



Thermodynamic Efficiency of Pumped Heat Electricity Storage

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Pumped heat electricity storage (PHES) has been recently suggested as a potential solution to the large-scale energy storage problem. PHES requires neither underground caverns as compressed air energy storage (CAES) nor kilometer-sized water reservoirs like pumped hydrostorage and can therefore be constructed anywhere in the world. However, since no large PHES system exists yet, and theoretical predictions are scarce, the efficiency of such systems is unknown. Here we formulate a simple thermodynamic model that predicts the efficiency of PHES as a function of the temperature of the thermal energy storage at maximum output power. The resulting equation is free of adjustable parameters and nearly as simple as the well-known Carnot formula. Our theory predicts that for storage temperatures above 400 °C PHES has a higher efficiency than existing CAES and that PHES can even compete with the efficiencies predicted for advanced-adiabatic CAES.

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The solution of the energy-storage problem is an outstanding challenge to our society. The integration of intermittent sources of electric power from renewable energies into the future energy infrastructure requires storage capacities on the order of gigawatt hours (GWh). There are currently two nonchemical techniques—pumped hydrostorage (PHS) and compressed air energy storage (CAES)—that could potentially solve this large-scale energy storage problem. However, PHS requires water reservoirs with volumes on the order of 10^7 m³ and CAES requires underground caverns with 10^6 m³ size. Hence the location of both PHS and CAES cannot be chosen freely and is constrained by geographical and geological limitations.

Recently, a new concept named pumped heat electricity storage (PHES) has been independently invented by several authors (see Refs. [1–7] with Wolf [1] having earliest priority) to overcome this drawback. The principle of the simplest possible PHES [1], differing from the concepts [2,3,6,8] in that it has only one thermal energy storage [9,10], is sketched in Fig. 1. Electric energy is converted to thermal energy using a heat pump cycle operating between the ambient temperature T_0 and a thermal energy storage with temperature T_1 as shown in Fig. 1(a). The thermal energy is then stored. When electric energy is required, a power cycle converts thermal energy back to work and electricity as illustrated in Fig. 1(b).

A thermodynamically identical variant of this technique is shown in Figs. 1(c) and 1(d) and shall be referred to as pumped cryogenic electricity storage [11] (PCES). Here electric energy is used to drive a refrigeration cycle which extracts thermal energy from a cryogenic energy storage. Of course, PCES does not necessarily involve cryogenic temperatures; hence, the abbreviation PCES can also be understood as pumped “cold” electricity storage thereby encompassing all temperatures below ambient. When

electric energy is required, a thermal power cycle operating between the cold storage and the environment converts the cold back to electric energy.

The storage of energy in the form of heat or cold has the advantage that thermal energy storage facilities can be constructed irrespective of the geological conditions of the location and are therefore geographically more flexible than CAES and PHS [9]. It is therefore important to answer the question about the thermodynamic efficiency (also called round-trip efficiency) of PHES and PCES defined as $\Psi = W^-/W^+$ where W^+ is the supplied and W^- the recovered mechanical or electrical energy.

Although the basic idea of PHES appears simple, the prediction of its actual efficiency is a subtle thermodynamic task. This is due to the fact that the computation of the efficiency of an ideal PHES consisting of a Carnot heat pump with efficiency $\beta = T_1/(T_1 - T_0)$ and a Carnot power cycle with efficiency $\eta = (T_1 - T_0)/T_1$ leads to the trivial result $\Psi = \beta\eta = 1$. PHES is one of the rare examples of a thermodynamic cycle for which a reversible theory leads to a trivial result and the calculation of a nontrivial efficiency requires taking into account irreversibilities.

It is obvious that a comprehensive analysis of the efficiencies of different kinds of PHES and PCES involving piston engines [5] Rankine cycles [1,7], gas turbine cycles [4,8], one-vessel storage systems [10] or two-vessel storage systems [4,5] requires full-scale numerical power-plant simulations. However, the purpose of the present work is not to formulate an all-embracing theory of PHES but rather to find the simplest nontrivial thermodynamic model with the smallest number of adjustable parameters that can describe the universal features of PHES.

