The 9th International Supercritical CO_2 Energy Technologies Symposium
March 2 - 5, 2026, Pittsburgh, PA
Paper 41

Evaluation of Off-Design Performance of sCO2 Pumped Thermal Energy Storage System

Diana Jones
Echogen Power Systems
365 Water St, Akron, OH 44308

Ladislav Vesely
Echogen Power Systems
365 Water St, Akron, OH 44308

Jason Mallinak
Echogen Power Systems
365 Water St, Akron, OH 44308

Timothy Held
Echogen Power Systems
365 Water St, Akron, OH 44308

Jason Miller
Echogen Power Systems
365 Water St, Akron, OH 44308

Bios

Diana Jones is a systems engineer at Echogen Power Systems. She earned her undergraduate degree in physics from John Carroll University and went on to complete a master's degree in Entrepreneurial Physics at Case Western Reserve University. Diana has worked at three startup companies since graduating with her bachelor's and has developed a strong appreciation for the fast-paced, collaborative culture and the excitement of building innovative technologies from the ground up.

Ladislav Vesely works as a senior system engineer at Echogen Power Systems. He received his Ph.D. in 2018 from Czech Technical University where he investigated the effect of various mixtures on the performance of sCO_2 power cycles. Since receiving his Ph.D., his work has focused on sCO_2 cycles, energy storage, waste heat, CCS, and ORC systems.

Jason Mallinak is a senior systems engineer at Echogen Power Systems. He studied mechanical engineering at The University of Akron (B.S. 2010) and University of Minnesota (M.S. 2012). His academic and professional focus has been on thermal energy systems, focusing on sCO_2 systems at Echogen since 2018.

Timothy Held is the Chief Technology Officer at Echogen Power Systems in Akron, Ohio, where he is responsible for development, maturation, and commercialization of supercritical CO_2 power cycles and energy storage systems. Prior to joining Echogen in 2008, Dr. Held was with GE Aviation for 13 years, where he held leadership positions in several combustor design and fuels technology teams. He received a BSAAE from Purdue University in 1987, and a Ph.D. in Mechanical and Aerospace Engineering from Princeton University in 1993. He has published several technical journal articles and book chapters and currently holds 50 U.S. patents.

Jason Miller is the Director of Engineering at Echogen Power Systems, where he leads multidisciplinary teams advancing sCO_2 power systems and thermal-energy technologies. He has over a decade of experience delivering DOE-funded programs, turbomachinery development efforts, and large-scale test initiatives. His work spans advanced heat-pump technologies, expander development, and next-generation energy-system modeling and testing. He earned his BSME in 2003 and pursued graduate studies at the University of Akron until 2008. He has published journal articles and holds multiple energy systems patents related to architecture and controls.

ABSTRACT

Pumped thermal energy storage (PTES) is a long-duration energy storage (LDES) technology that stores electricity as thermal energy by heating thermal reservoirs during the charging cycle and then converting that stored heat to electricity during the generating cycle. In PTES systems using supercritical carbon dioxide (sCO_2) as the working fluid, electrical energy is stored by heating and compressing sCO_2 and transferring the heat into high-temperature reservoirs. During discharge, sCO_2 is heated by the high-temperature thermal reservoirs and expanded through a turbine to produce mechanical work, which is then converted to electricity. LDES plays a critical role in enabling the large-scale integration of variable renewable energy sources such as solar and wind. By storing excess energy during periods of low demand and releasing it when generation is insufficient, LDES helps ensure grid reliability. Unlike short-duration storage solutions, LDES can balance energy supply and demand over extended periods of time, ranging from several hours up to multiple weeks, making it essential for decarbonizing the power sector, while reducing reliance on fossil fuel-based peaking plants. Because energy demand can vary over time, PTES systems must operate across a range of conditions, which directly influence their design and operational strategies. This study investigates the range of operating conditions a PTES system may encounter, focusing on variable load profiles, thermal energy balancing between hot and cold reservoirs to achieve synchronized depletion, and the impact of thermal degradation on system performance over time. A series of quasi steady-state simulations are performed using a detailed MATLAB-based cycle model. By systematically varying electrical load, reservoir thermal duties, and reservoir temperatures, the study evaluates key off-design performance metrics, including coefficient of performance in the CHG cycle and cycle efficiency in the GEN cycle.

INTRODUCTION

As global energy systems transition toward low-carbon energy generation technologies, such as wind turbines and photovoltaic solar cells, the addition of large-scale energy storage technologies is becoming increasingly important to combat intermittency challenges and to ensure electrical grid reliability. Simultaneously, the energy demand is increasing, which creates additional challenges in energy production and electrical grid stability. For example, in 2024, the energy demand increased by 2.2% and the electricity demand by 4.3% [1]. Moreover, the US electrical demand is predicted to rise 25% by 2035, and it is expected that 50% of the electrical generation will be provided by solar and wind systems [1]. Consequently, additional energy systems such as energy storage systems are necessary to ensure grid resiliency in this rapidly developing energy landscape.

One of the most widely deployed energy storage solutions on the market is lithium-ion battery technology; however, its applicability is constrained by its short durations of up to four hours [2, 3]. The need for longer-duration solutions arises from renewable energy generation variability. This necessitates the deployment of other forms of longer-duration energy storage technologies to fill in these gaps when renewable energy generation becomes insufficient. These technologies are known as long-duration energy storage (LDES) technologies. Some of the most prominent LDES technologies include pumped hydropower, compressed air energy storage (CAES), and pumped thermal energy storage (PTES) [2-4]. Hydropower energy storage is well-established, but has geographical limitations, as sites must be located near water streams with elevation differentials. Consequently, many of the suitable locations have already been exploited for this technology, establishing hydropower as a mature technology with a limited future for opportunity and growth. CAES exhibits high round-trip efficiencies, meaning that the majority of the energy stored in the system can effectively later be discharged to the grid. However, it faces geographical limitations as these technologies can only be deployed in sub-surface caverns and seabeds. Lastly, PTES, while a newer technology with a lower technology-readiness level, is considered a promising long-term solution to combat intermittency challenges due to its geographical flexibility, relatively small ecological footprint, use of non-toxic and readily available working fluids, and competitive round-trip efficiency [4].

Pumped thermal energy storage is a type of long-duration energy storage that relies on the transfer of heat to and from thermal reservoirs. PTES work by taking excess electricity from the grid, typically from renewable energy sources, to power a heat pump. This is known as the charging (CHG) cycle. In a heat pump, heat is transferred against its natural gradient, meaning it is transferred from a low-temperature source "uphill" to a higher-temperature source. That thermal energy is stored in thermal reservoirs until it is converted back to electricity via a turbine-generator system. This is known as the generating (GEN) cycle. In this paper, an evaluation of off-design performance of a PTES system using sCO_2 as the working fluid will be presented. The presented off-design results are based on a current project located in North Pole, Alaska for the Golden Valley Electric Association (GVEA). Several off-design strategies will be presented, including system turndown, thermal reservoir balancing, and reservoir temperature degradation, reflecting a realistic operational range of this PTES system as determined by GVEA.

