SAND2022-13972C

Placeholder Name of the conference

Placeholder Session title

https://doi.or/10........ DOI placeholder (WILL BE FILLED IN BY TIB Open Publishing)

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Analysis of Different Permutation of Hybrid Concentrated Solar Power (CSP) & Pumped Thermal Energy Storage (PTES) System

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Abstract.

~~Here a~~A hybrid Concentrated Solar Power (CSP) and Pumped Thermal Energy Storage (PTES) system design model is presented here, composed of three main energy storage vessels at different temperatures ~~is being analyzed~~. For this comparison ~~analysis it is taking into consideration~~ two different permutations of the hybrid system will be analyzed. ~~affecting where the~~The variation on these permutations consists of the location of the CSP component, resulting on different ~~determines the~~ temperatures and variations on the sizing of the energy storing components of the system ~~of the storing components and thus the sizing of it~~. ~~The a~~The ~~Analysis is~~ performed analysis ~~for the~~is oriented towards the development of a prototype lab-scale system ~~aim~~ to produce 2 kW$_e$ ~~for a discharge~~over a minimum period of four hours. The assessment of ~~both~~ the two presented system configurations will be executed under steady state operational conditions, as well as using idealized conditions for components. The factors to consider for the evaluation of the system are sizing of the different thermal energy storage containers and efficiency of energy production loop. Leading to an efficiency difference of more than 10% and size increase/reduction six times on space between the analyzed configurations for the hybrid CSP plus PTES system

Keywords: Concentrate Solar Power, Pumped Thermal Energy Storage, Round-Trip Efficiency.

1. Introduction

Pumped Thermal Energy Storage has been considered a promising approach for complementing established energy storing systems [1,2] or by repurposing waste heat from existing systems like natural gas plants [3]. Also, PTES systems have been considered for electricity production systems as standalones, where a heat pump and a heat engine interact via both hot and cold storage to produce electricity using a reciprocating Joule cycle [4]. A hybrid Concentrated Solar Power (CSP) plus PTES system design was developed at the National Solar Thermal Test Facility (NSTTF) at Sandia National Laboratories. Two different system arrangements were considered to determine the optimal pilot-scale demonstration

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configuration.

The system is composed of three thermal storage subsystems:

1. A High Temperature Storage (HTS) implementing solid particles at temperatures above 650℃.
2. A Medium Temperature Storage (MTS) with temperatures ranging from 25℃ to 170℃ or 750℃ depending on the configuration being analyzed.
3. A Low Temperature Storage (LTS) system at 0℃.

Energy stored in the HTS tanks is collected through a CSP particle receiver that allows for thermal storage at temperatures above 650℃. The heat energy stored in the MTS tank is obtained using a CO_2 heat pump through a heat exchanger as shown in Fig. 1. Finally, the LTS takes advantage of the latent heat produced and adsorbed during the liquid-solid phase transition of water. Here coils with CO_2 are immersed in the water tank for the CO_2 to absorb or deposit heat depending on the operation phase. This investigation assesses cycle performance of two thermal energy storage (TES) configurations, specifically for the MTS and HTS, under varying operational modes. The assessment is executed under steady operational conditions for each of the respective operational modes.

~~storagerepurposedwith A hybrid Concentrated Solar Power (CSP) plus Pumped Thermal Energy Storage (PTES) system design is being analyzedwas developed at the National Solar Thermal Test Facility (NSTTF), at Sandia National Laboratories. Two different system arrangements are were being considered to determine the optimal pilot-scale demonstration configuration. The system is composed of three thermal storage subsystems: 1. A High Temperature Storage (HTS) implementing solid particles at temperatures above 650℃. 2. A Medium Temperature Storage (MTS) with temperatures ranging from 25℃ to 170℃ or 750℃ depending on the configuration being analyzed, . Aand 3. A Low Temperature Storage (LTS) system at 0℃. The eEnergy stored in the HTS tanks is collected through a CSP particle receiver that allows for thermal storage at temperatures above 650℃. The Hheat energy stored in the MTS tank is obtained using a CO2 heat pump through a heat exchanger as shown in figure 1. Finally, the LTS will take advantage of the would utilize latent heat produced and adsorbed during the liquid-solid phase transition of water. Here coils with CO2 are submerged into the water tank for the CO2 to absorb or deposit heat depending on the operation phase. This investigation assesses cycle performance of two thermal energy storage (TES) configurations, specifically for the MTS and HTS, under varying operational modes. The assessment will be executed under steady operational conditions for each of the respective operational modes.~~

2. Configurations to Consider

The two permutations being considered for this investigation are shown in Fig.~~ure~~ 1.

