

Thermo-economic assessments of pumped-thermal electricity storage systems employing sensible heat storage materials

Yongliang Zhao ^{a, b}, Jian Song ^b, Ming Liu ^a, Yao Zhao ^b, Andreas V. Olympios ^b, Paul Sapin ^b, Junjie Yan ^{a, *}, Christos N. Markides ^{b, **}

^a State Key Laboratory of Multiphase Flow in Power Engineering, Xi'an Jiaotong University, Xi'an, 710049, China

^b Clean Energy Processes (CEP) Laboratory, Department of Chemical Engineering, Imperial College London, London, SW7 2AZ, UK

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ABSTRACT

Three distinct pumped-thermal electricity storage (PTES) system variants based on currently available sensible heat storage materials are presented: (i) Joule-Brayton PTES systems with solid thermal reservoirs; (ii) Joule-Brayton PTES systems with liquid thermal stores; and (iii) transcritical Rankine PTES systems with liquid thermal stores. Parametric design optimisation is performed for each PTES system variant considering various system configurations, working fluids and storage media from a thermodynamic perspective. The results show that amongst the investigated systems, the recuperative transcritical Rankine PTES system with CO₂ as the working fluid and Therminol VP-1 as the storage material achieves the highest roundtrip efficiency of 68%. Further to the optimal thermodynamic performance of these system, their corresponding capital costs are also evaluated. The economic performance comparisons of selected optimal PTES designs reveal that the recuperative transcritical Rankine PTES system with CO₂ and Therminol VP-1 exhibits the lowest capital cost of 209 M\$ for the given power capacity (50 MW) and discharge duration (6 h). The influences of the power capacity and discharge duration are also investigated, with results showing that the lowest power and energy capital costs are 3790 \$/kW (discharge duration of 2 h) and 396 \$/kWh (discharge duration of 12 h), respectively.

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1. Introduction

As the share of electricity generation from renewable energy sources (RESs), such as solar, wind, wave and tidal energy, continues to grow steadily worldwide [1], the increasing penetration of RES generation introduces increasingly substantial challenges in terms of grid stability, security and the economics of electric power systems, due to the variable, stochastic and reduced predictability nature of RESs [2]. Therefore, sufficient power system flexibility is required to allow a wider utilisation of these non-dispatchable sources.

Flexibility can be provided by different technological solutions placed at the supply, network or demand sides of the power system [3]. Supply-side improvements of operational flexibility, e.g., through thermal regulation of gas- and coal-fired power plants, is a

key measure for allowing an energy future with a growing share of renewables [4–7]. Moreover, advanced technologies for demand-side management, such as power-to-gas and power-to-heat, can provide flexibility in the management of surplus renewable generation [8–10], whilst grid-scale energy storage technologies are indispensable for the efficient integration of RESs into the grid and play a pivotal role in balancing inflexible or intermittent supply with fluctuating demand [11]. Luo et al. [12] reviewed the development of electrical energy storage technologies and their application potential in the operation of power systems. Although several conventional rechargeable batteries, flow batteries, flywheel energy storage, solar/fuel cells and high-power/high-energy supercapacitors have been proposed [13], these are still immature, short-duration and only available for small-scale applications at present [14]. For large-scale electricity storage, pumped hydro storage (PHS) [15] and compressed air energy storage (CAES) [16] are two commercially-available, widely-deployed and highly-durable electricity storage technologies. However, these have several noteworthy drawbacks, particularly, geographical and geological constraints, environmental issues, long construction

* Corresponding author.

** Corresponding author.

E-mail addresses: yanjj@mail.xjtu.edu.cn (J. Yan), c.markides@imperial.ac.uk (C.N. Markides).

Nomenclature*Symbols*

A	Heat transfer area [m^2]
C	Heat capacity [J/K]
C_{BM}	Bare module cost [$\text{\$}$]
c_p	Isobaric specific heat capacity [$\text{J}/(\text{kg}\cdot\text{K})$]
C_R	Heat capacity ratio [–]
C_T	Total capital cost [$\text{\$}$]
c_v	Isochoric specific heat capacity [$\text{J}/(\text{kg}\cdot\text{K})$]
d	Tube diameter [m]
E	Stored energy [J]
h	Specific enthalpy [J/kg]
I	Chemical engineering plant cost index [–]
k	Thermal conductivity [$\text{W}/(\text{m}\cdot\text{K})$]
$\dot{m}$	Mass flow rate [kg/s]
NTU	Number of transfer units [–]
P	Pressure [Pa]
$\dot{Q}$	Heat transfer rates [W]
r_{fs}	Fouling resistance on the shell side [$(\text{m}^2\cdot\text{K})/\text{W}$]
r_{ft}	Fouling resistance on the tube side [$(\text{m}^2\cdot\text{K})/\text{W}$]
T	Temperature [K]
t	Time [s]
U	Overall heat transfer coefficient [$\text{W}/(\text{m}^2\cdot\text{K})$]
V	Volume [m^3]
v	Specific volume [m^3/kg]
w	Specific work [J/kg]
α	Heat transfer coefficient [$\text{W}/(\text{m}^2\cdot\text{K})$]
γ	Specific heat ratio [–]
δ	Wall thickness [m]
ε	Heat transfer effectiveness [–]
ε_{VF}	Void fraction of the packed bed [%]
η_i	Isentropic efficiency [%]
η_p	Polytropic efficiency [%]
λ	Heat conductivity of wall material [$\text{W}/(\text{m}\cdot\text{K})$]
μ	Viscosity [$\text{Pa}\cdot\text{s}$]
π	Pressure ratio [–]
ρ	Density [kg/m^3]
χ	Roundtrip efficiency [%]
ψ	Temperature ratio [–]
ω	Work ratio [–]

Abbreviations

AHX	Auxiliary heat exchanger
CAES	Compressed air energy storage
CHEST	Compressed heat energy storage
CHX	Cold heat exchanger
COMP	Compressor
CR	Cold reservoir

ECC	Energy capital cost
EXP	Expander
HHX	Hot heat exchanger
HT	High temperature
HR	Hot reservoir
HX	Heat exchanger
IT	Intermediate temperature
LAES	Liquid air energy storage
LCOE	Levelised cost of energy
LCOS	Levelised cost of storage
LT	Low temperature
LTS	Liquid thermal store
ORC	Organic Rankine cycle
PCC	Power capital cost
PCM	Phase-change material
PHS	Pumped hydro storage
PPTD	Pinch point temperature difference
PTES	Pumped-thermal electricity storage
RES	Renewable energy source
SC	System configuration
STR	Solid thermal reservoir
TCC	Total capital cost
TMES	Thermo-mechanical energy storage

Subscript

c	Compressor
chg	Charge process
CR	Cold reservoir
cr	Critical
cs	Cold side
dis	Discharge process
e	Expander
g	Working fluid
HR	Hot reservoir
hs	Hot side
i	Inside
in	Inlet
is	Isentropic process
m	Mean
max	Maximum
min	Minimum
N	Nominal
o	Outside
out	Outlet
pack	Packed bed
s	Storage medium/material
ss	Shell-side
tank	Storage tank
ts	Tube-side

times and high initial capital costs [17,18]. In this context, a remarkable effort has been made to develop alternative electricity storage technologies, including two recently-proposed medium-to large-scale and longer-duration thermo-mechanical energy storage (TMES) technologies, namely liquid air energy storage (LAES) [19] and pumped-thermal electricity storage (PTES) [20–22]. LAES and PTES systems involve few geographical constraints, long lifetimes and flexible power ratings, which are characteristics that suggest they can be deployed as important components in future electric power systems [23].

PTES (also referred to as ‘Carnot battery’, ‘pumped heat electricity storage’, ‘electrothermal energy storage’, ‘thermo-electrical energy storage’ or ‘compressed heat energy storage’ in the literature) stores electricity in the form of sensible and/or latent heat in insulated thermal reservoirs containing appropriate storage media, such as solid packed beds or liquid storage tanks. A detailed list of PTES variants reviewed and their performance metrics are provided in Table 1, which indicates that four types of cycles are mainly being considered. With respect to the Carnot cycle, which is an idealisation made in several of the more theoretical studies in the

Table 1
Literature overview of PTES system variants and performance metrics.

Cycle	Reference(s)	Working fluid(s)	Thermal energy store	Performance metrics
Carnot	Thess [24]	Ideal gas	Solid/liquid storage materials	Roundtrip efficiency (34%–61%)
	Guo et al. [25,26]	Ideal gas	N.A.	Roundtrip efficiency (<80%)
Joule-Brayton	Desrues et al. [27]	Argon	Tank, refractory bricks	Storage efficiency (67%)
	Howes [28]	Argon	Packed bed, granite	Roundtrip efficiency (72%)
	White, McTigue, Farres-Antunez et al. [29–33]	Argon	Packed bed, gravel	Total cyclic losses (6%), roundtrip efficiency (~70%)
	Ni and Caram [34]	Argon and air	Tank, refractory materials	Roundtrip efficiency (64%–82%)
	Guo et al. [35]	Ideal gas	N.A.	Roundtrip efficiency (<40%)
	Smallbone et al. [36,37]	Argon	Packed bed	Roundtrip efficiency (52%–72%), LCOS (9–11 €/kWh)
	Benato and Stoppato [38–40]	Argon and air	Packed bed, nine solid storage materials	Roundtrip efficiency (70%–80%), energy capital cost (50–180 \$/kWh)
	Chen et al. [41]	Argon and air	Packed bed, α -Alumina	Roundtrip efficiency (42%–50%)
	Georgiou, Sapin, Markides et al. [42–45]	Argon	Packed bed, magnetite	Roundtrip efficiency (~80%), power capital cost (3.0–6.1 \$K/kW), energy capital cost (0.5–1.1 \$K/kWh), LCOS (0.4 \$/kWh)
	Wang et al. [46–48]	Argon and He	Packed bed, basalt	Roundtrip efficiency (39%–57%)
	Steinmann et al. [49]	Argon and air	Packed bed regenerator and storage tank, molten salt	Roundtrip efficiency (45%–57%)
	Dietrich et al. [50,51]	Air	Packed bed, basalt chips without quartz	Roundtrip efficiency (~43%)
	Laughlin [52]	Argon and N ₂	Storage tank, solar salt and hexane	Energy marginal cost (13 \$/kWh)
	Farres-Antunez et al. [33,53–57]	He, N ₂ and air	Storage tank, molten salt, thermal oil, and pentane	Roundtrip efficiency (55%–62%)
	Xue [58,59]	Argon and air	Tank, solar salt, mineral oil, propane, and methanol	Roundtrip efficiency (50%–60%)
	Salomone-González et al. [60]	Argon and air	Storage tank, solar salt and three cryogenic liquids	Roundtrip efficiency (35%–40%)
	Zhao et al. [61]	Argon	Storage tank, solar salt, Therminol 66 and butane	Modified overall exergetic efficiency (34%–57%)
	McTigue et al. [62]	CO ₂	Storage tanks, molten salt, and synthetic fluids	Roundtrip efficiency (60%–78%)
Transcritical Rankine	Mercangöz, Wright, Hemrle et al. [63,64]	CO ₂	Storage tanks, hot water and ice	Roundtrip efficiency (51% for pilot and 65% for commercial plants)
	Morandin et al. [65–67]	CO ₂	Storage tanks, hot water and ice	Roundtrip efficiency (48%–64%), total purchase costs (27–38 M\$)
	Kim et al. [68]	CO ₂	Storage tanks, hot water and ice	Roundtrip efficiency (65%–73%)
	Baik et al. [69]	CO ₂	Storage tanks, water	Roundtrip efficiency (<30%)
	Tauveron et al. [70,71]	CO ₂	Ground heat storage and PCMs	Roundtrip efficiency (43%–56%)
	Abarr et al. [72–74]	NH ₃	Tube-in-concrete hot TES	Stand-alone ES efficiency (52%–66%), LCOE (32–50 \$/MWh)
	Wang and Zhang [75]	CO ₂ and NH ₃	Storage tank and heat exchangers	Roundtrip energy ratio (139%) ^a
	Koen et al. [76]	CO ₂ and R1311	Tank, water and Therminol oils	Roundtrip efficiency (58%)
Subcritical Rankine	Steinmann et al. [77–79]	NH ₃ /water	Sensible/latent heat storage units	Roundtrip energy ratio (<1.25) ^a
	Vandersickela et al. [80]	R142b/R245fa	Pressurised water tank	Roundtrip efficiency (26%)
	Vinnemeier et al. [81,82]	CO ₂ /steam	Storage tank, water	Roundtrip efficiency (62%)
	Roskosch et al. [83,84]	Organic fluid, Butane, R245fa	Storage heat exchanger and the surrounding	Roundtrip efficiency (37%–56%)
	Staub et al. [85]	R365mfc	Storage tank, water	Power-to-power efficiency (50%–70%)
	Dumont et al. [86,87]	R1233zd(E) and R1234yf	Storage tank, water	Roundtrip energy ratio (70%–200%) ^a
	Eppinger et al. [88]	R1233zd(E), Novec 649 and R365mfc	Storage tank, sensible (pressurised water) and latent heat storage units	Roundtrip efficiency (62%–80%)
	Hassan, Payá et al. [89,90]	Organic fluids	Pressurised water tanks and PCMs	Net power ratio (0.8–1.3)
	Frate et al. [91–94]	Organic fluids	Storage tank, thermal integration	Roundtrip energy ratio (60%–130%) ^a , energy capital cost (136–484 €/kWh)

^a When additional external heat sources and/or are integrated with PTES systems, the defined roundtrip efficiency can exceed 100%, which is why the paper reports the thermodynamic performance using a so-called roundtrip energy ratio, which is the ratio of the electrical energy output recovered during discharge to the electrical energy input during charge.

literature, isothermal heat transfer is assumed between thermal reservoirs and working fluid in the cycle [24], with results showing roundtrip efficiencies in the range of 34%–80% depending on various parameters and conditions [24–26].

The Joule-Brayton cycle has been widely considered for high-temperature (HT) PTES, and the world's first pilot plant based on this cycle is being commissioned by Newcastle University [36,37,95]. The most widely used working fluids in Joule-Brayton cycles are argon [27–34,36,38–43,46,49,52,58], air [34,39,41,49,53,58], nitrogen [52], and helium [46,53] whilst supercritical carbon dioxide (sCO₂) is also being investigated [62]. Many researchers have focused on PTES systems with solid thermal reservoirs (STRs), usually employing packed beds as sensible solid

stores. Desrues et al. [27] was one of the first to propose such a type of PTES system and developed a simple numerical model to examine its feasibility, and obtained a storage efficiency of 67%. A more practical thermodynamic model was then used to derive the analytic expressions for the roundtrip efficiency and power output of these PTES systems [35], and optimal operating regions were obtained. Based on this, they suggested that the heat pump should be operated at or near a point of the maximum coefficient of performance and the heat engine at an operating point between the maximum efficiency and maximum power output. Focusing on how various sources of thermodynamic irreversibility affect roundtrip efficiency, White et al. [29–33,96–99] developed detailed models of such PTES systems and found that roundtrip

efficiency was particularly susceptible to compression and expansion polytropic efficiency, especially at low-temperature ratios. In conventional PTES systems, high temperatures mean high pressure ratios, which in turn cause high purchasing costs for the compressor and the hot reservoir (HR). To overcome these issues, an electric heater can be inserted after the compressor [38–40], which enables a considerable improvement in the roundtrip efficiency (70%–80%) and energy capital cost (50–180 \$/kWh) of these systems.

Organic Rankine cycles (ORCs) are an effective option for low-to-medium-grade heat conversion [100–102], and have been examined in the context of HT-PTES in Ref. [41], specifically for recovering and converting rejected heat produced due to system irreversibility. The simulation results showed that combining HT-PTES and parallel ORC systems can lead to maximum roundtrip efficiencies of almost 48%. Beyond the modelling and simulation studies, experimental work focusing on early prototypes was performed to assess the potential performance of PTES technology in Ref. [28], with a roundtrip efficiency stated as theoretically reaching up to 72%.

Because of the unsteady operation of the thermal reservoirs, transient analyses of these systems have also been performed. Wang et al. [46,47] proposed transient models for PTES systems coupling dynamics, heat transfer and thermodynamics. The results from this work were used to obtain optimal values of the delivery compression ratio, particle sizes, length-to-diameter ratios of the reservoirs, and discharging durations corresponding to the maximum roundtrip efficiency. Dietrich et al. [50,51] also considered the transient operation of these systems and reported that the roundtrip efficiencies reached close to 43%.

Exergo-economic analyses have suggested that these systems have higher power-specific costs than established storage technologies. Similarly, based on conventional costing methods, Markides et al. [43,45] presented a combined thermo-economic modelling as a unified basis to compare PTES and LAES systems in terms of system efficiency and costs. This effort showed that, as proposed by the developers, PTES had the potential to achieve higher roundtrip efficiencies, but LAES (12 MW, 50 MWh) potentially had lower capital costs and levelised costs of storage than PTES (2 MW, 11.5 MWh). Following this work, the proposed PTES and LAES technologies were integrated into a whole-electricity system model to assess their system-level value in various scenarios for system decarbonisation [44]. The results showed that the system value of PTES was slightly higher than that of LAES, promoted by a higher storage duration and efficiency, at least in the UK which was the focus of this study.

In Joule-Brayton PTES systems, thermal energy can also be stored in liquid thermal stores (LTSs). LTSs have fewer limitations in the choice of the cycle operating pressure without the need to employ pressure vessels (which are necessary when using STRs), which means that the costs of the thermal stores can be reduced. Laughlin [52] was the first to propose a Joule-Brayton PTES system with LTSs and considered a closed-cycle heat engine transferring heat from a cryogenic storage fluid to molten solar salt. Following this work, Salomone-González et al. [60] presented a thermodynamic model of a steady-state PTES system with liquid storage media, and an analytically derived global roundtrip efficiency in terms of the isentropic efficiencies of the compressor and turbine, pressure losses and the effectiveness of the heat exchangers, and the coupling between the working fluid and the storage liquid media. By using this expression, they obtained efficiencies in the range of 35%–40%. Detailed conventional and advanced exergy analyses of this PTES-LTS technology have also been performed [61], which indicated that improving the technological readiness level of the compressor and turbine (higher isentropic efficiency)

and heat exchangers (increased effectiveness, decreased pressure loss) to suggested maximum presently attainable values, led to improved overall exergetic efficiencies of up to 57%.

Argon and air are selected as the working fluids in most previous studies, and a PTES-LTS variant based on sCO₂ was proposed by McTigue et al. [62], who found that sCO₂-PTES cycles achieved high roundtrip efficiencies for a given hot storage temperature (up to 78% at 560 °C). Furthermore, a comparison of adiabatic compressed air energy storage (A-CAES) and PTES systems with LTSs [58,59] showed that A-CAES systems usually exhibited higher overall efficiencies and lower unit storage costs, but this must be seen in light of the geographical restrictions of such A-CAES systems that rely on compressed air storage in underground caverns.

The transcritical Rankine cycle is an engaging PTES variant due to the potential for an improved thermal match between the cycle temperature profile and that of the thermal storage material(s). A novel PTES system using hot water and ice storage combined with a transcritical CO₂ cycle was first proposed by Mercangöz et al. [63]. Based on this original concept, Morandin et al. [65–67] performed a thermo-economic optimisation of system configurations (SCs) and found that the roundtrip efficiency of the optimal PTES system would not exceed a maximum value of 64%, which could be reached with a maximum total purchasing cost of around 34 million \$ (2009 basis). A further case study by Baik et al. [69] showed there existed an optimal temperature (70 °C) of the lowest-temperature hot storage tank to achieve the maximum roundtrip efficiency. These studies of transcritical-cycle PTES systems were based on conventional (nominally adiabatic) compression and expansion processes. However, Kim et al. [68] proposed a novel isothermal PTES system, and the results suggested that the roundtrip efficiency of such a system was 3% higher (in absolute efficiency points) relative to previously proposed conventional PTES systems with the same design conditions. By using ammonia as the working fluid, Abarr et al. [72–74] presented the baseline performance, sensitivity and levelised cost of energy (LCOE) analyses of a PTES and bottoming system (Bot-PTES), and their results showed that the standalone energy storage efficiency and LCOE were in the ranges of 51%–66% and 32–50 \$/MWh, respectively. Moreover, a comparative investigation of transcritical PTES systems with different working fluids demonstrated that trifluoriodomethane (R13II) was a promising option with a peak roundtrip efficiency of 58% [76]. Although PTES has become a hot topic for large-scale energy storage, few systems have yielded roundtrip efficiencies that are comparable with those PHS (except for systems utilising external heat sources/sinks). By employing ambient thermal energy and liquid-natural-gas cold energy, Wang and Zhang [75] proposed a PTES system with a cascade design of a transcritical CO₂ Rankine cycle and a subcritical NH₃ Rankine cycle and the optimised baseline case showed that roundtrip efficiency could reach 139%.

Subcritical Rankine-cycle-based PTES systems are promising candidates for intermediate-temperature (IT) and low-temperature (LT) sensible and latent heat energy storage (<500 K). Using sensible heat storage tanks, Roskosch et al. [83,84] analysed the general thermodynamic potential and limits of a PTES system consisting of a compression heat pump, a thermal storage unit and an ORC system, with results showing roundtrip efficiencies in the range of 37%–56%. As with the transcritical systems mentioned above, the electrical roundtrip efficiency of classical PTES systems does not exceed ~60%, but when using low-grade thermal energy integration, the roundtrip energy ratio can exceed 1. Dumont et al. [87,103] investigated a thermally integrated PTES system using hot storage or cold storage and provided a performance map to estimate which type of configuration is more profitable depending on the glide, the waste heat temperature and the air temperature. Similarly, a thermally integrated PTES system combined with a heat

pump and an ORC system with a simple hot water storage tank was proposed [85], and the results showed that applications of the proposed system in the MW range with more efficient turbines and dynamic compressors increase the power-to-power efficiency to promising values over 70%, making them competitive with PHS. In addition, Frate et al. [91–93] designed a Rankine-based PTES system with low-grade thermal energy integration and performed a multi-objective optimisation of the system to maximise electrical roundtrip efficiency and minimise system cost. They found that large-size configurations could result in equipment purchasing costs as low as 140 €/kWh and roundtrip efficiencies of 60%.