Our model for the prediction of Ψ is based on Figs. 1(e) and 1(f) and rests on the following assumptions. (i) Both charging and discharging cycles are Carnot

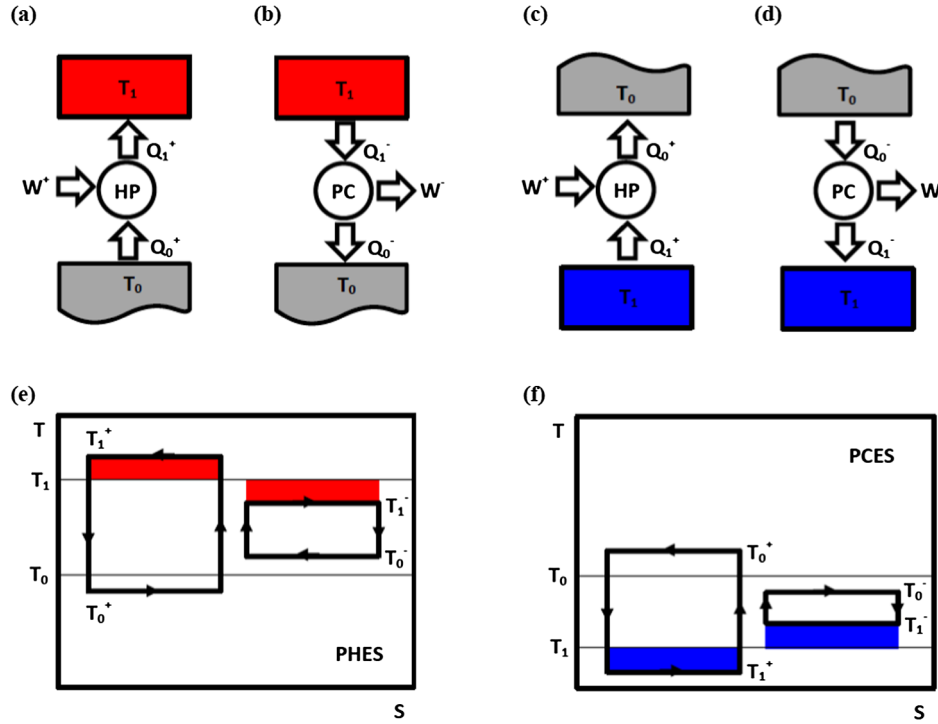


FIG. 1 (color online). Conceptual representation of PHES and PCES systems. (a) Charging and (b) discharging cycle of PHES involving (a) conversion of work W^+ into heat Q_1^+ using a heat pump cycle (HP), storage of the heat at temperature T_1^+ and (b) conversion of heat into work W^- using a power cycle (PC). (c) Charging and (d) discharging cycle of PCES involving (a) conversion of work W^+ into cold Q_1^+ using a HP or refrigeration cycle, storage of the cold at temperature T_1^+ and (b) conversion of cold into work W^- using a PC. (e) and (f) show the temperature-entropy charts corresponding to the assumption that charging and discharging are accomplished by Carnot cycles. Red (e) and blue (f) areas represent the amount of stored heat (e) and cold (f), respectively. The location of the Carnot cycles along the S axis is arbitrary; the cycles are also allowed to overlap. T_0 represents the ambient temperature.

cycles. (ii) The Carnot cycles interact with the environment (at temperature T_0) and the heat storage [at temperature T_1 , cf. Fig. 1(e)] or the cryogenic storage [at temperature T_1 , cf. Fig. 1(f)]. (iii) The only source of irreversibility is heat transfer across the temperature differences between the environment (T_0 , T_1) and the Carnot cycles (T_0^+ , T_1^+ , T_0^- , T_1^-). (iv) Each temperature difference is associated with a linear heat transfer law given by $Q_0^+ = \alpha_0^+(T_0 - T_0^+)$, $Q_1^+ = \alpha_1^+(T_1^+ - T_1)$, $Q_0^- = \alpha_0^-(T_0 - T_0^-)$, $Q_1^- = \alpha_1^-(T_1 - T_1^-)$. (v) The heat transfer coefficients in the heat exchangers are equal for the charging and discharging cycles, i.e., $\alpha_0^+ = \alpha_0^- = \alpha_0$ and $\alpha_1^+ = \alpha_1^- = \alpha_1$. (The quantities α_1 and α_2 differ from the conventional definition of the heat transfer coefficients in that they represent compound quantities involving the heat transfer coefficient, the area of the heat exchangers and the interaction times as in [12].) (vi) The interaction times in the sense of finite-time or endoreversible thermodynamics [13] between the Carnot cycle, the environment, and the energy storage are equal for the charging and discharging cycles. (vii) The discharging cycle operates at maximum power [12,14,15]. (Since there is no maximum-power heat pump cycle in the sense

of finite time thermodynamics, no further assumption on the charging cycle is necessary.)

