LATENT HEAT STORAGE FOR CONCENTRATING SOLAR POWER

by

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A thesis submitted to the Faculty and the Board of Trustees of the Colorado School of Mines in partial fulfillment of the requirements for the degree of Doctor of Philosophy (Materials Science).

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ABSTRACT

Due to the rising threat of climate change, there has been a significant effort to expand clean electricity generation from renewable resources - primarily solar photovoltaics and wind power. However, these resources provide variable generation that depends on when the sun shines and when the wind blows. To be effectively integrated into electric grids, this variable generation then needs complementary technologies that could enable continued growth. One potential enabling technology, concentrating solar power, is particularly attractive due to its ability to incorporate inexpensive thermal energy storage. Many different specific embodiments of concentrating solar power are still being considered, though significant recent attention has been given to systems using latent heat storage. These systems store thermal energy in the phase change of a material, which is an isothermal process that can reach high energy density and potentially low storage cost. To implement these theoretical advantages in practice, several aspects of latent heat storage for concentrating solar power are still being developed. Much of the focus has been on phase change material development, including integration of these materials into effective thermal storage subsystems and overall system designs. As a part of this growing field, in this thesis I have provided contributions in design, modeling, and experimental demonstration of latent heat storage systems with the end goal of widespread deployment of concentrating solar power technology.

In one broadly applicable contribution to this field, I developed a holistic system model to evaluate potential phase change materials based on the influence of their material properties on the overall levelized cost of a electricity of a concentrating solar power plant. Next, I focused on development of a new concentrating solar power concept in which latent heat storage and a power block are both held on top of a small scale solar power tower. This concept is unique for its size (∼100 kW), methodology (latent heat storage on top of the tower), and is ability to turn on and off on demand. To analyze the potential of this design, I
first developed a detailed thermal model to predict overall system performance and efficiency. I also combined this model with techno-economic analysis to evaluate the scenarios in which the concept could be cost-competitive with both conventional and alternative technologies. With this opportunity identified, my subsequent efforts were largely experimental, as I led construction and testing of two prototypes to provide a proof-of-concept of the technology. The second of these prototypes built upon all previous advancements in knowledge and design of the concept, and was able to demonstrate higher system efficiency than any previous experimental efforts to combine latent heat storage with power generation. The remaining work to be done to develop this concept is in reliable containment of the phase change material that we selected, aluminum-silicon. I performed experiments on several potential containment methods, but found mixed results, and further efforts are still required before commercial development of the technology.

The theoretical and experimental results of this thesis help advance the technology of latent heat storage for concentrating solar power. I have provided a better understanding of the impact of phase change material selection on performance and cost of an entire concentrating solar power plant, with detailed consideration of the importance of each phase change material property. I have demonstrated the advantages of metal phase change materials with modeling and experiments, reducing the risk of using these materials in latent heat storage systems. I also helped identify remaining issues in reliable containment of aluminum-silicon as a phase change material. Together, this thesis provides motivation for further progress in latent heat storage for concentrating solar power, which could enable significant growth of clean electricity around the world.
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For those that will suffer from climate change
and those that want to do something about it
CHAPTER 1
BACKGROUND INFORMATION

1.1 Motivation

With recognition that human activity will have a strong effect on future consequences of climate change, there has been a global effort to develop new technologies that reduce greenhouse gas emissions. The electricity sector contributes over 40% of worldwide energy-related carbon dioxide emissions [1], and thus clean electricity generation from renewable resources has been a focus of much of this global effort. Hydropower is the most established renewable electricity technology (it currently makes up nearly two thirds of renewable electricity generation), but it is geographically limited and has limited opportunity for future growth (in 2017, hydropower capacity increased by less than 2% [2]). Conversely, rapid technical progress and cost reductions have been recently realized for both wind power and solar photovoltaics (PV) [3, 4]. Together, these made up almost 60% of global power capacity additions in 2017 [2].

Unfortunately, the renewable electricity technologies that are currently growing fastest, PV and wind power, are inherently intermittent and therefore cannot alone be used to meet all of our electricity needs. The grid must always exactly match electricity supply with demand, but grid operators do not have the ability to control when the sun shines or when the wind blows. When PV and wind are added to the grid, then either other generators must reduce output, or excess generation must be curtailed. The consequence of this curtailment can be significant: the National Renewable Energy Laboratory estimated that if PV grows to meet 22% of total electricity demand in California, 50% of PV generation will be curtailed due to over-generation during periods of high solar resource or low electricity demand [5]. Not only would this be a lost opportunity to use available clean electricity, but it would also increase the effective cost of wind and solar, making them less economical and limiting their
growth. In addition to this curtailment issue, the variability of PV and wind also create secondary effects on the rest of the grid. When PV and wind output fall, other generators must increase their power output with high ramp rates, and must turn on and off more frequently. This reduces efficiency and increases the cost of grid operation, an effect that could prevent utilities from installing large amounts of PV and wind that otherwise would be cost effective [6, 7].

Several potential solutions may help to alleviate this issue and enable effective integration of clean electricity onto the grid. As a first low-cost solution, grid operators may vary output of what have historically been considered “baseload” generators, to reduce curtailment and satisfy a significant portion of ramping requirements [8]. Another contribution to the overall solution could be demand response: industrial and residential electricity consumers can use smart grid technologies to shift their demand to times when variable resources are available [9]. Finally, energy storage can be used to de-couple energy collection from electricity production. For grids with low levels of penetration of variable generation, short duration (<4 hour) storage may be sufficient to capture near-maximum value of PV and wind [8]. However, as grids shift to higher levels of penetration of variable generation, longer duration storage will become more valuable [10, 11]. To capture this growing value, many different methods of energy storage are being pursued.

As of May 2019, pumped hydro storage was by far the most widely deployed energy storage technology, making up 97.6% of globally operational energy storage capacity on a power basis and 98.8% on an energy basis [12]. However, pumped hydro storage (and similarly, compressed air energy storage) is geographically limited, and unlikely to provide significant additional storage in the future. Thus, attention has shifted to emerging technologies of electro-chemical, hydrogen, and thermal energy storage. Each of these options has certain advantages, and can be compared to each other for a given application on the basis of levelized cost of storage. In this comparison, electro-chemical storage, primarily with lithium ion batteries, has been proven as an effective and low-cost solution for short duration storage.
However, these batteries can become quite expensive for long duration storage due to a relatively high incremental storage cost of over 150 $/kWh [15, 16]. In this long duration storage application, or in some portable storage applications, hydrogen energy storage appears to be attractive, but is still being developed to achieve costs that will make the technology marketable [13, 17].

A final energy storage technology, thermal energy storage (TES), is the focus of the remainder of this thesis. Though it is sometimes overlooked as a grid storage option, TES currently provides more global energy storage capacity than any technology other than pumped hydro storage [12]. TES also has attractive prospects for the future, because of its low incremental storage cost of $\sim 15$ $$/\text{kWh}$$ [18]. That gives it an advantage for long durations of energy storage, and makes it particularly useful for the application of integrating high levels of variable renewable electricity onto the grid. Many different methods for employing TES exist, but here I focus on concentrating solar power, with detailed discussion in the following section.

1.2 Concentrating solar power

Concentrating Solar Power (CSP, also commonly referred to as “concentrated solar power” or “solar thermal power”) is a technology that converts sunlight to electricity indirectly, by collecting the sunlight as heat, then transferring that heat to a heat engine that converts the heat to electricity. The first step of sunlight collection is most often done with mirrors that track the sun and reflect sunlight onto a solar receiver. The mirrors may take several different forms, including parabolic troughs that focus sunlight on a receiver tube, parabolic dishes that focus sunlight onto a point receiver, or heliostats that focus sunlight to a receiver on top of a tower. Less commonly, Fresnel lenses may also be used to concentrate sunlight onto a receiver. Once sunlight is collected as heat at the solar receiver, the heat can be transferred to a power cycle that produces electricity immediately, or to a heat exchanger that transfers heat into a thermal storage system where the heat is stored before being transferred to a power cycle at a later time.
The first generation of CSP system designs were those that immediately transferred heat to a power cycle for electricity generation when sunlight was available. These systems directly competed with PV, whose power output also follows the solar resource. In this competition, PV had several advantages: high levels of investment, ability to use diffuse sunlight, and modularity, which allowed PV to target many different markets and use high throughput manufacturing techniques to achieve economies of scale. PV technology then progressed more rapidly and out-performed CSP, which had adapt and find another route to commercial deployment.

In the next phase of technology development, modern (generation 2) CSP designs incorporate thermal energy storage with molten salt as an energy storage material. In solar power tower designs, this salt (a mixture of nitrate salts) is pumped from a cold storage tank to a central receiver on top of a tower where it is heated by solar radiation from 290 to 560°C. The salt then returns to the ground and is delivered to a hot storage tank. Separate from this solar collection system, the salt can be pumped from the hot storage tank through a heat exchanger where it transfers heat to a steam Rankine cycle that converts thermal energy to electricity. The discharged salt then returns to the cold storage tank where it can be re-heated by solar radiation when it is available.