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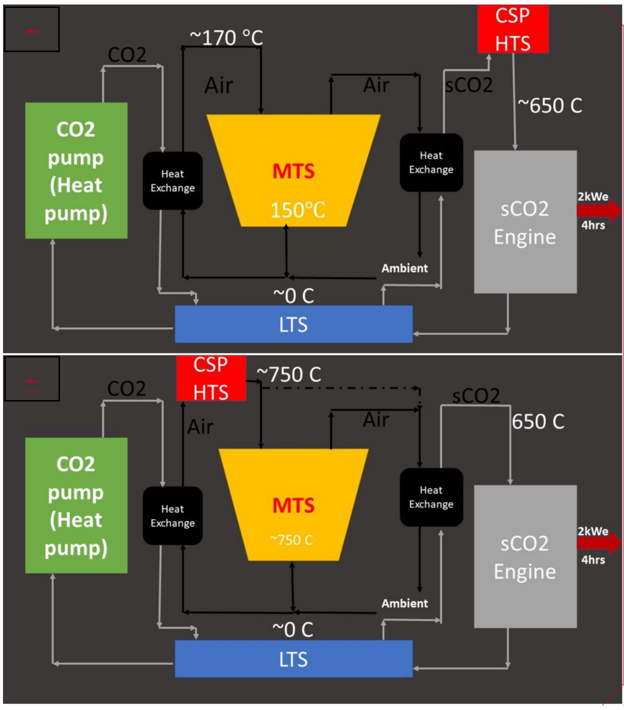

Figure 1: Permutations to be consider for the hybrid PTES+CSP system. Configuration A system located at the top and Configuration B located at the bottom of the figure.

Configuration A uses the HTS tank in direct contact with ~~the~~ a supercritical CO_2 (sCO_2) engine system. Also, for this system design, the MTS tank is heated by the CO_2 pump alone, reaching temperatures of approximately 150℃. Conversely, c~~C~~onfiguration B implements the HTS tank at two different points within the system. Configuration B uses the CO_2 pump and the CSP system connected to the HTS tank to heat up the MTS tank, increasing its temperatures above 650℃. In this second permutation for the CSP/HTS configuration, the heat exchanger for the sCO_2 engine bypasses the MTS tank heating ~~up~~ the air ~~that is~~ in direct contact with the heat exchanger for the sCO_2 engine. This later configuration could allow for longer discharge periods, but for the scope of this ~~paper~~ investigation the bypass will be disregarded ~~on~~ in the operation phases for this~~e~~ system design. This later configuration could allow for longer discharge periods, but for the scope of this investigation the bypass will be disregarded in the operation phases for this system design. There are two main phases we analyze for these systems:

1.Charging Phase.

2.Discharging Phase.~~There are two main phases being analyzed for these systems:~~; ~~p~~Phase 1~~:~~, denominated ~~"charging~~ Charging ~~p~~Phases~~"~~ and, ~~P~~phase 2~~:~~, denominated ~~"d~~Discharging ~~P~~phase.~~"~~

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3. Analysis

During the charging phase, different thermal storage components are thermal-hydraulically charged to ~~desired~~ design temperatures. For the HTS and MTS, heat is added~~,~~ and for LTS heat is extracted (pumped into the MTS), producing ice. Using the Engineering Equations Solver (EES) software [5], a high-level thermodynamic, steady state analysis was performed to compare ~~the~~ theoretical, idealized performance of both systems during charging and discharging.