Subcritical Rankine PTES systems can be used for the autarkic cogeneration of heat and power [80] and integrated into thermal plants (efficiency could reach 50%–60%) [81,82]. In combination with latent heat storage, subcritical PTES is a preferred choice in such applications. Eppinger et al. [88] presented a comparison of latent and sensible thermal storages for PTES with heat pump-ORC-systems, and their results showed that for the most sensible-storage cases, *trans*-1-Chloro-3,3,3-trifluoropropene (R1233zd(E)) was the best choice working fluid with a small trade-off between reduced efficiency for the benefit of better environmental sustainability and safety. For latent-heat storage, cyclopentane achieved the highest efficiency. A similar PTES system comprising an HT heat pump, an ORC system and an IT thermal energy storage also showed that for latent heat storage at 133 °C, R-1233zd(E) and R-1234ze(Z) presented the best performance [89,90]. Such PTES-ORC systems can be used as a sector-coupling technology for heat and electricity through LT heat from external sources [77–79].

As indicated by the aforementioned literature, research in this field has mainly focused on investigating how to improve the roundtrip efficiency of PTES systems whilst paying less attention to cost, which is arguably an even more considerable factor in the deployment of large-scale electricity storage systems. In addition, although the thermodynamic performance of PTES systems changes for different SCs, working fluids and storage media, comparison of optimised variants is scarce. In summary, most studies only focus on specific PTES SCs, such that a comprehensive comparison of the thermodynamic and economic performance of different PTES variants is still lacking from the literature.

In this paper, thermodynamic and economic performance of different PTES systems are investigated and compared using a unified methodology and based on a set of fixed criteria and assumptions. Three distinct PTES SCs based on available sensible heat storage materials are presented: (i) reversible Joule-Brayton PTES with STRs, (ii) reversible Joule-Brayton PTES with LTSs, and (iii) reversible transcritical Rankine PTES with LTSs. The main novelties are three-fold: (1) Parametric optimisation based on thermodynamic performance criteria is performed for each system type whilst considering different system layouts/configurations, working fluids and storage media. (2) The sizes of the main components of these optimal PTES systems are evaluated, and the corresponding capital costs are estimated using the same four different costing methods. This approach allows a more robust determination and fair comparison of the thermodynamic and economic performance of the various PTES system types for selected cases. (3) The effects of discharge duration and power capacity on the power and energy capital costs of the three PTES systems are investigated to assess the economic feasibility of PTES for large-scale bulk energy storage, and compared with other large-scale energy storage technologies. The present study provides detailed guidance to support the design and development of PTES technology from thermodynamic and economic perspectives.

2. Description of PTES system variants

2.1. PTES system configurations

Three main PTES system variants employing sensible heat storage are considered in this work: (i) reversible Joule-Brayton PTES systems with STRs, (ii) reversible Joule-Brayton PTES systems with LTSs, and (iii) reversible transcritical Rankine PTES systems with LTSs. Reversible transcritical Rankine PTES systems with STRs are not selected because of the limitation of high working pressures (up to 20.0 MPa) and corresponding costs in STRs. The detailed selection of investigated storage materials is described in [Section 2.2](#).

A generalised reversible Joule-Brayton PTES system with STRs is shown in [Fig. 1](#). The system consists of two thermal reservoirs (hot and cold packed beds), four turbomachines (one compressor/expander pair used during the charge process and another one during the discharge process), two heat exchangers (in the baseline design) and a recuperator. During the charge process, as shown in [Fig. 1\(a\)](#), the working fluid leaving the cold reservoir (CR) is first compressed in an electrically-driven compressor. Direct heat transfer then occurs from the compressed and hot working fluid to the storage material in the HR, such that thermal energy is stored in the HR whilst the temperature of the working fluid reduces as it flows through the HR. At the exit of the HR, the high-pressure, LT working fluid is expanded, producing useful power and reducing the temperature of the working fluid to its lowest value. Finally, the working fluid flows through and cools down by direct heat transfer to the cold storage material in the CR. Later, during the discharge process, as shown in [Fig. 1\(b\)](#), cold working fluid at the exit from the CR is first compressed and then flows through the HR where it is heated by absorbing thermal energy. At the exit of the HR, the expansion of the high-pressure, HT working fluid produces mechanical power, which is converted into electricity by an electrical generator. Subsequently, the working fluid is cooled down by transferring heat to the storage material in the CR. To keep the temperature profile of the HR and CR between charge and discharge processes similar, auxiliary heat exchangers are used to reject/absorb heat to/from the environment. Moreover, a recuperator can be added to (internally) heat and cool the working fluid prior to the compressor and expander during the charge and discharge processes.

A generalised reversible Joule-Brayton PTES system with LTSs is shown in [Fig. 2](#). Compared with the reversible Joule-Brayton PTES system with STRs (shown earlier in [Fig. 1](#)), the difference here is that the packed-bed thermal stores are replaced by gas-liquid heat exchangers coupled to two liquid storage tanks (referred to here as HT and LT tanks), thus indirect heat transfer occurs in each heat exchanger between the liquid storage material and the working fluid. One advantage of this design is that the pressures in the stores are decoupled from those in the cycle itself, such that the LTSs can operate at lower pressures. This design does not involve high-pressure vessels, and hence the costs of the thermal stores can be reduced. However, liquid storage materials have limited operating temperature ranges compared with solid storage materials, and the heat transfer performance between the liquid storage material and the working fluid deteriorates. Therefore, a more detailed investigation and comparison of the thermodynamic and economic performance of the two distinct PTES systems is performed in this work.

It has been suggested in Ref. [63] that transcritical cycles can be designed to have a better thermal match the temperature profiles

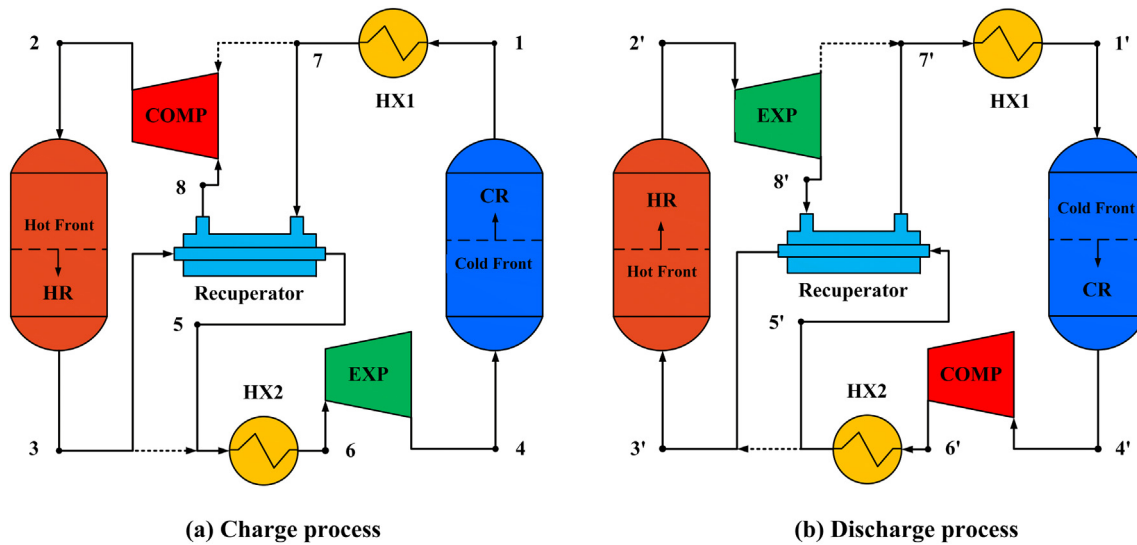


Fig. 1. Layout of generalised reversible Joule-Brayton PTES systems with STRs. HR/CR: hot/cold reservoir; COMP: compressor; EXP: expander; HX: heat exchanger.

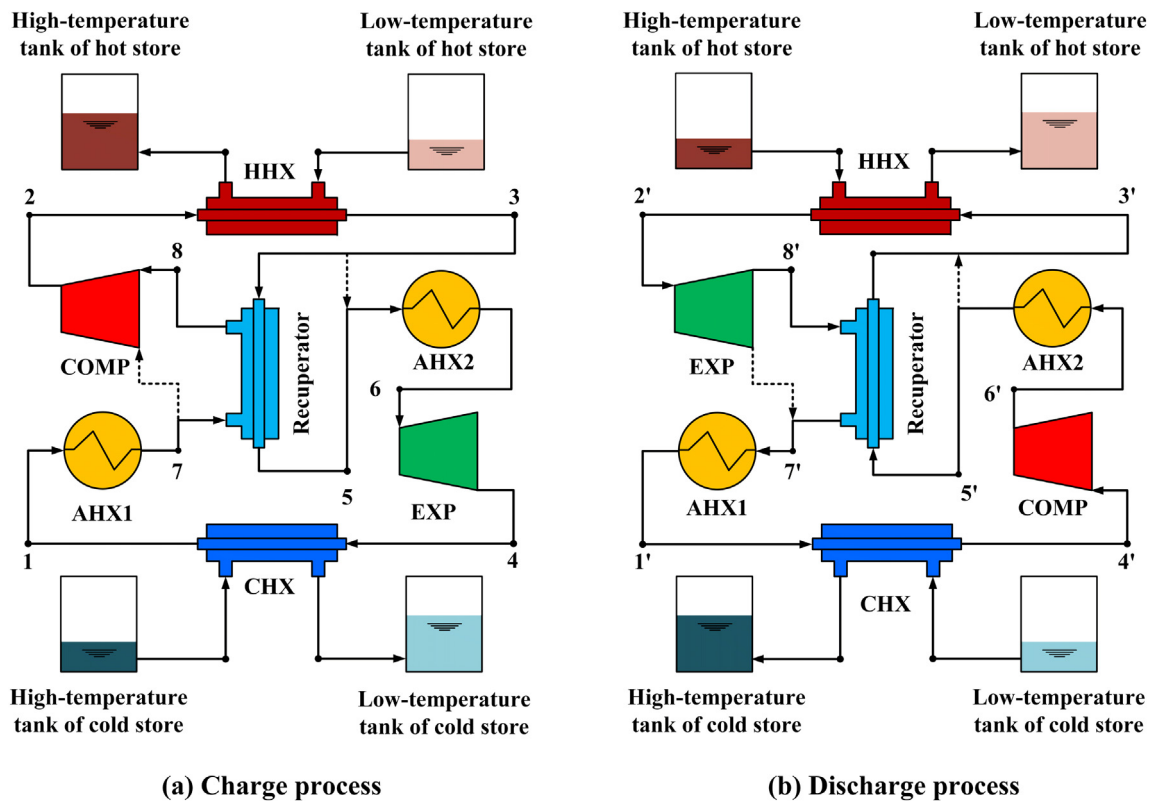


Fig. 2. Layout of generalised reversible Joule-Brayton PTES systems with LTSs. AHX: auxiliary heat exchanger; HHX/CHX: hot/cold heat exchanger.

through the stores. Therefore, a third potential system, which comprises a reversible transcritical Rankine PTES system with LTSs, is considered, as shown in Fig. 3. Compared with the Joule-Brayton PTES system with LTSs, the compressor previously required for the discharge cycle is replaced by a pump, which reduces the required power consumption, the single-phase cold-side heat exchanger is replaced by a heat exchanger with phase-change heat transfer, namely an evaporator during charge process and a condenser during discharge process, and the ambient environment is used as the heat sink. Moreover, the operating pressures (up to 20.0 MPa)

are typically higher than those of reversible Joule-Brayton PTES systems (~1.0 MPa).

2.2. Selection of storage media

There are three main categories of thermal energy storage: sensible, latent, and chemical heat storage [104,105]. In this investigation, sensible heat storage is focused, as sensible heat storage is the most technically and commercially mature option, with materials that are readily available, inexpensive, non-toxic,

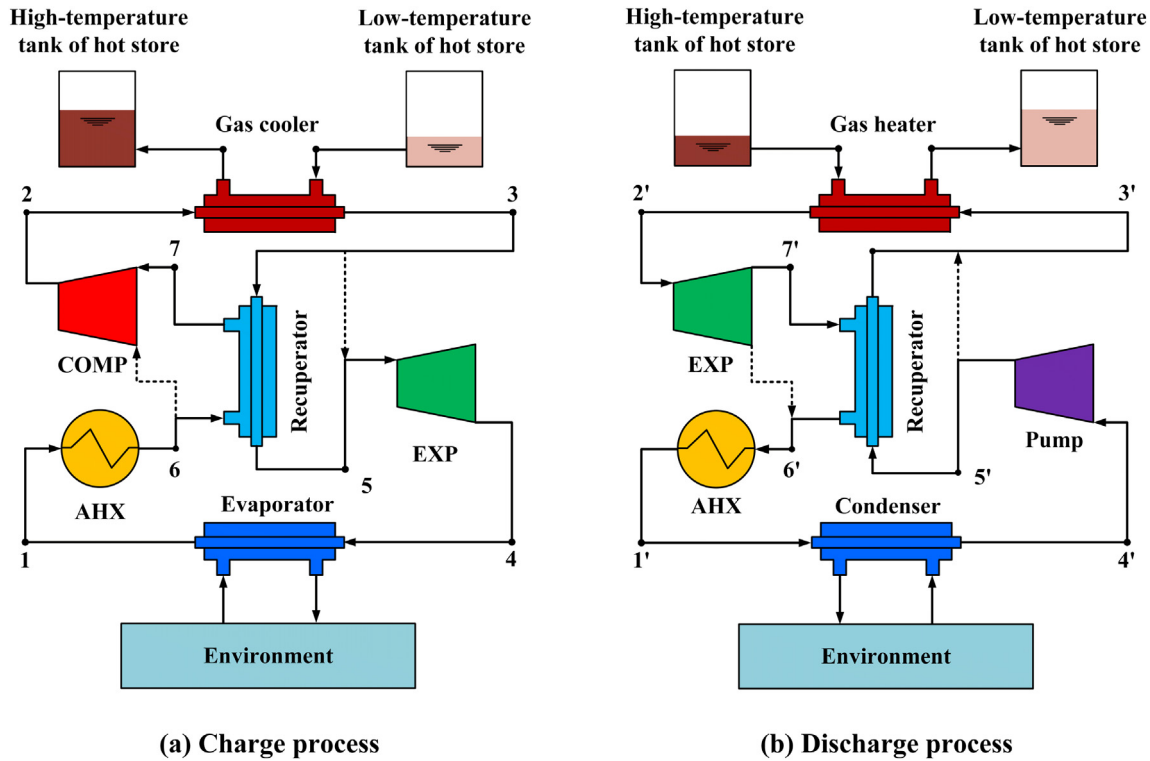


Fig. 3. Layout of generalised reversible transcritical Rankine PTES systems with LTSs.

non-flammable and with good thermal properties. Latent heat storage is becoming more widely available and phase-change materials (PCMs) can be used as storage media in the systems of interest, especially to store heat during the two-phase heat exchange processes in the transcritical Rankine cycle (N.B.: cascaded tanks can also be used to store heat along a temperature gradient), including at high temperatures [106], but this is out of the scope of the present work. Similarly, thermochemical heat storage is in the early stages of development and not considered here.

Packed-bed stores are a suitable option for STRs in PTES systems due to their low cost, wide operating temperature ranges, large specific surface areas (which results in small temperature differences during heat transfer), non-toxicity, long life and other practical advantages. Six potential solid storage materials have been selected for the packed-bed STRs by consulting the relevant literature and are summarised in Table 2. The specific heat capacity of a solid storage material is usually dependent on its temperature, such that detailed correlations for this property are obtained from the literature.

For the sensible liquid storage media used in LTSs, the investigation focuses on the most commercially mature options for PTES systems, as summarised in Table 3. For the HT zone (above 500 K), three molten salts, specifically HITEC, HITEC XL and solar salt, are

considered due to their desirable characteristics at high temperatures, low cost and low environmental effect compared with synthetic oils [114]. For the IT zone (from ambient temperature to ~500 K), Therminol 66, Therminol VP-1, rapeseed oil and pressurised water (0.5 MPa) are potential candidates. Although thermal oils exhibit lower heat capacities than water, they remain in the liquid phase at higher temperatures (up to 250 °C) at atmospheric/low pressures [111], which is a remarkable advantage because this allows higher-temperature operation whilst avoiding the need (and costs associated with) for pressurisation and sealing. To increase the roundtrip efficiency of the PTES system, the working temperature of the cold energy storage material should be below ambient temperature. In this paper, five different organic liquids, namely butane, pentane, hexane, heptane and ethanol, are examined as potential storage materials in this component because they enable operating temperatures below 200 K. However, although these fluids remain in the liquid state at a low vapour pressure (near or below 0.1 MPa), they are also highly flammable and neurotoxic [52]. For all hot and cold storage materials, thermophysical properties could not be directly obtained from the NIST database through REFPROP 9.1 [115] (which is a limitation often found in the recent literature focusing on the choice of storage materials), and corrections are applied to the thermophysical properties dependent on

Table 2
Main characteristics of sensible heat solid storage materials used in packed beds [38,40,41,107–113].

Storage material	Density ρ , [kg/m ³]	Specific heat capacity c_p , [J/(kg·K)]	Thermal conductivity k , [W/(m·K)]	Cost, [\$/kg]	References
Magnetite (Fe ₃ O ₄)	5081	851	4.91	0.5	[40,107]
Quartzite (SiO ₂)	2500	830	3.16	0.04	[107,108]
Alumina (Al ₂ O ₃)	3990	1167	11.1	1.5	[40,41]
Titanium oxide (TiO ₂)	4230	692	8.40	1.7	[38,40,109]
Hematite (Fe ₂ O ₃)	5240	628	12.6	0.5	[40,110]
Basalt	2640	1231	1.50	0.12	[46,111,112]

Table 3Main characteristics of sensible heat liquid storage materials. The values of the thermophysical properties correspond to intermediate temperatures between $T_{\min}$ and $T_{\max}$.

Storage material	$T_{\min}$ [K]	$T_{\max}$ [K]	Density ρ , [kg/m ³]	Specific heat capacity c_p , [J/(kg·K)]	Thermal conductivity k , [W/(m·K)]	Viscosity μ , [mPa·s]	Cost [\$/kg]	References
HITEC	415	808	1637	1562	0.382	2.60	0.93	[114,116,117]
HITEC XL	403	823	1959	1432	0.519	4.19	1.43	[114,117]
Solar salt	533	873	1817	1517	0.525	1.57	0.5	[53,114]
Therminol 66	264	616	909	2072	0.109	1.20	1.0	[75,116]
Therminol VP-1	285	673	909	2066	0.113	0.371	3.96	[114,117,118]
Rapeseed oil	280	523	843	2383	0.195	7.59	0.8	[108]
Pressurised water	273	486	896	4355	0.679	0.155	0.001	[75]
Butane (C ₄ H ₁₀)	135	323	646	2139	0.136	0.328	1.0	[113,119]
Pentane (C ₅ H ₁₂)	144	366	663	2141	0.129	0.338	1.0	[54,113]
Hexane (C ₆ H ₁₄)	178	403	662	2240	0.123	0.323	1.0	[52,113]
Heptane (C ₇ H ₁₆)	183	437	700	2292	0.119	0.346	1.0	[113,120]
Ethanol (C ₂ H ₆ O)	159	398	802	2292	0.168	1.62	1.0	[113,119]

the temperature to ensure the accuracy of the calculations, as detailed in **Supplementary Material A**. The cost data for various liquid storage materials are obtained following Ref. [113].

The detailed working temperature ranges of all the said liquid storage materials are shown in Fig. 4. For reversible Joule-Brayton PTES systems with LTSS, the selection of HT liquid storage material has a substantial effect on roundtrip efficiency. The lowest working temperatures of these materials are higher than the ambient temperature, and a combination of HT and IT storage materials is a possible choice for hot energy storage due to the overlap of their working temperatures. In this paper, different combinations of hot storage materials are compared when optimising the roundtrip efficiency. For the case of reversible transcritical Rankine PTES systems with LTSS, only an IT storage material is used. Based on the selected materials, the operating thermodynamic parameters of the different PTES systems are identified.

2.3. Working fluid selection

Following a comprehensive review of the literature, seven working fluids highlighted as suitable for PTES cycles are identified: argon, nitrogen, helium, air, carbon dioxide, ammonia and trifluoriodomethane. The critical temperatures and pressures of these fluids obtained from NIST REFPROP 9.1 [115] are summarised in Table 4. The critical temperatures of argon, nitrogen, helium and air are much lower than the ambient temperature, such that these

Table 4

Critical temperatures and pressures of selected working fluids [115].