The addition of storage significantly reduced the cost of CSP systems, primarily because it increased the capacity factor of these power plants (the percentage of time that the plant runs at full power). More electricity can be generated with the same power block, while only adding incremental costs from other subsystems (the storage system, receiver, and heliostat field). Adding storage also significantly increased the value of CSP technology, by making it capable of shifting electricity production to times when the sun is not shining. For some electric grids, this can make CSP more than twice as valuable as PV [19]. This has encouraged a second wave of commercial development of CSP, with solar power towers currently either planned or under construction in Australia, Chile, China, Greece, Morocco, South Africa, and the United Arab Emirates [2, 20, 21].
While generation 2 CSP has achieved some recent commercial success, additional improvements could be made for further cost reductions or value additions. Third generation CSP is currently under development, with several competing technology pathways being considered [22]. The first of these, the molten salt pathway, is an incremental change from current generation 2 technology. By increasing operating temperature and shifting from steam Rankine to a supercritical CO$_2$ Brayton cycle for power generation, this pathway could increase conversion efficiency and thereby reduce system cost. For this pathway, a number of advancements are under development: power cycles, high-temperature salt materials for sensible thermal energy storage, and other high-temperature receiver and heat transfer components. Second, the falling particle pathway is based on a concept of concentrating sunlight onto small particles that perform a dual role in both receiving and storing sensible thermal energy. Packed-bed or fluidized-bed heat exchangers can then transfer heat from the particles to a power block. Finally, the third pathway uses a receiver with a gas phase heat transfer fluid at high temperature and high pressure, and may be combined with many different methods of energy storage, one of which is latent heat storage. The technology proposed in this thesis can be most closely associated with the third pathway, but represents its own unique approach as an addition to other pathways being considered.

1.3 Modular concentrating solar power

In the previous discussion, typical CSP designs have historically been quite large (>100 MW$_{electric}$) so that they can utilize high efficiency and low cost power blocks. However, several advantages could be gained by shifting to a modular (<1 MW$_{electric}$) system design. (1) Small scale systems can target market applications that are not available to conventional CSP. Half of microgrids in the United States require power levels of less than 1 MW$_{electric}$ [23], and a large CSP plant would be under-utilized and excessively costly for these applications. (2) Small scale results in low capital cost, which makes crowd-sourced community-based solar farms or other funding options attainable. (3) Low capital cost reduces the risk of financing new projects compared to the billions of dollars of capital required for conventional CSP.
Finally, with evidence from the success of PV, (4) rapid technology progress and (5) high efficiency manufacturing processes are possible through higher throughput manufacturing of small plants.

Though they have not yet led to significant commercial deployment, these advantages have inspired several efforts to develop modular CSP systems. These efforts can be divided into two categories, based on their method of solar concentration. First, several efforts have been made to incorporate TES with dish Stirling systems [24]. Early efforts were motivated by remote power for space applications, and more recent efforts have focused on terrestrial applications. These used a wide variety of technologies for each system component: thermoelectric generators, Rankine cycles, and Stirling engines for power conversion; salt, sugar, or metal thermal storage materials; and pumped oils, heat pipes, and pool boilers for heat exchangers. Notably, Temple University and Infinia Corporation built a latent heat storage system using NaF-NaCl as a thermal storage material and a Stirling engine for electricity generation. They demonstrated some initial success, but experienced dry-out issues with their heat pipe design [25, 26]. Sandia National Laboratories also investigated latent heat storage with Dish Stirling [27, 28]. They found potential for cost-competitiveness, but required further work on flexible high temperature heat transfer pipes and material compatibility between their copper-magnesium-silicon thermal storage material and candidate containment materials.

In addition to efforts to incorporate TES with a dish concentrator, modular solar power towers have also been proposed by several companies and research groups. At a pilot plant scale, AORA developed a design called the “Solar Tulip” (Figure 1.1(a)), which collected sunlight in a compressed air receiver and did not incorporate any energy storage but hybridized solar heat input with hydrocarbon fuels for continuous production of electricity and heat from a micro-turbine. Graphite Energy (Figure 1.1(b)) developed an alternative concept based on sensible heating of graphite blocks held on top of a small scale solar power tower. Vast Solar (Figure 1.1(c)) also created a small power tower design, utilizing a sodium
receiver. Unlike AORA and Graphite Energy, this design located thermal energy storage and power conversion (a steam turbine) components on the ground. Another company still developing at a research level (design announced in 2018), Azelio (Figure 1.1(d)), designed a system with latent heat storage in an aluminum-silicon PCM, a sodium heat transfer fluid, and a Stirling engine for power conversion, with all components on top of the tower. While none of these companies have reached extensive commercial success, they provided technology advancements and significant motivation for continued development of modular solar power towers, as I have also pursued in this thesis.

![Previous small power tower designs include a variety of different solar receiver, thermal storage, and power generation methods. (a) Solar Tulip. (b) Graphite Energy. (c) Vast Solar. (d) Azelio. [29–32]](image)
1.4 Heat engines

Though not the primary focus of this thesis, the heat engine that converts heat to electricity (often referred to as the power block) is a critical component of a CSP system. Because its efficiency affects the size of all other system components, it has more influence on overall system performance and cost than any other subsystem. This means that it must be well understood when designing a thermal energy storage system for CSP, and to provide relevant background information, I further discuss heat engines in this section.

Fundamentally, heat engines are governed by the thermodynamic limit of Carnot efficiency, $\eta_{\text{carnot}}$:

$$\eta_{\text{carnot}} = 1 - \frac{T_{\text{cold}}}{T_{\text{hot}}}$$

(1.1)

where $T_{\text{cold}}$ is the temperature of the cold side of the heat engine, and $T_{\text{hot}}$ is the temperature of the hot side of the heat engine. In practice, the efficiency of heat engines is usually closer to 75% of Carnot efficiency [33], and is also often close to what is referred to as the Chambadal-Novikov limit. This limit (sometimes incorrectly attributed to Curzon and Ahlborn) is based on endoreversible thermodynamics and considers exergy destruction during heat transfer to and from the heat engine, resulting in a predicted efficiency of :

$$\eta_{\text{Chambadal-Novikov}} = 1 - \sqrt{\frac{T_{\text{cold}}}{T_{\text{hot}}}}$$

(1.2)

To apply these limits to a hypothetical scenario, we may consider a system that has a hot reservoir temperature of 550°C, and a cold reservoir temperature of 35°C. The 75% of Carnot efficiency rule-of-thumb predicts that this system could reach 37% efficiency, and the Chambadal-Novikov efficiency limit for this system is 39%. Both of these predictions are close to the efficiency of a practical heat engine with these operating conditions, and can be extended to other temperatures to determine approximate potential efficiency of a power block for a given operation temperature of a thermal storage system. This reveals that high temperature storage can lead to high thermal-to-electric conversion efficiency, which is
very important for overall system success. However, higher temperature operation can also require more expensive materials for the storage system and receiver, which is an important trade-off in overall system design.

Another important consideration when designing thermal energy storage systems is how heat is input to the thermodynamic cycle of a heat engine. Some heat engines operate with isothermal heat input and output (e.g. Stirling), while others use a working fluid that changes temperature during the cycle (e.g. Rankine, Brayton). For maximum exergetic efficiency, isothermal heat engines should be combined with isothermal energy storage. Similarly, heat engines with temperature changes are most effectively combined with thermal storage systems that use sensible heat. In any other combination, there will be a large temperature drop from the thermal storage to the working fluid of the power cycle. This exergy destruction during heat transfer represents a lost opportunity to produce useful work (i.e. electricity output). However, it is also important to note that ideal heat transfer performance is not necessarily required to develop a cost-effective system, so the rule of matching heat input to the thermodynamic cycle is not without exception.

Also relevant to ongoing development of CSP systems, we may consider the commercial maturity of different heat engine technologies. The most commonly used power generation cycle is the steam Rankine cycle, and as a result all components of this cycle have been heavily optimized to reach overall efficiencies of 35-45%. In CSP systems, heat is input to the steam Rankine cycle at temperatures of 250-550°C, which is primarily dependent on the operating temperature range of the nitrate salt material used for thermal energy storage. At lower temperatures, organic working fluids can be used as an alternative to steam, and organic Rankine cycles have also reached maturity in commercial development for applications such as geothermal power. Another cycle in which the working fluid changes temperature during heat input is the Brayton cycle. Though less mature than the Rankine cycle, the Brayton cycle has received recent interest, in particular with use of supercritical CO₂ as a working fluid, for its potential to reach 50% conversion efficiency [34].
Several heat engines that operate with isothermal heat input may also be considered for use with CSP systems. Stirling engines are the most mature of these options, due to previous development for dish Stirling applications. These have reached an efficiency of 40%, and have been used at temperatures over 700\(^\circ\)C [35]. Other emerging heat engines that use isothermal heat input include thermophotovoltaics and thermo-acoustic, thermoelectric, and thermionic generators. These technologies are all still under development, with lower efficiencies and higher costs compared to options previously discussed, due to limited commercial maturity. However, continued research and progress could allow any of these technologies to become viable options for use in CSP systems in the future.

### 1.5 Thermal energy storage

Both specific to CSP and general to TES for other applications, thermal energy may be stored in three different forms: sensible heat storage (energy is stored by increasing the temperature of a storage material), latent heat storage (energy is stored in the phase change of a material at a constant temperature), or thermochemical heat storage (energy is stored via a chemical reaction).