SYSTEM DESCRIPTION

GVEA project description

The GVEA PTES project is located in North Pole, Alaska and is designed to deploy 1200 MWhe of electrical energy using an sCO_2 PTES plant with expected operation by 2029. The generating side of the PTES system is designed for 50 MWe x 24 hours, and the charging side is designed for 25 MWe x 100+ hours. Minimum operating loads are 20% (CHG) and 10% (GEN).

PTES system description

PTES consists of two separate consecutive thermodynamic cycles; the first cycle, the CHG cycle, seeks to charge the system by storing thermal energy in high-temperature reservoirs. The second cycle, the GEN cycle, then converts the stored thermal energy into electricity to be dispatched to the grid. The CHG and GEN layout configurations are listed in Figure 1.

The CHG and GEN cycles employ three shared heat exchangers, the high-temperature heat exchanger (HTX), the medium-temperature heat exchanger (MTX), and the low-temperature heat exchanger (LTX), that facilitate heat transfer between the working fluid and thermal reservoirs, the high-temperature reservoir (HTR), medium-temperature reservoir (MTR), and low-temperature reservoir respectively (LTR) respectively. The HTR stores and releases high-grade heat across a specific temperature range, while the MTR handles a lower-grade temperature range. In this paper, the term ‘high-temperature reservoirs’ refers collectively to both the HTR and MTR, while individual reservoirs are identified by their acronyms. In the CHG cycle, the LTR serves as the heat source for the working fluid, and the high-temperature reservoirs are the heat sink. The opposite is true for the GEN cycle. Additionally, both cycles also share a fourth heat exchanger, the recuperating heat exchanger, or RCX, that is used to transfer heat internally to the working fluid to boost each cycle’s overall performance by pre-heating the fluid with excess heat from the cycle. The final heat exchanger, the air-cooled heat exchanger (ACX), is a cycle-specific heat exchanger that serves to reject excess heat in each cycle to maintain thermal balance between the reservoirs. Because PTES systems rely on thermal reservoirs to store and reject heat, maintaining an energy balance among the three reservoirs is critical. Each reservoir must charge and discharge thermal energy at coordinated rates such that all reservoirs reach accumulation and depletion simultaneously in the CHG and GEN cycles, ensuring balanced operation and optimal cycle performance.

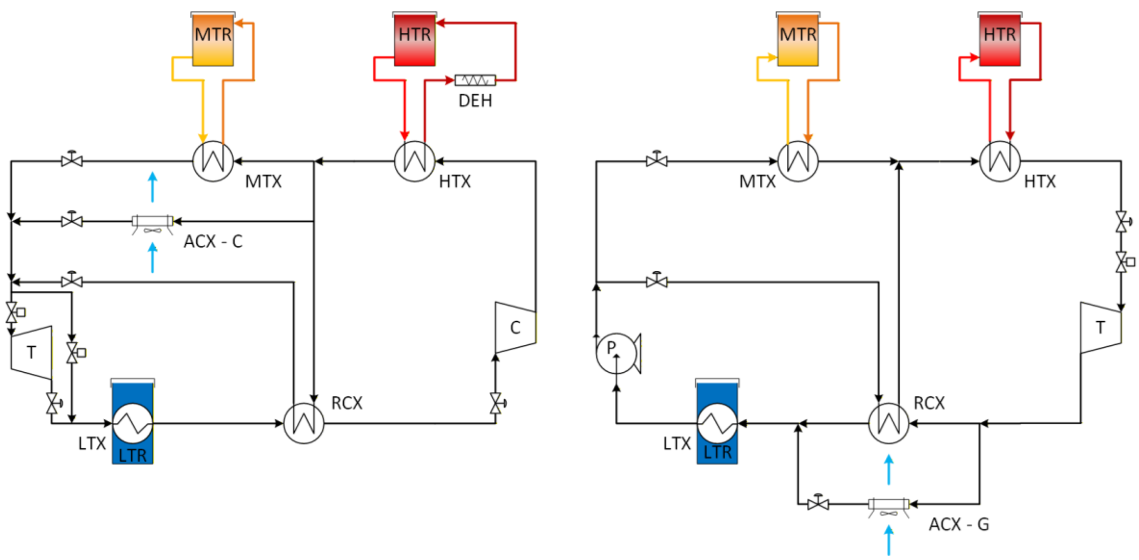

Figure 1: CHG (left) and GEN (right) cycle configurations.

Each cycle also utilizes its own set of turbomachinery equipment to compress and expand the working fluid, thereby controlling system pressure. The CHG cycle utilizes three (3) identical compressors in parallel to raise the pressure of the working fluid and two turbines of differing sizes to extract work out of the working fluid and achieve pressure reduction. Additionally, the CHG cycle deploys a direct-electric heater (DEH) at the inlet of the HTR to boost cycle performance in certain off-design scenarios. The GEN cycle consists of a single pump to pressurize the working fluid and a single turbine to generate electrical power for the grid.

Each cycle consists of its own set of design point operating parameters, namely mass flow, temperature, and pressure. Off-design parameters are dependent on the established design point for each cycle and vary according to changes in boundary conditions of the system itself. The CHG cycle is designed for a nominal operating capacity of 25 MWe, with a load range extending down to 5 MWe, or 20% of its design point. The GEN cycle is designed for a maximum load of 50 MWe, with its load range extending down to 5 MWe, or 10% of its design point.

OFF-DESIGN METHODOLOGY

Because energy demand and operating conditions vary [2], a PTES system needs to operate under a diverse set of conditions. Changes in operating conditions can include variations in load (i.e., turndown), initial reservoir temperatures (i.e., thermal degradation), reservoir duties (i.e., the rate at which power is extracted from the reservoirs), and ambient conditions. Ultimately, the system must remain resilient under changing operating conditions, continuing to store and extract the necessary power to and from the grid over a continuous duration. This resiliency becomes a key requirement that influences system design (e.g., component size and quantity), operating strategy, and performance metrics, specifically coefficient of performance (COP) for CHG and cycle efficiency (η) for GEN. To investigate changes in performance and develop operational strategies for the system, four modeling studies are investigated. The off-design modeling studies are as follows:

- Turndown (Load)
- Reservoir Balancing (Reservoir Duty Ratios)
- Reservoir Thermal Degradation (Reservoir Temperatures)
- Ambient (External Temperature)

Notably, the PTES system is designed to operate within a load range of 100% to 20% (CHG) or 10% (GEN) as determined by GVEA, so turndown studies will focus on these operating ranges. Reservoir balancing does not have an upfront defined operating range. Rather, the objective is to assess how deviations in the thermal ratios that correlate reservoir operation impact COP, cycle efficiency, and cycle duration, and how those relate to the overall energy available in the system. Thermal degradation refers to temperature losses within the reservoirs, which directly reduce the stored thermal energy. This study investigates how reservoir temperature degradation influences overall cycle performance. Lastly, the system will be affected by highly variable ambient temperature and weather conditions. The ambient temperature can range from -40°C to +30°C throughout the year in North Pole, Alaska. Rather than a stand-alone study, ambient may be combined with any of the three preceding studies to determine its impacts as other parameters also move off design. In investigating the results of these studies, the goal is to employ a control strategy that adapts PTES operation in response to real-time system conditions, with the objective of mitigating performance losses and maximizing energy out of the system.