Working fluid	T_{cr} [K]	P_{cr} [MPa]
Argon (Ar)	151	4.86
Nitrogen (N ₂)	126	3.40
Helium (He)	5.2	0.23
Air	133	3.79
Carbon dioxide (CO ₂)	304	7.38
Ammonia (NH ₃)	406	11.4
Trifluoro iodomethane (R1311)	396	3.95

gases can readily be used in Joule-Brayton PTES systems. On the other hand, the critical temperatures of carbon dioxide, ammonia, and trifluoriodomethane are slightly higher than the ambient temperature, which makes these working fluids suitable for transcritical Rankine PTES systems. The operating thermodynamic parameters of PTES systems depend considerably on the selected working fluid, which consequently affects the thermodynamic and economic performance of these systems. Therefore, in this paper, parametric optimisation and performance comparisons of PTES systems employing these working fluids are performed.

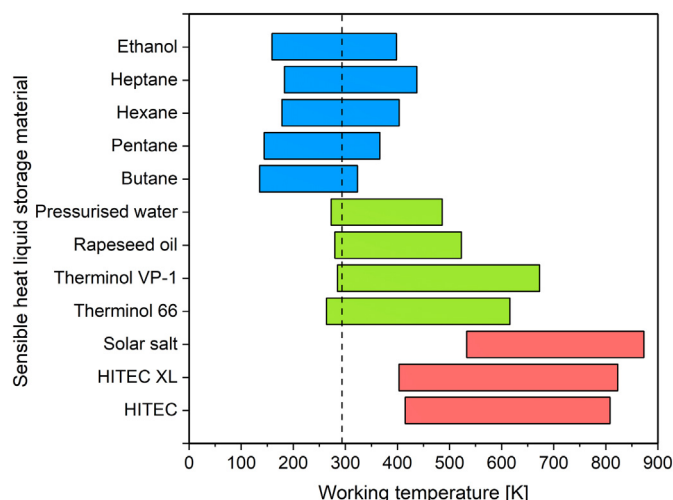
3. Modelling methodology

3.1. Main components

There are three main categories of components in PTES systems (see Section 2.2): (i) mechanical machineries (compressors, expanders/turbines, pumps), (ii) thermal storage units (packed-bed reservoirs and liquid tanks), and (iii) heat exchangers (hot/cold/auxiliary heat exchangers, gas heaters/coolers, evaporators/condensers and recuperators). Polytropic and isentropic efficiencies are used to characterise the performance of compression/expansion devices and pumps whilst the effectiveness-NTU (number of transfer units) method is used to predict heat transfer rates in all heat-exchange components. The fixed values of these thermodynamic performance indicators are used to predict the overall PTES technical performance (e.g., roundtrip efficiency). Their definitions are detailed in this section.

3.1.1. Compressors and expanders in Joule-Brayton cycles

For reversible Joule-Brayton PTES systems, although reciprocating-piston devices have a higher efficiency than turbomachinery and can reduce the total component and system costs, the maximum power for a single reciprocating machine is limited [22,28], and such devices are not available at present for large-scale PTES systems. Hence, the paper focuses on the use of turbomachines.

**Fig. 4.** Working temperature of different sensible heat liquid storage materials.

Consistent with many other studies [27,30,31,46], polytropic efficiency is used as the sole indicator to predict the thermodynamic performance of compression and expansions devices. The polytropic efficiency ($\eta_{p,c}$) of a compression process is defined as the ratio of the ideal work in a reversible isentropic process to the work in an adiabatic process for each infinitesimally small step:

$$\eta_{p,c} = \frac{v dp}{dh} \quad (1)$$

For an expansion process, the polytropic efficiency ($\eta_{p,e}$) is defined as the ratio of the actual work in an adiabatic process to the ideal work in an isentropic (and reversible) process for each infinitesimally small step:

$$\eta_{p,e} = \frac{dh}{v dp} \quad (2)$$

In the Joule-Brayton cycle, the specific heat ratio $\gamma = c_p/c_v$ is used for ideal gases, which is substituted into Eqs. (1) and (2), and after integration result in the following:

$$\frac{T_{c,out}}{T_{c,in}} = \left(\frac{P_{c,out}}{P_{c,in}} \right)^{\frac{\gamma-1}{\gamma} \cdot \frac{1}{\eta_{p,c}}} \quad (3)$$

$$\frac{T_{e,out}}{T_{e,in}} = \left(\frac{P_{e,out}}{P_{e,in}} \right)^{\frac{\gamma-1}{\gamma} \cdot \eta_{p,e}} \quad (4)$$

where $P_{c,in}$ and $P_{c,out}$ are the inflow and outflow pressures of the compressor, respectively; $T_{c,in}$ and $T_{c,out}$ are the inflow and outflow temperatures of the compressor, respectively; $P_{e,in}$ and $P_{e,out}$ are the inflow and outflow pressures of the expander, respectively; and $T_{e,in}$ and $T_{e,out}$ are the inflow and outflow temperatures of the expander, respectively.

Moreover, pressure ratio $\pi_c = P_{c,out}/P_{c,in}$ and temperature ratio $\psi_c = T_{c,out}/T_{c,in}$ across the compressor, and pressure ratio $\pi_e = P_{e,in}/P_{e,out}$ and temperature ratio $\psi_e = T_{e,in}/T_{e,out}$ across the expander, are defined as follows:

$$\psi_c = \pi_c^{\frac{\gamma-1}{\gamma} \cdot \frac{1}{\eta_{p,c}}} \quad (5)$$

$$\psi_e = \pi_e^{\frac{\gamma-1}{\gamma} \cdot \frac{1}{\eta_{p,e}}} \quad (6)$$

3.1.2. Machines in transcritical Rankine cycles

In transcritical Rankine cycles, the working fluid is near the two-phase region and cannot be considered an ideal gas. Isentropic efficiencies are used for all machines. Assuming no heat losses to the surroundings, isentropic efficiency can be defined for a compressor and a pump as follows:

$$\eta_{i,c} = \frac{h_{c,is,out} - h_{c,in}}{h_{c,out} - h_{c,in}} \quad (7)$$

where $h_{c,in}$, $h_{c,out}$ and $h_{c,is,out}$ are the inflow specific enthalpy, outflow specific enthalpy and outflow specific enthalpy, respectively, which would occur in an isentropic process of a compressor or pump.

For a turbine and an expander, isentropic efficiency can be defined as follows:

$$\eta_{t,e} = \frac{h_{e,in} - h_{e,out}}{h_{e,in} - h_{e,is,out}} \quad (8)$$

where $h_{e,in}$, $h_{e,out}$ and $h_{e,is,out}$ are the inflow specific enthalpy, outflow specific enthalpy and outflow specific enthalpy, respectively.

3.1.3. Packed-bed thermoclines and liquid tanks

For a packed-bed thermocline store, the volume of the reservoir V_{pack} is calculated as follows:

$$V_{pack} = \frac{\dot{m} c_{p,g} t_N}{\rho_s (1 - \varepsilon_{VF}) \bar{c}_s} \quad (9)$$

where $\dot{m}$ is the mass flow rate of the working fluid; $c_{p,g}$ the specific heat of the working fluid; t_N the nominal charge time; ρ_s the density of the storage material; ε_{VF} the void fraction of the packed bed; and $\bar{c}_s$ the average specific heat capacity of the storage material.

Stored energy in the reservoir E_{pack} is given by the following:

$$E_{pack} = \rho_s (1 - \varepsilon_{VF}) \bar{c}_s V_{pack} (T_1 - T_2) \quad (10)$$

where T_1 and T_2 are the charge and discharge temperatures, respectively.

For a liquid tank store, tank volume V_{tank} and stored energy E_{tank} are calculated by the following:

$$V_{tank} = \frac{\dot{m}_s t_N}{\rho_s} \quad (11)$$

$$E_{tank} = \rho_s \bar{c}_s V_{tank} (T_1 - T_2) \quad (12)$$

where $\dot{m}_s$ is the mass flow rate of the storage liquid.

3.1.4. Heat exchangers

In PTES systems, the heat exchangers play an important role and need to be carefully designed. Firstly, the ε -NTU method is used for the choice of thermodynamic parameters. Heat transfer effectiveness ε [121] is defined as follows:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \quad (13)$$

where $\dot{Q}$ and $\dot{Q}_{max}$ are the actual and maximum possible heat transfer rates, respectively.

Maximum possible heat transfer is defined as the maximum value of heat transfer that could be attained if one of the fluids undergoes a temperature change equal to the maximum temperature difference present in the exchanger:

$$\dot{Q}_{max} = (\dot{m}c)_{min} (T_{hs,in} - T_{cs,in}) \quad (14)$$

where $\dot{m}$ and c are the mass flow rate and the specific heat of the working fluid, respectively. The fluid with the minimum values of $\dot{m}c$ may be either the hot or the cold fluid, and $T_{hs,in}$ and $T_{cs,in}$ are the inlet temperatures of the hot-side and cold-side fluids, respectively.

NTU stands for number of transfer units and is representative of the heat exchanger size, defined as follows:

$$NTU = \frac{UA}{C_{min}} = \frac{UA}{(\dot{m}c)_{min}} \quad (15)$$

where U is the overall heat transfer coefficient, A is the heat transfer area, and C_{min} is the minimal heat capacity between hot and cold

fluid. In addition, heat capacity ratio is defined as follows:

$$C_R = \frac{C_{\min}}{C_{\max}} \quad (16)$$

The most commonly employed heat exchanger type is the shell-and-tube heat exchanger, which is selected here as the heat transfer device in all the considered PTES systems. For a counter-flow heat exchanger, the heat transfer effectiveness is associated with C_R and NTU as follows:

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C_R)]}{1 - C_R \exp[-NTU(1 - C_R)]} \quad (17)$$

TEMA standards are followed for the design of the shell-and-tube heat exchangers [122]. The detailed design procedures for structure parameters and several pressure losses are shown in the study [123]. In this paper, the main corrections of heat transfer coefficients of tube-side (α_{ts}) and shell-side (α_{ss}) for distinct heat exchangers (single and two-phase regions) are presented in **Supplementary Material B**. The overall heat transfer coefficient of a heat exchanger is calculated as follows:

$$U = \frac{1}{\frac{1}{\alpha_{ts}} \frac{d_o}{d_i} + r_{ft} \frac{d_o}{d_i} + \frac{\delta_w}{\lambda_w} \frac{d_o}{d_m} + r_{fs} + \frac{1}{\alpha_{ss}}} \quad (18)$$

where λ_w and δ_w denote the wall heat conductivity and the thickness, respectively; d_o , d_i and d_m are the outside, inside and mean tube diameters, respectively; and r_{ft} and r_{fs} are the fouling resistances on the tube and shell sides, respectively.

3.2. Conditions, parameters and assumptions

To establish a specific basis for the comparison of the thermodynamic and economic performance of the different PTES systems, the (discharge) power capacities (50 MW) and discharge durations (6 h) of all plants are fixed. These specific values are selected because they are intermediate values between values found in literature and are points of overlap for all of the described systems [20,22,44,46,64,67,93,96,124,125]. Given that the discharge capacity and storage duration of these plants will affect the relative comparison between the performance of these systems, these restrictions are relaxed towards the end of this paper (e.g., see Fig. 19); for further insight into the role of these parameters, the interested reader is referred to Olympios et al. [23].

The design optimisation of the thermodynamic parameters of the PTES systems is performed by selecting the roundtrip efficiency as an objective function, using optimisation tools in MATLAB whilst considering the limitations of the working temperatures of the storage materials and manufacturing capabilities of the main components in practical applications [126]. For the optimisation procedure, the initial input parameters include: (i) for the reversible Joule-Brayton PTES system: compressor and expander polytropic efficiencies of 90% according to suggested design parameters of a PTES system reported in the literature [27,46,54]; (ii) for the transcritical Rankine PTES system: isentropic efficiencies are again taken from the literature, with a value of 85% taken for the compressor [41,70], 85% for the expander [63,69], 88% for the turbine [41,89], and 80% for the pump [75,89]; (iii) for packed beds: a void fraction of 0.4 and an equivalent spherical particle diameter of 0.02 m are considered, which agree with the study of McTigue et al. [31]; (iv) for the heat exchangers: effectiveness is set to 0.95, which is consistent with Ref. [76] and a heat capacity ratio of unity; and (v) an ambient temperature of 25 °C (298 K) and inlet pressure to the compressor of 1.05 bar.

A sequential quadratic programming algorithm, which is a

highly reliable method for solving problems such as the one under investigation here [93], is adopted in this work. Furthermore, in order to prevent the solver from being trapped in local minimum solutions (which may happen as the design problem is highly non-linear), a multi-start strategy is adopted, starting from different initial points that are randomly selected. Furthermore, the following assumptions are made for the parametric optimisation of the PTES systems:

- (1) Heat leakage in the turbomachines is negligible due to their high rotational speed;
- (2) Pressure drops in all nominally isobaric components are neglected;
- (3) Heat losses to the surroundings in all components are ignored;
- (4) Working fluids used in the Joule-Brayton cycles are assumed to be ideal gases;
- (5) Thermodynamic calculations of all PTES systems are performed at the design point for the charging and discharging processes based on a full-charge-to-full-discharge cycle, although actual performance may depend on the variable operating conditions the energy storage will experience.

3.3. Performance indicators

Achieving high efficiencies along with low costs is vital for any electricity storage technology to compete with more established, mature technologies such as PHS and CAES. These indicators can be optimised by adjusting a variety of design and operational parameters, which in several cases is not achievable in practice due to trade-offs or conflicts. Here, the roundtrip efficiencies and capital costs of different PTES variants are investigated and compared. Firstly, these performance indicators are defined.

3.3.1. Roundtrip efficiency

Roundtrip efficiency χ is defined as the net power output during discharge divided by the net power input during charge:

$$\chi = \frac{W_{\text{dis}}^{\text{net}}}{W_{\text{chg}}^{\text{net}}} = \frac{\dot{m}_{\text{dis}} \cdot (w_{e,\text{dis}} - w_{c,\text{dis}}) \cdot t_{\text{dis}}}{\dot{m}_{\text{chg}} \cdot (w_{c,\text{chg}} - w_{e,\text{chg}}) \cdot t_{\text{chg}}} \times 100, [\%] \quad (19)$$

where $\dot{m}_{\text{chg}}$ and $\dot{m}_{\text{dis}}$ are the mass flow rates during the charge and discharge processes, respectively; $w_{c,\text{chg}}$ and $w_{e,\text{chg}}$ are the specific work of the compressor and the expander used during charge, respectively; $w_{e,\text{dis}}$ and $w_{c,\text{dis}}$ are the specific work of the expander and the compressor used during discharge, respectively; and t_{chg} and t_{dis} are the charge and discharge duration times, respectively.

To compare the work of the compression and expansion processes of different PTES systems at identical roundtrip efficiencies, a work ratio of the charge mode (ω_{chg}) is defined as the ratio of specific compression work to specific expansion work during charge, and a work ratio of the discharge mode (ω_{dis}) is defined as the ratio of specific expansion work to specific compression work during discharge:

$$\omega_{\text{chg}} = \frac{w_{c,\text{chg}}}{w_{e,\text{chg}}} \quad (20)$$

$$\omega_{\text{dis}} = \frac{w_{e,\text{dis}}}{w_{c,\text{dis}}} \quad (21)$$

3.3.2. Capital cost

The costing of emerging technologies is challenging and can involve substantial uncertainties. Which costing method available in the literature provides the most accurate estimation of the related equipment costs is currently not known. Therefore, preliminary cost estimation of PTES systems by using a variety of costing methods is obtained in this analysis. This approach involves obtaining a mean cost estimation from different methods, along with associated uncertainty ranges.

The total capital cost (TCC) includes the costs associated with the construction of a new plant or the modifications to an existing chemical manufacturing plant and is used to evaluate the systems' economic performance. Moreover, the power capital cost (PCC), which is the TCC per unit of rated power output, \$/kW, and the energy capital cost (ECC), which is the TCC per unit of energy storage capacity, \$/kWh, are also both evaluated for different plant designs, conditions and parameters.

The module costing technique is used to calculate the bare module cost of each component. The cost is then modified by the chemical engineering plant cost index to obtain the TCC of the system [127]:

$$C_T = \sum C_{BM} \cdot \left(\frac{I_a}{I_b} \right), [\$] \quad (22)$$

where I_a and I_b are dimensionless numbers employed to adjust the capital cost required to build a system based on inflation from a past date 'b' to a later time 'a'. The index was equal to 607.5 in 1919, and values for other years are obtained from Chemical Engineering Online [128] and reported in **Supplementary Material C**.

In this paper, four methods are used to calculate and compare the bare module cost of each component, obtained from Turton et al. [127], Seider et al. [129], Couper et al. [130] and Morandin et al. [67]. Included are costs of compression and expansion machines, pumps, storage tanks, pressurised vessels, heat exchangers and electrical generators, which are assumed to be the dominant component costs. The resulting 'total' costs are not necessarily intended to be accurate estimations of true final costs of these plants because other costs (land, legal, etc.) are not considered, and these additional costs are highly variable, regionally specific and can change appreciably over time, such that including them leads to larger uncertainties and a greater difficulty to perform like-for-like comparisons of the various options under examination. A detailed presentation of these methods is given in **Supplementary Material C**. In the corresponding results (Section 4), the variability of the cost methods for each PTES system reported here is indicated by cost ranges ('error bars') in all plots, which correspond to either the mean or the absolute deviation, as indicated.

4. Thermodynamic optimisation of PTES systems

In PTES systems, overall technical performance is closely associated with a number of thermodynamic parameters. These parameters are mainly determined by the properties of the working fluid, storage materials and conditions imposed by the processes taking place in the main system components (compressors, expanders, pumps, heat exchangers, and thermal reservoirs). Therefore, in this section, the results from design optimisation based on maximising system roundtrip efficiency as the thermodynamic performance indicator of interest and performed based on plants with a selected power capacity (50 MW) and discharge duration (6 h) for the thermodynamically optimal PTES systems are presented, followed by evaluating the economic performance of these systems in Section 5.

4.1. Reversible Joule-Brayton PTES with STRs

For reversible Joule-Brayton PTES systems with STRs, thermodynamic optimisation is performed by evaluating the performance of different SCs, working fluids and storage materials. The initial values, as well as the constraints (lower and upper bounds) of the optimised parameters, which include the pressure ratios of the charge and discharge processes, and working fluid temperatures at the HR and CR outlets during charge, are given in **Supplementary Material D**. The working fluid temperatures at the HR and CR outlets during charge only need to be optimised when a recuperator is considered in the system. The constraints used in this work are as follows: (a) The maximum working temperature should be below 850 K because this is the maximum operating temperature of the steel that is envisioned for the construction of the components. (b) The minimum temperature of the working fluid should be above 123 K, according to the grid-scale demonstration discussed in Ref. [36]. (c) The inlet/outlet temperatures of the HRs and CRs during charge are equal to those during discharge.

In the PTES systems shown in Fig. 1, the working fluid temperatures at the inlets of the compressor and expander during charge are set to 298 K (ambient temperature), as ensured by the auxiliary heat exchangers. When no recuperator is considered, the working fluid temperatures at the outlets of the HRs and CRs during charge are also set to 298 K. Moreover, auxiliary heat exchangers HX1 and HX2 are used to reject heat and keep the working fluid temperatures in the HRs and CRs stable during charge and discharge. Although the heat exchangers are simply auxiliary devices, the number of heat exchangers and the selected layout remarkably affect the thermo-economic performance of the system. Therefore, a detailed comparison of different PTES SCs with magnetite used as the packed-bed material and argon as the working fluid accounting for different heat exchanger layouts is performed. Corresponding T - s diagrams of the different SCs considered in this section are shown in Fig. 5, and the key thermodynamic performance indicators are shown in Fig. 6.

The results in Fig. 6 show that for SC1, SC2, SC1r and SC2r, the pressure ratios during charge are larger than those during discharge, whereas the trends are reversed for SC3 and SC3r. In addition, the work ratios during charge (2.94–3.22) are all larger than those during discharge (1.53–2.43). More importantly, the highest roundtrip efficiency (66.4%) is attained by SC1, which involves utilising HX1 and HX2, but not a regenerator. If only HX1 is used (SC2), the roundtrip efficiency is slightly lower than for SC1 but only marginally (66.2% for SC2 versus 66.4% for SC1), which is within the uncertainty of the present modelling/prediction methods. In SC3, which only involves HX2, and although the pressure ratio during discharge is the maximum possible within the operating limits, the maximum temperature of the working fluid is still lower than 850 K, and the roundtrip efficiency is lower than those of SC1 and SC2. This result suggests that, in the non-recuperated Joule-Brayton PTES system with STRs, the inclusion of HX1 is a key feature of the system if a high roundtrip efficiency is required, and HX2 does not play a substantial role. If a recuperator is added to SC1 (i.e. SC1r), the effect of this additional component on roundtrip efficiency is negligible (66.4% for SC1 versus 66.3% for SC1r), and the same results apply to SC2, for which the effect of recuperation on roundtrip efficiency can also be neglected (66.2% for SC2 versus 66.3% for SC2r). However, if a recuperator is added to SC3 (i.e. SC3r), then the working fluid temperature reaches its maximum possible value, and the roundtrip efficiency increases remarkably from 52.3% to 63.6%, indicating that this component is important in improving the roundtrip efficiency of this SC.

