Supercritical CO2 Heat Pumps and Power Cycles for Concentrating Solar Power

Preprint

Joshua D. McTigue,[1] Pau Farres-Antunez,[2] Ty Neises[1] and Alexander White[2]

1 National Renewable Energy Laboratory
2 University of Cambridge

Presented at the 26th SolarPACES Conference 2020
September 28 - October 2, 2020

NREL is a national laboratory of the U.S. Department of Energy
Office of Energy Efficiency & Renewable Energy
Operated by the Alliance for Sustainable Energy, LLC

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.

Contract No. DE-AC36-08GO28308

Conference Paper
NREL/CP-5700-77955
December 2020

Supercritical CO2 Heat Pumps and Power Cycles for Concentrating Solar Power

Preprint

Joshua D. McTigue,[1] Pau Farres-Antunez,[2] Ty Neises[1] and Alexander White[2]

1 National Renewable Energy Laboratory
2 University of Cambridge

Suggested Citation
McTigue, Joshua D., Pau Farres-Antunez, Ty Neises and Alexander White. 2020. *Supercritical CO2 Heat Pumps and Power Cycles for Concentrating Solar Power: Preprint*. Golden, CO: National Renewable Energy Laboratory. NREL/CP-5700-77955. https://www.nrel.gov/docs/fy21osti/77955.pdf.

NREL is a national laboratory of the U.S. Department of Energy
Office of Energy Efficiency & Renewable Energy
Operated by the Alliance for Sustainable Energy, LLC

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.

Contract No. DE-AC36-08GO28308

Conference Paper
NREL/CP-5700-77955
December 2020

National Renewable Energy Laboratory
15013 Denver West Parkway
Golden, CO 80401
303-275-3000 • www.nrel.gov

NOTICE

This work was authored in part by the National Renewable Energy Laboratory, operated by Alliance for Sustainable Energy, LLC, for the U.S. Department of Energy (DOE) under Contract No. DE-AC36-08GO28308. Funding provided the U.S. Department of Energy Office of Energy Efficiency and Renewable Energy Solar Energy Technologies Office. The views expressed herein do not necessarily represent the views of the DOE or the U.S. Government. The U.S. Government retains and the publisher, by accepting the article for publication, acknowledges that the U.S. Government retains a nonexclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this work, or allow others to do so, for U.S. Government purposes.

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.

U.S. Department of Energy (DOE) reports produced after 1991 and a growing number of pre-1991 documents are available free via www.OSTI.gov.

Cover Photos by Dennis Schroeder: (clockwise, left to right) NREL 51934, NREL 45897, NREL 42160, NREL 45891, NREL 48097, NREL 46526.

NREL prints on paper that contains recycled content.

Supercritical CO_2 Heat Pumps and Power Cycles for Concentrating Solar Power

Joshua D. McTigue,[1, a)] Pau Farres-Antunez[2], Ty Neises[1] and Alexander White[2]

[1] National Renewable Energy Laboratory, 15013 Denver West Parkway, Golden, CO, USA, 80401
[2] Cambridge University Engineering Department, Trumpington Street, Cambridge, UK, CB2 1PZ

[a)] Corresponding author: JoshuaDominic.McTigue@nrel.gov

Abstract. Pumped Thermal Energy Storage (PTES) is a promising technology for electricity storage applications. Grid electricity drives a heat pump which moves energy from a cold space to a hot space, thereby creating hot and cold thermal storage. The temperature difference between the storage is later used to drive a heat engine and return electricity to the grid. In this article, supercritical carbon dioxide (sCO_2) is chosen as the working fluid for PTES, and results are compared to 'conventional' systems that use an ideal gas. Molten salts are used for the hot storage which means that a CSP plant with thermal storage and an sCO_2 power cycle could potentially be hybridized with PTES by the addition of a heat pump. This article describes some of the benefits of this combined system which can provide renewable power generation and energy management services. Two methods by which an sCO_2 heat pump can be combined with an sCO_2 power cycle for CSP are described and techno-economic results are presented. Results indicate that these systems can achieve reasonable technical performance, but that costs are currently high.

INTRODUCTION

Pumped Thermal Energy Storage (PTES) is a grid-scale energy storage device that stores electricity in a thermal potential between hot and cold media. PTES has been investigated globally (under a variety of names, such as a Carnot Battery) and is receiving widespread commercial interest. PTES has several advantages compared to other electricity storage devices, including no geographical restrictions, long lifetimes, and the ability to use cheap, abundant, non-toxic materials as the storage media. PTES may use a variety of different power cycles, working fluids, and thermal storage systems. Commonly discussed concepts include Brayton cycles with packed beds or concrete storage [1–3], recuperated Brayton cycles with molten salt storage [4], and transcritical carbon dioxide cycles with liquid and ice storage [5,6].