Turndown strategy

Turndown strategy refers to the method used to reduce power or load of the cycle from its design point, such that performance penalties of the entire system are minimized. Because of its unique complexities, the turndown of the CHG cycle will specifically be investigated in this section. In order to operate across a large range of load conditions as specified by GVEA, the sizes of the compressor(s) and charging turbine(s) are critical design parameters to the CHG cycle. A common way to address the large operating range of this cycle is to use several varying sizes of compressors and turbines. However, due to specific turndown operating range limits for the pieces of turbomachinery, the CHG cycle requires a relatively large number of compressors and turbines to span the full operating range. This can negatively impact the CHG cycle performance and overall PTES plant cost. As a result, the main challenge lies in maximizing cycle performance while minimizing the number of components added to the system. The solution to this problem relies on the use of a direct electric heater, or DEH. The DEH is installed at the inlet of the HTR to provide supplemental thermal energy to the reservoir. As a direct electric heater, it converts electrical energy directly into heat and therefore operates with a COP of unity. Under certain off-design operating conditions, the system COP degrades significantly; in these cases, the inclusion of the DEH can partially offset this degradation and improve the COP, offering a smoother turndown curve.

Based on GVEA turndown requirements, the CHG cycle operates using three (3) different turndown methodologies (see Figure 2). Each one is designed according to component limitations, specifically for the CHG compressor, as this component tends to be the limiting factor in the CHG cycle. The turndown methods are as follows:

- *Primary Turndown* – includes the use of compressor inlet guide vanes (IGV)
- *Secondary Turndown* – includes the use of compressor IGVs, as well as compressor inlet throttling to modify inlet pressure
- *Tertiary/Thermal Turndown* – includes the use DEH at the inlet of the HTR

The CHG cycle consists of three identical compressors, each representing approximately 33.3% or one-third of the total compressor power. In primary turndown, changes in compressor power are dictated by IGVs controlling flow entering the compressor. The IGV angle range is typically between 20º to -60º, depending on the specific compressor design. This method can be used to turn down each individual compressor by up to 20%, or 80% of its compressor power. Because the cycle utilizes three compressors, this turndown method can realistically turn down the cycle to approximately 25% of total power with a single compressor engaged compared to all three compressors fully engaged. Refer to Figure 3 for a visual representation of the primary turndown with all three compressors compared to a single compressor. When the compressor reaches the IGV design limits, the compressor flow and related load cannot be reduced. To continue CHG compressor turndown, an additional step needs to be undertaken. While the compressor has reached its minimum volume flow, the mass flow may be further reduced by reducing the inlet density. The additional step requires a lower compressor inlet pressure compared to design point pressure via a throttling valve. This is known as secondary turndown. The use of throttling at the inlet of the compressor can extend the turndown range below the limits of primary turndown. The lower compressor inlet pressure affects the required compressor work and outlet pressure. As a result, the compressor can operate below the original IGV angle limits compared to primary turndown. The maximum compressor inlet pressure reduction is approximately 15% of the design compressor inlet pressure, meaning the inlet pressure to the compressor can be as low as 85% of its design value. However, turning down the CHG cycle in this manner has potential negative effects on COP. In order to achieve an acceptable performance drop, the secondary turndown should not be less than 60% of the total compressor load. This limit affects the selected number and size of the compressors. Hence, three identical compressors have been selected to use for the GVEA project application. To improve the COP performance across the system turndown an additional turndown scenario can be used to directly heat the heat transfer fluid entering the HTR. This is known as tertiary or thermal turndown. The main benefits of using DEH are as follows:

- Reduction of system performance drop between compressors switches, that is turning down from *three* to *two*, and from *two* to *one* compressor operation
- Reduction of compressor inlet throttling based on the size of the compressor and CHG load
- Compressor(s) size selection for the 20% system load compared to design system load

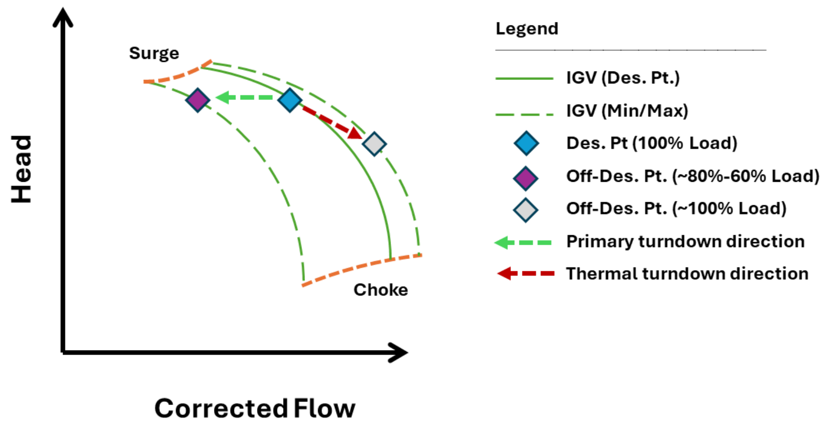

Figure 2: Compressor turndown scenarios.

Similar to CHG turndown strategy, the GEN turndown strategy is created to respect operating range requirements in the range of 100% to 10% load, or 50 to 5 MWe. Compared to CHG cycle turbomachines, the GEN pump and turbine have much larger operating ranges, and therefore the GEN cycle only requires one of each component. For this reason, GEN cycle turndown strategy is much simpler and does not require various turndown methods. Instead, turndown is based on a pump equipped with a variable frequency drive (VFD) [3]. A VFD pump can convert fixed-frequency electrical power from the grid into variable frequency to control and change the speed of an AC motor. In doing so, the mass flow of the pump can be manipulated to control pressure ratio across the pump and ultimately power output of the cycle [3]. While the GEN cycle turndown is relevant and results will be presented, it is not the primary focus of the turndown analysis.