Based on the SC1 variant of PTES systems with STRs, the influence of the working fluid on the key thermodynamic performance

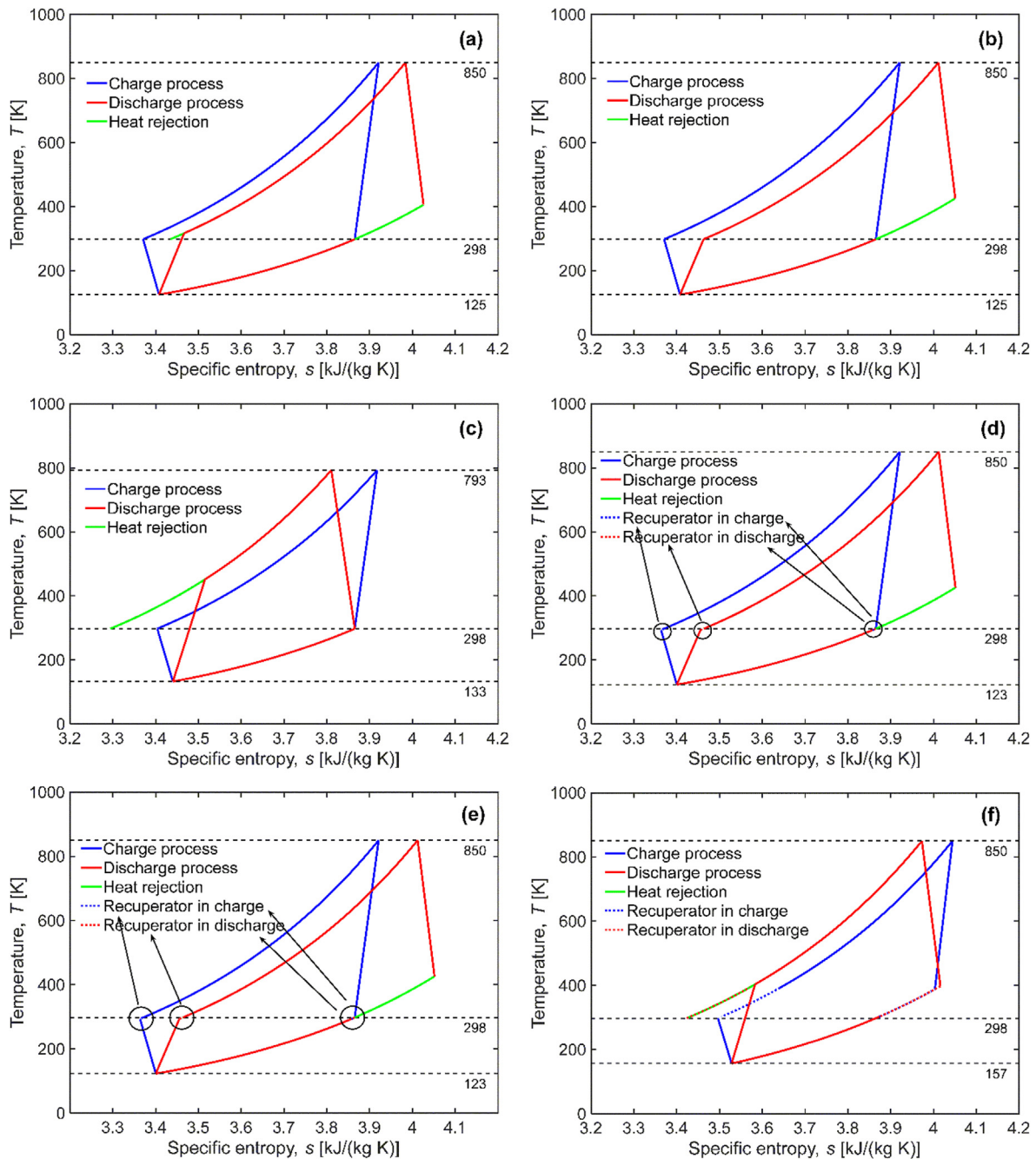


Fig. 5. T - s diagrams of reversible Joule-Brayton PTES systems with argon as the working fluid and magnetite as the packed-bed material: (a) SC1 – both HX1 and HX2 are used, but no recuperator; (b) SC2 – only HX1 is used; (c) SC3 – only HX2 is used; (d) SC1r – HX1, HX2 and a recuperator are used; (e) SC2r – HX1 and a recuperator are used; and (f) SC3r – HX2 and a recuperator are used (see also Fig. 1).

indicators is shown in Fig. 7, where systems with helium as the working fluid achieve the highest roundtrip efficiency (66.5%), and those with argon have a similar roundtrip efficiency (66.4%) that is within the uncertainty of the present predictions. However, when nitrogen or air is the working fluids, the pressure ratio during discharge reaches its maximum possible value within the operating limits, but the roundtrip efficiency (57%–58%) is lower than when using helium or argon.

The roundtrip efficiency and total reservoir volume of PTES systems when using different packed-bed materials and helium as the working fluid are summarised in Table 5. The results indicate that the choice of solid storage material does not affect roundtrip

efficiency but has a substantial influence on total reservoir volume. This finding is not important from a thermodynamic perspective, but can have space and cost implications.

In summary, the above results suggest that, if only the thermodynamic performance of these systems (i.e. here, their roundtrip efficiency) is considered, the combination of PTES SC1 or SC2, and helium or argon as the working fluid are the best choices for Joule-Brayton PTES systems with STRs.

4.2. Reversible Joule-Brayton PTES with LTSs

Design optimisation of the thermodynamic performance of

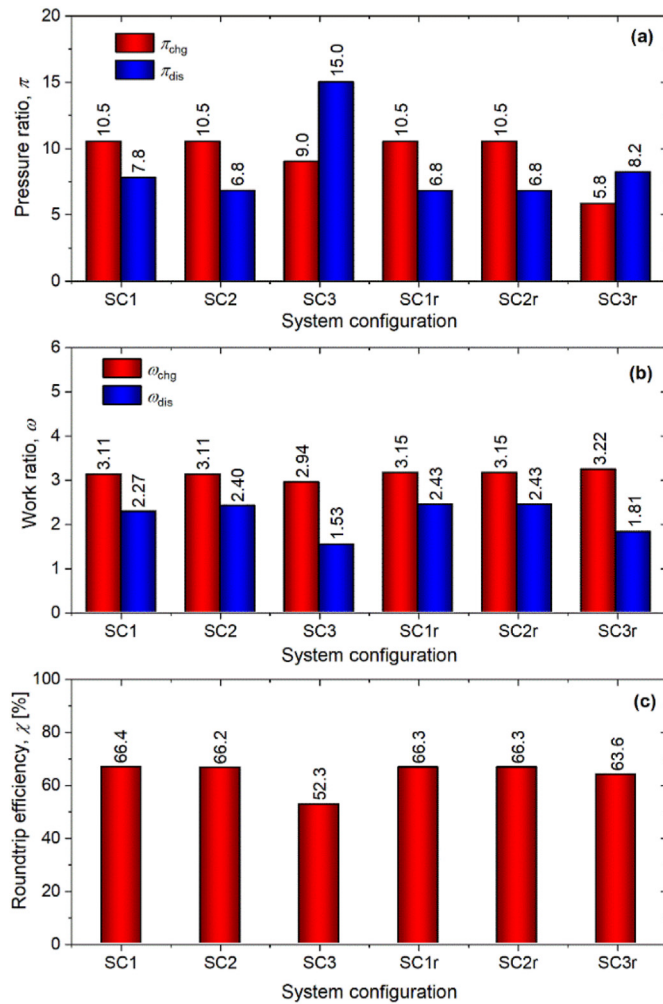


Fig. 6. Thermodynamic performance comparison of the six 50 MW Joule-Brayton PTES system configurations with 6 h discharge duration, when using argon as the working fluid and magnetite as the packed-bed material: (a) pressure ratio; (b) work ratio; and (c) roundtrip efficiency, where SC1 – both HX1 and HX2 are used, but no recuperator; SC2 – only HX1 is used; SC3 – only HX2 is used; SC1r – HX1, HX2 and a recuperator are used; SC2r – HX1 and a recuperator are used; and SC3r – HX2 and a recuperator are used (see also Figs. 1 and 5).

reversible Joule-Brayton PTES systems with LTSs is also performed, by considering different SCs, working fluids and storage materials. As described in Section 2.1, the combination of HT and IT liquid storage materials would be a feasible method for hot energy storage, but the working fluid temperatures at the inlets of IT and HT liquid tanks need to be optimised. The initial values, lower and upper bounds of 10 optimised parameters are summarised in **Supplementary Material D**. The main constraints are as follows: (a) The maximum working temperature should be lower than 850 K because this is the maximum operating temperature of the steel that is used in the construction of the components. (b) The operating temperatures of the hot and cold storage materials must be limited within the feasible working temperature ranges, as reported in Table 3. (c) The inlet/outlet temperatures of the HT hot, IT hot and cold liquid tanks during charge are equal to those during discharge.

In the PTES system shown in Fig. 2, when no recuperator is considered, the working fluid temperature at the compressor inlet during charge is fixed at 298 K (ambient temperature). Moreover, auxiliary heat exchangers AHX1 and AHX2 are used to reject heat

and keep the working fluid temperatures at the inlet/outlet of hot and cold liquid stores stable during charge and discharge. As with the case of the Joule-Brayton PTES system with STRs, a detailed comparison of different PTES SCs using solar salt, Therminol 66 and pentane for storage materials and argon used as the working fluid, accounting for different heat exchanger layouts, is performed. The associated T - s diagrams and the thermodynamic performance of system variants are shown in Fig. 8 and Fig. 9, respectively.

The work ratios during charge and discharge are all lower than those in Joule-Brayton PTES with STRs for all SCs. Moreover, the highest roundtrip efficiency (56.6%) is attained by SC3r, which features a recuperator and AHX2. Amongst the non-recuperated systems, the roundtrip efficiency is highest for SC1 and SC2, i.e. when AHX1 and AHX2, or only AHX1 is used, respectively; the latter being marginally lower. For SC3 (only AHX2 used), the roundtrip efficiency is lower than SC1 and SC2 because the maximum temperature is still lower than 850 K. This finding implies that in the non-recuperated Joule-Brayton PTES system with LTSs, similar to the case of STRs, the inclusion of AHX1 is important for a high roundtrip efficiency. If a recuperator is added to SC1 (i.e. SC1r), its effect on the system's roundtrip efficiency is negligible (54.6% for SC1 versus 54.5% for SC1r), and the same conclusions are drawn for SC2, for which the effect of recuperation on roundtrip efficiency can also be neglected (53.7% for SC2 versus 54.4% for SC2r). However, if a recuperator is added to SC3 (i.e. SC3r), roundtrip efficiency increases considerably, from 38.3% to 56.6%, which as with the case of STRs, indicates that the inclusion of this component can improve the system's overall roundtrip efficiency. Given the uncertainty of these predictions, SC1, SC2, SC1r, SC2r and SC3r have similar efficiencies.

In the Joule-Brayton PTES system with LTSs, three heat exchangers transfer heat/cold between the working fluid and the liquid storage materials, including an HT hot heat exchanger, an IT hot heat exchanger and a cold heat exchanger. In this investigation, different combinations of storage materials are used. The pinch point temperature difference (PPTD), which is the minimum temperature difference between heat exchanger fluids, of each heat exchanger should be checked carefully because their thermodynamic properties are all dependent on the working temperature. As an example, the T - Q diagram of hot and cold heat exchangers during charge and discharge is shown for the optimised Joule-Brayton PTES system (based on SC1 and using argon as the working fluid and solar salt, Therminol 66 and pentane as the HT hot, IT hot and cold storage materials, respectively) in Fig. 10.

This figure shows that for the HT hot heat exchanger, the working temperature range of the HT storage material (solar salt) is 584–837 K during charge and discharge. The PPTDs of the HT hot heat exchanger, which occur at the inlets of the HT storage material during charge and discharge, are all equal to 13 K. For the IT hot heat exchanger, the working temperature range of the IT storage material (Therminol 66) is 331–584 K during charge and discharge. The PPTDs of the IT hot heat exchanger, which occur at the inlet of the IT storage material during charge and discharge, are all equal to 13 K. For the cold heat exchanger, the working temperature range of the cold storage material (pentane) is 153–306 K during charge and discharge. The PPTDs of the cold heat exchanger, which occur at the inlet of the cold storage material during charge and discharge, are all equal to 8 K.

Based on PTES SC1 using solar salt, Therminol 66 and pentane as the storage materials, the influence of selecting different working fluids on the systems' thermodynamic performance is shown in Fig. 11. The results show that for all the working fluids, the work ratios during charge are higher than those during discharge, and the work ratios of Joule-Brayton PTES systems with LTSs are lower than those of Joule-Brayton PTES systems with STRs (shown in

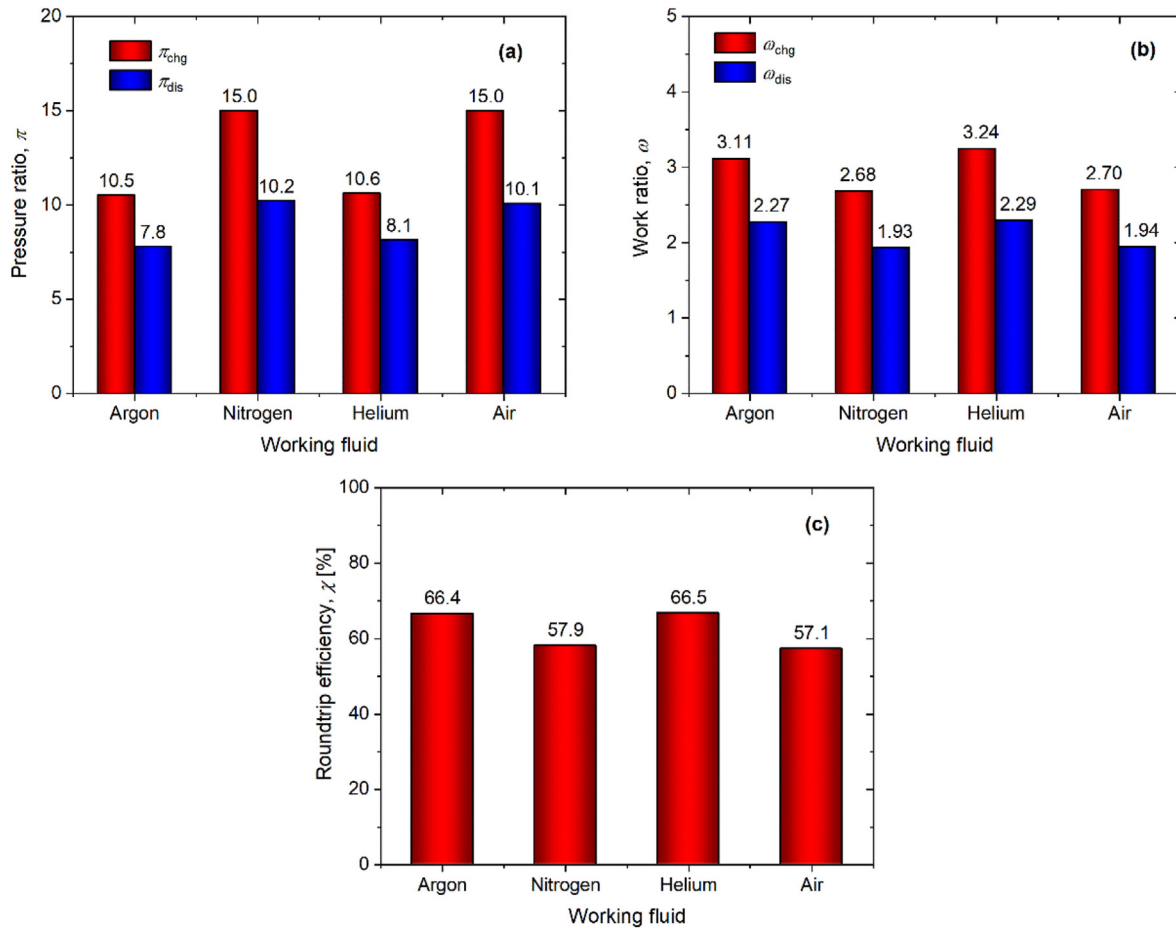


Fig. 7. Thermodynamic performance comparison of 50 MW Joule-Brayton PTES systems based on SC1 with 6 h discharge duration (both HX1 and HX2 used), when using different working fluids and magnetite as the packed-bed material: (a) pressure ratio; (b) work ratio; and (c) roundtrip efficiency (see also Figs. 1 and 5).

Table 5

Roundtrip efficiency and total reservoir volume of Joule-Brayton PTES systems based on SC1, when using helium as the working fluid and different packed-bed materials. Best value is shown in **bold**.

Total reservoir volume [m ³] (Roundtrip efficiency, χ [%])	SC1	SC2	SC3	SC1r	SC2r	SC3r
Magnetite	3537 (66.5)	3565 (66.0)	5163 (53.3)	3492 (66.5)	3492 (66.5)	4258 (64.2)
Alumina	3223 (66.5)	3248 (66.0)	4670 (53.3)	3185 (66.5)	3185 (66.5)	3919 (64.2)
Quartzite	6836 (66.5)	6890 (66.0)	9894 (53.3)	6732 (66.5)	6732 (66.5)	8396 (64.2)
Titanium oxide	4846 (66.5)	4884 (66.0)	7014 (53.3)	4772 (66.5)	4772 (66.5)	5952 (64.2)
Hematite	4310 (66.5)	4344 (66.0)	6239 (53.3)	4245 (66.5)	4245 (66.5)	5294 (64.2)
Basalt	6030 (66.5)	6078 (66.0)	8789 (53.3)	6038 (66.5)	6038 (66.5)	7032 (64.2)

Fig. 7). Moreover, the highest roundtrip efficiencies are 54.6% and 54.4% with argon and helium as the working fluid, respectively. When using nitrogen or air as the working fluid, although the pressure ratio reaches the maximum possible value, the roundtrip efficiency is still lower than the systems that use argon or helium.

A comprehensive comparison of the roundtrip efficiency of Joule-Brayton PTES systems with different combinations of storage materials and argon used as the working fluid is shown in Table 6. When using solar salt, Therminol 66 and butane as the storage materials, the PTES system achieves the highest roundtrip efficiency (55.3%), but other combinations of hot and cold stores have similar thermodynamic performance (efficiency), e.g. other Therminol or HITEC variants in the hot store and pentane in the cold store. Given that the minimum working temperatures of heptane and hexane are higher than those of the other cold storage

materials (pentane, ethanol and butane), the roundtrip efficiency associated with these two cold storage materials is remarkably lower than those associated with the other options.

In summary, the best roundtrip efficiencies of Joule-Brayton PTES systems with LTSs are associated with SC1, SC2, SC1r, SC2r and SC3r and argon or helium as the working fluid, with efficiencies in the range 54%–57%. A combination of solar salt + Therminol or HITEC + Therminol as the hot storage materials and butane or pentane as the cold storage material allows the highest roundtrip efficiencies of within $\pm 1\%$.

4.3. Reversible transcritical Rankine PTES with LTSs

Similar to the two other PTES system variants, the optimisation of the thermodynamic performance of reversible transcritical

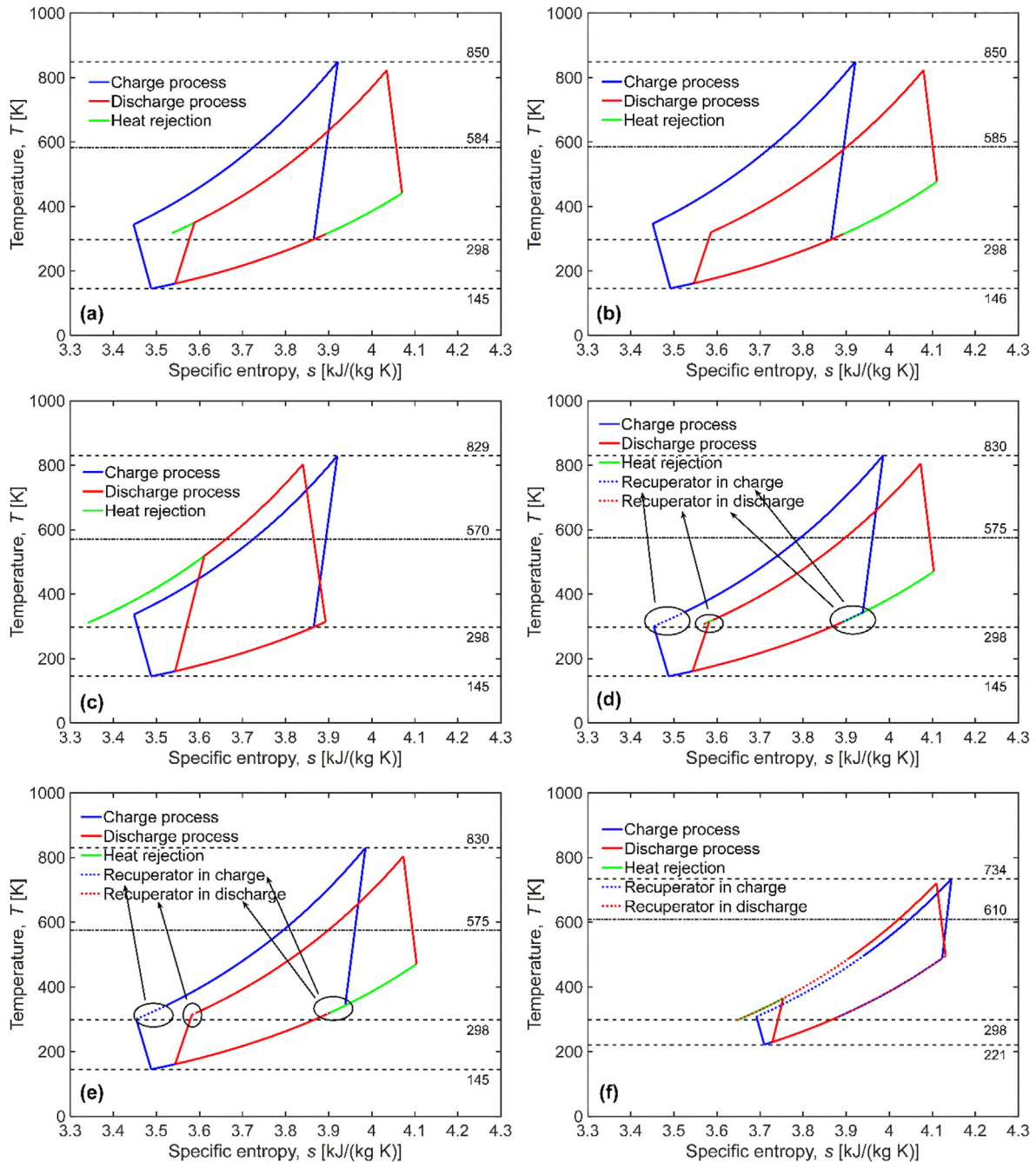


Fig. 8. T-s diagrams of reversible Joule-Brayton PTES systems with argon as the working fluid and solar salt, Therminol 66 and pentane as the storage materials: (a) SC1 – both AHX1 and AHX2 are used, but no recuperator; (b) SC2 – only AHX1 is used; (c) SC3 – only AHX2 is used; (d) SC1r – AHX1, AHX2 and a recuperator are used; (e) SC2r – AHX1 and a recuperator are used; and (f) SC3r – AHX2 and a recuperator are used (see also Fig. 2).