PTES and Concentrating Solar Power (CSP) systems both use similar components such as high temperature thermal storage and power cycles. This work aims to describe systems that hybridize PTES with CSP; components are shared between the two systems to reduce costs, and the combined system can generate both renewable electricity and provide electricity storage services.

Previous work has described how PTES can be integrated with conventional CSP plants that use a steam Rankine cycle [7], as well as methods of using a heat pump to enhance the power output of low-temperature solar systems [8]. A growing body of work is exploring whether existing fossil fuel generating plants can be retrofitted with high-temperature heat pumps and thermal storage in order to give a 'second lease of life' to existing infrastructure [9]. Thus, the integration of heat pumps and thermal storage with power generation systems is a promising concept for flexible power generation and consumption. At last year's solarPACES conference, supercritical carbon dioxide (sCO_2) PTES cycles for CSP integration were introduced, and these cycles were found to potentially have several advantages compared to PTES using ideal gases, such as large power densities and high round-trip efficiencies [10]. In this article, refinements are made to the computational models that improve the accuracy and detail of results. Technical and economic results of sCO_2-PTES cycles that are integrated with CSP plants are presented.

Conventional PTES using ideal gases is first described to provide a benchmark for comparison. Two methods of integrating PTES with CSP are then introduced. Firstly, a high-temperature heat pump takes low-value electricity from the grid and converts it into hot and cold thermal energy. The hot energy is stored in molten salt thermal storage which is shared with a concentrating solar power plant. The cold energy is stored in low-cost water storage. The stored energy is later discharged through an sCO_2 power cycle which is also shared with the CSP system. The second method "time-shifts" the recompression process in an sCO_2 power cycle to periods of low-value power, and the generated heat is stored. Later, during high-value periods, solar heat is converted to electricity in the sCO_2 power cycle which bypasses the recompressor and uses the stored heat instead, thereby increasing the net work output at valuable times.

PUMPED THERMAL ENERGY STORAGE

PTES takes low-value power off the grid to create a temperature difference between two reservoirs using a heat pump, and later exploits this temperature difference using a heat engine to produce electricity. In this article, a PTES system that uses an ideal-gas recuperated Brayton cycle with two-tank liquid storage is described [4]. A schematic of the charging cycle is shown in Fig. 1. During charge, grid electricity is used to compress gas to high pressure and temperature, states 1→2. The hot gas transfers its energy to a thermal storage media such as a molten salt (2→2b) before entering a recuperator and being cooled to ambient temperature (2b→3). The gas is then expanded (3→4) to its original pressure and cold temperatures. The cold gas next exchanges heat with a cold storage media (4→4b) before being returned to the compressor inlet temperature in the recuperator. The charging process thus creates a cold store and a hot store. Energy is extracted during discharge by reversing the direction of the gas flow. Cold gas is compressed before heat is transferred from the hot store. The hot gas is expanded to generate electricity and is cooled in the cold store.

Increasing the temperature difference between the hot and cold storage leads to higher efficiencies and energy densities [2]. Molten salts have typically been chosen for the hot storage since these fluids have been demonstrated for long-duration large-scale energy storage in CSP plants. However, molten salts have a limited operating temperature range, and therefore the recuperator is introduced into the cycle in order to maximize the temperature difference. The cold storage is typically at sub-ambient temperatures, and suitable liquids include glycol and methanol (the latter has an inherent fire risk, however) [11].