Reservoir balancing strategy

The PTES utilizes three thermal reservoirs as mentioned above, the two high-temperature reservoirs and the LTR. The thermal reservoirs are linked to each other via two thermal balancing ratios, which ultimately tie the reservoir duties to each other. Reservoir duty refers to the rate at which thermal energy is extracted or supplied to the thermal reservoirs. Each reservoir stores a specified quantity of thermal energy. To ensure that the reservoirs charge synchronously during the CHG cycle and deplete synchronously during the GEN cycle, pre-defined balancing ratios are used to coordinate their respective energy duties. The ratios are $\Psi_{m\text{-}mh}$ and $\Psi_{h\text{-}c}$ and are defined in the equations below. The $\Psi_{m\text{-}mh}$ ratio ties the MTR duty to the high-temperature reservoir duties. The $\Psi_{h\text{-}c}$ ratio ties the HTR duty to the LTR duty. As the system moves off-design, whether that be from load turndown and the use of DEH or changes in heat rejection as a consequence of fluctuating ambient conditions, the system may not simultaneously charge or deplete the reservoirs, leaving thermal energy availability in one more of the reservoirs below its specified design. To assess the impacts of changes in these ratios, $\Psi_{h\text{-}c}$ was swept in a fine step size for a given $\Psi_{m\text{-}mh}$ value. This procedure was repeated for a large range of $\Psi_{m\text{-}mh}$ values above, below and at its design. The result was matrices with $\Psi_{h\text{-}c}$ along one axis and $\Psi_{m\text{-}mh}$ the other, populated with COP values for the CHG cycle and cycle efficiency values for the GEN cycle.

$$\boldsymbol{\Psi}_{\boldsymbol{m-mh}} = \frac{\dot{Q}_{MTR}}{\dot{Q}_{MTR}+\dot{Q}_{HTR}} \qquad \text{(Eq. 1)}$$

$$\boldsymbol{\Psi}_{h-c} = \frac{\dot{Q}_{HTR}}{\dot{Q}_{LTR}} \quad \text{(Eq. 2)}$$

Reservoir thermal degradation strategy

The three thermal reservoirs each have a specific design hot temperature, T_h and a specific design cold temperature, T_c. At the start of the CHG cycle, the high-temperature reservoirs begin with their design T_c. values, and as the system is charged to full capacity, they reach their design T_h values. The GEN cycle starts with the reservoirs at design T_h, and as heat is transferred back to the working fluid via the heat exchangers, the reservoirs end at their design T_c values, corresponding to the initial CHG temperature. At this point, the system is ready to begin the CHG cycle again.

The final off-design analysis is focused on the reservoir temperature degradation, formally known as thermal degradation. Thermal degradation primarily focuses on the GEN cycle, specifically the HTR T_h, as this is the primary driver in delivering heat to the working fluid, which ultimately gets converted to electric power for the grid. The net power extracted from the turbine is a function of the working fluid's enthalpy, which is dependent on the temperature and pressure of the working fluid at the inlet of the turbine. As the fluid expands through the turbine, its enthalpy is converted into mechanical torque on the rotor [5, 6]. This rotational energy is transmitted through a gearbox to a generator shaft, which rotates a magnet that induces an electro-motive force via Faraday's Law to deliver power to the grid [6]. Any reduction in pressure or temperature of the working fluid directly impacts power output to the grid [5]. High-side pressure is controlled via the VFD pump, where head rise can easily be manipulated to meet necessary pressure ratios across the generating turbine [3]. However, the thermal input from the thermal reservoirs may vary due to heat losses to the environment or off-design charging conditions. Based on site requirements as determined by GVEA, the GEN cycle is required to produce design power across a range of HTR T_h temperatures. More specifically, the GEN cycle is required to dispatch 50 MWe of power, even with a temperature loss of 20°C. This loss in temperature corresponds to approximately a 10% loss in stored enthalpy of the HTR. To meet this requirement, a new oversizing configuration has been investigated that centers around a lower HTR temperature. MTR T_h and CHG cycle HTR T_c thermal degradation will also be analyzed, but the primary focus of the results is related to defining a new design configuration that is an alternative to the current design point.

RESULTS AND DISCUSSION

Several quasi-steady state modeling studies were run using proprietary MATLAB-based models. There are two separate cycle models for CHG and GEN, and both include all components as described in *PTES system description*. The CHG cycle is a simple recuperative heat pump cycle with two high-temperature reservoirs that serve as the heat sink and one low-temperature reservoir that serves as the heat source. It utilizes three identical compressors and two turbines of differing sizes. The GEN cycle is a simple recuperative heat engine cycle; this cycle also contains two high-temperature reservoirs, as well as a single low-temperature reservoir. The high-temperature reservoirs are the heat source, and the low-temperature reservoir is the heat sink. This cycle utilizes a single pump and a single turbine. The in-house code uses the NIST - Standard Reference Database 23: Reference fluid thermodynamic and transport properties (REFPROP), Version 9.1 [7] and includes preliminary component sizing and performance information based on vendor input to ensure the simulation results remain within a physically realistic operating space. Note that normalized values in the following results are calculated by taking the off-design modeling value and dividing by its nominal design value, unless otherwise stated.

System turndown

The first off-design analyses focus on the system turndown. As mentioned previously, the CHG cycle consists of specific components (three identical compressors, two turbines of differing sizes, and one DEH) to provide a turndown load in range of 100% to 20% CHG load. To achieve the required turndown range with acceptable CHG performance, three turndown procedures were introduced: primary, secondary, and tertiary/thermal. The CHG system turndown results are presented in Figure 3 for design day ambient conditions (0°C)

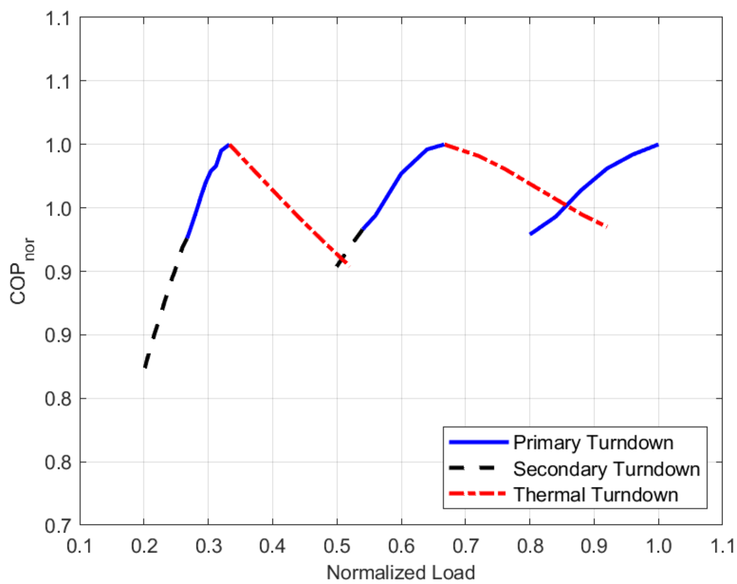

Figure 3: Normalized COP as a function of normalized load for design ambient conditions.

According to the results in Figure 3, the CHG cycle can operate in full system turndown range with the normalized COP between 100% and 82%, corresponding to 20% system load. The results also indicate that thermal turndown can improve the COP across the compressor operating ranges, specifically the transition from three to two compressors and from two to one. Simultaneously, the thermal turndown can completely eliminate secondary turndown in the transition from three to two compressors. To achieve the 20% system load, the single compressor needs to use full range of the secondary turndown procedure. Similar trends are observed in the hot (30ºC) and cold (-40ºC) day turndown scenarios. An example of these trends is shown in Figure 4. The ambient conditions have impact on the COP; the cold day conditions increase COP marginally and the hot day conditions decrease COP in a range of 5 to 10% compared to design day conditions.