Using EES, ~~the~~ state thermodynamic properties of the working fluids, air and CO_2~~,~~ were obtained ~~and implemented~~ to develop the idealized models using the following equations:~~.~~

$$\dot{W}_{turbine} = m_x * (H_i - H_f) \tag{1}$$

$$\dot{W}_{pump} = m_x * (H_f - H_i) \tag{2}$$

$$\dot{W}_{compressor} = \dot{m}_x * (H_f - H_i) \tag{3}$$

$$\dot{Q} = m_x * \Delta H \tag{4}$$

~~Where~~ In Eqs. 1 and 2, $\dot{m}_x$ is the the mass flow rate of the working fluid, and H_x represents the enthalpy value of the fluid at the given state conditions. Eqs. 1 and 2 describe the major components of the system that are being considered in this analysis. From Eqs. 1-4, the turbine, pump, compressor, and heat exchangers have efficiencies 88%, 86%, 86%, and 100% respectively~~From Eqns. 1 and 2, $\dot{m}_x$ is the the mass flow rate of the working fluid, and H_x represents the enthalpy value of the fluid at the given state conditions. Using the above shown equations~~~~Eqns. 1 and 2 it is possible to describe the major components of the system that are being considered oin this analysis. Along with Eqns. the equations 1-4 the turbine, pump, compressor, and heat exchangers have efficiencies 88%, 86%, 86%, and 100%, respectively~~.

~~For both system the start point for the analysis is the turbine use to achieve the desired energy production for the system.~~ For both system configurationss, the inlet conditions of the turbine ~~have been~~are assumed to be identical ~~for the two different system~~. The inlet has a temperature of 650 °C at a pressure of 30 MPa, and a 200 °C temperature drop through the turbine has been assumed. ~~Using this information~~ Subsequently, the outlet condition can be determined using the EES real fluids database. To find the thermophysical properties of the CO_2 at the outlet (state two in Fig. 2), the EES database was used. We determined the CO_2 characteristics at this state point considering the temperature drop across the turbine along with the calculated enthalpy (obtained using Eq. 5.c and the assumed isentropic efficiency for the turbine), as the EES data base requires two thermophysical properties to acquire all the desired information~~Subsequently, the outlet condition can be determined using the EES real fluids database. To find the thermoptometries of the CO_2 at the outlet, using the EES data base it is necessary to providedwas used to determine the two state characteristics, in this case, the temperature is obtained from the temperature drop across the turbine, and the second property can be calculated using equation (1) and the assumed isentropic efficiency of the turbine.~~.

$$\eta = \frac{\dot{W}_{pump}}{\dot{W}_{pump_{ideal}}} \tag{5.a}$$

$$\eta = \frac{\dot{m}(h_{in} - h_{out})}{\dot{m}(h_{in} - h_{outideal})} \tag{5.b}$$

$$h_{out} = \eta * (h_{in} - h_{outideal}) \tag{5.c}$$

Then, the enthalpy can be used as an input argument to obtain the rest of the state thermal proprieties of the CO_2 at the turbine outlet. Also, the required mass flow rate can be determine based on using ~~equation~~ Eq. ~~(~~1~~)~~ and rearranging it as follows.

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$$\dot{W}_{turbine} = \dot{m}_x * (H_i - H_f) \tag{1}$$

$$\dot{m}_{sCO_2} = \frac{(H_i - H_f)}{\dot{W}_{turbine}} \tag{6.a}$$

~~Leading to a~~A mass flow rate of 9.916 g/s was determined ~~on~~for the energy production ~~side~~ needed to achieve the target energy production goals of the system design objectives.

3.1. Configuration A

~~This c~~Configuration A for the energy production loop can be observed in ~~figure~~ Fig. 2. ~~Here, as previously mentioned, t~~The CSP component of the system is represented by a particle to sCO_2 heat exchanger at the HTS, getting the sCO_2 to ~~the~~ expected temperatures to enter the turbine and reach ~~the~~ production goals.

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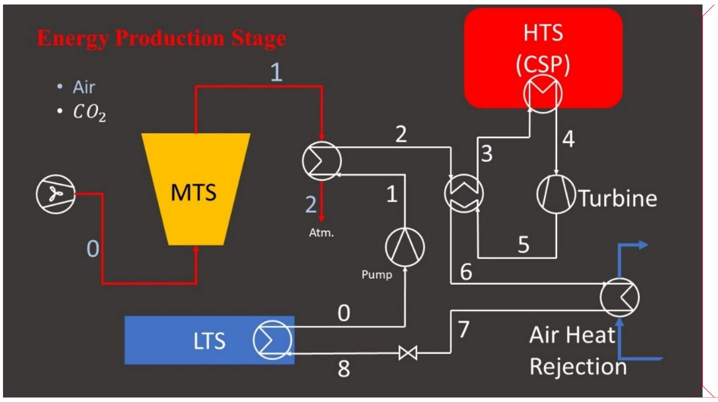

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Figure 2: ~~Energy production for c~~Configuration A process flow diagram~~of the system~~ with marked state points~~ marked~~.