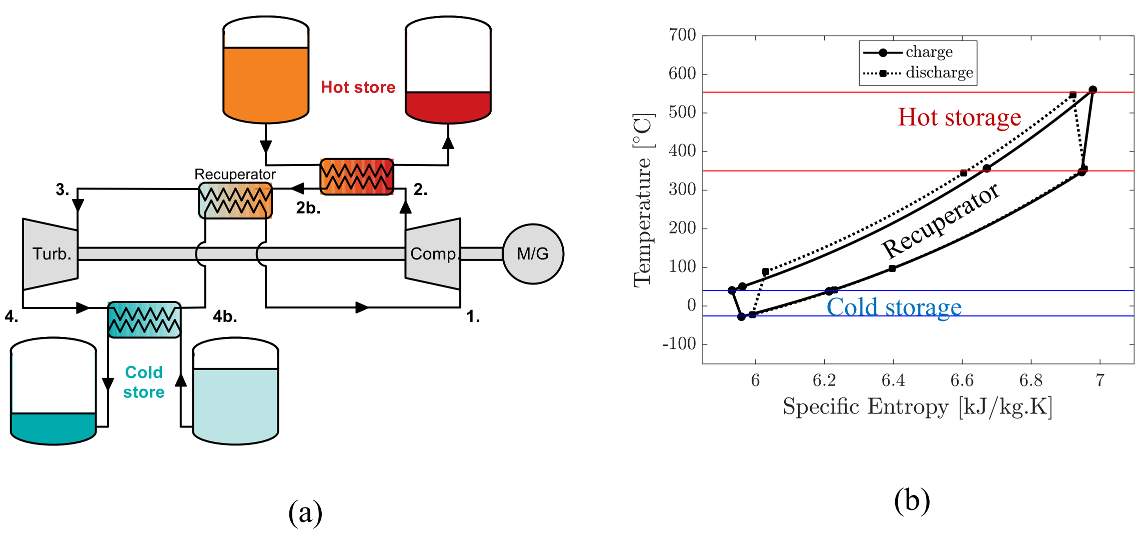

FIGURE 1: (a) Schematic layout of the charging cycle of PTES. (b) Temperature-entropy diagram of the charging and discharging processes.

PTES CYCLES WITH SCO_2 WORKING FLUID

Supercritical carbon dioxide power cycles are considered to have the potential to reach high efficiencies with compact turbomachinery, due to the high density of the working fluid. These power cycles are being considered for the next generation of CSP plants. Integrating PTES with a CSP plant could have several advantages, as the hybrid plant would be able to generate renewable power as well as providing electricity storage services. In this section, PTES cycles with a supercritical-CO_2 working fluid are considered. These systems may either form a stand-alone PTES system, or could be integrated into a CSP plant that uses an sCO_2 power cycle. In the latter case, molten salt storage tanks could be charged either by solar heat or electricity that is used to drive an sCO_2-based heat pump. Thus, the hot storage tanks are shared between the CSP system and the PTES system. Furthermore, the same sCO_2 recompression power cycle is used to convert the stored thermal energy into electricity. Since the CSP and PTES systems share several key components, this hybrid system should require a lower capital investment than two separate systems.

Supercritical-CO_2 PTES cycles and their hybridization with CSP was introduced at the 2019 SolarPACES conference [10], and this paper provided a simplified analysis and considered several key performance indices, such as the round-trip efficiency, the work ratio, and the heat-to-work ratio. The round-trip efficiency, η_{rt} is simply the work recovered during discharge as a fraction of the work input during charge:

$$\eta_{rt} = \frac{W_{\text{dis}}}{W_{\text{chg}}} \tag{1}$$

The work ratio is the ratio between the compressor work input and the expander work output during charge:

$$W_R = \frac{W_{\text{comp}}}{W_{\text{exp}}} \tag{2}$$

The work ratio can be rewritten in terms of the net charging work [12], such that $W_{\text{net}} = (W_R - 1)W_{\text{exp}}$. This expression indicates that high work ratios are preferable: a low work ratio requires more work to be 'processed' to provide the required net work. Thus, high work ratios reduce the sensitivity of a cycle to compression and expansion irreversibilities.

The heat-to-work ratio is a similar metric that quantifies the heat that is processed per unit work input during charge, such that

$$Q_R = \frac{|Q_{\text{in}}| + |Q_{\text{out}}|}{W_{\text{chg}}} \tag{3}$$

Note, that this metric involves all heat exchange processes in and out of the cycle, and also within the cycle (e.g. within recuperators). High heat-to-work ratios imply that large quantities of heat must be exchanged for a given work input, and that the cycle will be more sensitive to heat transfer irreversibilities.

The previous study found that sCO_2-PTES cycles had relatively high work ratios and high heat-to-work ratios compared to 'conventional' ideal-gas PTES. That is, sCO_2-PTES was less sensitive to turbomachinery inefficiencies, but more sensitive to heat exchanger losses. Furthermore, the study found that sCO_2-PTES could potentially achieve very high round-trip efficiencies but that this was contingent on minimizing temperature differences within the heat exchangers.

The thermodynamic models have been substantially improved, and updated results are presented in this section. The previous study modelled turbomachinery with isentropic efficiencies and heat exchangers with a fixed temperature difference between the two fluids. This second assumption limited the accuracy of the study since the properties of sCO_2 can vary significantly over a given temperature range. The improved models now define heat exchanger performance in terms of an effectiveness ε and a pressure loss fraction f_p. The variable properties of the working fluid and storage fluids are included and the model calculates the temperature-heat profile in the heat exchanger such that the required effectiveness is obtained. The turbomachinery is modelled using polytropic efficiencies η. Other updates to the model include the calculation of parasitic losses that arise due to liquid pumps, air fans, motors and generators.