The GEN turndown procedure is simpler compared to CHG. The GEN cycle consists of a single turbine and pump. Because of this, a single turndown procedure is used (see Figure 5). Although a significant reduction in cycle efficiency is observed as turbomachinery efficiency degrades at lower loads, it is important to note that the cycle efficiency remains relatively stable between 80% and 100% turndown. Beyond this point, the cycle efficiency falls off substantially, with only 40% of normalized cycle efficiency seen at the lowest load. Based on these results, it is ideal to run the GEN cycle at higher loads as the stored thermal energy from the CHG cycle is more efficiently converted to electrical power in the GEN cycle. Similar trends can be observed for the hot (30ºC), and cold (-40ºC) day system turndowns. The results comparison for hot, cold and design day GEN turndown performance is listed in Figure 5. The effect of the hot and cold day conditions has negligible effects on the GEN turndown performance. Hypothetically, the GEN cycle could utilize additional turbomachinery. However, the observed impacts of an additional pump (2x50%, as opposed to 1x100%), on the GEN cycle turndown performance is negligible. An additional turbine may have a positive effect on the cycle efficiency for the lower loads, but it will likely reduce the cycle efficiency for higher loads. The primary selection criteria are turbine size and turbine count, as these factors directly influence system cost and footprint.

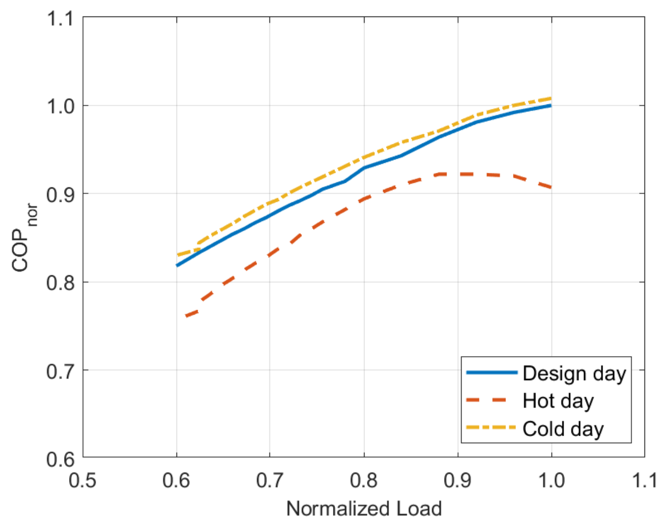

Figure 4: Normalized COP as a function of normalized load under varying ambient conditions.

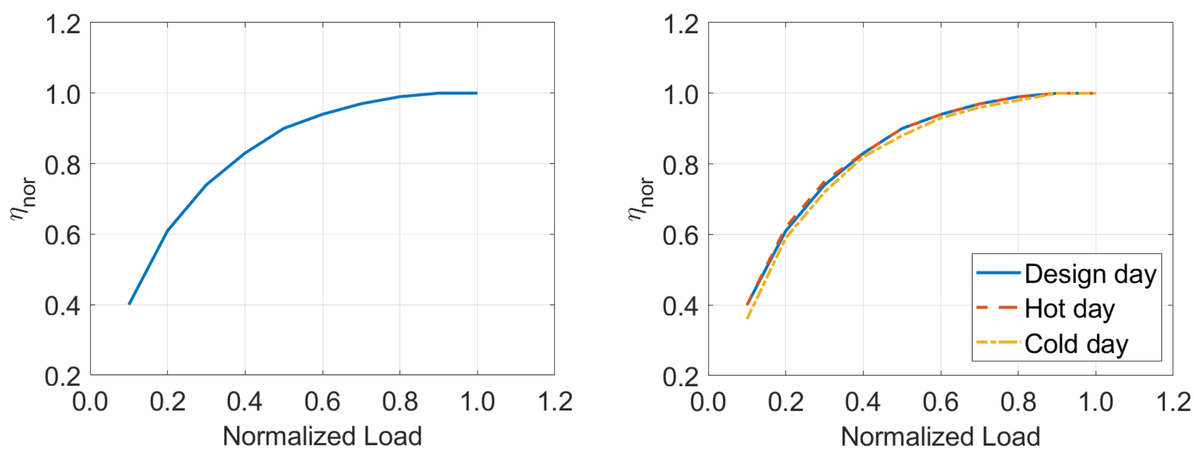

Figure 5: Normalized cycle efficiency as a function of normalized load for design (right) and off-design (left) ambient conditions.

Reservoir balancing

The second of the off-design analyses are focused on the thermal reservoir balancing. According to Figure 1, the PTES system operates with three thermal reservoirs, and these reservoirs are connected via thermal balancing ratios, $\Psi_{h\text{-}c}$ and $\Psi_{m\text{-}mh}$. Changes in these ratios compared to its design point are investigated to determine where each cycle operates to maximize its cycle performance. The impact on the CHG cycle performance, normalized COP, is shown in Figure 6. The results are for design day (0ºC).

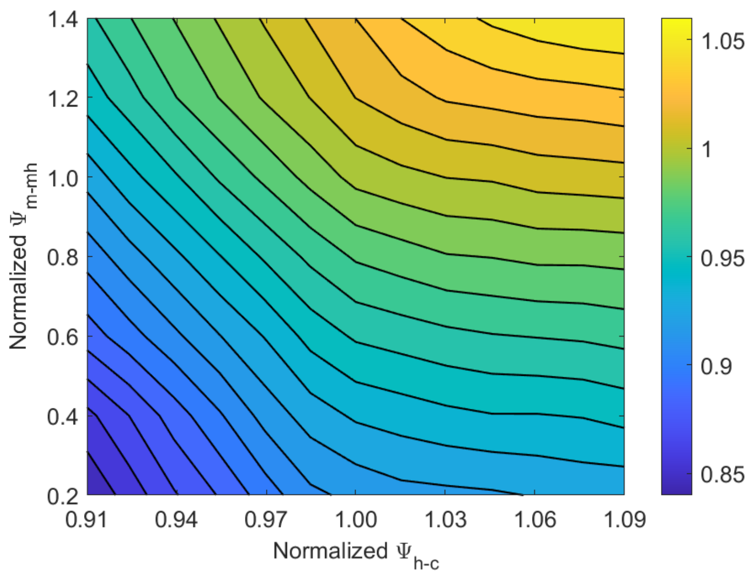

Figure 6: CHG cycle reservoir balancing – normalized COP operating map.

According to the normalized COP operating map, increasing both thermal ratios has a positive impact on performance. More specifically, COP can increase up to 6% compared to design point. The opposite is true for the lower thermal balancing ratios. The lower Ψ ratios decrease the normalized COP by up to 15% (corresponding to normalized $\Psi_{h\text{-}c} = 0.9$ and $\Psi_{m\text{-}mh} = 0.2$ compared to design point ratios). Operating the system in different ambient scenarios has additional effect on the performance. In the case of a cold day scenario, the performance is marginally improved for the higher Ψ ratios. For the lower Ψ ratios, the COP improvement can be in range of up to several percent. In the case of the hot days, the performance is dramatically reduced, especially for the lower Ψ ratios by an additional several percent compared to the design day operating conditions.