Rankine PTES systems with LTSs is performed by evaluating different SCs (with and without recuperation), working fluids and storage materials. The main constraints are as follows: (a) The operating temperatures of the hot and cold storage materials must be within their feasible working temperature ranges. (b) The inlet/outlet temperatures of the hot and cold liquid tanks during charge are equal to those during discharge. The initial values, lower and upper bounds of the optimised parameters for three working fluids are summarised in **Supplementary Material D**.

Cycle (T-s) diagrams of transcritical Rankine PTES systems using CO₂ as the working fluid and Therminol 66 for the storage material are shown in Fig. 12. For the PTES system without a recuperator,

although the working fluid pressure at the compressor outlet during charge reaches its upper limit, the working fluid temperature at the compressor outlet during charge is much lower than the maximum working temperature of Therminol 66 due to the heat transfer/thermal match between CO₂ and Therminol 66 in the gas cooler (during charge) and the gas heater (during discharge), whereas for the PTES system with a recuperator, the working fluid temperature at the compressor outlet during charge and the temperature of Therminol 66 reach their upper working temperature limits.

The thermodynamic performance of transcritical Rankine PTES systems using different storage materials is summarised in Table 7.

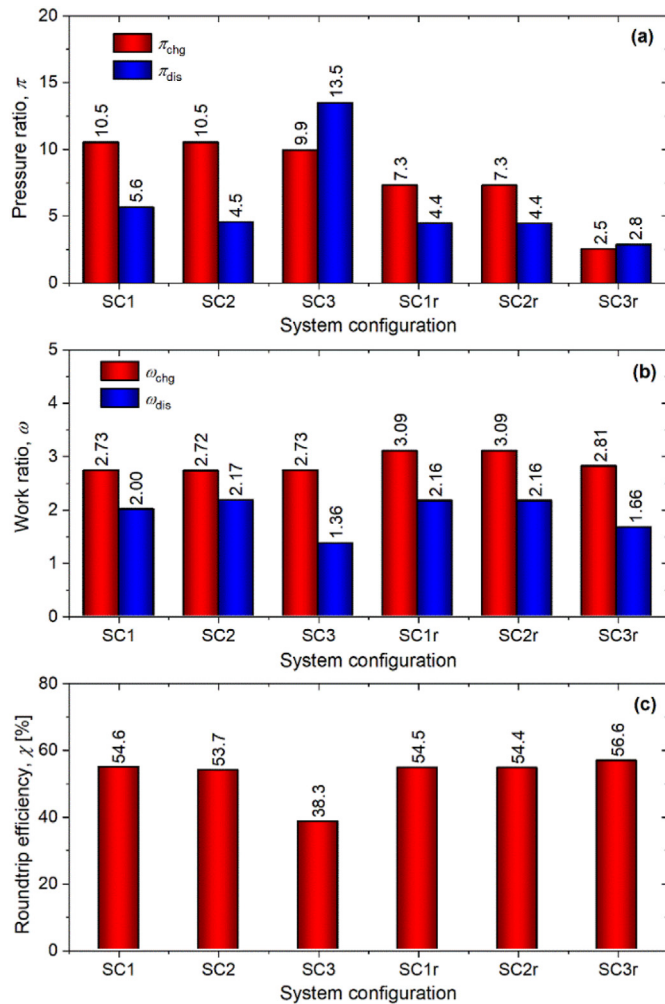


Fig. 9. Thermodynamic parameter comparison of the six 50 MW Joule-Brayton PTES system configurations with 6 h discharge duration, when using argon as the working fluid and solar salt, Therminol 66 and pentane as the storage materials: (a) pressure ratio; (b) work ratio; and (c) roundtrip efficiency, where SC1 – both AHX1 and AHX2 are used, but no recuperator; SC2 – only AHX1 is used; SC3 – only AHX2 is used; SC1r – AHX1, AHX2 and a recuperator are used; SC2r – AHX1 and a recuperator are used; and SC3r – AHX2 and a recuperator are used (see also Figs. 2 and 8).

For all four storage materials considered, without a recuperator, the pressure at the outlet of the compressor during charge is higher than that of the pump during discharge, whereas for systems with a recuperator, the pressure at the outlet of the compressor during charge is lower than that of the pump during discharge. Compared with Joule-Brayton PTES systems with STRs and LTSs, transcritical Rankine PTES systems with LTSs experience larger work ratios during charge and discharge, especially when using a recuperator, which has a positive effect on roundtrip efficiency. Moreover, adding a recuperator increases the roundtrip efficiency of the transcritical systems for all storage materials. Amongst the considered storage materials, the system achieves the highest roundtrip efficiency (68.0%) when using Therminol VP-1 and when a recuperator is included. This value, and the system that it corresponds to, is considered promising and worthy of further examination as it is marginally higher than the efficiencies of the best subcritical PTES systems presented earlier in Sections 4.1 and 4.2.

Ammonia (NH_3) is a potential choice as the working fluid of transcritical Rankine PTES systems. However, its critical temperature and pressure are 406 K and 11.4 MPa, respectively, such that pressurised water and rapeseed oil are excluded as storage materials in this case, due to limited working temperatures. Therefore, only Therminol 66 and Therminol VP-1 are selected for the design of these PTES systems.

Cycle (T - s) diagrams when using Therminol 66 as the storage material and NH_3 as the working fluid are shown in Fig. 13. Here, the temperature of NH_3 at the outlet of the compressor, even without a recuperator, is near its imposed operating limit, such that the effect of adding a recuperator is expected to be limited. Table 8 shows that when using NH_3 as the working fluid, the work ratios during charge and discharge are larger than those with CO_2 as the working fluid. Moreover, the system using Therminol VP-1 as the storage material and including a recuperator achieves the highest roundtrip efficiency (53.9%), but this is lower than the subcritical PTES system efficiencies presented earlier in Sections 4.1 and 4.2.

Koen et al. [76] suggested that trifluoriodomethane (R1311) is a promising working fluid for transcritical Rankine PTES systems with LTSs. To select the best combination of working fluid and storage materials, R1311 is also included in this investigation and corresponds to PTES systems designed to use three different storage materials: Therminol 66, Therminol VP-1 and rapeseed oil. The cycle (T - s) diagrams in Fig. 14 show that when using Therminol 66 as the storage material, the pressure at the outlet of the compressor during charge is slightly higher than that of the pump during discharge when the system does not include a recuperator, whereas with a recuperator, the pressure at the outlet of the compressor during charge is slightly lower than that of the pump during discharge. Moreover, the thermodynamic performance of these PTES systems with different storage materials is shown in Table 9. When using Therminol VP-1 as the storage material and including a recuperator, the system achieves the highest roundtrip efficiency (63.5%), which is similar (only marginally lower) to the efficiencies of the PTES systems with STRs presented earlier in Section 4.1 and higher than the efficiencies of the PTES systems with LTSs presented in Section 4.2.

Based on the most promising hot-store material (i.e. Therminol VP-1), the thermodynamic performance comparison of the transcritical Rankine PTES systems with different working fluids performed in this work is shown in Fig. 15. The results show that the PTES system which uses CO_2 as the working fluid and includes a recuperator achieves the highest roundtrip efficiency (68.0%). In general, adding a recuperator has a remarkable positive effect on the thermodynamic performance (here, roundtrip efficiency) of these transcritical Rankine PTES systems, especially for systems that use CO_2 as the working fluid.

Finally, the roundtrip efficiencies of the presently considered PTES systems are compared with other TMES systems and large-scale storage flow batteries and Li-ion batteries in Fig. 16 (see Fig. 15). The results show that most TMES technologies, specifically, CAES, LAES and PTES here, have overall efficiencies in the range from 45% to 70% [23]. Substantial efficiency improvement is promised by advanced CAES systems (60% for diabatic-CAES versus 70% for adiabatic-CAES), whereas standalone LAES systems show (relatively) lower roundtrip efficiencies (approximately 50%). Although compressed heat energy storage (CHEST) systems often operate at lower maximum temperatures than Joule-Brayton PTES systems, they show similar roundtrip efficiencies (about 65%). Although PHS, VRB flow batteries and Li-ion batteries show slightly

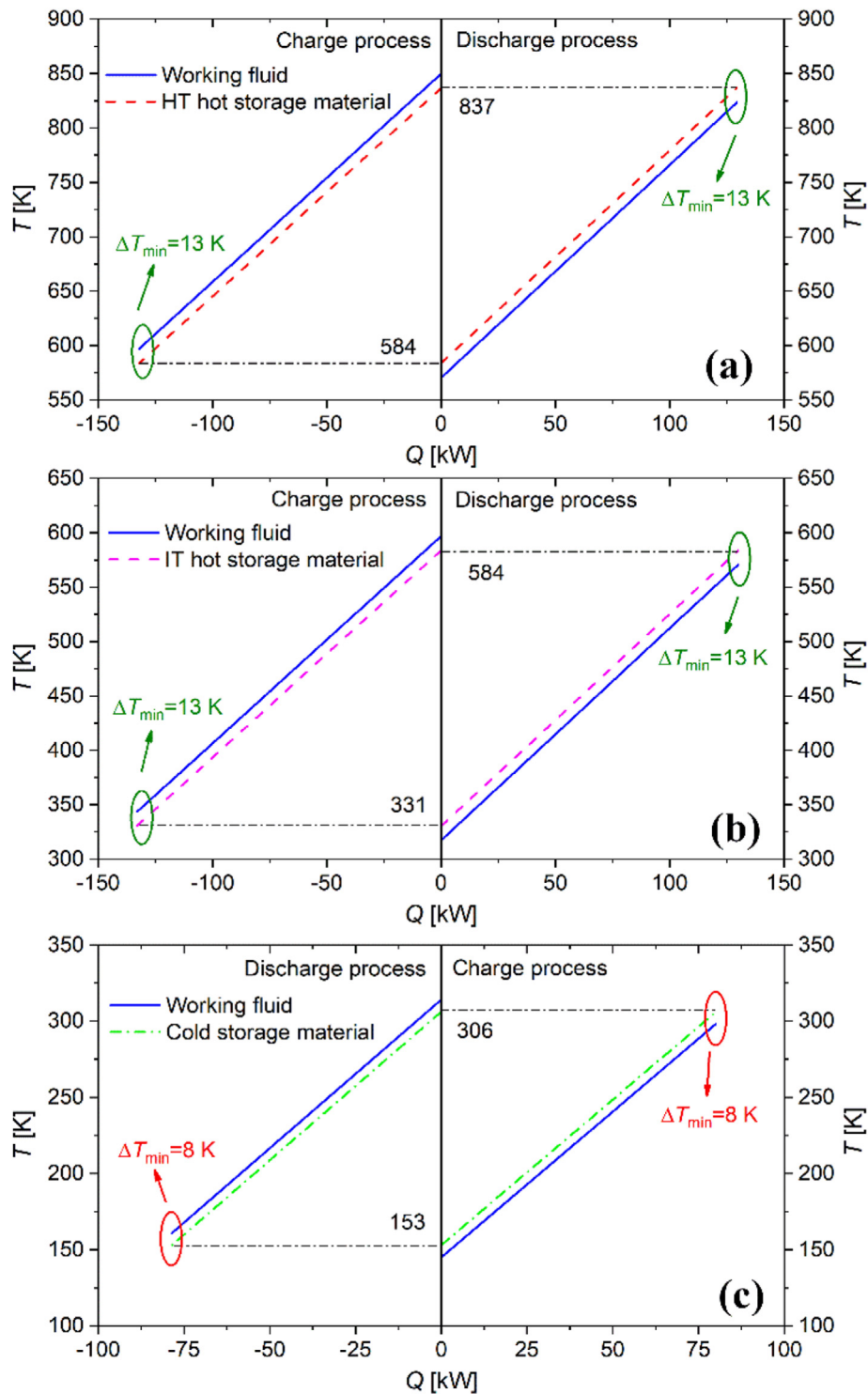


Fig. 10. T-Q diagrams in hot and cold heat exchangers of Joule-Brayton PTES systems based on SC1 (both HX1 and HX2 used) during charge and discharge: (a) HT hot heat exchanger with argon used as the working fluid and solar salt as the HT hot storage material; (b) IT hot heat exchanger with argon used as the working fluid and Therminol 66 as the IT hot storage material; and (c) cold heat exchanger with argon used as the working fluid and pentane as the cold storage material (see also Figs. 2 and 8). N.B.: For $Q < 0$ the working fluid releases heat to the store, and for $Q > 0$ the working fluid absorbs heat from the store.

higher efficiencies (60%–90%) [23] than TMES systems, the efficiency gap among these technologies is not substantial. The roundtrip efficiencies of PTES systems in this paper are close to 70%, and with technological improvements in key components, this technology has a promising potential in relevant applications, i.e.

grid-scale, medium/long-duration electricity storage with few geographical constraints, low environmental impact and long lifetimes, whilst providing additional advantages in providing thermal energy along with electrical energy during discharge, and to be integrated with other (e.g., solar or waste) heat or source. For a

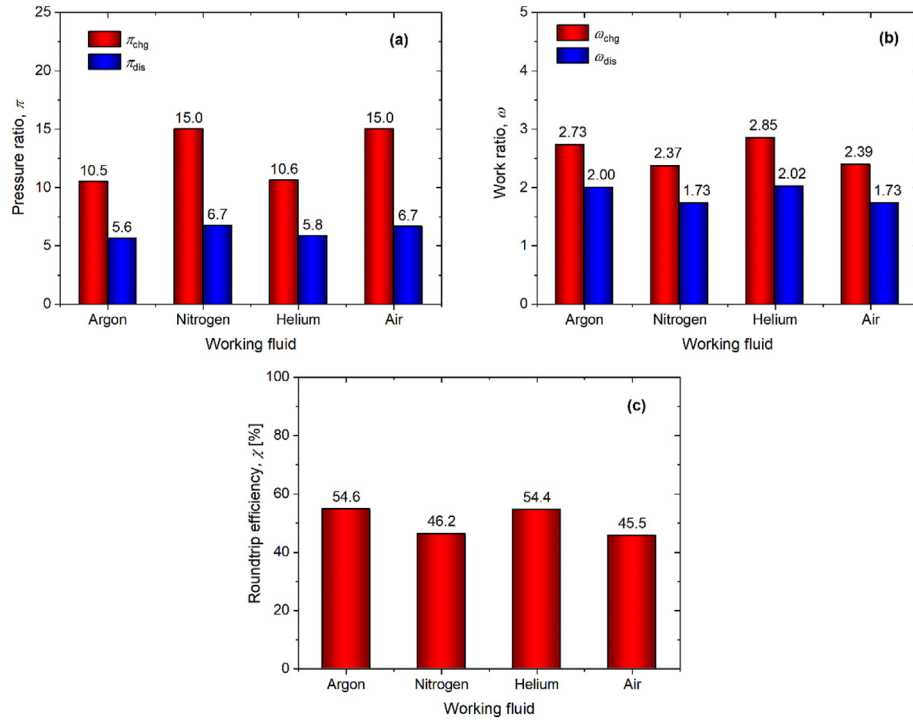


Fig. 11. Thermodynamic parameter comparison of 50 MW Joule-Brayton PTES systems based on SC1 with 6 h discharge duration (both HX1 and HX2 used), when using different working fluids and solar salt, Therminol 66 and pentane as the storage materials: (a) pressure ratio; (b) work ratio; and (c) roundtrip efficiency (see also Figs. 2 and 8).

Table 6

Roundtrip efficiencies of 50 MW Joule-Brayton PTES systems based on SC1 with 6 h discharge duration, and when using argon as the working fluid and different liquid storage materials. Best value is shown in **bold**.

Hot storage materials	Cold storage fluids				
	Pentane	Ethanol	Butane	Heptane	Hexane
Solar salt + Therminol 66	54.6	50.6	55.3	44.3	46.0
HITEC + Therminol 66	53.7	50.5	54.2	42.4	44.2
HITEC XL + Therminol 66	53.9	50.5	54.5	42.4	44.2
Solar salt + Therminol VP-1	54.6	50.6	55.3	44.2	45.6
HITEC + Therminol VP-1	53.7	50.5	54.2	42.4	44.2
HITEC XL + Therminol VP-1	53.9	50.5	54.5	42.4	44.2
HITEC + Rapeseed oil	51.6	49.8	51.6	42.4	44.2
HITEC XL + Rapeseed oil	51.6	49.8	51.6	42.4	44.2

more detailed discussion and comparison of PTES and, by extension, TMES systems to PHS and other battery technologies the interested reader is referred to the review by Olympios et al. [23].

5. Combined thermodynamic and economic performance comparisons of PTES systems

Based on the optimal thermodynamic designs of the different PTES systems in Sections 4.1–4.3, the economic performance of eight PTES system design cases based on a pre-selected power capacity (50 MW) and discharge duration (6 h) are evaluated and compared here. The results of the optimised Joule-Brayton PTES systems with STRs in Section 4.1 indicate that the highest roundtrip efficiency is attained by configuration SC1 with helium as the working fluid. Hence, this system is selected for further study and referred to here as Case 1. In addition, a full capital cost analysis of all SCs (shown in Table 10) reveals that configuration SC2 has a roundtrip efficiency close to that of SC1 but the lowest TCC, PCC and ECC; thus, this system is also selected to investigate the role of different SCs and working fluids further. This PTES system is referred to when using helium and nitrogen as the working fluids in Cases 2 and 3, respectively. Finally, because the storage medium

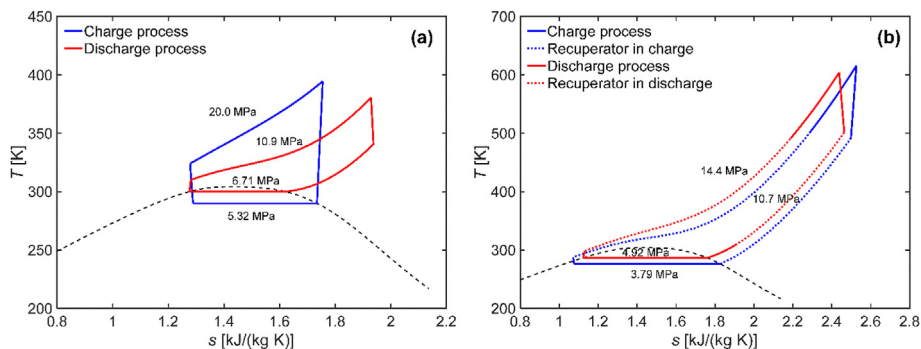


Fig. 12. T-s diagrams of reversible transcritical Rankine PTES systems with CO₂ as the working fluid and Therminol 66 (LTS) as the storage material: (a) without; and (b) with a recuperator (see also Fig. 3).

Table 7

Thermodynamic performance of 50 MW transcritical Rankine PTES systems with 6 h discharge duration, when using CO₂ as the working fluid and different storage materials. Best values are shown in **bold**.

Performance parameter	Pressurised water		Therminol 66		Therminol VP-1		Rapeseed oil	
	No	Yes	No	Yes	No	Yes	No	Yes
Recuperator included	No	Yes	No	Yes	No	Yes	No	Yes
Pressure at compressor inlet during charge [MPa]	5.32	5.31	5.32	3.79	4.27	3.74	5.32	4.33
Pressure at compressor outlet during charge [MPa]	20.0	9.41	20.0	10.7	20.0	12.0	20.0	10.5
Pressure at pump inlet during discharge [MPa]	6.71	6.71	6.71	4.92	6.12	4.85	6.71	5.52
Pressure at pump outlet during discharge [MPa]	10.9	11.7	10.9	14.4	10.6	16.6	10.9	13.8
Work ratio, ω_{chg} [–]	3.35	11.4	3.35	18.0	3.82	17.8	3.35	14.2
Work ratio, ω_{dis} [–]	2.93	4.00	2.92	6.81	3.57	6.59	2.92	5.11
Roundtrip efficiency, χ [%]	39.1	55.1	38.6	67.0	40.7	68.0	38.6	62.3

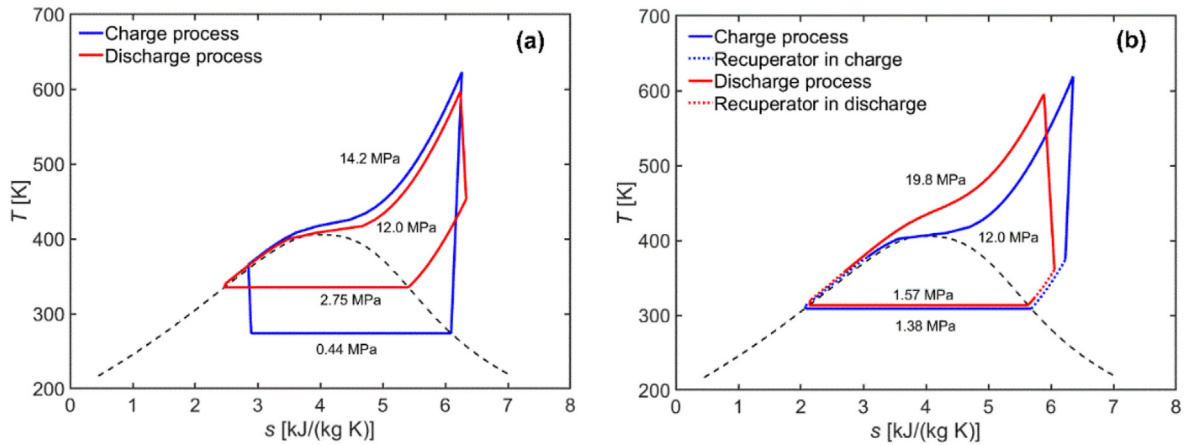


Fig. 13. T - s diagrams of reversible transcritical Rankine PTES systems with NH₃ as the working fluid and Therminol 66 (LTS) as the storage material: (a) without; and (b) with a recuperator (see also Fig. 3).