The economic performance is also evaluated. Several capital cost correlations for each component were gathered from a variety of sources, including [13–19]. The levelized cost of storage (LCOS) is calculated using the fixed charge rate (FCR) method [20], where the LCOS is defined as

$$\text{LCOS} = \frac{\text{FCR} \cdot \text{C}_{\text{cap}} + \text{O\&M} + E_{\text{price}} W_{\text{in}}}{W_{\text{out}}} \tag{4}$$

Where FCR is the fixed charge rate, C_{cap} is the capital cost, O&M is the annual operations and maintenance cost, E_{price} is the electricity price, W_{in} is the annual electricity into the system, and W_{out} is the annual electricity delivered by the system. The capital cost, LCOS and the uncertainty of these terms are evaluated using a Monte Carlo approach, whereby the capital cost is calculated thousands of times using a random set of suitable cost correlations. A probability distribution of costs is then obtained, from which statistics such as the mean and standard deviation (or confidence intervals) can be found.

This techno-economic computational model will be fully documented in a forthcoming journal article.

Numerous configurations for sCO_2-PTES exist, which can employ different combinations of recuperators or multiple storage tanks [10]. In this article, the system is intended to be integrated with a CSP plant that uses an sCO_2 recompression cycle as the heat engine and nitrate molten salts for the hot storage. The charging cycle is therefore configured to share the storage components and recuperators in an effort to reduce the cost of the system. The temperature-entropy diagram is illustrated in Fig. 2 and the charging cycle follows the same process as the ideal-gas PTES cycle: the only difference is that two recuperators are used rather than one. Since the expander inlet is close to the carbon dioxide critical point, the minimum temperature of the cycle is higher than in ideal-gas PTES, and as a result a cheap cold storage fluid, such as water, can be used.

During discharge, the thermal energy in the hot storage is discharged through an sCO_2-recompression cycle. Rather than rejecting heat to the environment, waste heat is instead transferred into the cold storage, therefore completing the PTES cycle. Using cold storage as the heat sink has the advantage that the temperature is more stable (and often lower) than the ambient temperature, and that pumping the liquid storage fluid incurs lower parasitic losses than air fans in an air-cooled heat rejection system.

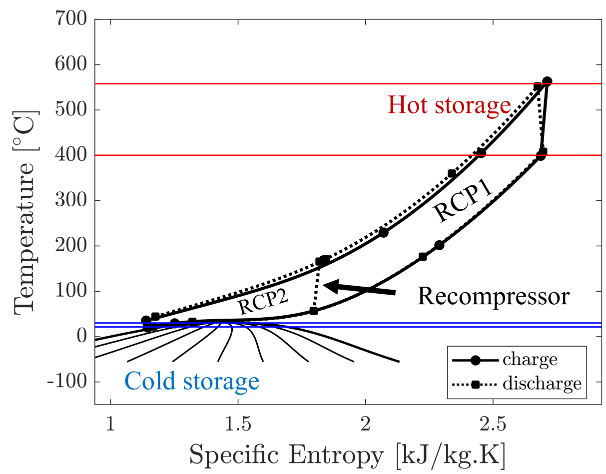

FIGURE 2: Temperature-entropy diagram of sCO2-PTES which uses an sCO2¬ recompression cycle during discharge.

TABLE 1: Assumptions made for nominal PTES designs

Assumptions		
Polytropic efficiency, η	%	90.0
Pressure loss, f_p	%	1.0
Effectiveness, ε	-	0.97
Power output	MW_e	100
Storage duration	h	10

TABLE 2: Techno-economic results for ideal-gas PTES and sCO_2-PTES

Performance		**Ideal-gas PTES**	**sCO_2-PTES**
T_{max}	°C	570.0	563.0
T_{min}	°C	-53.7	20.0
Pressure ratio	-	3.6	3.1
Maximum pressure	bar	25.0	250.0
Hot storage fluid	Molten salt	Nitrate	Nitrate
Cold storage fluid	-	Methanol	Water
Work ratio	-	3.5	11.0
Heat-to-work ratio	-	4.5	8.5
Energy density	kWh_e/m^3	16.3	7.6
Coefficient of performance	-	1.3	1.2
Heat engine efficiency	%	43.9	45.4
Round-trip efficiency	%	58.2	52.6
Cap. Cost per energy discharged	$\$/kWh_e$	311.3 ± 86.4	719.0 ± 303.8
LCOS	$\$/kWh_e$	0.14 ± 0.03	0.27 ± 0.10

Design assumptions for a nominal PTES system are shown in Table 1, and corresponding results for an ideal-gas PTES and sCO_2-PTES cycle are shown in Table 2. Note, that a high heat exchanger effectiveness is chosen. Using a high effectiveness is crucial to achieve reasonable round-trip efficiencies, and while this leads to higher capital costs, a better round-trip efficiency typically leads to a lower LCOS.