To compute the round trip efficiency Ψ we need to determine the individual efficiencies $\beta = T_1^+/(T_1^+ - T_0^+)$ and $\eta = (T_1^- - T_0^-)/T_1^-$ of the heat pump and power cycles, respectively. The latter quantity is readily identified as the Curzon-Ahlborn efficiency [12] $\eta = 1 - (T_0/T_1)^{1/2}$ by virtue of assumption (vii). By contrast, no optimization is possible for the heat pump cycle. Hence, the computation of $\beta = T_1^+/(T_1^+ - T_0^+)$ requires the explicit evaluation of both T_1^+ and T_0^+ . As detailed in the Supplemental Material [16], T_1^+ and T_0^+ can be expressed in terms of T_0 , T_1 , α_0 , and α_1 by invoking the conservation of energy in the thermal energy storage in the form $Q_1^+ = Q_1^-$ (assumptions iv and vi) together with the property $Q_1^+/T_1^+ = Q_1^-/T_1^-$ (assumption i) of the heat pump process. Inserting the resulting expressions for T_1^+ and T_0^+ into the definition of β and using the abbreviation $\gamma = \alpha_1/\alpha_0$ it is straightforward to show that the desired quantity Ψ is given by

$$\Psi = \frac{\left(\frac{T_1}{T_0}\right)^{1/2} - \frac{(1+\gamma)}{(2+\gamma^{1/2}+\gamma)}}{\left(\frac{T_1}{T_0}\right)^{1/2} + \frac{(1+\gamma^{1/2})}{(2+\gamma^{1/2}+\gamma)}}. \quad (1)$$

Equation (1) describes the round trip efficiency of PHES as a function of the heat transfer ratio γ . It is interesting to note that the analysis for PCES leads to the same result, i.e., the single equation (1) embraces both PHES and PCES.

For a prediction to be universal it is desirable that it be independent on adjustable parameters like γ . We therefore inquire for which particular value of γ the round trip efficiency attains its maximum. Starting from the condition $\partial\Psi/\partial\gamma = 0$ we obtain, after a lengthy but straightforward computation, that Ψ is highest when $\gamma = 3 - \sqrt{8} \approx 0.172$. This implies that the maximum of Ψ is reached when the heat transfer coefficients obey the relation

$$\alpha_1 = 0.172\alpha_0, \quad (2)$$

i.e., when the heat transfer coefficient to the environment is approximately 5.81 times higher than the heat transfer to the thermal energy storage. It is noteworthy that this value of γ is independent of the temperatures. For the optimum value of γ the round trip efficiency attains its highest possible value. This optimum is given by

$$\Psi = \frac{\theta^{1/2} - \theta_0^{1/2}}{\theta^{1/2} - \theta_0^{1/2} + 1}, \quad (3a)$$

where $\theta = T_1/T_0$ is the temperature ratio and $\theta_0^{1/2} = (\sqrt{8} - 2)/(\sqrt{8} - 1) \approx 0.453$ a numerical constant. Equation (3a) can be rewritten as

$$\Psi = \frac{\left(\frac{T_1}{T_0}\right)^{1/2} - 0.453}{\left(\frac{T_1}{T_0}\right)^{1/2} + 0.547} \quad (3b)$$

and is plotted in Fig. 2. Equations (3) represent the central result of the present work. They describe the roundtrip efficiency of both PHES and PCES as a function of the temperatures of the thermal energy storage and the environment. Observe that Eq. (3) does not contain any adjustable parameters; it is conceptually as simple as the well-known Carnot formula $\eta = 1 - T_0/T_1$ and the Curzon-Ahlborn efficiency [12] $\eta = 1 - (T_0/T_1)^{1/2}$.