The GEN cycle displays different trends in the same operating map compared to the CHG cycle (see Figure 7). The highest cycle efficiency naturally occurs for the design Ψ ratios. Rather than a single point, the peak cycle efficiency forms a diagonal line across the center of the matrix, corresponding to a specific $\Psi_{h\text{-}c}$ value for each $\Psi_{m\text{-}mh}$ value. Along this diagonal line, the optimal $\Psi_{h\text{-}c}$ value increases as $\Psi_{m\text{-}mh}$ decreases. Deviations from this diagonal in either direction resulted in a reduction in cycle efficiency. This result is noteworthy, as it suggests a potential linkage between the two ratios in the GEN cycle. While further investigation is needed to fully characterize this relationship, it is evident that the diagonal represents the combination of ratios that yields the most efficient GEN cycle, directly influencing the cycle duration.

A comparison of the two cycles reveals conflicting results. In the CHG cycle, cycle efficiency is highest in the top right corner of Figure 6, corresponding to the highest normalized thermal balancing ratios. In the GEN cycle, cycle efficiency falls off completely, resulting in non-convergence in this corner due to the inability to maintain power with the given ratio constraints. Where the CHG cycle wants to operate is precisely where the GEN cycle cannot operate. While this result is not ideal for the combined cycle system, the results are consistent with thermodynamic cycle behavior. A heat pump's performance (CHG cycle) improves when the cycle can deliver more heat for the same amount of work input. A heat engine's performance (GEN cycle) improves when it needs less heat to produce the same amount of work output. Because of this inverse relationship between the two cycles, the result is not surprising. Because this is a two-cycle system, in which the combined goal is to maximize round-trip system efficiency, both CHG COP and GEN η need to be balanced. This means selecting an operating point does not necessarily maximize one performance metric over the other but instead harmonizes the two cycles together.

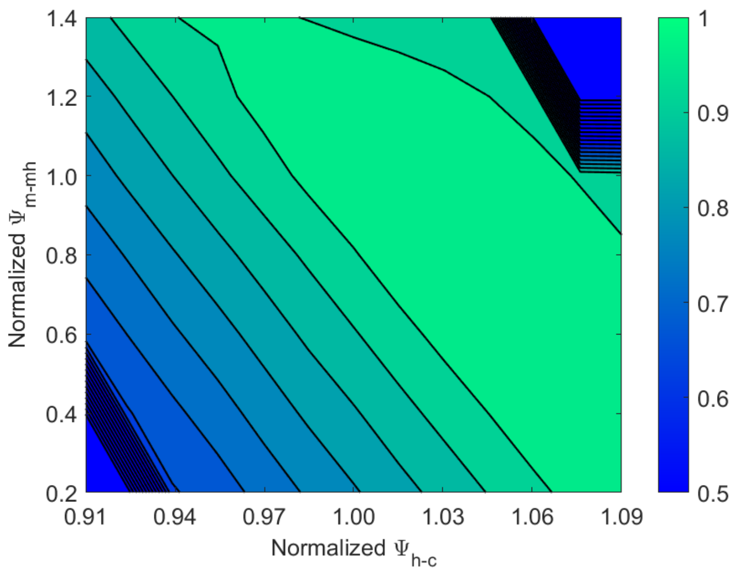

Figure 7: GEN cycle reservoir balancing – normalized cycle efficiency operating map.

Thermal degradation

According to GVEA system requirements, the GEN cycle is required to output its rated power, even with an HTR Th temperature loss of 20°C, corresponding to approximately 90% of its total stored enthalpy. Initial modeling revealed that the current design point could not maintain 50MWe of power under any HTR thermal degradation. This case is known as the *Baseline* case, and these initial results can be seen in Figure 8. To meet this requirement then, a new design configuration, known as the *Alternative* case in the figures below, has been investigated. In this *Alternative* case, the HTR centers around a lower HTR temperature, while still maintaining design thermal balancing ratios. It is understood that total power is a function of both mass flow and change in enthalpy of the CO_2 across the turbine. The inlet pressure to the pump, i.e. low-side pressure, will remain steady even in off-design conditions as the low-side pressure of the system is effectively controlled by a phase-change material in the LTR, effectively maintaining vapor-liquid equilibrium of the CO_2 working fluid. Resultingly, outlet enthalpy is controlled and will not change in off-design situations. However, with HTR thermal degradation, the inlet enthalpy of the working fluid will decrease, resulting in a smaller change in enthalpy across the turbine. The goal is then to redefine the design point around a lower temperature to understand how losses in HTR T_h affect mass flow and head rise of the cycle. By doing so, oversizing requirements of the turbomachinery can be defined. In other words, the *Alternative* case becomes an oversizing case.

In the *Baseline* case, the outlet pressure to the pump is constrained by its design value, so there is no room to increase head rise of the cycle. In the *Alternative* case, the pressure constraint is released, while still obeying system piping pressure constraints. By releasing the constraint, the cycle can produce a larger head rise across the pump, resulting in a higher pressure into the inlet of the turbine, and correspondingly a larger cycle mass flow. The *Alternative* design case can be seen in Figure 8. In both design cases, *Baseline* and *Alternative*, the cycle efficiency is optimized on cycle efficiency with a power constraint of 50MWe. Note that the *Alternative* design point is centered around an HTR enthalpy loss of 15% (or, 85% of total enthalpy) for two reasons. This design can maintain 50 MWe, despite the greater loss, and it also provides the system margin to output the necessary power should additional parameters deviate off-design.

In evaluating HTR temperature as a function of power output in Figure 8, the *Baseline* configuration exhibits an immediate loss in power output following the initial temperature drop, thereby failing to satisfy the 50 MWe requirement under a 20°C reduction in HTR temperature. In contrast, the *Alternative* configuration is designed to accommodate reduced HTR temperatures and is therefore capable of sustaining the design power at the lowered HTR temperature. Consistent with expectations, this case also continues to meet the 50 MWe requirement as the HTR temperature increases toward the *Baseline* design condition. Beyond its design point, moving in the opposite direction, the *Alternative* case falls off as it can no longer sustain the continued HTR thermal degradation. However, it continues to produce significantly higher power output compared to the *Baseline* case under substantial losses in reservoir temperature. While it is unlikely that the HTR should ever endure that significant of a drop in temperature, it remains a positive result for the GEN cycle to obtain an additional 7-10% power relative to the *Baseline* case at these lower HTR temperatures. Normalized HTR temperature is defined in the following equation:

$$\boldsymbol{T_{norm}} = \frac{\boldsymbol{T_h - T_{c,DP}}}{\boldsymbol{T_{h.DP} - T_{c,DP}}} \qquad \text{(Eq. 3)}$$

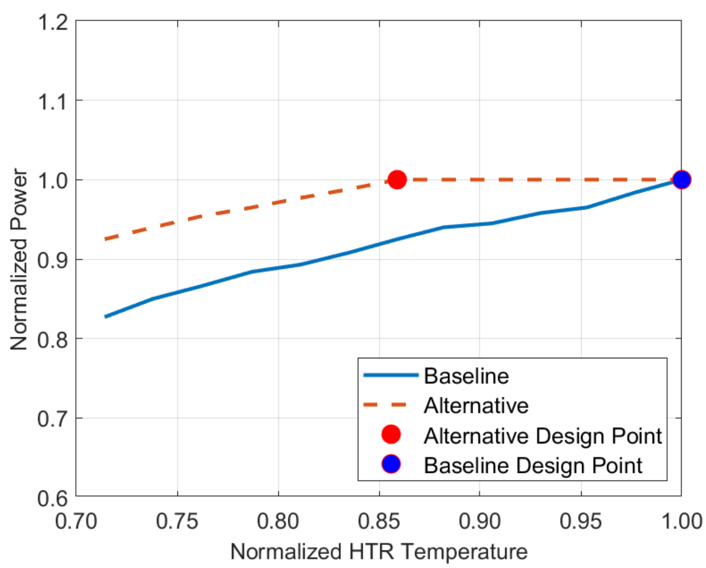

Figure 8: Normalized power as a function of normalized HTR temperature.