The results in Table 2 indicate that ideal-gas PTES outperforms sCO_2-PTES in terms of both efficiency and cost. The lower sCO_2-PTES efficiencies are primarily the result of larger losses in the heat exchangers: the ideal gas has an almost constant heat capacity so that temperature differences in the heat exchangers are quite small. On the other hand, the variable heat capacity of sCO_2 leads to pinch points and larger temperature differences in some parts of the heat exchanger, thereby leading to larger losses. The ideal-gas PTES has a larger temperature difference between the hot and cold storage. This has been shown to lead to higher efficiencies [2], as well as larger energy densities which therefore reduces the storage volume and the cost.

The capital cost and LCOS of sCO_2-PTES is nearly double the ideal-gas PTES values. This may be attributed to the lower energy density and the higher costs of developing new technologies for sCO_2 power cycles. However, there may be some scope of cost reductions as these technologies are advanced and commercialized. On the other hand, ideal-gas PTES cycles are based on existing technologies such as gas turbines, and there may be more limited opportunities for cost reductions in these well-developed components.

The costs in Table 2 account for the cost of all components in the PTES cycles and do not consider the additional value that is achieved by sharing several components (discharging turbomachinery, hot storage, recuperators) between the PTES and CSP systems. Further analysis is required to understand the full benefits of 'generation-integrated electricity storage'.

THE "TIME-SHIFTED" RECOMPRESSION SCO_2 POWER CYCLE

A second method of hybridizing PTES concepts sCO_2 power cycles for CSP is presented in this section. Rather than using the heat pump to charge the high temperature molten salt storage, in this concept, the heat pump charges a lower temperature storage.

While sCO_2 recompression cycles can achieve reasonably good efficiencies, it is notable that the requirement for a recompressor has a significant impact on the performance as it consumes roughly 30-40% of the work input. The recompressor could instead be used as the compressor in the charging heat pump of a PTES device. In this case, electricity at low-value periods is used to drive the heat pump and store the medium temperature thermal energy. During periods of high-value electricity, the solar heat in the molten salt storage is discharged through the sCO_2 cycle. However, the recompressor is bypassed so that the additional heat requirement for recuperation is gathered from the medium temperature storage. As a result, the net work output at high-value periods is larger than with a conventional sCO_2 recompression cycle. By changing the time at which the recompressor operates, this cycle has been dubbed the "time-shifted recompression" (TSRC) sCO_2 cycle.

Schematic cycle layouts and a T-s diagram are shown in Fig. 3 and Fig. 4, respectively. The charging heat pump is a closed cycle, therefore an expander and cold storage are also required in addition to the recompressor and medium temperature storage. Unlike the above heat pumps, this cycle is not recuperated. A suitable fluid for the medium temperature storage is mineral oil, while water can be used for the cold storage. The discharging cycle is similar to an sCO_2 recompression cycle, albeit with some important differences. In a recompression cycle, the low-pressure flow splits after the low-temperature recuperator. One fraction goes through the heat rejection system, the pump and the low-temperature recuperator, while the other fraction is re-compressed. The two fractions mix before entering the high-pressure side of the high-temperature recuperator. In the proposed cycle, all the low-pressure fluid goes through pump. The flow then splits with one fraction going through the high-pressure side of the low-temperature recuperator, and the rest being heated by the medium-temperature storage. The flows mix before entering the high-pressure side of the high-temperature recuperator. Another difference is that the flow splits before the heat rejection system. One fraction uses the atmosphere as a heat sink, while the other fraction uses the cold storage. The flows then mix before the pump. As a result, this cycle requires some modifications to the sCO_2 recompression cycle and a larger pump may be necessary.

Additional metrics are introduced here to evaluate the performance of the time-shifted sCO_2 recompression (TSRC-sCO_2) cycle. Comparing the electrical work input during charge to the electrical work output during discharge provides valuable information about the rate at which electricity can be stored and dispatched. However, using the conventional definition of round-trip efficiency (Eq. 1) leads to values greater than 100% due to the solar heat input. An exergetic round-trip efficiency may be defined by considering the maximum work that can be extracted from the solar heat input to the cycle. The exergetic round-trip efficiency $\eta_{rt,x}$ is given by:

$$\eta_{rt,x} = \frac{W_{\text{dis}}}{W_{\text{chg}} + \Delta\text{Ex}_{\text{solar}}} \tag{5}$$

where $\Delta\text{Ex}_{\text{solar}} = \dot{m}\big(h_{in} - h_{out} - T_o(s_{in} - s_{out})\big)\Delta t$ is the maximum work that could be extracted from the solar heat input.