As can be inferred from Fig. 2, the round trip efficiency is a monotonically increasing function of the energy storage temperature. It starts with $\Psi = 0$ at $T_1 = [(\sqrt{8} - 2)/(\sqrt{8} - 1)]^2 T_0$, it equals $\Psi \approx 0.354$ at $T_1 = T_0$ and approaches $\Psi = 1$ as $T_1 \rightarrow \infty$.

The fact that the round trip efficiency tends to zero as the storage temperature of PCES approaches its lower limit $T_1 \approx 0.453T_0$ is counterintuitive and is in contrast to the claims of Ref. [11]. (For $T_0 = 20^\circ\text{C}$ this corresponds to a lowest possible temperature $T_1 = -140^\circ\text{C}$, which is above the boiling temperature of liquid N_2 at atmospheric pressure.) However, it is readily inferred from Fig. 1(f) that this feature is a consequence of the equality of the blue areas describing the heat exchange of the Carnot cycles with the cryogenic energy storage. Hence, T_1^+ approaches zero for $T_1 \rightarrow 0.453T_0$. The collapse of the round trip

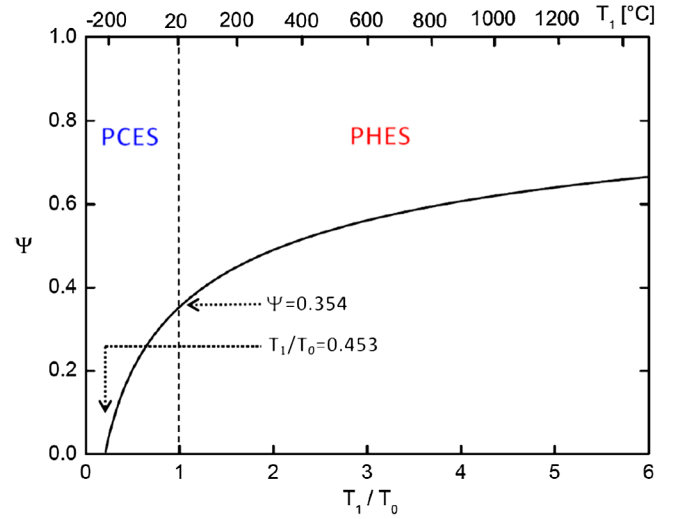


FIG. 2 (color online). Thermodynamic efficiency of PHES and PCES. Round trip efficiency as a function of the storage temperature as computed from Eq. (3). The lower abscissa represents the nondimensional temperature ratio, the upper abscissa represents dimensional temperature for the case $T_0 = 20^\circ\text{C}$.

efficiency at low temperature does not imply that PCES becomes impossible for $T_1 < 0.453T_0$. It rather implies that there is no maximum-power PCES in the sense of finite-time thermodynamics with Carnot cycles satisfying assumption (vi). If this assumption is dropped, a more general model would emerge. It would be interesting to study this generalized case. However, the result would depend on adjustable parameters and the conceptual simplicity of Eq. (3) would be lost.

Another subtle feature of our model is the prediction that for vanishing temperature differences ($\theta \rightarrow 1$) the efficiency attains a nonzero value of about 35%. This implies that the way to cost-effective PHES does not necessarily involve high-temperature heat storage. In line with Refs. [1,7] it is conceivable to design PHES systems in which large quantities of water or seawater are heated from ambient temperature to temperatures of about 90°C using heat pump systems. The thermal energy would be converted back to work using either CO_2 -Rankine [1], ORC [17], or modified OTEC [18] cycles. Although the roundtrip efficiency would be smaller than for high-temperature systems, it is possible that the storage of large quantities of inexpensive water is more cost effective than the storage of costly high-temperature materials.

In Table I we present some key parameters for selected PCES and PHES systems as predicted by our theory for a hypothetical storage system charged with $W^+ = 1$ GWh of electric energy. The first example refers to a PCES system involving a latent heat energy storage using ice. Although the temperature difference to the environment is only 20 K, the roundtrip efficiency amounts to almost 35%. It would be interesting to apply thermoeconomic analysis in order to investigate under which circumstances such systems would