Sensible heat storage is the most mature technology for use with CSP, and involves storing energy in a material that increases in temperature as it receives heat input. Historically, sensible heat storage been done with sodium-potassium nitrate salt as a thermal storage material. This salt melts at 220\(^\circ\)C, and begins to degrade slightly below 600\(^\circ\)C, meaning that its operating temperature range is typically 250-550\(^\circ\)C [22]. The salt also serves as a heat transfer fluid in the solar receiver, which reduces system complexity but also has some disadvantages. If the material solidifies, it expands, which can lead to failure of pipes and difficulty in re-starting the system. Therefore, these systems use heat tracing to ensure that the salt never solidifies during operation. Alternative to molten salts, solid materials have also been considered for sensible heat storage, including ceramic particles and other inexpensive materials such as rocks. Solid materials cannot be pumped, but can be physically moved by other means, or can be placed in fixed beds while a fluid (gas or liquid) is used.
to transfer heat into or out of the solid material. This creates potential for very low cost systems, though also lower energy density than alternative thermal storage approaches, which increases the cost of other system components (such as heat exchangers and storage tanks) [22].

Latent heat storage, as an alternative to conventional sensible heat storage, has not been deployed in any commercial CSP systems but is an attractive primarily for its high energy density and potential for low overall storage system cost. This has led to recent interest in international research and commercial development, and provides the basic motivation for focus on latent heat storage in the rest of this thesis.

1.6 Latent heat storage

As discussed previously, latent heat thermal energy storage (LHTES) stores energy in the isothermal phase change of a material. This provides the advantage of high energy density, which leads to a potential for low cost thermal energy storage systems. The most common phase change considered for these systems is between solid and liquid phases, because of its high energy density, though solid-solid phase change storage may also be considered in some applications. Storing energy in a phase change from liquid to gas is generally not feasible for large scale energy storage systems, because this either has very low storage density or poses the difficulty of very high pressure in the gas phase.

A key attribute of latent heat storage systems is isothermal operation, which as previously discussed, makes it ideal for applications that involve heat engines that receive isothermal heat input. However, phase change can also happen over a range of temperatures, leading to the possibility of combining latent and sensible heat storage for use with other heat engines as well. For a more detailed discussion, consider Figure 1.2. This figure provides a phase diagram of the aluminum-silicon system, with increasing silicon content from left to right. This phase diagram describes what phases are thermodynamically stable for a given composition and temperature. At the far left of the diagram, below 660°C, is a single phase region of nearly pure solid aluminum. Depending on the temperature, a small amount of
silicon may be incorporated without changing the phase, though saturation and movement into two-phase regions occurs with relatively small amounts of silicon. Similarly, at the far right of the diagram, below 1414°C, is a single phase region of nearly pure solid silicon. From here, even less aluminum may be added before moving into a two-phase region, and so the width of this single phase region is very small (less than the width of the lines in the diagram). The final single phase region in this phase diagram is the liquid region at the top, which exists at all compositions but at varying temperatures depending on composition. All other regions in the diagram represent mixtures of multiple phases. At the bottom of the diagram, we have solid Al-Si, where each location in this region represents a mixture of solid aluminum and silicon phases. Above this are regions of solid aluminum with liquid aluminum-silicon (left), and solid silicon with liquid aluminum-silicon (right).

In a typical latent heat storage system, the eutectic composition is used, such that phase change occurs at a single temperature. In the case of aluminum-silicon, the eutectic composition is 12% silicon, as shown in green in Figure 1.2. Starting in the liquid phase region at this composition and decreasing in temperature, we reach the eutectic point, where three phases are in equilibrium: solid aluminum (with a small amount of silicon), solid silicon (with a small amount of aluminum), and liquid aluminum-silicon. To decrease temperature from this point, a significant amount of energy must be removed from the material. If this is done, solid aluminum and solid silicon simultaneously form, with a structure dependent on how the material is cooled (most common is a lamellar structure). The composition of each component of the solid aluminum-silicon material is then given by the purple lines in Figure 1.2, which follow the saturation lines for the two single phase regions as cooling continues. However, during typical operation, the temperature will remain near the phase change temperature, such that heat can be input and removed with high exergetic efficiency in combination with an isothermal heat engine.

The above discussion is pertinent because eutectic compositions are the most commonly considered for phase change materials for latent heat storage. However, off-eutectic com-
Figure 1.2: Aluminum-Silicon phase diagram. If the eutectic composition (12% silicon, in green) is used for latent heat storage, phase change will occur at a single temperature of 577°C. However, if an off-eutectic composition is used, phase change occurs over a temperature range (for the blue composition with 27% silicon, this range is from 577−~800°C), which could be desirable for certain applications.
positions could also be used for non-isothermal operation, as shown by the blue line that represents aluminum-27% silicon (an arbitrary composition) in Figure 1.2. In this case, starting in the liquid phase region and decreasing temperature, we reach a two-phase region of solid silicon and liquid aluminum-silicon when we intersect the liquidus line near 800°C. At this point, regions of solid silicon begin to form within the aluminum-silicon liquid. Removing more heat leads to further temperature reduction, as more solid silicon forms and the liquid aluminum-silicon becomes more aluminum-rich. This continues until the temperature reaches the eutectic phase change temperature of 577°C. Here, solid aluminum begins to form, in addition to the other two phases in equilibrium (solid silicon and liquid aluminum-silicon with 12% silicon). Cooling beyond this point, we enter the two phase solid region, which has a higher concentration of solid silicon than in the case of the eutectic composition.

Considering the full phase transition from liquid aluminum-silicon to the solid solution of aluminum and silicon, this discussion illustrates how choosing an off-eutectic composition allows for combination of both latent and sensible heat storage in a single material, and interestingly shows that phase change can occur over a range of temperatures, which could be attractive for implementation in a variety of systems.

While the above analysis considers different ways that phase change material could be deployed, in the rest of this thesis, I focus on systems that use isothermal heat engines, and therefore on the first case that uses eutectic phase change material compositions. For this application, several reviews provide abundant data for phase change material properties [36–40]. Among available materials, salts and metals have received the most attention, but each have their own advantages and challenges. Salts can be low cost materials, but have low thermal conductivity, which increases the cost of heat exchangers and has encouraged research on advanced heat exchanger designs and development of other thermal conductivity enhancements [38, 39, 41–43]. Metals solve this issue because they have high thermal conductivity, but generally have higher material cost and are very corrosive: no high temperature metal phase change material systems have been demonstrated with enough reliability for
long-term operation. For both salts and metals, the solid-liquid phase change also presents
difficulty for heat exchanger development, because the conventional method of pumping a
sensible liquid storage material cannot be used. Thus, several alternative solutions have
been experimentally explored, including tube-in-tank heat exchangers using pumped fluids
[44–46], encapsulated PCMs [47–49], pool boilers [50, 51], and heat pipes [24, 25, 27, 28, 52–
56]. However, both material development and system design centered around latent heat
storage are under significant development, leaving a significant opportunity for technology
advancement in this thesis.

1.7 Scientific advancements from this thesis

This thesis has contributed to the advancement of latent heat storage for concentrating
solar power in a variety of ways:

- A broad contribution to the field of CSP is given in Chapter 2. In this paper, I
developed a new methodology for evaluating phase change materials for latent heat
storage. My holistic modeling method took in critical PCM properties as inputs,
and used these to calculate LCOE of a solar power tower. I applied the model to
hypothetical materials to quantify the value of each material property, then to real
candidate materials to identify some of the most promising candidates that should be
considered for future development, and also considered the influence of heat exchanger
design on overall system success. Importantly, this model was adaptable to many
different CSP design configurations, making it widely useful for researchers working on
variations of solar power towers.

- Narrowing down from this broad perspective, I then completed performance modeling
and techno-economic analysis for a new specific concept for concentrating solar power
in Chapter 3. This system design combined latent heat storage, a valved thermosyphon,
and a heat engine all on top of a modular solar power tower. My performance analysis
predicted a potential for high exergetic efficiency, and an annual system efficiency of
18%. Most importantly, my techno-economic analysis demonstrated that the concept had potential for cost-competitiveness with alternative peaking plant technologies.

- In Chapter 4, I built a first prototype system to prove that the above-modeled concept would work in practice. The positive results of this prototype included minimal temperature gradients within the PCM, effective on/off control with high exergetic efficiency in the valved thermosyphon, and effective integration of subsystems with no degradation in performance at interfaces.

- A second prototype system, presented in Chapter 5, transitioned from thermoelectric generators to a Stirling engine for electricity generation, and also improved heater design and insulation for thermal management. With these changes, I demonstrated 18.5% efficiency in converting stored heat to electricity, and 8.6% system efficiency on a daily basis. This was a $\sim200\times$ improvement over the first prototype, and higher efficiency than any previously demonstrated latent heat storage system combined with electricity generation.

- In Chapter 6, I performed experiments to screen potential containment materials for Al-Si as a PCM. I helped evaluate thermodynamic and mechanical stability of metals, ceramics, and coatings when exposed to a variety of conditions on a variety of surface geometries. We found that plasma-sprayed ceramic coatings were effective for a short duration or under constant temperature conditions, but produced mixed results when exposed to thermal cycles. We were able to use these coatings for initial prototype experiments, but long-term reliability is still a necessary development for future work.
CHAPTER 2
GUIDELINES FOR PHASE CHANGE MATERIAL SELECTION BASED ON A
HOLISTIC SYSTEM MODEL

Jonathan E Rea,1,2 Eric S Toberer,2 Nathan P Siegel1

The idea for this paper arose from my frustration as I read the existing literature on PCMs for CSP. Researchers working in this field appeared to be approaching their material selection from a very narrow view. They looked at a confined class of materials, down-selected to a desired temperature range based on their chosen power block, then picked a material with what seemed like high energy density because that would have supposed performance and cost benefits. Some papers would make qualitative comments and would list important material properties, but still did not perform any quantitative analysis to truly determine the best material for their system.