The TSRC-sCO_2 cycle uses grid electricity to charge the hot and cold storage. This electricity consumption should be considered when assessing the net electricity generation of the system. The TSRC-sCO_2 cycle can be compared to the conventional use of solar heat in a heat engine with the net efficiency, which is defined as

$$\eta_{\text{net}} = \frac{W_{\text{dis}} - W_{\text{chg}}}{Q_{\text{solar}}} \tag{6}$$

where Q_{solar} is the solar heat added to the system. The net efficiency effectively compares the storage system to a conventional solar heat engine under the assumption that the value of electricity is always constant. However, the TSRC-sCO_2 system may be able to take advantage of electricity price fluctuations as well as providing electricity storage services.

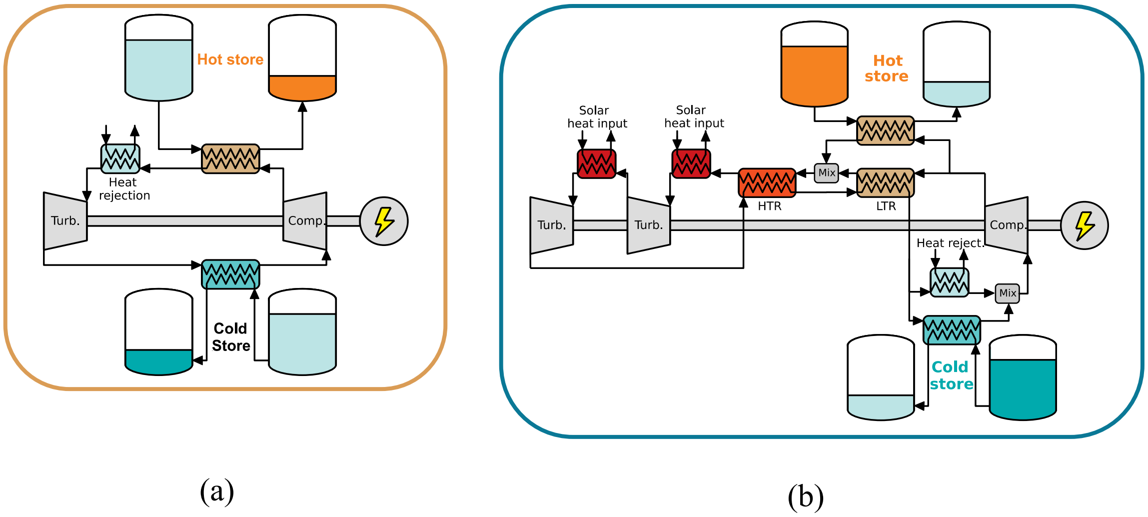

FIGURE 1: Schematic layout of the (a) charging and (b) discharging cycles for a time-shifted recompression sCO_2 power cycle

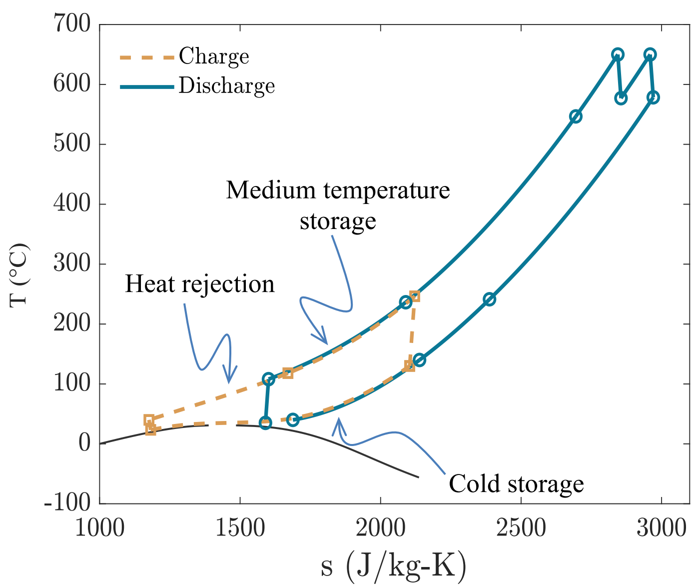

FIGURE 2: Temperature-entropy diagram showing the charge and discharge of a time-shifted recompression sCO_2 cycle. (Note, this diagram shows two stages of expansion whereas the following results are for a single expansion).