Implementing isentropic efficiencies for the components shown here, along with Eqs. (1-4) leads to the state point thermophysical properties for the energy generation loop, summarized in Table 1.

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~~Implementing the mentioned isentropic efficiencies for the here used components the following along with equations (1-4) the state points thermophysical properties were obtained for the energy generation loop and can be seen in table 1.~~

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Table 1: Thermo-properties of the carbon dioxide at representative points on the energy production stage of the system.

Stage	Temperature ($°C$)	Pressure (kPa)	Entropy $\left(\frac{kJ}{kg \cdot K}\right)$	Specific Volume $\left(\frac{m^3}{kg}\right)$	Enthalpy $\left(\frac{kJ}{kg}\right)$
0	3.305	3800	-1.71	0.001103	-298.6
1	24.62	30000	-1.695	0.001033	-266.3
2	90	30000	-1.299	0.001422	-135.6
3	347	30000	-0.4448	0.003915	264.7
4	650	30000	0.06021	0.006209	649.3
5	450	7106	0.06021	0.01927	420.1
6	100	7106	-0.6975	0.008196	19.82
7	29	7106	-1.432	0.001554	-213.4

8	3.305	3800	-1.402	0.003936	-213.4

Using the ~~above~~ data from Table 1~~information~~, ~~it is possible to calculate~~ the round-trip efficiency was calculated for~~of~~ this loop. There are two ~~heat sources~~heat exchangers (HEX) in thise system, the air to CO_2 ~~heat exchanger~~HEX (between points 1 and 2), and the particles to CO_2 ~~heat exchanger~~HEX (between points 3 and 4). ~~Also~~Additionally, there is a pump (between points 0 to 1) ~~to be consider~~. Taking these components in consideration with an energy production of 2 kW, the efficiency of the idealized energy production ~~loop~~ is calculated, as prescribed in Eqn. 7~~follows~~.

$$\eta_{EP_{loop}} = 100\% * \frac{\dot{W}_{turbine} - \dot{W}_{pump}}{\dot{Q}_{12} + \dot{Q}_{34}} = 32.88\% \tag{7}$$

~~The information data inon tTable 1 can also be used to produce the T-s and P-v diagrams for the system, figures 3 and 4 respectively as shown in Figs. 3 and 4.~~

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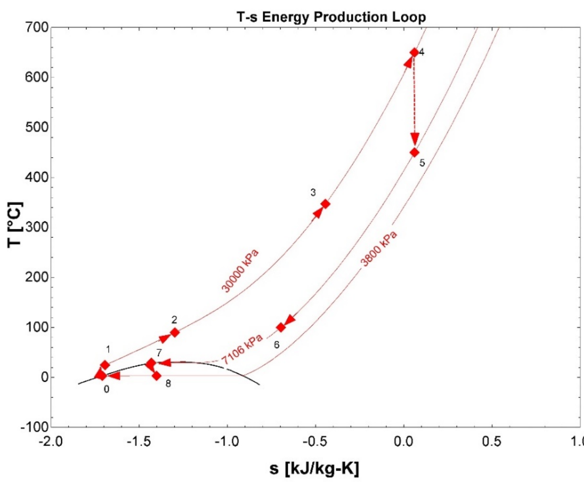

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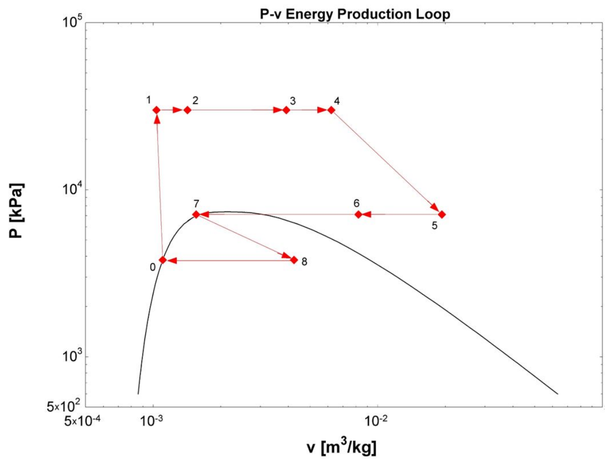
~~**Figure 3: T-s diagram for the energy production loop. Configuration A**~~



