP_{th} —a measure of how effectively the solar heat is utilised. This quantity is defined as the marginal increment of $W = W_T - W_C$ divided by the increment of Q . The marginal approach produces clear results that depend on the operating regime. The overall ratio (i.e., W/Q instead of $\delta W/\delta Q$) is more ambiguous, since W can be both positive and negative in different regimes, while $W(Q)$ is a consistently monotonic function. Note that the increase δW involves a combination of several factors, such as improved thermal performance and the reduction in losses associated with the storage system, while only overall (not marginal) characteristics are conventionally constrained by thermodynamic inequalities.

The expressions for CoP_{th} in (20) and (27) are summarised in Table 2 below.

Table 2. Thermal CoP_{th} for isentropic compressor and turbine.

Regime	CoP_{th}	Q Range
1	$1 - \frac{1}{r_p} = 1 - \frac{T_0}{T_{TES}}$	$0 \leq \frac{Q}{NC_p T_0} \leq \Theta_1$
2	$\frac{r_p - 1}{r_p - 1 + 1/N}$	$\Theta_1 \leq \frac{Q}{NC_p T_0}$

Note that, for regime 1, CoP_{th} strongly resembles the efficiency of an ideal Brayton cycle,

$$\eta_{\text{Brayton}} = 1 - \frac{1}{r_{\text{Brayton}}}, \quad r_{\text{Brayton}} = \frac{T_2}{T_1} = \frac{T_3}{T_4} \quad (30)$$

using the conventional notation for the node temperatures $T_1, \dots, T_4$. Note that this equation can compare with the efficiency of a Carnot cycle $\eta_{\text{Carnot}} = 1 - T_0/T_{TES}$ operating between temperatures T_{TES} and T_0 . This is because CoP_{th} characterises marginal properties of a complex system such as CAES and not overall efficiency performance of a heat engine. These comparisons nevertheless show the benefits of solar boosting, which can utilise additional solar heat efficiently. Our analysis of CoP_{th} indicates that:

Proposition 2. *The marginal thermal coefficient of performance, CoP_{th} , of a solar-boosted CAES system with single TES characterises the incremental conversion of additional solar heat into additional useful work. Under the ideal reversible assumptions adopted in the present analysis, this marginal efficiency is constrained by the Carnot limit associated with the temperatures at which heat is supplied and rejected.*

4.3. Second-Law Efficiency

While the previous sections demonstrate relatively high values of CoP associated with two different aspects of the practical performance of the system, namely energy return and heat-to-work conversion, a proper thermodynamic efficiency of the solar-boosted CAES system also needs to be analysed. Since two different energy inputs are used, namely work W and heat Q , the second-law efficiency must be employed to characterise properly the quality of these inputs. Specifically, the second-law efficiency is defined as

$$\eta_{II} = \frac{E_r}{E_s} = \frac{W_T}{W_C + Qx}, \quad x = 1 - \frac{T_0}{T_{TES}} \quad (31)$$

where E_s is spent exergy, E_r is recovered exergy, and x characterises the quality of the heat. In this section, we prove the following theorem, which establishes a fundamental constraint that is clearly consistent ($\eta_{II} < 1$) with the second law of thermodynamics:

Theorem 1. *The second-law efficiency of an A-CAES system with solar-boosted single TES is subject to the following upper bound,*

$$\eta_{II} \leq \left(1 + (r_p - 1) \frac{T_0}{T_{TES}}\right)^{-1} < 1 \quad (32)$$

which can be written for regime 1 as

$$\eta_{II} \leq \frac{T_{TES}}{T_{TES} + (T_{TES} - T_0)} < 1 \quad (33)$$

Proof of Theorem 1. The proof below demonstrates that

$$\frac{1}{\eta_{II}} \geq 1 + \frac{r_p - 1}{y}, \quad y = \frac{T_{TES}}{T_0} \quad (34)$$

equivalently to (32). It is clear that $\eta_{II} < 1$ in the above equation since $r_p > 1$ and $y > 0$.

For regime 1: The second-law efficiency is given by:

$$\eta_{II} = \frac{NC_p(T_{TES} - T_0)}{C_p(T_1 - T_0) + (N - 1)C_p(T_3 - T_2) + Q(1 - \frac{T_0}{T_{TES}})} \quad (35)$$

$$\eta_{II} = \frac{NC_p T_{TES}(1 - T_0/T_{TES})}{C_p T_1(1 - T_0/T_1) + (N - 1)C_p T_3(1 - T_2/T_3) + Q(1 - \frac{T_0}{T_{TES}})} \quad (36)$$

Using the ratio r_p of regime 1, we have:

$$\eta_{II} = \frac{NC_p T_{TES}}{C_p T_1 + (N - 1)C_p T_3 + Q} \quad (37)$$

Using Q , which is given by Equation (13), resulting in the following value of the second law for regime 1:

$$\eta_{II} = \frac{T_{TES}}{T_{TES} + (T_{TES} - T_0)} = \frac{r_p}{2r_p - 1} \quad (38)$$

We obtain the exact value of the second law for regime 1.