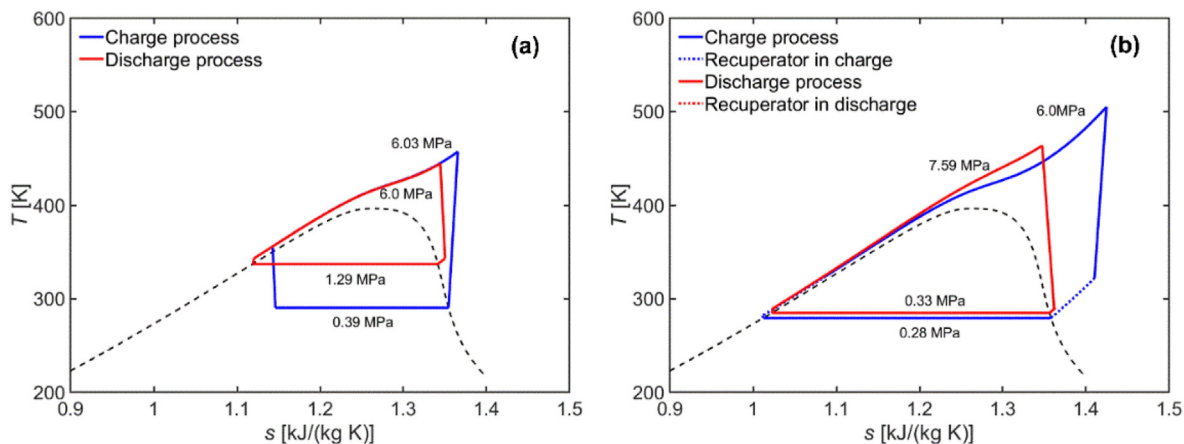


Fig. 14. T - s diagrams of reversible transcritical Rankine PTES systems using R1311 as the working fluid and Therminol 66 (LTS) as the storage material: (a) without; and (b) with a recuperator (see also Fig. 3).

does not affect roundtrip efficiency, the storage material with the smallest reservoir volume (alumina, according to Table 5) is selected for Cases 1 to 3 as this has the lowest cost thermal reservoirs.

For the Joule-Brayton PTES systems with LTSs, the results in Section 4.2 show that systems based on SC1, SC2, SC1r, SC2r and SC3r, which use argon as the working fluid and a combination of solar salt, Therminol 66 and butane as the storage materials, have (similar) high roundtrip efficiencies. However, a full capital analysis of all SCs (shown in Table 11) reveals that SC1 has the lowest TCC, PCC and ECC. Therefore, this system is taken as Case 4 for the

thermo-economic investigation. Furthermore, to perform a wider comparison of Joule-Brayton PTES systems with LTSs, Cases 5 and 6, which involve helium and nitrogen as the working fluids, respectively, are included in the analysis. Finally, for the transcritical Rankine PTES systems with LTSs, the results in Section 4.3 indicate that Therminol VP-1 is the best choice for the storage material. Thus, Cases 7 (with CO₂ as the working fluid) and 8 (with NH₃ as the working fluid) are included in the analysis. All eight investigated cases are summarised in Table 12.

Before proceeding to the economic results, the roundtrip efficiencies of all eight PTES system cases considered in this section are

Table 8

Thermodynamic performance of 50 MW transcritical Rankine PTES systems with 6 h discharge duration, when using NH_3 as the working fluid and different storage materials. Best values are shown in **bold**.

Performance parameter	Therminol 66		Therminol VP-1	
	No	Yes	No	Yes
Recuperator included	No	Yes	No	Yes
Pressure at compressor inlet during charge [MPa]	0.44	1.38	0.43	0.76
Pressure at compressor outlet during charge [MPa]	14.2	12.0	14.5	12.0
Pressure at pump inlet during discharge [MPa]	2.75	1.57	2.14	0.76
Pressure at pump outlet during discharge [MPa]	12.0	19.8	20.0	20.0
Work ratio, ω_{chg} [–]	10.4	35.2	11.3	47.0
Work ratio, ω_{dis} [–]	14.1	11.1	14.7	15.3
Roundtrip efficiency, χ [%]	42.2	46.4	53.7	53.9

summarised in Fig. 17, which shows that amongst all investigated cases, the transcritical system achieves the highest roundtrip efficiency (68.0% for Case 7). This result is higher than that of Morandin et al. [67] and can be explained in terms of the maximum working temperature of Therminol VP-1, which is higher than that of pressurised water. When using the same working fluid in Joule–Brayton PTES systems with STRs and LTSs, the efficiencies of the former (Cases 1 and 3) are higher than those of the latter (Cases 5 and 6), with the highest value attained with helium in Case 1 (66.5%). Joule–Brayton PTES systems with STRs or LTSs have the lowest efficiencies when using nitrogen as the working fluid (Cases 3 and 6).

Based on the optimal thermodynamic parameters obtained for each case, the sizes of the main components (compressors, expanders, pumps, storage tanks, pressurised vessels, heat exchangers and electrical generators) of the PTES systems are identified using the methods described in Section 3.1. These parameters are then used to calculate the capital cost of the components, and subsequently, the whole systems. A mean cost estimation is employed for each component using four costing methods (described in Section 3.3), along with associated uncertainty ranges, as it is still unknown that which costing method available in the literature provides the most accurate estimation. Finally, an economic performance comparison of the investigated thermodynamically optimal designs is performed and the results are shown in Fig. 18.

The results show that Case 7 (transcritical cycle with a recuperator and using CO_2 as the working fluid) is associated with the lowest TCC of 209 M\$, whereas Case 6 (Joule–Brayton cycle with LTSs and nitrogen as the working fluid) achieves the highest TCC, equal to 409 M\$. Focusing on the Joule–Brayton PTES systems with STRs (Cases 1 to 3), it shows that the TCC of SC2 is slightly lower than that of SC1, which is attributed to the absence of an additional heat exchanger. The TCC when using nitrogen as the working fluid is higher than when using helium due to the higher cost of the HR, which is attributed to the higher working pressure, as shown in Case 3 in Fig. 18(b). Moreover, the main components of the TCC

Table 9

Thermodynamic performance of 50 MW transcritical Rankine PTES systems with 6 h discharge duration, when using R1311 as the working fluid and different storage materials. Best values are shown in **bold**.

Performance parameter	Therminol 66		Therminol VP-1		Rapeseed oil	
	No	Yes	No	Yes	No	Yes
Recuperator included	No	Yes	No	Yes	No	Yes
Pressures at the inlet of the compressor in charge [MPa]	0.39	0.28	0.23	0.27	0.41	0.27
Pressures at the outlet of the compressor in charge [MPa]	6.03	6.0	20.0	6.0	6.0	6.0
Pressures at the inlet of the pump in discharge [MPa]	1.29	0.33	0.92	0.32	1.43	0.32
Pressures at the outlet of the pump in discharge [MPa]	6.0	7.59	15.3	7.67	6.11	7.51
Work ratio, ω_{chg} [–]	6.56	22.6	4.74	22.8	5.94	22.9
Work ratio, ω_{dis} [–]	4.50	7.03	3.33	6.98	4.23	7.06
Roundtrip efficiency, χ [%]	39.4	63.4	39.9	63.5	37.3	62.6

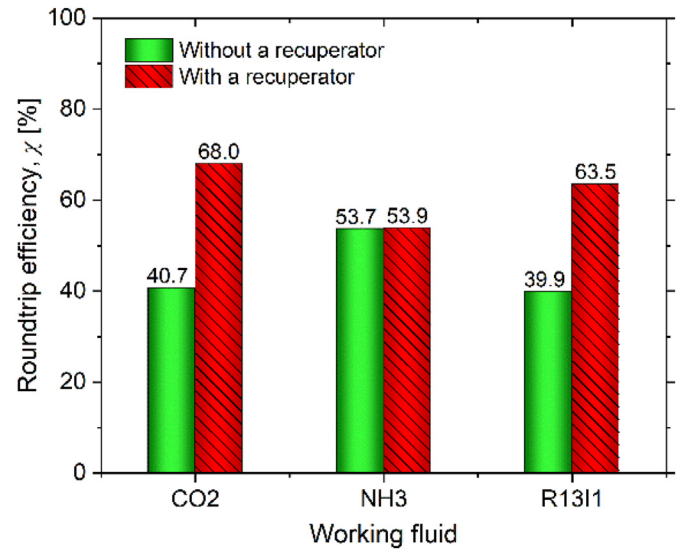


Fig. 15. Roundtrip efficiency comparison of 50 MW transcritical Rankine PTES systems with 6 h discharge duration, when using different working fluids and Therminol VP-1 (LTS) as the storage material.

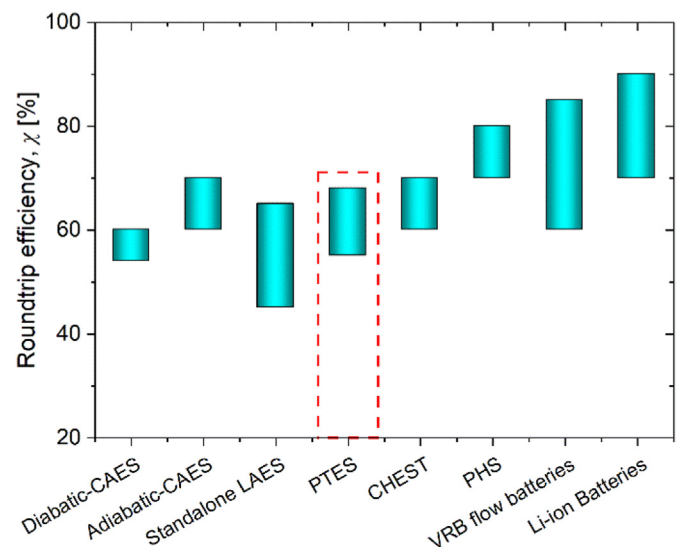


Fig. 16. Comparison of roundtrip efficiency of the PTES systems with other TMES systems and large-scale flow and Li-ion batteries. Data for technologies other than PTES are from the review by Olympios et al. [23].

involve the HR (23%–37%), the compressor required for charge (22%–29%) and the expander required for discharge (16%–22%).

Table 10

Thermo-economic performance comparison of all Joule-Brayton PTES systems with STRs when using helium as the working fluid and alumina as the packed-bed material, where SC1 – both HX1 and HX2 are used, but no recuperator; SC2 – only HX1 is used; SC3 – only HX2 is used; SC1r – HX1, HX2 and a recuperator are used; SC2r – HX1 and a recuperator are used; and SC3r – HX2 and a recuperator are used (see also Figs. 1 and 5).

System configurations	SC1	SC2	SC3	SC1r	SC2r	SC3r
Roundtrip efficiency, χ [%]	66.5	66.0	53.3	66.5	66.5	64.2
Total capital cost, TCC [M\$]	250	245	410	254	254	302
Power capital cost, PCC [\$/kW]	5000	4910	8200	5080	5080	6030
Energy capital cost, ECC [\$/kWh]	833	818	1370	846	846	1010

Table 11

Thermo-economic performance comparison of all Joule-Brayton PTES systems with LTSs when using argon as the working fluid and a combination of solar salt, Therminol 66 and butane as the storage materials, where SC1 – both AHX1 and AHX2 are used, but no recuperator; SC2 – only AHX1 is used; SC3 – only AHX2 is used; SC1r – AHX1, AHX2 and a recuperator are used; SC2r – AHX1 and a recuperator are used; and SC3r – AHX2 and a recuperator are used (see also Figs. 2 and 8).

System configurations	SC1	SC2	SC3	SC1r	SC2r	SC3r
Roundtrip efficiency, χ [%]	55.3	53.7	38.8	55.7	55.7	56.6
Total capital cost, TCC [M\$]	307	317	492	450	378	768
Power capital cost, PCC [\$/kW]	6140	6340	9830	9000	7560	15400
Energy capital cost, ECC [\$/kWh]	1020	1060	1640	1500	1260	2560

Comparing the Joule-Brayton PTES systems with STRs (Cases 1 and 3) to the PTES systems with LTSs (Cases 5 and 6) shows that the latter systems are more expensive when using the same working fluid. Furthermore, Fig. 18(b) shows that the main components that contribute to the TCC of Joule-Brayton PTES systems with LTSs are the compressor required for charge (30%–35%), the expander required for discharge (17%–23%) and the heat exchangers (13%–24%). For the transcritical Rankine PTES systems with LTSs (Cases 7 and 8), the heat exchangers are the most expensive components due to the large heat transfer areas of the evaporator and condenser, which account for 33%–41% of the TCC. However, the TCCs of the transcritical Rankine PTES systems are all lower than those of the Joule-Brayton PTES systems.

The PCCs and ECCs of three shortlisted PTES systems (Cases 2, 5 and 7) are compared in Fig. 19 over a wider range of discharge durations (2–12 h) and power capacities (10–100 MW). These three cases were selected because they have the lowest TCCs compared with other PTES system cases, as shown in Fig. 18(a). For a fixed power capacity of 50 MW, discharge duration has a positive effect on the PCC and a negative effect on the ECC. Transcritical Rankine PTES systems with LTSs (Case 7) have the lowest PCC and ECC, the latter marginally so. When the discharge duration is shorter than 8 h, the highest PCC and ECC are obtained for the

Table 12

Description of eight PTES system cases used for thermo-economic performance evaluation.

Case	Cycle	Working fluid	Storage type	Storage material(s)	Configuration ^a
Case 1	Joule-Brayton	Helium	Packed-bed	Alumina	SC1
Case 2	Joule-Brayton	Helium	Packed-bed	Alumina	SC2
Case 3	Joule-Brayton	Nitrogen	Packed-bed	Alumina	SC2
Case 4	Joule-Brayton	Argon	Liquid-tank	Solar salt + Therminol 66 + Butane	SC1
Case 5	Joule-Brayton	Helium	Liquid-tank	Solar salt + Therminol 66 + Butane	SC1
Case 6	Joule-Brayton	Nitrogen	Liquid-tank	Solar salt + Therminol 66 + Butane	SC2
Case 7	Transcritical Rankine	CO ₂	Liquid-tank	Therminol VP-1	With recuperator
Case 8	Transcritical Rankine	NH ₃	Liquid-tank	Therminol VP-1	With recuperator

^a See Figs. 1–3, plus Figs. 5 and 8 and Figs. 12–14 for the definitions of all PTES system variant/configuration cases.

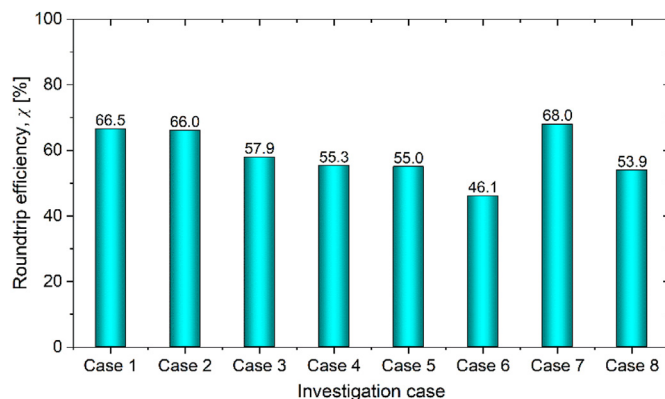


Fig. 17. Thermodynamic performance comparison of 50 MW PTES system designs under investigation with 6 h discharge duration. All cases (PTES system variants) are defined in Table 12.

Joule-Brayton PTES system with LTSs (Case 5), whereas for higher discharge durations (>8 h), the PCC and ECC of Joule-Brayton PTES system with STRs (Case 2) surpass those of the system with LTSs (Case 5). Within the investigated range of discharge durations, the lowest PCC and ECC are 3790 \$/kW (discharge duration is equal to 2 h) and 396 \$/kWh (discharge duration is equal to 12 h), respectively, which both correspond to the transcritical Rankine PTES systems with LTSs (Case 7). Fig. 19(b) shows the PCCs and ECCs of the three shortlisted cases all decrease with the system power capacity. If the power capacity is high enough, the PCCs and ECCs of all PTES systems no longer change appreciably with changes in the power capacity.

In conclusion, the analysis shows that when sensible heat storage media are selected, transcritical Rankine PTES systems using CO₂ as the working fluid and Therminol VP-1 as the storage material appears as a promising choice when considering thermodynamic and economic performance. Predictably, a suitable selection of discharge duration and power capacity makes PTES systems more economically competitive against other large-scale electricity storage technologies, such as CAES and PHS. Comparisons of PCCs and ECCs of these PTES systems with other TMES systems as well as large-scale storage flow batteries and Li-ion batteries are shown in Fig. 20, although these results do not reflect the much longer lifetimes, e.g. Li-ion batteries. Bearing this in mind, PHS and CAES are associated with the lowest ECC (<80 \$/kWh) [23] but are more technologically and commercially mature and available for larger-scale electricity storage; VRB flow batteries and Li-ion batteries have good PCCs but suffer from high ECCs compared with TMES systems (~600 \$/kWh and ~2000 \$/kWh, respectively) [23]. Furthermore, the PCCs and ECCs of LAES, CHEST,

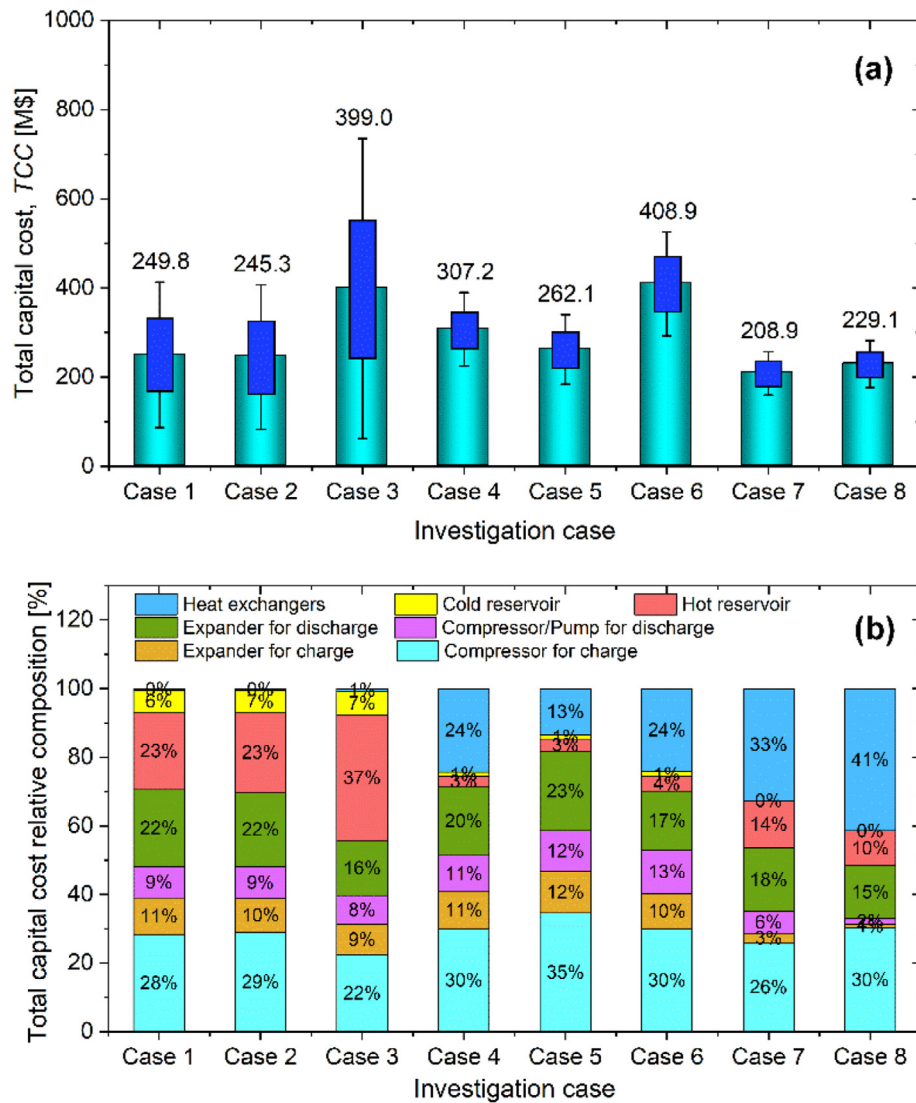


Fig. 18. Economic performance comparison of 50 MW PTES system designs under investigation with 6 h discharge duration: (a) total capital cost; and (b) capital cost breakdown. Costs are calculated using mean estimation from four different costing methods; associated lower and upper bounds are provided (error bars), along with the mean absolute errors (solid bars). All cases (PTES system variants) are defined in Table 12.