One significant take-away from this thesis is the importance of holistic system modeling. The design or performance of one component of a system can have a large influence on other components, and thus when designing complex systems, it is very important to consider the entire system as a whole. To apply this lesson to PCM selection, in this paper I built a new model to evaluate PCMs for their full impact on LCOE of a CSP plant. This allowed me to quantify the value of each PCM property, identify leading candidate materials, and illustrate my point about the importance of holistic system modeling. I intentionally made my model as simple as possible, and added detailed discussion of trade-offs between material selection and full system design, in the hope that this paper would provide a useful guide to a wide audience in future research into PCMs for CSP.

1Designed model, performed calculations
2Wrote manuscript
2.1 Abstract

Concentrating solar power (CSP) has potential to increase the amount of renewable energy on electric grids and reduce global carbon emissions, in particular because of its capability to incorporate inexpensive thermal energy storage. To realize this potential, development of latent heat storage with phase change materials (PCMs) is attractive because of its high energy density and low material cost. Research in this area has identified many promising materials, and has developed effective storage system designs that could be used to deploy PCMs. However, the field has lacked a rigorous methodology to select the optimal material for a particular CSP system. To address this need, here we present a holistic model that evaluates all important influences of PCM properties on the performance and cost of an entire CSP plant. We apply this model to over 100 of the most widely considered candidate materials, including both salts and metals. We quantify the relative value of each PCM property, and identify leading materials for application in CSP systems. This method and our results may aid future efforts to develop latent heat storage by providing a more informed view of how material selection impacts an entire CSP system.

2.2 Introduction

Compared to other carbon-free electricity generation technologies, concentrating solar power (CSP) can provide very high-value electricity due to its dispatchability. For some electric grids, CSP has been found to be more than twice as valuable as photovoltaics [19]. This advantage of CSP comes from its ability to incorporate inexpensive thermal energy storage, which currently costs near 20 $/kWh, and may reach future costs of below 15 $/kWh [18]. Based on these low storage system costs, the United States Sunshot program has set targets for levelized cost of electricity (LCOE) at 5 ¢/kWh for “baseload” CSP plants and 10 ¢/kWh for “peaking” CSP plants [57]. If successful, these targets could allow CSP to compete with other electricity generation or storage technologies, whose levelized costs of storage are likely to remain near or above 10 ¢/kWh when operated for purposes of peaker
To achieve Sunshot cost targets, three different technology pathways are being pursued for the next generation of CSP [22]. The first is an extension of modern state-of-the-art technology, which uses molten salt as both a heat transfer fluid and storage media, and a steam Rankine cycle to convert heat to electricity. To improve upon this technology, a transition to higher temperatures (requiring new materials) and a super-critical CO$_2$ Brayton cycle could increase conversion efficiency and therefore reduce system costs. In a second CSP technology pathway, solid particles are used both to collect and store heat. This technology has not yet been demonstrated commercially, but appears to have potential for future success [59]. The third CSP technology pathway uses a gas-phase receiver, and may incorporate a variety of methods for energy storage and power generation. Here, latent heat storage with phase change materials (PCMs) has received attention due to its high energy density and resulting potential for low material and storage system cost. This type of system with latent heat storage would be best matched to an isothermal heat engine for electricity generation, but cascaded systems with multiple PCMs at different temperatures could also be implemented with other power cycles (such as the super-critical CO$_2$ Brayton cycle) with high exergetic efficiency [60, 61].

Within latent heat storage research, many previous studies have established material properties for potential PCMs [36–40]. Other studies have investigated methods for effectively integrating PCMs into storage systems. Heat exchanger design is critical, and several concepts have been proposed that use heat pipes [24], graphite or metal foams [62, 63], or encapsulation [48, 49, 64] to enhance the thermal conductivity of PCMs or reduce heat exchanger costs. Also critical are reliable containment systems that can ensure long system lifetime. As such, additional studies have explored material compatibility between PCMs and containment materials [40, 65–67].

While this previous work has effectively identified some potential materials and storage system designs, what the field lacks is a holistic methodology for selecting the optimal PCM
for a particular system. Previous efforts have generally selected candidate materials based on limited comparisons of some, but not all important material properties [68, 69]. Several studies have extended their analysis to the entire storage system [70, 71], but still did not consider the influence of PCM selection on the rest of the CSP plant. Wei et al [72] provided a comprehensive review of this field of research, but importantly identified the lack of an integrated study for PCM selection that considers all properties and their influence on the entire system.

Ideally, material selection would be based on a holistic model that optimizes a detailed system design to minimize LCOE. This model would be applied to each available material to make an even comparison between options. However, detailed system design is costly, takes a long time to develop, and may be influenced by the material selection itself. As an alternative solution, in this paper we present a simpler modeling methodology that is computationally inexpensive but still considers the influence of all material properties on the most important overall system metric, LCOE. This provides rapid insight regarding the value of different material properties, and can be used to improve selection of a PCM for CSP.

In Section 2.3 of this paper, we qualitatively discuss the critical properties of PCMs, and their general influence on overall CSP system performance and cost. In Section 2.4, we establish a performance and cost model for a solar power tower that uses latent heat storage. We then execute this model in Section 2.5 with hypothetical ranges for PCM properties, in order to quantify the influence of each property on the overall system LCOE. Finally, in Section 2.6, we apply our model to real materials, in order to identify leading candidate materials and further investigate the effects of material selection on CSP systems.

2.3 Critical phase change material properties

When selecting phase change materials for concentrating solar power applications, several material properties are critical for successful system design and minimization of overall system cost. These properties, and their influence on the overall system, are:
• **Melting temperature.** The melting temperature of the PCM must match the desired temperature of the system, including the power block, receiver, and other components. One trade-off is that higher temperature operation improves thermal-to-electric conversion efficiency, but reduces receiver and storage efficiencies (or increases cost of mitigating losses). High temperatures can also be cost-prohibitive because they require expensive high-temperature materials for containment or heat transfer components.

• **Material cost.** The storage material can be a significant portion of the storage cost, and many materials are too expensive for a successful overall system, regardless of their thermal properties.

• **Energy density** primarily on a volumetric basis, though gravimetric energy density may be an important consideration in some cases (for example, if storage is located on top of a tower [73]). A high energy density means that a smaller volume of storage material is required; this reduces both containment and heat exchanger costs.

• **Thermal conductivity.** A high thermal conductivity material requires less heat exchanger surface area or leads to a smaller temperature drop through the PCM while charging or discharging the storage system. This reduces heat exchanger cost or improves power block and receiver efficiency.

• **Material compatibility.** The cost of containment can be a primary driver of storage cost, if expensive coatings must be applied or specialty materials must be used to prevent corrosion. Some materials have no existing solution for long-term reliable containment.

A wide array of materials have been studied for latent heat thermal energy storage, though within the temperature range typically considered for concentrating solar power, the two primary classes of materials are salts and metals. Figure 2.1 shows the critical properties for a near comprehensive list of these materials in a temperature range between $\sim 300$-1500
Figure 2.1: Critical thermal properties for salts and metals that may be considered as PCMs for CSP. Filled circles indicate materials with thermal conductivity above 10 W/mK (metals), and open circles represent materials with thermal conductivity below 10 W/mK (salts, mostly below 2 W/mK). Within these two material classes, a large number of materials are available with a wide range of thermal properties. However, the relative value of each property is difficult to quantify, and the trade-offs between different material choices are not evident from simply comparing these properties. Thus, a cursory look at these properties is insufficient to identify the best material for a specific application.
˚C [36]-[40],[74]-[75]. For salts, we include most materials with cations of Li, Be, Na, Mg, K, Ca, Rb, Sr, Cs, and Ba, and anions of hydride, hydroxide, carbonate, nitrate, phosphate, sulfate, fluoride, chloride, bromide, and iodide. For metals, we include Be, Mg, Al, Si, Ca, Mn, Fe, Co, Ni, Cu, Zn, Ge, As, Sr, Ag, Sb, Te, Au, and Pb. For both salts and metals, we also include many binary eutectic mixtures of these materials, and select ternary eutectics as well.

Figure 2.1 shows that there are a large number of candidate materials available for consideration, with a wide range of thermal properties. Using this data, a simple approach to selecting a material for a CSP application would be to constrain the search to a desired temperature range, then choose a material that appears to have a good combination of low material cost and high energy density. However, this method will not necessarily choose the best material, because it does not quantify the relative value of each property, or consider the influence of all material properties on the overall performance and cost of the system. To make a fully informed material selection requires development of a holistic system model that takes all material properties as inputs and determines levelized cost of electricity (LCOE) as an output. That is what we aim to provide in this paper: a simple full system model that can be used to improve searches for PCMs for CSP.