Results for a TSRC-sCO_2 cycle are compared to an equivalently sized recompression (RC) sCO_2 cycle in Table 3. These systems use nitrate molten salts for the hot storage and are therefore limited to maximum temperatures of 565°C. By avoiding the recompression during discharge, the power cycle increases the net power output during discharge by nearly 8%, which corresponds to the heat engine efficiency increasing from 45.5% to 49.0%. The TSRC-sCO_2 represents a good use of grid electricity since high values of exergetic round-trip efficiency are achieved (72.9%). On the other hand, consuming grid electricity during charge reduces the net work dispatched to the grid over a single charge-discharge cycle. As a result, the net efficiency is somewhat lower than the conventional RC-sCO_2 cycle. This implies that the TSRC-sCO_2 cycle generates less work per unit of solar heat than the conventional cycle. Therefore, the benefit of providing electricity storage services and being able to take advantage of price fluctuations should be considered.

Economic results are also presented in Table 3. These values assume that a CSP plant has already been installed with an RC-sCO_2 power cycle and molten salt storage. Therefore, the cost only includes the cost of the medium temperature and cold storage, the charging expander, a heat rejection unit, a motor, and additional pumping during discharge. It is assumed that solar heat is stored at the same time that the charging heat pump operates: if it is economically preferable to store electricity, then it is unlikely that the discharging power cycle would dispatch electricity to the grid. As a result, the solar heat in the molten salt storage is assumed to be 'free' and does not incur a cost. Thus, the energy output from the system (used in the LCOS calculation) considers all the electricity generated during discharge. Consequently, the TSRC-sCO_2 cycle achieves very low *energy storage* costs and this system appears to be competitive with other electricity storage systems.

A thorough economic assessment should also consider the cost of the CSP system, sCO_2 power block, and molten salt thermal storage, as well incorporating the renewable electricity dispatched to the grid. Calculating the *value* that this combined generation and storage system provides to the grid would provide a more meaningful evaluation of its potential than just considering the cost of the components.

TABLE 3: Results comparing the performance of a conventional sCO_2 recompression cycle with a time-shifted recompression sCO_2 cycle

		sCO2	TSRC
Charging power input, $\dot{W}_{\mathrm{net}}^{\mathrm{chg}}$	kJ/kg	-	15.8
Discharging power output, $\dot{W}_{\mathrm{net}}^{\mathrm{dis}}$	kJ/kg	100.0	107.8
Heat engine efficiency, η_{HE}	%	45.5	49.0
Exergetic roundtrip efficiency, $\eta_{rt,x}$	%	-	72.9
Net efficiency, η_{net}	%	45.5	41.8
Capital cost per unit energy discharged	$/kWh$_e$	-	100.8±30
LCOS	$/kWh$_e$	-	0.032±0.011

CONCLUSIONS

In this article, Pumped Thermal Energy Storage (PTES) based on supercritical carbon dioxide (sCO_2) Brayton cycles are described. Previously reported results have been updated with an improved modelling methodology that provides a techno-economic assessment of these devices. For the nominal cases studies, sCO_2-PTES is found to have a lower round-trip efficiency (52.6%) and higher levelized cost of storage (LCOS = 0.27 ± 0.10 $/kWh$_e$) than PTES using an ideal gas which achieves 58.2% and 0.14 ± 0.03 $/kWh$_e$, respectively. The lower efficiencies are predominantly the result of large losses in the heat exchangers due to the variable thermal properties of sCO_2. The sCO_2-PTES cost may reduce as sCO_2 technologies are advanced and commercialized. This study did not conduct an exhaustive design optimization investigation, and further improvements to PTES performance may be obtainable.

SCO_2-PTES systems have several components in common with next generation Concentrating Solar Power (CSP) plants, namely molten salt thermal storage and an sCO_2 power cycle. A hybrid plant that combines PTES with CSP may be able to provide several value streams at lower cost than separate systems. Two methods of hybridizing CSP with PTES are described. The first involves using an sCO_2 heat pump to charge the molten salt storage of a CSP plant. The second is known as a "Time-Shifted Recompression" sCO_2 power cycle. In this concept a heat pump replaces the recompressor in a conventional sCO_2 recompression power cycle. This system is found to have a favorable thermodynamic performance with an *exergetic* round-trip efficiency of 72.9% and a low LCOS of 0.032±0.011 $/kWh$_e$ – although this value can only be obtained under an optimistic set of assumptions.

ACKNOWLEDGEMENTS

This work was authored in part by the National Renewable Energy Laboratory, operated by Alliance for Sustainable Energy, LLC, for the U.S. Department of Energy (DOE) under Contract No. DE-AC36-08GO28308. Funding provided by U.S. Department of Energy Office of Energy Efficiency and Renewable Energy Solar Energy Technologies Office. The views expressed in the article do not necessarily represent the views of the DOE or the U.S. Government. The U.S. Government retains and the publisher, by accepting the article for publication, acknowledges that the U.S. Government retains a nonexclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this work, or allow others to do so, for U.S. Government purposes.