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~~**Figure 4: P-v diagram for the energy production loop. Configuration A**~~

3.1.1. Sizing Thermal Energy Storage Components.

~~Another important piece of information that can be obtained from the data shown on~~From Table 1, ~~is~~ the total heat capacity requirements for two thermal energy storage containers can be determined for~~,~~ the LTS and HTS for this configuration. Assuming an idealized 100% heat transfer efficiency from particles to CO_2 (state 3 to state 4, Figure 2) and a 100 °C particle temperature drop through the heat exchanger, the total mass of particles required for this configuration is 366.8 kg. The density of the particles being considered for this component is 2,150 kg/m^3 with an estimated void fraction of 0.2. Thus, ~~the~~ volume for the HTS in this configuration is 0.2133m^3.

Similarly, the required volume for the LTS can be calculated~~,~~ using the latent heat of melting water and the information provided ~~on~~ in Table 1. ~~t~~The total mass of ice ~~is~~was then calculated ~~that~~ for what would be melted during the energy production cycle ~~can be easily calculated~~. The latent heat of ~~melting~~ water is 334 kJ/kg. Assuming 100% efficiency at the heat exchanger (state 0 to state 1, Figure 2), the total amount of ice being melted ~~at~~ during ~~the~~ energy production ~~loop~~ with this configuration is 36.43 kg. ~~Then u~~Using the density of the ice (en~~making~~ suringe only 2/3 of the total LTS volume get melted ~~(to ensure~~while the LTS remains at 0°C at all times) the volume of the LTS ~~comes to be~~is 0.05958 m^3.

~~Lastly, the i~~Information from~~on~~ Table 1 was then~~can be~~ used to ~~indirectly~~ calculate the approximate sizing of the MTS packed bed. Assuming an idealized 100% efficiency at the air to CO_2 heat exchanger (state 1 to state 2, Figure 2), a temperature drop of 70 °C on the air side, and air temperature of 150°C at the ~~i~~inlet of the heat exchanger, ~~it is possible to~~ then determines the required air mass flow rate. For this configuration the air mass flow rate ~~comes to be~~is 18.27 g/s. Using granite as a low-cost~~the~~ material for the packed bed~~,~~ with a specific heat capacity of 0.79 kJ/kg-K and a density of 2691 kg/m^3, along with the previously mentioned temperature range ~~previously mentioned on the introduction section~~, the total mass needed ~~to meet the energy requirements can be calculated. The required mass is~~ was 323.5 kg. ~~of granite, t~~ Then taking into account its density and the thermocline ~~present~~ within~~on~~ the packed bed, the volume of the MTS packed bed ~~comes to be~~was 0.229 m^3.

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3.2. Configuration B

~~This~~ For ~~c~~Configuration B, ~~for~~ the process flow diagram ~~energy production loop~~ can be observed on ~~figure~~ Fig. 5. In this configuration all ~~the~~ heat input for the loop comes from the air to CO_2 ~~heat exchanger~~HEX. ~~Thus~~Here, ~~having~~ the MTS ~~be~~is at a high temperature~~s~~ above ~~the~~ 650 °C.

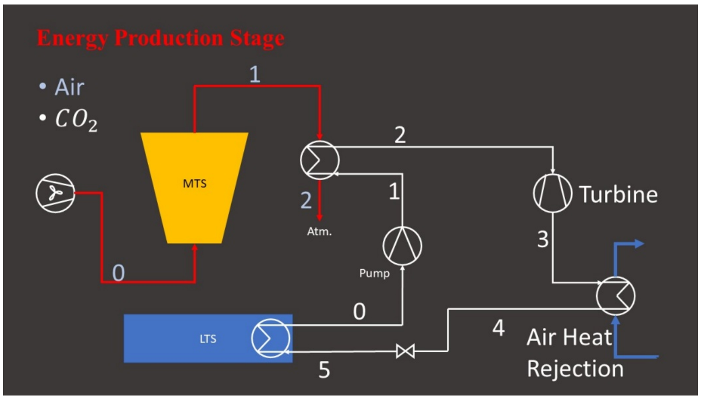

Figure 3: Energy production for configuration B of the system with state points marked.