2.4 Holistic system model

In this section, we develop a system-level performance and cost model of a solar power tower plant that incorporates latent heat thermal energy storage with a single PCM. Model inputs include PCM properties and several performance and cost assumptions; the key output is LCOE. The model is simple enough to be easily reproduced or modified, but sufficiently refined such that potential PCMs may be compared and each critical PCM property may be evaluated for its influence on overall system performance and cost.
2.4.1 LCOE

The LCOE of the solar power tower plant is given in Equation 2.1:

\[
LCOE = \frac{C_{\text{total}} CRF + FO&M}{8760 CF} + VO&M
\]  

(2.1)

where \(C_{\text{total}}\) is the total system capital cost, \(CRF\) is the capital recovery factor, \(FO&M\) is the fixed operation and maintenance cost, \(CF\) is the capacity factor, and \(VO&M\) is the variable operation and maintenance cost. In this model, \(CF\) is taken as an assumption, and \(CRF\) is calculated as

\[
CRF = \frac{i (1 + i)^n}{(1 + i)^n + 1}
\]

(2.2)

where \(i\) is the annual interest rate and \(n\) is the number of years that the plant operates. The remaining terms in Equation 2.1 (\(C_{\text{total}}\), \(FO&M\), and \(VO&M\)) depend on several more assumptions, as well as the influence of PCM properties on performance and cost of many components. We model and discuss these effects in the following subsections.

2.4.2 Total system capital cost

The total system capital cost, \(C_{\text{total}}\), is the sum of several subsystem costs, scaled by the design power output of the plant, \(\dot{W}_e\), and a capital cost factor, \(CCF\), that includes tax, engineering, procurement, and contingency costs:

\[
C_{\text{total}} = \frac{(C_{\text{storage}} + C_{\text{PB}} + C_{\text{collection}})}{\dot{W}_e} CCF.
\]

(2.3)

In Equation 2.3, \(C_{\text{storage}}\) is the cost of the storage system, \(C_{\text{PB}}\) is the cost of the power block, and \(C_{\text{collection}}\) is the cost of the solar collection system. The solar collection system cost is divided into several components in Equation 2.4:

\[
C_{\text{collection}} = C_{\text{land}} + C_{\text{field}} + C_{\text{tower}} + C_r
\]

(2.4)

where \(C_{\text{land}}\) is the cost of the land where the CSP plant is placed, \(C_{\text{field}}\) is the cost of the heliostat field, \(C_{\text{tower}}\) is the cost of the tower, and \(C_r\) is the cost of the receiver. Similarly,
storage cost is divided into its component costs in Equation 2.5:

\[ C_{\text{storage}} = C_{\text{PCM}} + C_{\text{HX}} + C_{\text{tank}} \] (2.5)

where \( C_{\text{PCM}} \), \( C_{\text{HX}} \), and \( C_{\text{tank}} \) represent the costs of the PCM material, heat exchanger, and storage tank(s), respectively.

### 2.4.3 PCM cost

The cost of the PCM is simply the specific cost of the material, \( c_{\text{PCM}} \), in units of $/kWh, scaled by the amount of thermal storage, \( E_{\text{storage}} \), in units of kWh:

\[ C_{\text{PCM}} = c_{\text{PCM}} E_{\text{storage}}. \] (2.6)

The amount of thermal storage in the plant can be estimated from the design thermal power of the power block, \( \dot{Q}_{PB} \), and the duration of storage, \( d_{\text{storage}} \):

\[ E_{\text{storage}} = \dot{Q}_{PB} d_{\text{storage}} \] (2.7)

where \( \dot{Q}_{PB} \) depends on the design power output \( \dot{W}_e \) and efficiency \( \eta_{PB} \) of the power block:

\[ \dot{Q}_{PB} = \frac{\dot{W}_e}{\eta_{PB}}. \] (2.8)

In this estimate of \( E_{\text{storage}} \), we have neglected any losses from the thermal storage system during discharge, which would typically be very small (~1%) for CSP systems.

For most power block technologies, efficiency is relatively constant as a percentage of Carnot efficiency, \( f_{PB} \) [33]. This allows the power block efficiency to be described as a function of its hot and cold side temperatures, \( T_h \) and \( T_c \):

\[ \eta_{PB} = f_{PB} \left(1 - \frac{T_c}{T_h}\right). \] (2.9)

The hot side temperature of the engine depends on the melting temperature of the PCM, \( T_m \), the temperature drop through the PCM during discharge, \( \Delta T_{\text{PCM,d}} \), and the temperature drop through the heat exchanger between the PCM and the power block during discharge, \( \Delta T_{\text{HX}} \):

\[ T_h = T_m - \Delta T_{\text{HX}} - \Delta T_{\text{PCM,d}}. \] (2.10)
Similarly, the cold side temperature of the engine depends on the ambient temperature, $T_{amb}$, and the temperature difference from the heat exchanger at the cold side of the power block, $\Delta T_{HX, rej}$:

$$T_c = T_{amb} + \Delta T_{HX, rej}.$$  

(2.11)

Combining Equations 2.6 - 2.11, the cost of the PCM is then

$$C_{PCM} = c_{PCM} \frac{\dot{W}_d storage}{f_{PB} \left(1 - \frac{T_{amb} + \Delta T_{HX, rej}}{T_m - \Delta T_{HX} - \Delta T_{PCM,d}}\right)}.$$  

(2.12)

In Equation 2.12, most parameters are input as assumptions, with the exception of the PCM properties that are unique to each material and $\Delta T_{PCM,d}$, which we optimize for each scenario and discuss in Subsection 2.4.11. Then, Equation 2.12 shows that the cost of the PCM depends not only on specific material cost, but also on melting temperature and the temperature drop through the PCM during discharge, which is influenced by the thermal conductivity of the material. This illustrates the importance of considering the influence of all PCM properties on the full system, rather than limiting our study for PCM selection to just the candidate material itself or the storage subsystem.

### 2.4.4 Heat exchanger cost

In this model, we estimate the cost of the heat exchanger, $C_{HX}$, based on the mass of heat exchanger, $m_{HX}$, the heat exchanger material cost, $c_{HX,mat}$, and a manufacturing cost factor, $c_{HX,mfg}$:

$$C_{HX} = m_{HX} c_{HX,mat} c_{HX,mfg}.$$  

(2.13)

where heat exchanger mass can be determined from the volume, porosity, and density of the heat exchanger, $(V_{HX}, p_{HX},$ and $\rho_{HX},$ respectively):

$$m_{HX} = V_{HX}(1 - p_{HX})\rho_{HX}.$$  

(2.14)
To determine the volume of the heat exchanger, we introduce a definition for the area density (or surface area to volume ratio) of the heat exchanger, $\alpha_{HX}$, with Equation 2.15:

$$V_{HX} = \frac{A_{HX}}{\alpha_{HX}}$$

(2.15)

The final term needed to estimate heat exchanger cost is the required surface area of the heat exchanger, $A_{HX}$. For a given temperature drop through the PCM during discharge, this can be derived with an assumption that all heat exchanger surfaces are equidistant from each other. The temperature drop is proportional to the heat transfer rate through the PCM to the power block, $\dot{Q}_{PB}$, and the thermal resistance through the PCM, $R_{PCM}$:

$$\Delta T_{PCM,d} = \dot{Q}_{PB}R_{PCM}.$$  

(2.16)

The thermal resistance through the PCM is

$$R_{PCM} = \frac{L_{PCM}}{\kappa_{PCM}A_{HX}}$$

(2.17)

where $\kappa_{PCM}$ is the thermal conductivity of the PCM and $L_{PCM}$, the characteristic length for conduction through the PCM, is halfway between heat exchanger surfaces:

$$L_{PCM} = \frac{V_{PCM}}{2A_{HX}}.$$  

(2.18)

The volume of the PCM, $V_{PCM}$, can be calculated as the ratio of the amount of energy storage in the plant to the volumetric energy density of the PCM, $u_{PCM}$:

$$V_{PCM} = \frac{E_{storage}}{u_{PCM}}.$$  

(2.19)

Combining Equations 2.7 and 2.16-2.19, and rearranging, the required surface area of the heat exchanger is:

$$A_{HX} = \dot{Q}_{PB}\sqrt{\frac{d_{storage}}{2\kappa_{PCM}u_{PCM}\Delta T_{PCM,d}}}.$$  

(2.20)
Then, combining Equation 2.20 with Equations 2.8 and 2.13-2.15, the cost of the heat exchanger is:

\[
C_{HX} = \left( \frac{\dot{W}_e}{\eta_{PB} \alpha_{HX}} \sqrt{\frac{d_{storage}}{2 \kappa_{PCM} u_{PCM} \Delta T_{PCM,d}}} \right) \left( \rho_{HX} (1 - p_{HX}) c_{HX_{mat}} c_{HX_{mfg}} \right) 
\] (2.21)

Our heat exchanger cost model in Equation 2.21 is based on two critical features of the heat exchanger: area density (or surface area to volume ratio) \( \alpha_{HX} \), and porosity \( p_{HX} \). This general treatment allows the model to be applied to a wide range of heat exchanger designs, but still retains two important influences of PCM properties on system cost. First, high PCM thermal conductivity allows heat exchanger surfaces to be placed further apart, leading to a lower heat exchanger cost. Second, both high thermal conductivity and high energy density reduce the required heat exchanger volume, which reduces tank size and therefore tank cost. We also note that the input of \( \Delta T_{PCM,d} \) strongly influences the required surface area and resulting cost of the heat exchanger, but also influences the temperature of the solar receiver and the temperature of the heat delivered to the power block. To make an even comparison among scenarios with different PCMs, our model optimizes this input as discussed in Subsection 2.4.11.