TABLE I. Predicted performance of PHES and PCES systems. Selected parameters of hypothetical storage systems as computed from Eq. (3) using the assumption $W^+ = 1$ GWh. In the first column (l) and (s) refer to latent and sensible thermal energy storage, respectively. The example “salt” refers to LiNO_3 with melting temperature $T_1 = 254$ °C, latent heat of melting $h = 360$ kJ/kg and density $\rho = 1780$ kg/m³. The examples “granite” and “refractory” refer to the systems discussed in Refs. [4,5], respectively. The quantities m , e_m , V , and e_v (mass of storage material, effective specific electric storage capacity, volume of storage material, effective volumetric electric storage capacity, respectively) for these two examples are in brackets because they comprise only the masses of the high-temperature thermal energy storage. The figures for the liquid metal latent thermal energy storage (aluminium) are computed for $h = 400$ kJ/kg and $\rho = 2700$ kg/m³. The temperatures T_1 for the sensible thermal energy storages are taken as arithmetic means between 20 °C and 100 °C (water), 55 °C and 500 °C (granite) [5], and 500 °C and 1000 °C (refractory) [4]. The temperatures T_0 for the sensible storages are taken as the environmental temperature (water) or as the arithmetic means between -166 °C and 55 °C (granite) and -70 °C and 30 °C (refractory).

	T_0 [°C]	T_1 [°C]	Ψ [%]	m [tons]	e_m [Wh/kg]	V [m ³]	e_v [Wh/m ³]
PCES Ice (l)	20	0	34	106 000	3.2	106 000	3.2
PHES Water (s)	20	60	38	66 000	5.8	66 000	5.8
PHES Salt (l)	20	254	47	18 500	26	10 400	45
PHES Granite (s)	-55.5	277.5	53	(11 600)	(46)	(4600)	(114)
PHES Aluminum (l)	20	660	57	11 700	49	4300	133
PHES Refractory (s)	-20	750	61	(8780)	(137)	(3500)	(174)

be commercially feasible. It can also be seen from Table I that PHES using hot water energy storage has a significantly lower efficiency than PHS. On the other hand, the effective electric energy storage density of hot water (roughly 6 Wh/kg) is almost twenty times higher than of cold water in PHS elevated by 100 meters. This implies that hot-water PHES requires significantly less space than PHS. The remaining examples from Table I indicate that high storage temperature is beneficial both for achieving a high roundtrip efficiency and a high storage density. The examples labeled granite and refractory correspond to the PHES systems discussed in Refs. [4,5], respectively. The example labeled aluminium indicates that liquid metal latent heat storages [19–21] have the potential to further enhance the efficiency of PHES while retaining the temperature at a lower level than the maximum temperature of the solid-state systems [2,3,22]. For $T_1 > 345$ °C the round trip efficiency of PHES exceeds that of CAES (50%) and for $T_1 > 1000$ °C it is of the same order as the efficiency predicted for advanced-adiabatic CAES [23].

A final comment is in order about the influence of heat losses from the thermal energy storage. The present model can be generalized to take into account these losses by replacing the energy-conservation condition $Q_1^- = Q_1^+$ with $Q_1^- = (1 - \varepsilon)Q_1^+$ where ε is a phenomenological coefficient describing the energy losses. Equation (3) from the Supplemental Material [16] then takes the form $(1 - \varepsilon)T_1^+ = (2 - \varepsilon)T_1 - T_1^-$ and the round-trip efficiency becomes an additional function of ε . We find that the resulting expression for Ψ is not universal: it depends explicitly on the heat transfer coefficients and cannot be simplified to the canonical form $\Psi(\theta, \varepsilon)$. However, an asymptotic analysis for the case $\varepsilon \ll 1$ demonstrates that the deterioration of Ψ can be roughly described by multiplying Eq. (3) with a factor $(1 - \varepsilon)$.

In summary, we have formulated a simple finite-time thermodynamics model that predicts the scaling of the performance of PHES and PCES. Finite-time thermodynamics [12–15,24] have been repeatedly criticized [24] for providing oversimplified models of complex power plant systems, for making exaggerated claims of universality and for lacking predictive power. We admit that our model can be criticized for its simplicity. However, we would like to emphasize that the criticism about the lack of specific predictions does not apply to the present work. Whereas previous finite-time thermodynamic models have successfully *postdicted* the performance of existing power plants, our Eqs. (2) and (3) make unbiased *predictions* which will either stand the test of time or be humiliated by future measurements and full-scale simulations.

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