Following the same approach ~~as that~~ used for configuration A, ~~equations~~ Eqns. (1-4) where used along with the ~~mentioned~~ isentropic efficiencies for the components to obtain the thermophysical properties ~~for the energy production loop.~~, Table 2 ~~contains the obtained information~~.

Table 2: Thermo-properties of the carbon dioxide at ~~representative~~ state points ~~on~~for the energy production stage of the system design.

Stage	Temperature $(°C)$	Pressure (kPa)	Entropy $\left(\frac{kJ}{kg \cdot K}\right)$	Specific Volume $\left(\frac{m^3}{kg}\right)$	Enthalpy $\left(\frac{kJ}{kg}\right)$
0	3.305	3800	-1.71	0.001103	-298.6
1	24.62	30000	-1.695	0.001033	-266.3
2	650	30000	0.06021	0.006209	649.3
3	450	7106	0.06021	0.01927	420.1
4	29	7106	-1.432	0.001554	-213.4
5	3.305	3800	-1.402	0.004236	-213.4

Similar to ~~that approach of~~ system configuration A, ~~the~~ information contained ~~on the above~~ in ~~t~~Table 2 ~~can was~~be utilized to calculate the efficiency of th~~is~~e ~~energy production~~ loop design ~~for the current configuration~~. Unlike configuration A, this configuration obtains all ~~the~~ heat energy ~~that it requires~~ from a single air to CO_2 ~~heat exchanger~~HEX, ~~as~~ and the CSP component is not present on this loop. ~~Thus, t~~The efficiency for ~~a~~this 2-kW energy production loop ~~with this configuration yield the following~~is determined by Eqn. 8:~~.~~

$$\eta_{EP_{loop}} = 100\% * \frac{\dot{W}_{turbine} - \dot{W}_{pump}}{\dot{Q}_{12}} = 18.51\% \quad (8)$$

~~Using the information from Ttable 2, the T-s and P-v diagrams for this configuration were determined and can be observed oin figuresFigs. 6 and 7 respectively.~~

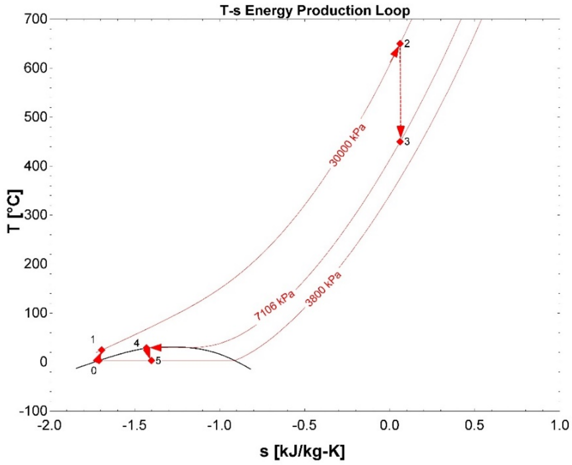

Figure 6: Configuration B T-s diagram for the energy production loop. ~~Configuration B~~

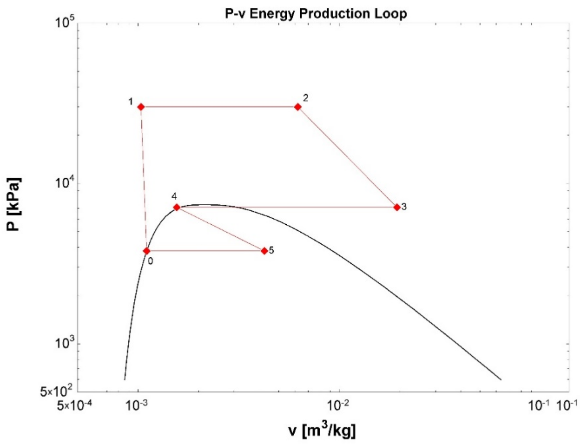

Figure 7: Configuration B P-v diagram ~~for the energy production loop. Configuration B~~

3.2.1. Sizing Thermal Energy Storage Components.

~~Using the information f~~From Table 2 the current configuration (configuration B) only allows for the direct sizing of the LTS component, which can be determined following the same approach shown ~~on~~ in section 3.1.1. Sizing Thermal Energy Storage Components. Similarly to ~~what was done for~~ configuration A, assuming a 100% heat transfer efficiency at the LTS HEX~~heat exchanger with the LTS component~~ (state 0 to state 5, Figure 3) the total ~~amount~~ volume of ice being melted ~~at~~ during the energy production loop with this configuration is 36.43 kg (same as for configuration A).