### 2.4.5 Tank cost

For this analysis, we determine tank cost based on the internal volume of the tank and the specific tank cost on a per-volume basis, \( c_{tank} \). The total volume inside the tank is the sum of the volumes of the PCM and heat exchanger, so:

\[
C_{tank} = (V_{HX} + V_{PCM}) c_{tank} 
\] (2.22)

where the volumes of the PCM and heat exchanger were previously determined in Equations 2.15 and 2.19. Inspection of these equations shows that high PCM volumetric energy density and high PCM thermal conductivity lead to low tank costs. Thus, similar to our general treatment of the heat exchanger, this simple model is adaptable to different tank designs,
but still retains the important influences of the PCM properties on system cost.

In addition to this tank cost, our model also includes the cost of insulation that surrounds the tank, to quantify the impact of tank size and PCM temperature on both insulation cost and storage efficiency. This insulation cost, $C_{\text{insulation}}$, is based on the volume of insulation, $V_{\text{insulation}}$, and the specific cost of the insulation on a volumetric basis $c_{\text{insulation}}$:

$$C_{\text{insulation}} = V_{\text{insulation}}c_{\text{insulation}} \tag{2.23}$$

Assuming all sides and the top and bottom of the storage tank are insulated, the volume of insulation is:

$$V_{\text{insulation}} = \pi \frac{h_{\text{tank}}^2}{4} \left( f_{\text{ins}}^3 - 1 \right) \tag{2.24}$$

where $h_{\text{tank}}$ is the tank height and $f_{\text{ins}}$ is the non-dimensional ratio of the outer radius of the insulation to the outer radius of the tank. To determine $h_{\text{tank}}$, we assume that tank height is equal to tank diameter (which minimizes surface area for heat losses), such that:

$$h_{\text{tank}} = \left[ \frac{4}{\pi} \left( V_{\text{PCM}} + V_{\text{HX}} \right) \right]^{1/3} \tag{2.25}$$

Then, using Equations 2.23-2.25 and previously calculated $V_{\text{HX}}$ and $V_{\text{PCM}}$, our model estimates the cost of insulation and includes this with tank cost.

### 2.4.6 Power block cost

In most cost models, power block cost is expressed on a cost-per-watt-electric basis. From an alternative perspective, a power block may be designed for a thermal power rating, and its power output depends on its efficiency, which in turn depends on its hot and cold side temperatures. In this method, we have:

$$C_{PB} = \dot{Q}_{PB}c_{PB} \tag{2.26}$$
where $c_{PB}$ is the power block cost on a per-watt-thermal basis. Combining Equation 2.26 with previous Equations 2.8-2.11, we then have:

$$C_{PB} = \frac{\dot{W}_e}{f_{PB} \left(1 - \frac{T_{amb} + \Delta T_{HX, rej}}{T_m - \Delta T_{HX} - \Delta T_{PCM,d}}\right)} c_{PB}. \tag{2.27}$$

In Equation 2.27, our model quantifies the impact of two PCM properties on power block cost: (1) higher melting temperature leads to higher efficiency and therefore lower cost, and (2) higher thermal conductivity reduces the temperature drop through the PCM during discharge, which increases the temperature delivered to the power block, thereby again increasing efficiency and reducing system costs.

### 2.4.7 Receiver cost

To determine receiver cost, we adopt the cost model used in the System Advisor Model developed by the National Renewable Energy Lab [76]:

$$C_r = C_{r, ref} \left(\frac{A_r}{A_{r, ref}}\right)^{0.7} \tag{2.28}$$

where $A_r$ is the area of the receiver, and $C_{r, ref}$ and $A_{r, ref}$ are the cost and area of a reference receiver design. We estimate the area of the receiver to be:

$$A_r = \frac{\dot{Q}_{receiver}}{\dot{q}_{r, max}/2} \tag{2.29}$$

where $\dot{Q}_{receiver}$ is the design thermal power of the receiver and $\dot{q}_{r, max}$ is the maximum allowable heat flux of the receiver. The design thermal power of the receiver depends on the design power output, the efficiencies of the power block and storage system, ($\eta_{PB}$ and $\eta_{storage}$), and the solar multiple, $SM$:

$$\dot{Q}_{receiver} = \frac{\dot{W}_e}{\eta_{PB}\eta_{storage}}SM. \tag{2.30}$$

Combining Equations 2.28 - 2.30, the cost of the receiver is:

$$C_r = C_{r, ref} \left(\frac{2\dot{W}_e SM}{\dot{q}_{r, max} A_{r, ref} \eta_{PB} \eta_{storage}}\right)^{0.7}. \tag{2.31}$$
Now, to estimate $C_r$, the remaining term that has not previously been calculated is $\eta_{\text{storage}}$. Storage efficiency is the ratio of annual energy that goes to the power block, $Q_{PB}$, to the energy that goes into the storage system, which is the sum of $Q_{PB}$ and the energy lost from the storage system, $Q_{\text{storage\_loss}}$:

$$
\eta_{\text{storage}} = \frac{Q_{PB}}{Q_{PB} + Q_{\text{storage\_loss}}}.
$$

(2.32)

Thermal losses occur from the storage system at all times, while heat is only transferred to the power block while it is running. The fraction of time that the power block runs is given by the capacity factor, $CF$, so:

$$
\eta_{\text{storage}} = \frac{\dot{Q}_{PB} CF}{Q_{PB} CF + \dot{Q}_{\text{storage\_loss}}}
$$

(2.33)

where $\dot{Q}_{\text{storage\_loss}}$ is the rate of heat loss from the storage system. Based on a 1-dimensional heat loss calculation (assuming that insulation is the primary thermal resistance for heat loss), we estimate this loss as:

$$
\dot{Q}_{\text{storage\_loss}} = \frac{T_m - T_{amb}}{R_{\text{insulation}}}.
$$

(2.34)

where $R_{\text{insulation}}$ is the thermal resistance of the insulation that surrounds the storage tank. Based on parallel thermal resistances on the sides, bottom, and top of the storage tank, we find:

$$
R_{\text{insulation}} = \left(\frac{2\pi \kappa_{\text{ins}} h_{\text{tank}}}{\ln(f_{\text{ins}})} + \frac{\pi \kappa_{\text{ins}} h_{\text{tank}}}{f_{\text{ins}} - 1}\right)^{-1}.
$$

(2.35)

Combining equations 2.33-2.35 and rearranging, we then have storage efficiency:

$$
\eta_{\text{storage}} = \left[1 + \frac{\eta_{PB}}{\dot{W}_c CF} \left(\frac{T_m - T_{amb}}{\ln(f_{\text{ins}})} + \frac{2\pi \kappa_{\text{ins}} h_{\text{tank}}}{f_{\text{ins}} - 1}\right)\right]^{-1}.
$$

(2.36)

We note that the value of $f_{\text{ins}}$ influences both storage efficiency and insulation cost, and must be optimized for each PCM and design scenario in order to make an even comparison between potential PCMs. This is discussed in Section 2.4.11. In combination with the previously calculated $\eta_{PB}$ and $h_{\text{tank}}$, Equation 2.36 can then be used with Equation 2.31
for calculating receiver cost. Examination of these equations shows that all PCM thermal properties have an influence on receiver cost, which is lowest for a PCM with intermediate melting temperature, high energy density, and high thermal conductivity.

2.4.8 Tower cost

Following the tower cost model in the System Advisor Model:

\[
C_{\text{tower}} = C_{\text{tower, ref}}e^{0.0112h_{\text{tower}}}
\]

(2.37)

where \(C_{\text{tower, ref}}\) is a constant with units of dollars and \(h_{\text{tower}}\) is the height of the tower in meters. Based on simulations that we ran using the publicly available software SolarPILOT [77], we approximate \(h_{\text{tower}}\) as a function of the thermal power rating of the heliostat field, \(\dot{Q}_{\text{field}}\):

\[
h_{\text{tower}} = 15.36(\dot{Q}_{\text{field}})^{0.4}
\]

(2.38)

where \(\dot{Q}_{\text{field}}\), in units of MW, is based on the system design power output, all subsystem efficiencies, and the solar multiple, \(SM\):

\[
\dot{Q}_{\text{field}} = \frac{\dot{W}_e}{\eta_P \eta_{\text{storage}} \eta_r} SM.
\]

(2.39)

The only term in Equation 2.39 that we have not previously derived is receiver efficiency, \(\eta_r\). For this calculation, we use Equation 2.40:

\[
\eta_r = \frac{\dot{Q}_{\text{receiver}}}{\dot{Q}_{\text{receiver}} + \dot{Q}_{r,\text{loss}}}
\]

(2.40)

where \(\alpha_r\) is the absorptivity of the receiver surface, \(\dot{Q}_{\text{receiver}}\) is the heat transfer rate from the receiver to the storage system, and \(\dot{Q}_{r,\text{loss}}\) is receiver thermal loss rate. This loss can be calculated as the sum of convective and radiative losses from the receiver:

\[
\dot{Q}_{r,\text{loss}} = h_r A_r (T_r - T_{\text{amb}}) + \sigma \epsilon_r A_r (T_r^4 - T_{\text{amb}}^4).
\]