Figure 9 illustrates the increased cycle mass flow rate required to obtain design power for the given temperature reduction and increased high-side pressure. With a modest 8.4% increase in mass flow, the *Alternative* case is able to maintain the required power output. As this case approaches the *Baseline* design point, the mass flow rate gradually decreases, returning to the *Baseline* design point mass flow. In the *Baseline* case, the mass flow increases slightly, about 1.5%, but is constrained by the *Baseline* outlet pump pressure. To compensate for the reduction in temperature while maintaining the required turbine work, the pressure across the turbine must increase. Accordingly, a 4.5% increase in the turbine inlet pressure, i.e. high-side pressure, is sufficient to offset the enthalpy losses associated with the temperature reduction as seen in Figure 10. The impact of the redesigned turbine inlet conditions on overall GEN cycle efficiency is minimal between both HTR design cases as shown in Figure 11. With this new design, the pump head rise will need to increase to provide the higher mass flow rate to compensate for thermal degradation, but has minimal effect on cycle performance. This result is favorable, as three parameters were modified to meet the same power requirement.

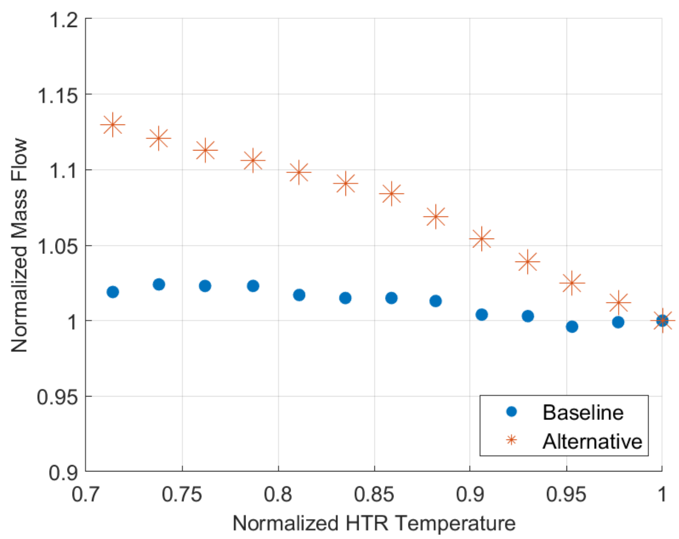

Figure 9: Normalized mass flow as a function of normalized HTR temperature.

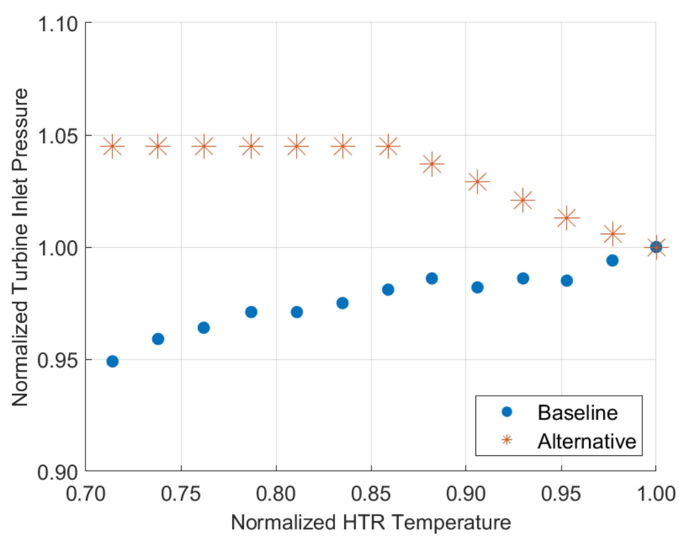

Figure 10: Normalized turbine inlet pressure as a function of normalized HTR temperature.

While HTR thermal degradation is the primary factor affecting the cycle's power output due to its role in storing high-grade heat, the thermal degradation of the MTR was also examined. In the CHG cycle, CO_2 first transfers high-grade heat to the HTR before flowing downstream to a split between the MTR/MTX, RCX, and ACX. Here, the working fluid transfers lower-grade heat; a portion is absorbed by the MTR, another portion is internally recuperated, and any residual low-grade heat that is not required is rejected to the ambient. In the GEN cycle, after being pressurized by the pump, CO_2 absorbs lower-grade heat from the MTR/MTX and through internal recuperation then enters the HTR/HTX to absorb high-grade heat. An analogous study was performed for the MTR in the GEN cycle, where the design-point temperature was intentionally lowered by a defined decrement, and the effects on cycle performance were assessed. Unlike GEN cycle HTR thermal degradation, there is no alternative design point to explore as the MTR is only responsible for pre-heating and storing low-grade heat. Three different load cases were examined, namely 50 MWe, 25 MWe, and 5 MWe. It was found that even after losing 25% of normalized MTR temperature, there was a minimal impact in cycle efficiency loss for both 25 MWe and 5 MWe cases. The 50 MWe case exhibits a slight loss in cycle efficiency, with a loss of 1.3% in cycle performance from design MTR temperature as seen in Figure 12. These results demonstrate that the MTR has minimal impact on cycle performance for both power output and η especially at lower loads and therefore does not play a crucial role in over-sizing or power output.

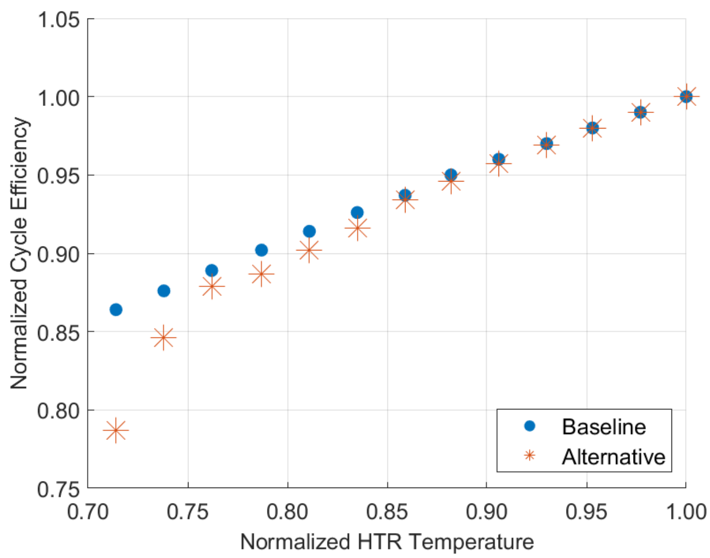

Figure 11: Normalized cycle efficiency as a function of normalized HTR temperature.