For regime 2: The second-law efficiency is given by:

$$\eta_{II} = \frac{NC_p(T_{TES} - T_4)}{NC_p(T_1 - T_0) + Q(1 - \frac{T_0}{T_{TES}})} \quad (39)$$

For this regime, let

$$y = \frac{T_{TES}}{T_0}, r_p = \frac{T_1}{T_0} = \frac{T_{TES}}{T_4}, \frac{T_0}{T_4} = \frac{r_p}{y} \quad (40)$$

Using Q for the regime 2 from Equation (23):

$$\eta_{II} = \frac{NC_p(T_{TES} - T_4)}{NC_p(T_1 - T_0) + (NC_p(T_{TES} - T_4) + C_p(T_4 - T_0))(1 - \frac{T_0}{T_{TES}})} \quad (41)$$

$$\frac{1}{\eta_{II}} = \frac{NC_p(T_1 - T_0)}{NC_p(T_{TES} - T_4)} + \frac{C_p(T_4 - T_0)(1 - \frac{T_0}{T_{TES}})}{NC_p(T_{TES} - T_4)} + (1 - \frac{T_0}{T_{TES}}) \quad (42)$$

$$\frac{1}{\eta_{II}} = \frac{T_0(T_1/T_0 - 1)}{T_4(T_{TES}/T_4 - 1)} + \frac{1}{N} \frac{T_0}{T_{TES}} \frac{(T_4/T_0 - 1)}{(1 - T_4/T_{TES})} \left(1 - \frac{T_0}{T_{TES}}\right) + \left(1 - \frac{T_0}{T_{TES}}\right) \quad (43)$$

$$\frac{1}{\eta_{II}} = \frac{r_p}{y} + \frac{1}{N} \frac{1}{y} \frac{(y/r_p - 1)}{(1 - 1/r_p)} \left(1 - \frac{1}{y}\right) + \left(1 - \frac{1}{y}\right) \quad (44)$$

The second term in the above equation is positive; therefore, for any value of y , we have:

$$\frac{1}{\eta_{II}} \geq \frac{r_p}{y} + \left(1 - \frac{1}{y}\right) = 1 + \frac{r_p - 1}{y} = 1 + (r_p - 1) \frac{T_0}{T_{TES}}, \quad (45)$$

which proves (34).

In regime 1, $T_4 = T_0$ and thus $r_p = T_{TES}/T_0 = y$, so (34) reduces to

$$\frac{1}{\eta_{II}} = 2 - \frac{T_0}{T_{TES}} \quad (46)$$

□

The derived upper bound on second-law efficiency serves as a thermodynamic benchmark for future numerical and experimental studies. Any practical solar-assisted A-CAES system operating under real losses, finite-temperature heat transfer, pressure drops, and storage inefficiencies must necessarily operate below this limit. The bound therefore, provides a useful reference for evaluating the thermodynamic quality of practical designs.

5. Conclusions

This work presents a thermodynamic analysis of an ideal advanced compressed air energy storage (A-CAES) system with a single TES integrated with an external heat booster supplied by concentrated solar energy or other thermal sources such as combustion or residual heat.

The main results can be summarised as follows:

- Closed-form analytical expressions are derived for the work and heat interactions of the integrated system, leading to an explicit relationship between the round-trip coefficient of performance, pressure ratio, and external heat input.
- Two distinct operating regimes are identified and characterised analytically according to the magnitude of the external heat input, together with the transition condition separating them.
- The analysis demonstrates that, in the presence of external heat addition, the round-trip coefficient of performance may exceed unity while remaining fully consistent with the first and second laws of thermodynamics when interpreted as a work–recovery metric rather than a thermal efficiency.
- A marginal thermal coefficient of performance is introduced to quantify the incremental conversion of externally supplied heat into additional useful work.
- A second-law efficiency formulation is established for the integrated system. Under the ideal reversible assumptions adopted throughout the analysis, this efficiency attains its theoretical upper bound and provides a rigorous thermodynamic benchmark for assessing the performance of practical systems.
- The resulting analytical framework establishes the maximum reversible performance in Regime 1 and derives corresponding second-law bounds for Regime 2, providing a thermodynamically rigorous benchmark for assessing solar-boosted A-CAES systems.

The principal contribution of this study is the establishment of a fully reversible reference model for externally heated A-CAES systems. Although idealised, such models play an important role in thermodynamic analysis because they define the maximum attainable performance permitted by the second law and provide a consistent benchmark against which the effects of irreversibilities in practical systems can be quantified.

Limitations: The present study is based on an idealised lumped-parameter thermodynamic model. Effects such as component irreversibilities, pressure losses, heat-transfer limitations, thermal leakage, storage losses, and transient operating behaviour are not considered. In addition, the mechanical behaviour and long-term stability of underground storage formations are beyond the scope of the present analysis. Therefore, the results represent fundamental thermodynamic limits rather than direct predictions of a specific engineering system.

Future work: The developed framework can be extended by incorporating realistic component characteristics, multi-temperature thermal energy storage models, and dynamic charging–discharging operation. Further studies may also include the effects of underground storage conditions to evaluate practical solar-assisted A-CAES systems more comprehensively.

Overall, this work provides a clear thermodynamic basis for understanding the performance limits of externally heated A-CAES systems and establishes a benchmark for comparing future practical implementations.

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Abbreviations

The following abbreviations are used in this manuscript:

A-CAES	Advanced Compressed Air Energy Storage
CAES	Compressed Air Energy Storage
TES	Thermal Energy Storage
CoP	Coefficient of Performance
CoP _{rt}	Round-trip Coefficient of Performance
CoP _{th}	Marginal Thermal Coefficient of Performance
T–S diagram	Temperature–Entropy Diagram

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