(2.41)

In Equation 2.41, \(h_r\) is the heat transfer coefficient for convection from the receiver, \(\sigma\) is the Stefan-Boltzmann constant, \(\epsilon_r\) is the emissivity of the receiver surface, and \(T_r\) is the temperature of the receiver. Combining Equations 2.29 and 2.40-2.41, receiver efficiency is
\[ \eta_r = \alpha_r \left[ 1 + \left( \frac{2}{q_{r,max}} \right) \left( h_r (T_r - T_{amb}) + \sigma \epsilon (T_r^4 - T_{amb}^4) \right) \right]^{-1}. \] (2.42)

The only parameter in Equation 2.42 that is not taken as an assumed input is receiver temperature. This temperature depends on the melting temperature of the PCM, the temperature drop from the receiver to the thermal storage system, \( \Delta T_r \), and the temperature drop through the PCM while charging, \( \Delta T_{PCM,c} \):

\[ T_r = T_m + \Delta T_r + \Delta T_{PCM,c}. \] (2.43)

We take \( \Delta T_r \) as a fixed input, and calculate the temperature drop through the PCM while charging as:

\[ \Delta T_{PCM,c} = \Delta T_{PCM,d} \frac{SM}{\eta_{storage}}. \] (2.44)

Equations 2.38-2.44 provide all required inputs to tower cost in Equation 2.37. This cost is lowest when the combination of power block, storage, and receiver efficiency is highest. This occurs for PCMs with an intermediate melting temperature, high energy density, and high thermal conductivity.

### 2.4.9 Solar field cost

The cost of the solar field, \( C_{field} \), is the product of the reflective area of the heliostat field, \( A_{field} \), and the specific cost of the field, \( c_{field} \):

\[ C_{field} = A_{field} c_{field}. \] (2.45)

We determine heliostat field area based on the design power output, all subsystem efficiencies, the solar multiple, and the design solar resource, \( \dot{q}_{solar} \):

\[ A_{field} = \frac{\dot{W}_e}{\dot{q}_{solar} \eta_{PB} \eta_{storage} \eta_r \eta_{field}} SM. \] (2.46)
To estimate the efficiency of the heliostat field, $\eta_{\text{field}}$, we use the same SolarPILOT simulations that we used to determine $h_{\text{tower}}$, and find:

$$\eta_{\text{field}} = 0.7e^{-0.000183\dot{Q}_{\text{field}}}$$

(2.47)

where $\dot{Q}_{\text{field}}$ is input in units of MW. Now, the cost of the field can be written using terms that have all been previously calculated:

$$C_{\text{field}} = \frac{\dot{W}_e}{\dot{q}_{\text{solar}}\eta_B\eta_{\text{storage}}\eta_r\eta_{\text{field}}}SMc_{\text{field}}.$$  

(2.48)

In addition to this field cost, our model includes the cost of preparing land for placement of heliostats, and the cost of the land itself. The site preparation cost, $C_{\text{siteprep}}$, is:

$$C_{\text{siteprep}} = A_{\text{field}}c_{\text{siteprep}}$$

(2.49)

where $c_{\text{siteprep}}$ is the specific cost of leveling and other necessary preparation for the heliostat field installation. Finally, our model estimates the cost of land, $C_{\text{land}}$, based on land area, $A_{\text{land}}$, and the specific cost of land, $c_{\text{land}}$:

$$C_{\text{land}} = A_{\text{land}}c_{\text{land}}.$$  

(2.50)

In Equation 2.50, our estimate of land area is based on the same SolarPILOT simulations previously mentioned for calculating $h_{\text{tower}}$ and $\eta_{\text{field}}$, with $\dot{Q}_{\text{field}}$ input in units of MW:

$$A_{\text{land}} = 1.37(\dot{Q}_{\text{field}})^{1.13}.  

(2.51)

Equations 2.48 - 2.51 show that the cost of the heliostat field depends on all subsystem efficiencies. This means that all of the PCM thermal properties (melting temperature, energy density, and thermal conductivity) have an important influence on field cost.

2.4.10 Operation & maintenance costs

Both fixed and variable operation and maintenance costs for CSP systems depend on the size of several system components. Following the cost model in the System Advisor Model, some of these costs are linearly dependent on subsystem size, and others scale less than linearly such that normalized cost is lower for larger systems. Variable operation and
maintenance cost, VO&M, includes utilities and other miscellaneous costs. We calculate this cost based on several reference costs (C_1-C_5) and the corresponding design power output, heliostat field area, receiver thermal power rating, and energy stored for a reference system (\dot{W}_{e,ref}, A_{field,ref}, \dot{Q}_{r,ref}, and E_{storage,ref} respectively):

\[ VO&M = C_1 \left( \frac{\dot{W}_e}{\dot{W}_{e,ref}} \right) + C_2 \left( \frac{A_{field}}{A_{field,ref}} \right) + C_3 \left( \frac{\dot{Q}_r}{\dot{Q}_{r,ref}} \right) + C_4 \left( \frac{E_{storage}}{E_{storage,ref}} \right)^{0.7} + C_5 \left( \frac{\dot{W}_e}{\dot{W}_{e,ref}} \right)^{0.7} \] (2.52)

Fixed operation and maintenance cost, FO&M, is primarily made up of labor costs, but also includes the cost of mirror washing and other service contracts, and additional miscellaneous costs. The full equation we use for FO&M has a large number of terms, so rather than replicate it here, we simply state that this cost depends on the sizes of the heliostat field, receiver, storage system, and power block in a similar fashion to Equation 2.52.

### 2.4.11 Optimization of \(\Delta T_{PCM}\) and \(f_{ins}\)

In our model, two important inputs that influence overall system performance and cost are the temperature drop through the PCM during discharge, \(\Delta T_{PCM,d}\), and the ratio of insulation radius to tank radius, \(f_{ins}\). A larger \(\Delta T_{PCM,d}\) reduces heat exchanger cost, but also reduces the efficiencies of the power block and receiver. A larger insulation thickness increases insulation cost, but improves storage efficiency. Both of these trade-offs depend on the temperature and size of the storage system, and thus PCM properties \(T_m\), \(u_{PCM}\), and \(\kappa_{PCM}\). Therefore, the best values of \(\Delta T_{PCM,d}\) and \(f_{ins}\) are different for each PCM, and these parameters should be optimized so that the model can provide an even comparison based on how the system would be designed for each PCM. To execute this optimization, our model runs simulations for a wide range of values of \(\Delta T_{PCM,d}\) (in increments of 1°C) and \(f_{ins}\) (in non-dimensional increments of 0.01), and chooses the design that results in the
lowest LCOE. Our results are then based on the optimal high-level design for each PCM, allowing for an even comparison among candidate materials.

2.4.12 Model limitations

This model is built to be as simple as possible, while retaining the influence of all PCM properties on performance and cost of all major system components. This makes the model adaptable and rapid to execute, but means that the model relies on simplifications that limit its accuracy. We make several high-level assumptions, though details in a real application are much more nuanced. Therefore, the results of our model are useful for exploring general trends and initial down-selection of potential PCMs, but not for final design of a CSP plant.

One potential contribution to cost that is ignored in this model is the cost involved with PCM chemistry. Some PCMs are highly corrosive to standard construction materials (in particular, metal PCMs), and additional cost for specialty containment materials or coatings could have an effect on the overall cost results [78]. Other materials are toxic or flammable, requiring costly handling procedures. For stability, some PCMs may also need to be contained within inert atmospheres, which would increase containment cost.

2.4.13 Summary of model and assumptions

Table 2.1 provides a summary of assumptions we use in our model. These assumptions are generally based on the 2030 targets for the Sunshot program, and represent technical and cost progress that is expected to be achieved by that time. In this paper, we choose to simulate a “peaking” CSP plant with 6 hours of storage and a capacity factor of 0.4, as one potential design for future CSP. However, our model could easily be applied to alternative configurations as well.

For some assumptions in Table 2.1, we use cost ranges rather than single input values. For these assumptions, we use uniform probability distributions to select input values, indicating our uncertainty in component technology progress by 2030. Further, some of our cost inputs depend on operation temperature, which we divide into three temperature ranges that
correspond to limitations of common structural materials. Steel may be used up to 400°C, stainless steel may be used up to 650°C, and specialty materials (inconel or ceramics) must be used for higher temperatures.