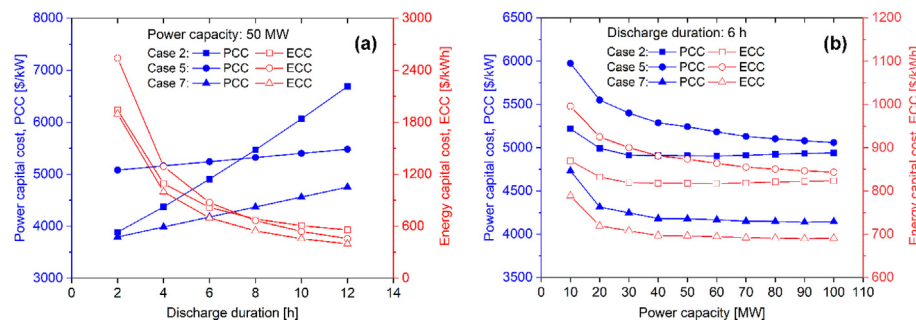


Fig. 19. Power and energy capital costs of PTES system designs under investigation for varying: (a) discharge duration; and (b) power capacity. All cases (PTES system variants) are defined in Table 12.

and PTES systems are very close to one another (~3500 \$/kW and ~400 \$/kWh, respectively) [23]. PTES and LAES technologies are still under development. The future uptake can affect costs through

economies of scale, and the costs depend strongly on the intended power capacity and discharge duration of the systems.

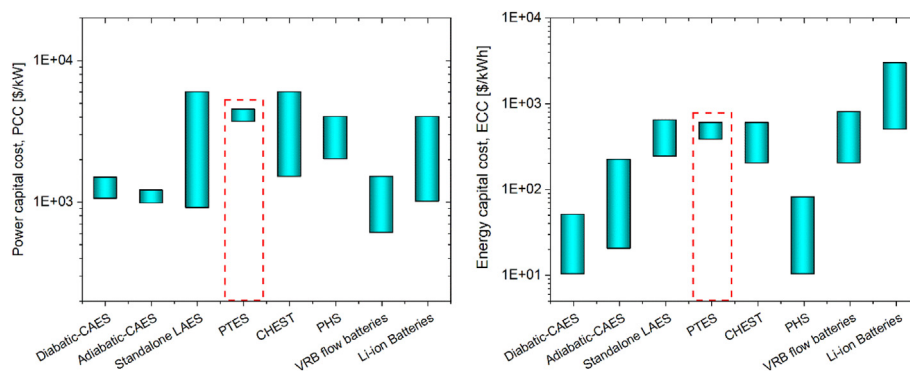


Fig. 20. Comparisons of power and energy capital costs of the PTES systems with other thermo-mechanical energy storage systems and large-scale storage flow batteries and Li-ion batteries. Data for technologies other than PTES are from the review by Olympios et al. [23].

6. Conclusions

Pumped-thermal electricity storage (PTES) is a medium-to large-scale, longer-duration thermo-mechanical energy storage technology that presents the advantages of few geographical constraints, low capital costs, long lifetimes and flexible power rating. Therefore, it exhibits great potential as a key component of future electric power systems. This paper focussed on presently available sensible heat storage materials and analysed three distinct PTES system variants: reversible Joule-Brayton PTES systems with solid thermal reservoirs (STRs), reversible Joule-Brayton PTES systems with liquid thermal stores (LTSs), and reversible transcritical Rankine PTES systems with LTSs. Comprehensive investigations and comparisons of thermodynamic and economic performance of these systems were presented for a power rating of 50 MW and discharge duration of 6 h.

Parametric design optimisation of each PTES system variant considering different system configurations, working fluids and storage media was initially performed from a thermodynamic perspective. The results revealed that for Joule-Brayton PTES systems with STRs, the best thermodynamic performance (roundtrip efficiency) were associated with system configurations with two auxiliary heat exchangers or an auxiliary heat exchanger after the expander in discharge, and helium or argon as the working fluid; for Joule-Brayton PTES systems with LTSs, most of the system configurations, with either argon or helium as the working fluid, showed comparable best roundtrip efficiencies, with values in the range 54–57%. A combination of solar salt + Therminol or HITEC + Therminol as the hot storage material and butane or pentane as the cold storage material allowed the highest roundtrip efficiencies, to within $\pm 1\%$. Turning to the transcritical Rankine PTES systems with LTSs, the highest roundtrip efficiency (68%) was exhibited by systems which featured a recuperator and that employed CO_2 as the working fluid, respectively. When using the same working fluid in Joule-Brayton PTES systems, the roundtrip efficiencies of systems that employed STRs were higher than those of equivalent systems with LTSs.

Among all the investigated PTES systems, the predicted roundtrip efficiency reached a (highest) value of 68%, which was similar to other TMES systems but slightly lower than large-scale PHS, VRB flow batteries and Li-ion batteries. This needs to be carefully considered as PTES systems offer added possibilities of providing and/or utilising thermal energy along with electrical energy during operation, when appropriately integrated with external sources and sinks (e.g., solar or waste heat).

Based on the aforementioned optimal thermodynamic parameters, the sizes of the main components of the considered PTES

systems were identified, and the corresponding capital costs were evaluated using mean estimations from four different costing methods to provide a unified costing framework. Based on these, the economic performance of eight selected designs based on the three types of PTES systems were analysed and compared. The main observations were that transcritical Rankine PTES systems that employ CO_2 as the working fluid, Therminol VP-1 as the storage material and include a recuperator had the lowest total capital cost (209 M\$) for the specific system specification (50 MW capacity and 8 h discharge duration). Finally, the power and energy capital costs of three optimised designs were compared for discharge durations varying over a wider range, and the results demonstrated that increasing the discharge duration has a positive effect on the power capital cost and a negative effect on the energy capital cost. Within the investigated range of discharge durations (2–12 h), the lowest power and energy capital costs were 3790 \$/kW (discharge duration is equal to 2 h) and 396 \$/kWh (discharge duration is equal to 12 h), respectively, which both corresponded to the transcritical Rankine PTES system with LTSs.

Compared with other TMES systems and large-scale PHS, VRB flow batteries and Li-ion batteries, the power and energy capital costs were very close to those of LAES and CHEST systems, lower than VRB flow batteries and Li-ion batteries but higher than commercially available, more mature technologies, such as CAES and PHS. With technological improvements to the key components of PTES systems, it can be expected that these systems will appear as a promising solution for grid-scale, medium/long-duration electricity storage with low environmental effects, lifetimes, and with fewer geographical constraints than CAES that relies on underground storage or PHS.

Overall, the transcritical Rankine PTES system using CO_2 as the working fluid and Therminol VP-1 as the storage material appears promising within the envelope of this investigation, when considering both thermodynamic and economic performance. The present paper provides detailed guidance for the design, optimisation and further development of PTES technology, and will be of interest to component manufacturers, technology developers, energy system operators, whole-energy system modellers and policymakers.

CRediT authorship contribution statement

Yongliang Zhao: Data curation, Software, Writing – original draft, Writing – review & editing. **Jian Song:** Conceptualization, Methodology, Writing – original draft, Writing – review & editing. **Ming Liu:** Conceptualization, Methodology, Writing – original draft, Writing – review & editing. **Yao Zhao:** Investigation,

Validation, Writing – review & editing. **Andreas V. Olympios:** Writing – review & editing. **Paul Sapin:** Writing – review & editing. **Junjie Yan:** Conceptualization, Funding acquisition, Supervision. **Christos N. Markides:** Conceptualization, Project administration, Supervision.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have influenced the work reported in this paper.

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Appendix A. Supplementary data

Supplementary data to this article can be found online at <https://doi.org/10.1016/j.renene.2022.01.017>.

References

- [1] J. Olauson, M.N. Ayob, M. Bergkvist, et al., Net load variability in Nordic countries with a highly or fully renewable power system, *Nat. Energy* 1 (175) (2016) 1–8, <https://doi.org/10.1038/nenergy.2016.175>.
- [2] J. Li, F. Liu, Z.Y. Li, et al., Grid-side flexibility of power systems in integrating large-scale renewable generations: a critical review on concepts, formulations and solution approaches, *Renew. Sustain. Energy Rev.* 93 (2018) 272–284, <https://doi.org/10.1016/j.rser.2018.04.109>.
- [3] M.R.M. Cruz, D.Z. Fitiwi, S.F. Santos, et al., A comprehensive survey of flexibility options for supporting the low-carbon energy future, *Renew. Sustain. Energy Rev.* 97 (2018) 338–353, <https://doi.org/10.1016/j.rser.2018.08.028>.
- [4] M.A. Gonzalez-Salazar, T. Kirsten, L. Prchlik, Review of the operational flexibility and emissions of gas- and coal-fired power plants in a future with growing renewables, *Renew. Sustain. Energy Rev.* 82 (2018) 1497–1513, <https://doi.org/10.1016/j.rser.2017.05.278>.
- [5] Y.L. Zhao, C.Y. Wang, M. Liu, et al., Improving operational flexibility by regulating extraction steam of high-pressure heaters on a 660 MW supercritical coal-fired power plant: a dynamic simulation, *Appl. Energy* 212 (2018) 1295–1309, <https://doi.org/10.1016/j.apenergy.2018.01.017>.
- [6] Y.L. Zhao, M. Liu, C.Y. Wang, et al., Increasing operational flexibility of supercritical coal-fired power plants by regulating thermal system configuration during transient processes, *Appl. Energy* 228 (2018) 2375–2386, <https://doi.org/10.1016/j.apenergy.2018.07.070>.
- [7] Y.L. Zhao, M. Liu, C.Y. Wang, et al., Exergy analysis of the regulating measures of operational flexibility in supercritical coal-fired power plants during transient processes, *Appl. Energy* 253 (2019) 113487, <https://doi.org/10.1016/j.apenergy.2019.113487>.
- [8] A. Bloess, W.-P. Schill, A. Zerrahn, Power-to-heat for renewable energy integration: a review of technologies, modeling approaches, and flexibility potentials, *Appl. Energy* 212 (2018) 1611–1626, <https://doi.org/10.1016/j.apenergy.2017.12.073>.
- [9] L.X. Zhang, N. Good, P. Mancarella, Building-to-grid flexibility: modelling and assessment metrics for residential demand response from heat pump aggregations, *Appl. Energy* 233–234 (2019) 709–723, <https://doi.org/10.1016/j.apenergy.2018.10.058>.
- [10] L. Söder, P.D. Lund, H. Koduvere, et al., A review of demand side flexibility potential in Northern Europe, *Renew. Sustain. Energy Rev.* 91 (2018) 654–664, <https://doi.org/10.1016/j.rser.2018.03.104>.
- [11] O. Schmidt, A. Hawkes, A. Gambhir, et al., The future cost of electrical energy storage based on experience rates, *Nat. Energy* 2 (8) (2017), <https://doi.org/10.1038/nenergy.2017.110>.
- [12] X. Luo, J.H. Wang, M. Dooner, et al., Overview of current development in electrical energy storage technologies and the application potential in power system operation, *Appl. Energy* 137 (2015) 511–536, <https://doi.org/10.1016/j.apenergy.2014.09.081>.
- [13] I. Hadjipaschalis, A. Poullikkas, V. Efthimiou, Overview of current and future energy storage technologies for electric power applications, *Renew. Sustain. Energy Rev.* 13 (6–7) (2009) 1513–1522, <https://doi.org/10.1016/j.rser.2008.09.028>.
- [14] T.M. Gür, Review of electrical energy storage technologies, materials and systems: challenges and prospects for large-scale grid storage, *Energy Environ. Sci.* 11 (10) (2018) 2696–2767, <https://doi.org/10.1039/c8ee01419a>.
- [15] S. Hameer, J.L. van Niekerk, A review of large-scale electrical energy storage, *Int. J. Energy Res.* 39 (9) (2015) 1179–1195, <https://doi.org/10.1002/er.3294>.
- [16] W. He, J.H. Wang, Optimal selection of air expansion machine in compressed air energy storage: a review, *Renew. Sustain. Energy Rev.* 87 (2018) 77–95, <https://doi.org/10.1016/j.rser.2018.01.013>.
- [17] H.L. Ferreira, R. Garde, G. Fulli, et al., Characterisation of electrical energy storage technologies, *Energy* 53 (2013) 288–298, <https://doi.org/10.1016/j.energy.2013.02.037>.
- [18] J. Mouli-Castillo, M. Wilkinson, D. Mignard, et al., Inter-seasonal compressed-air energy storage using saline aquifers, *Nat. Energy* 4 (2) (2019) 131–139, <https://doi.org/10.1038/s41560-018-0311-0>.
- [19] I. Lee, F.Q. You, Systems design and analysis of liquid air energy storage from liquefied natural gas cold energy, *Appl. Energy* 242 (2019) 168–180, <https://doi.org/10.1016/j.apenergy.2019.03.087>.
- [20] A. Benato, A. Stoppato, Pumped thermal electricity storage: a technology overview, *Therm. Sci. Eng. Prog.* 6 (2018) 301–315, <https://doi.org/10.1016/j.tsep.2018.01.017>.
- [21] A. Benato, A. Stoppato, A. Mirandola, State-of-the-art and future development of sensible heat thermal electricity storage systems, *Int. J. Heat Technol.* 35 (2017) S244–S251, <https://doi.org/10.18280/ijht.35Sp0134>.
- [22] W.-D. Steinmann, Thermo-mechanical concepts for bulk energy storage, *Renew. Sustain. Energy Rev.* 75 (2017) 205–219, <https://doi.org/10.1016/j.rser.2016.10.065>.
- [23] A.V. Olympios, J.D. McTigue, P. Farres-Antunez, et al., Progress and prospects of thermo-mechanical energy storage – a critical review, *Prog. Energy* 3 (2020) 100033, <https://doi.org/10.1088/2516-1083/abdbba>.
- [24] A. Thess, Thermodynamic efficiency of pumped heat electricity storage, *Phys. Rev. Lett.* 111 (11) (2013) 110602, <https://doi.org/10.1103/PhysRevLett.111.110602>.
- [25] J.C. Guo, L. Cai, J.C. Chen, et al., Performance optimization and comparison of pumped thermal and pumped cryogenic electricity storage systems, *Energy* 106 (2016) 260–269, <https://doi.org/10.1016/j.energy.2016.03.053>.
- [26] J.C. Guo, L. Cai, H.X. Yang, et al., Performance characteristics and parametric optimizations of a weak dissipative pumped thermal electricity storage system, *Energy Convers. Manag.* 157 (2018) 527–535, <https://doi.org/10.1016/j.enconman.2017.12.031>.
- [27] T. Desrues, J. Ruer, P. Marty, et al., A thermal energy storage process for large scale electric applications, *Appl. Therm. Eng.* 30 (5) (2010) 425–432, <https://doi.org/10.1016/j.applthermaleng.2009.10.002>.
- [28] J. Howes, Concept and development of a pumped heat electricity storage device, *Proc. IEEE* 100 (2) (2011) 493–503, <https://doi.org/10.1109/JPROC.2011.2174529>.
- [29] A.J. White, Loss analysis of thermal reservoirs for electrical energy storage schemes, *Appl. Energy* 88 (11) (2011) 4150–4159, <https://doi.org/10.1016/j.apenergy.2011.04.030>.
- [30] A. White, G. Parks, C.N. Markides, Thermodynamic analysis of pumped thermal electricity storage, *Appl. Therm. Eng.* 53 (2) (2013) 291–298, <https://doi.org/10.1016/j.applthermaleng.2012.03.030>.
- [31] J.D. McTigue, A.J. White, C.N. Markides, Parametric studies and optimisation of pumped thermal electricity storage, *Appl. Energy* 137 (2015) 800–811, <https://doi.org/10.1016/j.apenergy.2014.08.039>.
- [32] J.D. McTigue, Doctoral dissertation, in: Analysis and Optimisation of Thermal Energy Storage, Department of Engineering, University of Cambridge, 2016, <https://doi.org/10.17863/CAM.7084>.
- [33] P. Farres-Antunez, Modelling of thermal energy storage systems for bulk electricity storage, <https://doi.org/10.17863/CAM.42388>, 2015.
- [34] F. Ni, H.S. Caram, Analysis of pumped heat electricity storage process using exponential matrix solutions, *Appl. Therm. Eng.* 84 (2015) 34–44, <https://doi.org/10.1016/j.applthermaleng.2015.02.046>.
- [35] J.C. Guo, L. Cai, J.C. Chen, et al., Performance evaluation and parametric choice criteria of a Brayton pumped thermal electricity storage system, *Energy* 113 (2016) 693–701, <https://doi.org/10.1016/j.energy.2016.07.080>.
- [36] A. Smallbone, V. Jülch, R. Wardle, et al., Levelised cost of storage for pumped heat energy storage in comparison with other energy storage technologies, *Energy Convers. Manag.* 152 (2017) 221–228, <https://doi.org/10.1016/j.enconman.2017.09.047>.
- [37] A. Smallbone, Carnot Battery Energy Storage, 2019, <https://www.rushlightevents.com/wp-content/uploads/2019/02/RUSHLIGHT-SHOW-2019-CLEANTECH-CONF-PANEL-1-NEWCASTLE-UNIV.pdf>.
- [38] A. Benato, Performance and cost evaluation of an innovative pumped thermal electricity storage power system, *Energy* 138 (2017) 419–436, <https://doi.org/10.1016/j.energy.2017.07.066>.
- [39] A. Benato, A. Stoppato, Heat transfer fluid and material selection for an innovative pumped thermal electricity storage system, *Energy* 147 (2018) 155–168, <https://doi.org/10.1016/j.energy.2018.01.045>.
- [40] A. Benato, A. Stoppato, Energy and cost analysis of a new packed bed pumped thermal electricity storage unit, *J. Energy Resour. Technol.* 140 (2) (2018) 1–7, <https://doi.org/10.1115/1.4038197>.