This work is supported in part under the EPSRC funded Generation Integrated Energy Storage project, number EP/P023320/1.

REFERENCES

[1] T. Desrues, J. Ruer, P. Marty, J.F. Fourmigué, A thermal energy storage process for large scale electric applications, Appl. Therm. Eng. 30 (2010) 425–432. doi:10.1016/j.applthermaleng.2009.10.002.

[2] A. White, G. Parks, C.N. Markides, Thermodynamic analysis of pumped thermal electricity storage, Appl. Therm. Eng. 53 (2013) 291–298. doi:10.1016/j.applthermaleng.2012.03.030.

[3] J.D. McTigue, A.J. White, C.N. Markides, Parametric studies and optimisation of pumped thermal electricity storage, Appl. Energy. 137 (2015) 800–811. doi:10.1016/j.apenergy.2014.08.039.

[4] R.B. Laughlin, Pumped thermal grid storage with heat exchange, J. Renew. Sustain. Energy. 9 (2017). doi:10.1063/1.4994054.

[5] M. Morandin, F. Maréchal, M. Mercangöz, F. Buchter, Conceptual design of a thermo-electrical energy storage system based on heat integration of thermodynamic cycles - Part B: Alternative system configurations, Energy. 45 (2012) 386–396. doi:10.1016/j.energy.2012.03.033.

[6] M. Morandin, M. Mercangöz, J. Hemrle, F. Maréchal, D. Favrat, Thermoeconomic design optimization of a thermo-electric energy storage system based on transcritical CO2 cycles, Energy. 58 (2013) 571–587. doi:10.1016/j.energy.2013.05.038.

[7] P. Farres-Antunez, J.D. McTigue, A.J. White, A pumped thermal energy storage cycle with capacity for concentrated solar power integration, in: Offshore Energy Storage Conf., Brest, France, 2019.

[8] S. Henchoz, F. Buchter, D. Favrat, M. Morandin, M. Mercangöz, Thermoeconomic analysis of a solar enhanced energy storage concept based on thermodynamic cycles, Energy. 45 (2012) 358–365. doi:10.1016/j.energy.2012.02.010.

[9] M. Geyer, Carnot Batteries for the decarbonization of coal fired power plants using high temperature thermal storage technologies from solar power plants, Int. Work. Carnot Batter. (2018).

[10] J.D. McTigue, P. Farres-Antunez, K. Ellingwood, T. Neises, A.J. White, Pumped Thermal Electricity Storage with Supercritical CO2 Cycles and Solar Heat Input, in: SolarPACES, Daegu, S. Korea, 2019.

[11] P. Farrés-Antúnez, H. 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. doi:10.1016/j.est.2018.04.016.

[12] P. Farrés-Antúnez, Modelling and development of thermo-mechanical energy storage, University of Cambirdge, 2018. doi:https://doi.org/10.17863/CAM.38056.

[13] N.T. Weiland, B.W. Lance, S.R. Pidaparti, SCO2 Power Cycle Component Cost Correlations from DOE Data Spanning Multiple Scales and Applications, Proc. ASME Turbo Expo 2019. (2019).

[14] M.D. Carlson, B.M. Middleton, C.K. Ho, Cycles Using Component Cost Models Baselined With Vendor Data, Proc. ASME 2017 Power Energy Conf. (2017) 1–7.

[15] J. Couper, W.R. Penney, J. Fair, S. Walas, Chemical Process Equipment: Selection and Design, 3rd Editio, 2012.

[16] W. Seider, D. Lewin, J.D. Seader, S. Widagdo, R. Gani, K.M. Ng, Product and Process Ddesign Principles: Synthesis, Analysis, and Evaluation, 4th Editio, Wiley, 2017.

[17] A. Agazzani, A.F. Massardo, A tool for thermoeconomic analysis and optimization of gas, steam and combined plants, ASME 1996 Int. Gas Turbine Aeroengine Congr. Exhib. GT 1996. 3 (1996). doi:10.1115/96-GT-479.

[18] S.G. Hall, S. Ahmad, R. Smith, Capital cost targets for heat exchanger networks comprising mixed materials

of construction, pressure ratings and exchanger types, Comput. Chem. Eng. 14 (1990) 319–335. doi:10.1016/0098-1354(90)87069-2.
[19] M.S. Peters, K.D. Timmerhaus, Plant design and economics for chemical engineers, McGraw-Hill, Inc., 1990.
[20] W. Short, D.J. Packey, A Manual for the Economic Evaluation of Energy Efficiency and Renewable Energy Technologies, NREL Tech. Rep. NREL/TP-462-5173. (1995).