Lastly, CHG cycle thermal degradation is investigated for the HTR. The CHG cycle's primary function is to increase the thermal energy of the reservoirs rather than produce power. If T_c decreases, the reservoir begins at an initial colder state, allowing the CO_2 to transfer more heat to it. Note that the normalized temperature of the HTR in the CHG cycle relies on an arbitrary initial reference temperature. Because the cold-side temperature is the parameter being varied, no fixed reference temperature exists for normalization, so an appropriate reference value was defined for consistency, using Equation 3. Figure 13 presents the results of CHG cycle COP_{nor} as a function of HTR temperature. Interestingly, the CHG cycle COP_{nor} increases as the reservoir T_c decreases, with an increase of 1.7% from its design point. This somewhat counterintuitive behavior is caused by the modest reduction in the Lorenz mean temperature of the heat sink as T_c decreases. Because the average heat sink temperature decreases with T_c, the COP_{nor} of a heat pump actually increases.

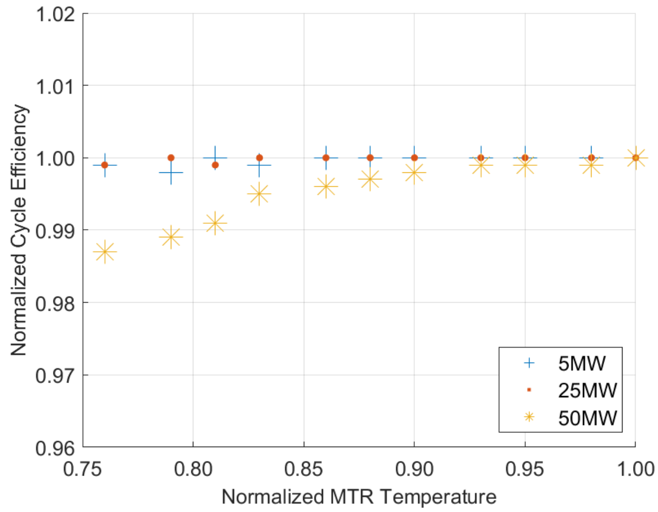

Figure 12: Normalized cycle efficiency as a function of normalized MTR temperature for three load cases (5, 25, and 50 MWe).

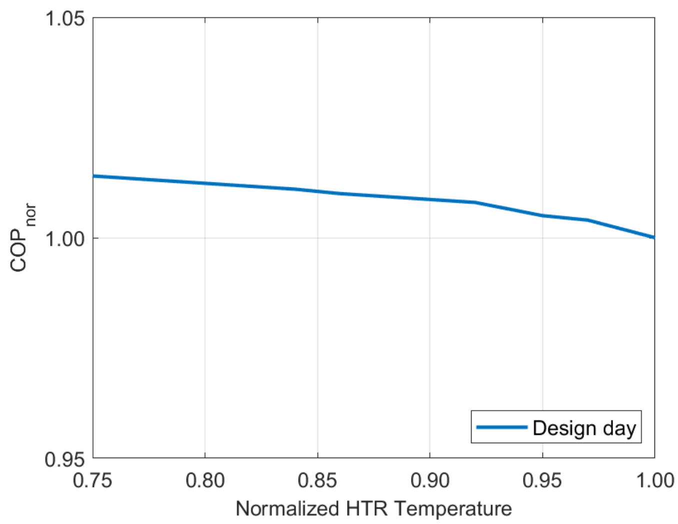

Figure 13: Normalized COP as a function of HTR T_c.

CONCLUSION

In this study, off-design analysis of the PTES system located in North Pole, Alaska, has been performed for three off-design scenarios (i.e., turndown, reservoir balancing, and thermal degradation). Each off-design scenario can provide insight for control and operating strategies for the system as a whole. According to presented results, the PTES system can operate in full operating range with acceptable penalties for the system performance. The penalties depend on the number of components (i.e., compressors, turbines) and required system load (see Figure 3 and Figure 5). Moreover, the CHG cycle can operate with off-design reservoirs ratios with potential positive effects on COP_{nor} with higher Ψ ratios and negative effects with lower Ψ ratios. In the GEN cycle, it was found that there is a distinct diagonal corresponding to the highest efficiency cycle across the same operating space as the CHG cycle. Further investigation is warranted to determine the linkage between the two thermal balancing ratios in off-design scenarios for the GEN cycle. Lastly, in re-designing the design point of the GEN cycle around a lowered HTR temperature, the GEN cycle can maintain 50 MWe of power even under immense normalized temperatures losses, with minimal impacts to cycle efficiency at the design HTR temperature.

REFERENCES

[1] IEA (2025), Global Energy Review 2025, IEA, Paris https://www.iea.org/reports/global-energy-review-2025, Licence: CC BY 4.0.

[2] Environmental and Energy Study Institute (EESI) (2019), *Fact Sheet: Energy Storage*, EESI, Washington, DC. Available at: https://www.eesi.org/papers/view/energy-storage-2019, Accessed 4 December 2025.

[3] Huggins, Robert. Energy storage: Fundamentals, materials and applications. Springer, 2016. https://doi.org/10.1007/978-3-319-21239-5.

[4] Sumit S., Mehdi M., Pumped thermal energy storage: A review, International Journal of Heat and Mass Transfer, Volume 213, 2023, 124286, ISSN 0017-9310, https://doi.org/10.1016/j.ijheatmasstransfer.2023.124286.

[5] Shankar, A., Kalaiselvan, V., Umashankar, S., Paramasivam, S. & Hanigovszki, N. (2016), *A comprehensive review on energy efficiency enhancement initiatives in centrifugal pumping system*, Applied Energy, Vol. 181, pp. 495–513, https://doi.org/10.1016/j.apenergy.2016.08.070

[6] U.S. Energy Information Administration (EIA) (n.d.), *How Electricity Is Generated*, U.S. EIA, Washington, DC. Available at: https://www.eia.gov/energyexplained/electricity/how-electricity-is-generated.php (Accessed: 4 January 2026).

[7] Lemmon, E. , Huber, M. and McLinden, M. (2013), NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, Natl Std. Ref. Data Series (NIST NSRDS), National Institute of Standards and Technology, Gaithersburg, MD, [online], https://tsapps.nist.gov/publication/get_pdf.cfm?pub_id=912382 (Accessed January 30, 2026)

ACKNOWLEDGMENT

This material is based upon work supported by the U.S. Department of Energy, Office of Clean Energy Demonstrations, Long-Duration Energy Storage program under Award Number DE-CD0000033.