- [41] L.X. Chen, P. Hu, P.P. Zhao, et al., Thermodynamic analysis of a high temperature pumped thermal electricity storage (HT-PTES) integrated with a parallel organic Rankine cycle (ORC), *Energy Convers. Manag.* 177 (2018) 150–160, <https://doi.org/10.1016/j.enconman.2018.09.049>.
- [42] S. Georgiou, M. Aunedi, G. Strbac, et al., Application of liquid-air and pumped-thermal electricity storage systems in low-carbon electricity systems, in: *Heat Powered Cycles - HPC-2018*, Bayreuth, Germany, 2018, pp. 1–8, <https://spiral.imperial.ac.uk/handle/10044/1/62577>.
- [43] S. Georgiou, N. Shah, C.N. Markides, A thermo-economic analysis and comparison of pumped-thermal and liquid-air electricity storage systems, *Appl. Energy* 226 (2018) 1119–1133, <https://doi.org/10.1016/j.apenergy.2018.04.128>.
- [44] S. Georgiou, M. Aunedi, G. Strbac, et al., On the value of liquid-air and Pumped-Thermal Electricity Storage systems in low-carbon electricity systems, *Energy* 193 (2019) 1119–1133, <https://doi.org/10.1016/j.energy.2019.116680>.
- [45] P. Sapin, M.C. Simpson, A.V. Olympios, et al., Cost-benefit analysis of reversible reciprocating-piston engines with adjustable volume ratio in pumped thermal electricity storage, in: *Proceedings of ECOS 2020 - the 33rd International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems*, Osaka, Japan, 2020, pp. 1–12, in: https://www.researchgate.net/publication/342764994_Cost-benefit_analysis_of_reversible_reciprocating-piston_engines_with_adjustable_volume_ratio_in_pumped_thermal_electricity_storage.
- [46] L. Wang, X.P. Lin, L. Chai, et al., Cyclic transient behavior of the Joule–Brayton based pumped heat electricity storage: modeling and analysis, *Renew. Sustain. Energy Rev.* 111 (2019) 523–534, <https://doi.org/10.1016/j.rser.2019.03.056>.
- [47] L. Wang, X.P. Lin, L. Chai, et al., Unbalanced mass flow rate of packed bed thermal energy storage and its influence on the Joule–Brayton based pumped thermal electricity storage, *Energy Convers. Manag.* 185 (2019) 593–602, <https://doi.org/10.1016/j.enconman.2019.02.022>.
- [48] H. Zhang, L. Wang, X. Lin, et al., Combined cooling, heating, and power generation performance of pumped thermal electricity storage system based on Brayton cycle, *Appl. Energy* 278 (2020) 115607, <https://doi.org/10.1016/j.apenergy.2020.115607>.
- [49] W.D. Steinmann, H. Jockenhöfer, D. Bauer, Thermodynamic analysis of high-temperature Carnot battery concepts, *Energy Technol.* 8 (3) (2019) 1900895, <https://doi.org/10.1002/ente.201900895>.
- [50] A. Dietrich, Doctoral dissertation, in: *Assessment of Pumped Heat Electricity Storage Systems through Exergoeconomic Analyses*, Department of Mechanical Engineering, Technische Universität Darmstadt, 2017, <https://tuprints.ulb-tu-darmstadt.de/id/eprint/6752>.
- [51] A. Dietrich, F. Dammal, P. Stephan, Exergoeconomic analysis of a pumped heat electricity storage system based on a Joule/Brayton cycle, *Energy Sci. Eng.* 9 (2020) 645–660, <https://doi.org/10.1002/ese3.850>.
- [52] R.B. Laughlin, Pumped thermal grid storage with heat exchange, *J. Renew. Sustain. Energy* 9 (4) (2017), 044103, <https://doi.org/10.1063/1.4994054>.
- [53] P. Farres-Antunez, H.B. Xue, A.J. White, Thermodynamic analysis and optimisation of a combined liquid air and pumped thermal energy storage cycle, *J. Energy Storage* 18 (2018) 90–102, <https://doi.org/10.1016/j.est.2018.04.016>.
- [54] P. Farres-Antunez, J.D. McTigue, A.J. White, A pumped thermal energy storage cycle with capacity for concentrated solar power integration, in: *2019 Offshore Energy and Storage Summit (OSES)*, BREST, France, 2019, pp. 1–10, <https://doi.org/10.1109/OSES.2019.8867222>.
- [55] P. Farres-Antunez, A.J. White, Effect of heat capacity variation on high-performance heat exchangers for thermo-mechanical energy storage, in: *Offshore Energy and Storage Symposium*, Valetta, Malta, 2016, pp. 1–13, <https://www.researchgate.net/publication/339883115>.
- [56] P. Farres-Antunez, Doctoral dissertation, in: *Modelling and Development of Thermo-Mechanical Energy Storage*, Department of Engineering, University of Cambridge, 2018, <https://doi.org/10.17863/CAM.38056>.
- [57] P. Farres-Antunez, A.J. White, Thermodynamic strategies for pumped thermal exergy storage (PTES) with liquid reservoirs, in: *UK Energy Storage Conference*, 2016, pp. 1–46, <https://doi.org/10.13140/RG.2.2.29880.19207>, Birmingham, UK.
- [58] H.B. Xue, A comparative study of the adiabatic compressed air energy storage (A-CAES) and pumped thermal energy storage (PTES) systems, in: *2019 Offshore Energy and Storage Summit (OSES)*, BREST, France, 2019, pp. 1–10, <https://ieeexplore.ieee.org/abstract/document/8867354>.
- [59] H.B. Xue, A Comparative Analysis and Optimisation of Thermo-Mechanical Energy Storage Technologies (Doctoral Dissertation), Department of Engineering, University of Cambridge, 2018, <https://doi.org/10.17863/CAM.37776>.
- [60] D. Salomone-González, J. González-Ayala, A. Medina, et al., Pumped heat energy storage with liquid media: thermodynamic assessment by a Brayton-like model, *Energy Convers. Manag.* 226 (2020) 113540, <https://doi.org/10.1016/j.enconman.2020.113540>.
- [61] Y.L. Zhao, M. Liu, J. Song, et al., Advanced exergy analysis of a Joule–Brayton pumped thermal electricity storage system with liquid-phase storage, *Energy Convers. Manag.* 231 (2021) 113867, <https://doi.org/10.1016/j.enconman.2021.113867>.
- [62] J.D. McTigue, P. Farres-Antunez, K. Ellingwood, et al., Pumped thermal electricity storage with supercritical CO₂ cycles and solar heat input, in: *2019 Solar Power and Chemical Energy Systems Conference*, Daegu, South Korea, 2019, p. 190024, <https://www.osti.gov/biblio/1570183>.
- [63] M. Mercangöz, J. Hemrle, L. Kaufmann, et al., Electrothermal energy storage with transcritical CO₂ cycles, *Energy* 45 (1) (2012) 407–415, <https://doi.org/10.1016/j.energy.2012.03.013>.
- [64] S.A. Wright, A. Z'Graggen, J. Hemrle, Control of a supercritical CO₂ electro-thermal energy storage system, in: *ASME Turbo Expo 2013: Turbine Technical Conference and Exposition*, 2013, pp. 1–9, <https://doi.org/10.1115/gt2013-95326>, San Antonio, Texas, USA.
- [65] M. Morandin, F. Maréchal, M. Mercangöz, et al., Conceptual design of a thermo-electrical energy storage system based on heat integration of thermodynamic cycles – Part A: methodology and base case, *Energy* 45 (1) (2012) 375–385, <https://doi.org/10.1016/j.energy.2012.03.031>.
- [66] M. Morandin, F. Maréchal, M. Mercangöz, et al., Conceptual design of a thermo-electrical energy storage system based on heat integration of thermodynamic cycles – Part B: alternative system configurations, *Energy* 45 (1) (2012) 386–396, <https://doi.org/10.1016/j.energy.2012.03.033>.
- [67] M. Morandin, M. Mercangöz, J. Hemrle, et al., Thermoeconomic design optimization of a thermo-electric energy storage system based on transcritical CO₂ cycles, *Energy* 58 (2013) 571–587, <https://doi.org/10.1016/j.energy.2013.05.038>.
- [68] Y.-M. Kim, D.-G. Shin, S.-Y. Lee, et al., Isothermal transcritical CO₂ cycles with TES (thermal energy storage) for electricity storage, *Energy* 49 (2013) 484–501, <https://doi.org/10.1016/j.energy.2012.09.057>.
- [69] Y.-J. Baik, J. Heo, J. Koo, et al., The effect of storage temperature on the performance of a thermo-electric energy storage using a transcritical CO₂ cycle, *Energy* 75 (2014) 204–215, <https://doi.org/10.1016/j.energy.2014.07.048>.
- [70] F. Ayachi, N. Tauveron, T. Tartiére, et al., Thermo-electric energy storage involving CO₂ transcritical cycles and ground heat storage, *Appl. Therm. Eng.* 108 (2016) 1418–1428, <https://doi.org/10.1016/j.applthermaleng.2016.07.063>.
- [71] N. Tauveron, E. Macchi, D. Nguyen, et al., Experimental study of supercritical CO₂ heat transfer in a thermo-electric energy storage based on Rankine and heat-pump cycles, *Energy Proc.* 129 (2017) 939–946, <https://doi.org/10.1016/j.egypro.2017.09.121>.
- [72] M. Abarr, B. Geels, J. Hertzberg, et al., Pumped thermal energy storage and bottoming system part A: concept and model, *Energy* 120 (2017) 320–331, <https://doi.org/10.1016/j.energy.2016.11.089>.
- [73] M. Abarr, J. Hertzberg, L.D. Montoya, Pumped thermal energy storage and bottoming system Part B: sensitivity analysis and baseline performance, *Energy* 119 (2017) 601–611, <https://doi.org/10.1016/j.energy.2016.11.028>.
- [74] M.L. Abarr, Modeling Pumped Thermal Energy Storage with Waste Heat Harvesting (Doctoral Dissertation), Mechanical Engineering, University of Colorado Boulder, 2016, https://scholar.colorado.edu/mcen_gradetds/124.
- [75] G.B. Wang, X.R. Zhang, Thermodynamic analysis of a novel pumped thermal energy storage system utilizing ambient thermal energy and LNG cold energy, *Energy Convers. Manag.* 148 (2017) 1248–1264, <https://doi.org/10.1016/j.enconman.2017.06.044>.
- [76] A. Koen, P. Farres-Antunez, A.J. White, A study of working fluids for transcritical pumped thermal energy storage cycles, in: *2019 Offshore Energy and Storage Summit (OSES)*, BREST, France, 2019, pp. 1–7, <https://doi.org/10.1109/OSES.2019.8867217>.
- [77] W.D. Steinmann, The CHEST (Compressed Heat Energy STORAGE) concept for facility scale thermo mechanical energy storage, *Energy* 69 (2014) 543–552, <https://doi.org/10.1016/j.energy.2014.03.049>.
- [78] W.D. Steinmann, D. Bauer, H. Jockenhöfer, et al., Pumped thermal energy storage (PTES) as smart sector-coupling technology for heat and electricity, *Energy* 183 (2019) 185–190, <https://doi.org/10.1016/j.energy.2019.06.058>.
- [79] H. Jockenhöfer, W.D. Steinmann, D. Bauer, Detailed numerical investigation of a pumped thermal energy storage with low temperature heat integration, *Energy* 145 (2018) 665–676, <https://doi.org/10.1016/j.energy.2017.12.087>.
- [80] A. Vandersickela, A. Aboueldhab, H. Spliethoff, Small-scale pumped heat electricity storage for decentralised combined heat and power generation: cost optimal design and operation, in: *Proceedings of ECOS 2016-the 29th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems*, Portorož, Slovenia, 2016, pp. 1–13, <https://pdfs.semanticscholar.org/eafe/9833b61ba92b44f4a6756bf52347b6d83acb.pdf>.
- [81] P. Vinnemeier, M. Wirsum, D. Malpiece, et al., Integration of heat pumps into thermal plants for creation of large-scale electricity storage capacities, *Appl. Energy* 184 (2016) 506–522, <https://doi.org/10.1016/j.apenergy.2016.10.045>.
- [82] P. Vinnemeier, M. Wirsum, D. Malpiece, et al., Integration of pumped-heat-electricity-storage into water/steam cycles of thermal power plants, in: *5th International Symposium-Supercritical CO₂ Power Cycles*, 2016, San Antonio, Texas, San Antonio, Texas, USA, <http://sco2symposium.com/papers2016/HeatRejection/082paper.pdf>.
- [83] D. Roskosch, V. Venzik, B. Atakan, Potential analysis of pumped heat electricity storages regarding thermodynamic efficiency, *Renew. Energy* 147 (2020) 2865–2873, <https://doi.org/10.1016/j.renene.2018.09.023>.
- [84] D. Roskosch, B. Atakan, Pumped heat electricity storage: potential analysis and orc requirements, *Energy Proc.* 129 (2017) 1026–1033, <https://doi.org/10.1016/j.egypro.2017.09.235>.
- [85] S. Staub, P. Bazan, K. Braimakis, et al., Reversible heat pump–organic Rankine

- cycle systems for the storage of renewable electricity, *Energies* 11 (6) (2018) 1–17, <https://doi.org/10.3390/en11061352>.
- [86] O. Dumont, R. Dickes, M. Ismael, et al., Mapping of performance of pumped thermal energy storage (Carnot battery) using waste heat recovery, in: ORC 2019 Proceedings, 2019, pp. 1–9. Athens, Greece, https://orbi.uliege.be/bitstream/2268/239271/1/orc_ptes_review.pdf.
- [87] O. Dumont, G.F. Frate, A. Pillai, et al., Carnot battery technology: a state-of-the-art review, *J. Energy Storage* 32 (2020) 101756, <https://doi.org/10.1016/j.est.2020.101756>.
- [88] B. Eppinger, L. Zigan, J. Karl, et al., Pumped thermal energy storage with heat pump-ORC-systems: comparison of latent and sensible thermal storages for various fluids, *Appl. Energy* 280 (2020) 115940, <https://doi.org/10.1016/j.apenergy.2020.115940>.
- [89] A.H. Hassan, L. O'Donoghue, V. Sánchez-Canales, et al., Thermodynamic analysis of high-temperature pumped thermal energy storage systems: refrigerant selection, performance and limitations, *Energy Rep.* 6 (2020) 147–159, <https://doi.org/10.1016/j.egy.2020.05.010>.
- [90] V. Sánchez-Canales, A.H. Hassan, J.M. Corberán, et al., Excess electricity storage via thermal energy storage, in: Eurotherm Seminar (Advances in Thermal Energy Storage), Lleida, Spain, 2019, pp. 1–12. https://www.researchgate.net/publication/333402646_Excess_electricity_storage_via_thermal_energy_storage.
- [91] G.F. Frate, M. Antonelli, U. Desideri, A novel pumped thermal electricity storage (PTES) system with thermal integration, *Appl. Therm. Eng.* 121 (2017) 1051–1058, <https://doi.org/10.1016/j.applthermaleng.2017.04.127>.
- [92] G.F. Frate, L. Ferrari, U. Desideri, Multi-criteria investigation of a pumped thermal electricity storage (PTES) system with thermal integration and sensible heat storage, *Energy Convers. Manag.* 208 (2020) 112530, <https://doi.org/10.1016/j.enconman.2020.112530>.
- [93] G.F. Frate, L. Ferrari, U. Desideri, Multi-criteria economic analysis of a pumped thermal electricity storage (PTES) with thermal integration, *Front. Energy Res.* 8 (2020) 1–19, <https://doi.org/10.3389/fenrg.2020.00053>.
- [94] G.F. Frate, L. Ferrari, U. Desideri, Rankine Carnot batteries with the integration of thermal energy sources: a review, *Energies* 13 (18) (2020) 1–28, <https://doi.org/10.3390/en13184766>.
- [95] T. Roskilly, Pumped heat energy storage. https://d2umxnkyjne36n.cloudfront.net/documents/10Yol_ESD_TonyRoskilly_NewcastleUni.pdf?mtime=20171124104241, 2017.
- [96] J.D. McTigue, “Carnot batteries” for electricity storage. <https://www.nrel.gov/docs/fy20osti/75559.pdf>, 2019.
- [97] A. White, J. McTigue, C. Markides, Wave propagation and thermodynamic losses in packed-bed thermal reservoirs for energy storage, *Appl. Energy* 130 (2014) 648–657, <https://doi.org/10.1016/j.apenergy.2014.02.071>.
- [98] A.J. White, J.D. McTigue, C.N. Markides, Analysis and optimisation of packed-bed thermal reservoirs for electricity storage applications, in: Proceedings of the Institution of Mechanical Engineers Part A-Journal of Power and Energy, 2016, pp. 739–754, <https://doi.org/10.1177/0957650916668447>.
- [99] J.D. McTigue, C.N. Markides, A.J. White, Performance response of packed-bed thermal storage to cycle duration perturbations, *J. Energy Storage* 19 (2018) 379–392, <https://doi.org/10.1016/j.est.2018.08.016>.
- [100] J. Song, Y. Song, C.-w. Gu, Thermodynamic analysis and performance optimization of an Organic Rankine Cycle (ORC) waste heat recovery system for marine diesel engines, *Energy* 82 (2015) 976–985, <https://doi.org/10.1016/j.energy.2015.01.108>.
- [101] J. Song, P. Loo, J. Teo, et al., Thermo-economic optimization of organic Rankine cycle (ORC) systems for geothermal power generation: a comparative study of system configurations, *Front. Energy Res.* 8 (2020), <https://doi.org/10.3389/fenrg.2020.00006>.
- [102] Y. Wang, J. Song, M.A. Chatzopoulou, et al., A holistic thermoeconomic assessment of small-scale, distributed solar organic Rankine cycle (ORC) systems: comprehensive comparison of configurations, component and working fluid selection, *Energy Convers. Manag.* 248 (2021), <https://doi.org/10.1016/j.enconman.2021.114618>.
- [103] O. Dumont, V. Lemort, Mapping of performance of pumped thermal energy storage (Carnot battery) using waste heat recovery, *Energy* 211 (2020) 118963, <https://doi.org/10.1016/j.energy.2020.118963>.
- [104] A. Gil, M. Medrano, I. Martorell, et al., State of the art on high temperature thermal energy storage for power generation. Part 1—concepts, materials and modelling, *Renew. Sustain. Energy Rev.* 14 (1) (2010) 31–55, <https://doi.org/10.1016/j.rser.2009.07.035>.
- [105] S. Kuravi, J. Trahan, D.Y. Goswami, et al., Thermal energy storage technologies and systems for concentrating solar power plants, *Prog. Energy Combust. Sci.* 39 (4) (2013) 285–319, <https://doi.org/10.1016/j.peccs.2013.02.001>.
- [106] Y. Zhao, C.Y. Zhao, C.N. Markides, et al., Medium- and high-temperature latent and thermochemical heat storage using metals and metallic compounds as heat storage media: a technical review, *Appl. Energy* 280 (2020), <https://doi.org/10.1016/j.apenergy.2020.115950>.
- [107] Y. Filali Baba, H. Ajdad, A.A.L. Mers, et al., Multilevel comparison between magnetite and quartzite as thermocline energy storage materials, *Appl. Therm. Eng.* 149 (2019) 1142–1153, <https://doi.org/10.1016/j.applthermaleng.2018.12.002>.
- [108] A. Bouguila, R. Said, Performance investigation of a 100-kW_{th} thermocline packed bed thermal energy storage system: comparison between synthetic oil and vegetable oil, *Adv. Mech. Eng.* 12 (4) (2020) 1–10, <https://doi.org/10.1177/1687814020905746>.
- [109] S. Arafin, R.N. Singh, Thermal and physical properties of titanium oxide at high pressure, *J. Phys. Chem. Solid.* 99 (2016) 86–91, <https://doi.org/10.1016/j.jpcs.2016.07.015>.
- [110] Y. Grosu, A. Faik, I. Ortega-Fernández, et al., Natural Magnetite for thermal energy storage: excellent thermophysical properties, reversible latent heat transition and controlled thermal conductivity, *Sol. Energy Mater. Sol. Cell.* 161 (2017) 170–176, <https://doi.org/10.1016/j.solmat.2016.12.006>.
- [111] G. Alva, Y.X. Lin, G.Y. Fang, An overview of thermal energy storage systems, *Energy* 144 (2018) 341–378, <https://doi.org/10.1016/j.energy.2017.12.037>.
- [112] P. Hartlieb, M. Toifl, F. Kuchar, et al., Thermo-physical properties of selected hard rocks and their relation to microwave-assisted comminution, *Miner. Eng.* 91 (2016) 34–41, <https://doi.org/10.1016/j.mineng.2015.11.008>.
- [113] Chemical Book. https://www.chemicalbook.com/ProductIndex_EN.aspx, 2020. (Accessed 25 June 2020).
- [114] H. Benoit, L. Spreafico, D. Gauthier, et al., Review of heat transfer fluids in tube-receivers used in concentrating solar thermal systems: properties and heat transfer coefficients, *Renew. Sustain. Energy Rev.* 55 (2016) 298–315, <https://doi.org/10.1016/j.rser.2015.10.059>.
- [115] E.W. Lemmon, M.L. Huber, M.O. McLinden, NIST standard reference database 23: reference fluid thermodynamic and transport properties-REFPROP, version 9.1. <https://www.nist.gov/publications/nist-standard-reference-database-23-reference-fluid-thermodynamic-and-transport>, 2013. (Accessed 7 May 2020).
- [116] A. Modi, C.D. Pérez-Segarra, Thermocline thermal storage systems for concentrated solar power plants: one-dimensional numerical model and comparative analysis, *Sol. Energy* 100 (2014) 84–93, <https://doi.org/10.1016/j.solener.2013.11.033>.
- [117] M. Liu, N.H. Steven Tay, S. Bell, et al., Review on concentrating solar power plants and new developments in high temperature thermal energy storage technologies, *Renew. Sustain. Energy Rev.* 53 (2016) 1411–1432, <https://doi.org/10.1016/j.rser.2015.09.026>.
- [118] X.L. Li, E. Xu, S. Song, et al., Dynamic simulation of two-tank indirect thermal energy storage system with molten salt, *Renew. Energy* 113 (2017) 1311–1319, <https://doi.org/10.1016/j.renene.2017.06.024>.
- [119] I. Sarbu, C. Sebarhievici, A comprehensive review of thermal energy storage, *Sustainability* 10 (2) (2018) 1–32, <https://doi.org/10.3390/su10010191>.
- [120] G. Li, Y. Hwang, R. Radermacher, et al., Review of cold storage materials for subzero applications, *Energy* 51 (2013) 1–17, <https://doi.org/10.1016/j.energy.2012.12.002>.
- [121] R.K. Shah, D.P. Sekulic, *Fundamentals of Heat Exchanger Design*, John Wiley & Sons, New York, 2003.
- [122] T. Kuppan, *Heat Exchanger Design Handbook*, CRC Press, New York, 2013.
- [123] X.Y. Li, J. Song, G.P. Yu, et al., Organic Rankine cycle systems for engine waste-heat recovery: heat exchanger design in space-constrained applications, *Energy Convers. Manag.* 199 (2019) 111968, <https://doi.org/10.1016/j.enconman.2019.111968>.
- [124] T.C. Allison, Thermal-mechanical-chemical energy storage technology overview, in: TMCES Workshop, 2020, pp. 1–21. Pittsburgh, PA, USA, https://www.netl.doe.gov/sites/default/files/2020-02/TMCES/Tim%20Allison%20Energy%20Storage%20Technology%20Overview_TMCES%20Workshop%202020_rev2.pdf.
- [125] T.C. Allison, J. Moore, N.R. Smith, et al., Grid-scale energy storage systems and technologies, in: 2019 Gas/Electric Partnership Conference XXVII, 2019. Houston, Texas, USA, http://www.gaselectricpartnership.com/NEnergy%20Storage_SwRI.pdf.
- [126] J. Song, X.Y. Li, X.D. Ren, et al., Thermodynamic and economic investigations of transcritical CO₂-cycle systems with integrated radial-inflow turbine performance predictions, *Appl. Therm. Eng.* 165 (2019) 114604, <https://doi.org/10.1016/j.applthermaleng.2019.114604>.
- [127] R. Turton, R. Bailie, W. Whiting, et al., *Analysis, Synthesis and Design of Chemical Processes*, Pearson Education International, London (UK), 2013.
- [128] S. Jenkins, Chemical engineering plant cost index annual average. <https://www.chemengonline.com/2019-chemical-engineering-plant-cost-index-annual-average/>, 2019. (Accessed 20 March 2020), 2020.
- [129] W. Seider, J. Seader, D. Lewin, et al., *Product and Design Principles: Synthesis Analysis, and Evaluation*, John Wiley & Sons Inc., New York, 2013.
- [130] J. Couper, W. Penney, J. Fair, et al., *Chemical Process Equipment: Selection and Design*, Butterworth-Heinemann, Elsevier Inc., Burlington, 2010.