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~~As f~~For configuration A, the information i~~o~~n Table 2 ~~can be~~is used to indirectly to calculate ~~the~~ sizing requirements for the MTS packed bed. Assuming a 100% efficiency at the air to CO_2 ~~heat exchanger~~ HEX (state 1 to state 2,Figure 1), a temperature drop of 70 °C on the air side, and air temperature of 750°C at the HEX inlet ~~of the heat exchanger it is possible to determine the require air mass flow rate. For this configuration~~ the air mass flow rate was determined~~comes~~ to be 113.9 g/s. Using the same TES material as ~~as~~ for configuration A, the total mass needed to meet the thermal energy requirements for the packed bed on ~~this~~ configuration B is 2219 kg of granite. ~~Then, c~~Considering its density and the thermocline present ~~on~~ in the packed bed, the volume of the MTS packed bed comes to be 1.57 m^3.

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4. Conclusion

The energy production cycle performance of two hybrid Concentrated Solar Power (CSP) and Pumped Thermal Energy Storage (PTES) were assessed in this ~~investigation~~paper. ~~Even w~~With both systems operating at similar input conditions(i.e., efficiency of key components, operating temperature for turbine, and idealized assumptions), configuration A had~~s~~ a ~~better~~~~higher~~14.37% higher thermal to electrical efficiency. However, configuration B offered~~s~~ the advantage of possible longer discharging periods as well as the capabilities of keeping the sCO2 engine operating during the charging phase, while configuration A only allows for energy production during the discharging phase. ~~Non the less~~However, ~~the sizing aspect of the~~this analysis showed~~s~~ that configuration A required~~s~~ less physical space (and possible costs) to achieve the same electric energy production, which is an important aspect to consider when considering ~~scaling~~ scalability of the system to~~e~~ industrial proportions.

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Data availability statement

All data here presented can be accessed through common research sources. Properties of materials here mentioned are of public domain.

~~Please include a statement on how the data supporting the results of your article/contribution can be accessed. If the submission is not based on data or the data it is based on is restricted (third-party data, legal or ethical constraints), this has to be explained in the data availability statement, too. Ideally, data should be deposited in a FAIR-aligned public repository. A registry to find suitable data repositories is re3data.org. Reciprocal linking of data and the article/contribution through persistent identifiers (e.g. DOIs) is best practice.~~

Author contributions

Authors contributions as per CreDIT guidelines for the here presented work are as follows.

Guillermo Anaya: Writing -original draft, Formal Analysis, Investigation, Methodology, Data curation, Software, and Visualization.

Kenneth Armijo: Conceptualization, Resources, Software, Supervision, Funding acquisition, Project administration, and Writing – review and editing.

David Wait: Conceptualization, Resources, Software, Funding acquisition, Project administration.

Aaron Overacker: Formal Analysis, Investigation, and Writing – review and editing.

Dimitri Madden: Formal Analysis, Investigation, and Writing – review and editing.

Ansel Blumenthal: Formal Analysis, Investigation, and Writing – review and editing.

Peter Vorobieff: Writing – review and editing.

Mohan Gowtham: Writing – review and editing.

Competing interests

The authors declare no competing interests.

Funding

This work is funded in part or whole by the U.S. Department of Energy Solar Energy Technologies Office under DOE-1697-1544.

Sandia National Laboratories is a multi-mission laboratory managed and operated by National Technology & Engineering Solutions of Sandia, LLC, a wholly owned subsidiary of Honeywell International Inc., for the U.S. Department of Energy's National Nuclear Security Administration under contract DE-NA0003525.

Acknowledgement

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Special acknowledgement to the NSTTF Sandia team that collaborated with the analysis of the work here presented. Also, special acknowledgement to the CHRES organization for supporting the first author of this paper through his grad studies.

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