Table 2.1: Summary of model assumptions, most of which align with Sunshot targets for 2030 CSP peaking plants. Parameters on the left side of the table have a single input value for all simulations. Parameters on the right side of the table are assigned a random value within the given ranges for each new simulation with uniform probability. In this way, our results provide cost ranges that represent uncertainty in future CSP progress.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
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<td>$T_{amb}$</td>
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<td>°C</td>
<td>$\alpha_{HX}$</td>
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<td>$m^2/m^3$</td>
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<td>$\rho_{HX}$</td>
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<td>$MW$</td>
<td>$f_{PB}$</td>
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</tr>
<tr>
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<td>hours</td>
<td>$\Delta T_r$</td>
<td>15 +/− 5</td>
<td>°C</td>
</tr>
<tr>
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<td>$C_{field}$</td>
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<td>$c_{HXmfg}$</td>
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<td>$c_{siteprep}$</td>
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<td>$/m^2$</td>
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<td>$/acre$</td>
</tr>
<tr>
<td>$h_r$</td>
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<td>$W/m^2K$</td>
<td>$c_{insulation}$</td>
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<td>$/m^3$</td>
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<td>$C_{tower,ref}$</td>
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<td>$c_{PB}$</td>
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<td>$/W_t$</td>
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</tr>
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<td>$/kg$</td>
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<td>$W/mK$</td>
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<td>if $T_m &lt; 400°C$, 0.62 +/− 25%</td>
<td></td>
</tr>
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<td>years</td>
<td>$c_{tank}$</td>
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<td></td>
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<td>$$/L$</td>
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<td>$CCF$</td>
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<td>$C_{r,ref}$</td>
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<td>$</td>
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<td></td>
<td></td>
<td></td>
<td>if $T_r &lt; 650°C$, 85,000,000 +/− 10%</td>
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</tr>
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<td></td>
<td></td>
<td></td>
<td>if $T_r &gt; 650°C$, 159,375,000 +/− 10%</td>
<td>$</td>
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</table>

2.5 Model results for varied PCM properties

Here, we apply the modeling framework established in Section 2.4 to a single solar power tower system design with specifications and assumptions given in Table 2.1. Before considering real candidate materials, in this section we run >100,000 simulations for cases within ranges of possible material properties. This provides results that quantify the relative value of each PCM property based on its impact on overall system LCOE.
Figure 2.2: LCOE results for general variation of material properties. (a) Surfaces of constant LCOE for a material with a melting temperature of 600°C show that all material properties are important considerations for the overall system LCOE. The slopes in the contour lines of (b-d) quantify the relative value of each PCM property for a certain situation. (b) assumes a melting temperature of 600 °C and a thermal conductivity of 0.5 W/mK. (c) assumes a melting temperature of 600°C and an energy density of 500 kWh/m$^3$. (d) assumes a thermal conductivity of 0.5 W/mK and an energy density of 500 kWh/m$^3$. 
In Figure 2.2, we show results of LCOE as a function of the four critical PCM properties $T_m$, $u_{PCM}$, $\kappa_{PCM}$, and $c_{PCM}$ (melting temperature, energy density, thermal conductivity, and material cost). Figure 2.2(a) shows surfaces of constant LCOE, assuming a PCM melting temperature of 600°C. The shape of each surface then illustrates the relative importance of each material property for minimizing LCOE. Material cost has a linear influence on LCOE, while the non-linear slopes of the surfaces with respect to energy density and thermal conductivity reveal diminishing returns for improvements in each of these properties. The symmetry of the surfaces with respect to material cost, energy density and thermal conductivity also demonstrate that each of these material property inputs have an important impact on overall system LCOE. When selecting a PCM for CSP applications, none of these properties should be neglected.

Figure 2.2(b) shows a vertical slice of Figure 2.2(a) at a thermal conductivity of 0.5 W/mK. Following the trend from one contour line to another, high energy density and low material cost lead to low LCOE. Following along a single contour line of constant LCOE (the slope of the line) quantifies the value of energy density for the overall system. For example, a PCM with an energy density of 300 kWh/m$^3$ and a material cost of 5.3 $$/kWh results in an LCOE of 7.2 \(\text{¢/kWh}\). If comparing with a material that has an energy density of 400 kWh/m$^3$, the same LCOE corresponds to a material cost of 6.9 $$/kWh. Thus, for a starting material with $T_m = 600\degree\text{C}$, $\kappa_{PCM} = 0.5$ W/mK, and $u_{PCM} = 300$ kWh/m$^3$, increasing energy density is worth approximately 1 $$/kWh in material cost per 60 kWh/m$^3$ in energy density.

Similarly, Figure 2.2(c) shows a vertical slice of Figure 2.2(a) at an energy density of 500 kWh/m$^3$. Following the trend from one contour line to another, high thermal conductivity and low material cost lead to low LCOE. Following along a single contour line of constant LCOE quantifies the value of thermal conductivity for the overall system. For example, the contour line for an LCOE of 7 \(\text{¢/kWh}\) intersects the thermal conductivity axis near 0.36 W/mK. This contour line shows that an equivalent LCOE could be achieved by increasing
thermal conductivity to 0.5 W/mK while also increasing material cost to near 1.5 $/kWh. Therefore, for a material with \( T_m = 600^\circ C \), \( u_{PCM} = 500 \) kWh/m\(^3\), and \( \kappa_{PCM} = 0.36 \) W/mK, increasing thermal conductivity by 40% is worth about 1.5 $/kWh in material cost.

Lastly, Figure 2.2(d) shows the influence of PCM melting temperature on LCOE. The black vertical lines in this figure occur at the discontinuities in LCOE that are caused by transitioning from carbon steel to stainless steel (400°C) and from stainless steel to higher temperature materials (650°C) for the storage tank and heat exchanger. In addition, as the PCM melting temperature approaches these transition points, receiver temperature increases such that the receiver must also be made of more expensive materials, which increases LCOE sharply. Across the full range of temperatures that we consider, the other primary trade-off is that as melting temperature increases, power block efficiency increases while receiver efficiency decreases. Considering all of these effects, our model shows that while higher temperature generally reduces cost within a temperature range, the lowest cost occurs at intermediate temperatures, where relatively low-cost stainless steel may be used as a material for storage tank, heat exchanger, and receiver components. A final note on Figure 2.2(d) is that the slopes of the contour lines quantify the value of changing melting temperature. As an example, within the range of 400-600°C these slopes show that increasing melting temperature is worth between 0.1-0.2 $/kWh per °C.

The same process followed in the above discussion for Figure 2.2(a)-(d) could be applied to any material within the range that we consider. For a given set of material properties, our system model and Figure 2.2 provides the relative value of each material property. This visualization is valuable for comparing two materials that may be very similar with respect to two material properties, as it would quantify the value of the trade-off between the other two properties.

### 2.6 Model results for candidate PCMs

Here we apply our cost model to candidate materials that may be considered for latent heat storage for CSP. The materials that we consider here are the same as in Figure 2.1,
with focus on two classes of materials - metals and salts.

### 2.6.1 Cost breakdown

Figure 2.3 shows the general breakdown of costs for select PCMs that illustrate the impact of material properties on costs of different subsystems. Comparing the first two materials, Al-Si is one of the most promising metal PCMs, and NaCl represents a characteristic salt. Considering the storage system costs in Figure 2.3(a), NaCl has a lower material cost than Al-Si (0.6 $/kWh compared to 15 $/kWh), but much higher heat exchanger cost because of its lower thermal conductivity (0.49 W/mK compared to 180 W/mK for Al-Si). While both materials have a relatively similar energy density (289 kWh/m$^3$ for NaCl and 365 kWh/m$^3$ for Al-Si), NaCl has a significantly higher tank cost because its higher melting temperature (802 vs 577°C, respectively) requires more expensive structural materials. Together, this leads NaCl to a storage cost of 30 $/kWh, and Al-Si to a storage cost of 16 $/kWh; for our modeled system, these storage costs contribute about 1 and 0.5 ¢/kWh to overall system LCOE, respectively. Moving to power block cost in Figure 2.3(b), using NaCl as a PCM results in a lower power block cost because its higher melting temperature leads to higher power block efficiency (44% compared to 40%). However, this higher temperature also reduces receiver efficiency, and the high operation temperature of NaCl means that more expensive materials must be used for the receiver. Thus, in Figure 2.3(c), the primary difference in collection costs between the two materials is that using Al-Si results in a much lower receiver cost. Based on the sizes of the storage system, power block, receiver, and heliostat field, the two materials also have very similar operation and maintenance (O&M) costs. The final result in Figure 2.3(b) shows that Al-Si results in a lower overall LCOE than NaCl (7.0 ¢/kWh compared to 8.0 ¢/kWh).

The third material shown in Figure 2.3, Zn-Al, has a very high material cost (72 $/kWh), and because its melting temperature is relatively low (381°C), choosing this PCM results in low power block efficiency (33%). This leads to high power block cost (1.4 $/W) and a large (and high cost) receiver, storage system, and heliostat field. However, low operation
Figure 2.3: Cost breakdown for select candidate PCMs: Al-Si, NaCl, and Zn-Al. Al-Si has an intermediate melting temperature (577 °C) and high thermal conductivity (160 W/mK), leading to low heat exchanger (HX) and tank costs. NaCl has a much lower material cost, but higher melting temperature and lower thermal conductivity. This low thermal conductivity leads to high heat exchanger cost, and the high temperature results in high tank and receiver costs, though it does improve power block efficiency and therefore reduce power block cost. In total, this results in a lower LCOE for Al-Si (7.0 ¢/kWh) compared to NaCl (8.0 ¢/kWh). The third material shown, Zn-Al, has a much higher material cost, which is its primary component of storage cost. In addition, its low melting temperature allows for use of low cost heat exchanger, tank, and receiver materials, but also leads to low power block efficiency and therefore high power block costs.
temperature also means that lower cost structural materials may be used. Overall, choosing Zn-Al as a PCM leads to the highest LCOE (10.2 ¢/kWh) of these three materials.

It is important to note that both the overall LCOE results and the comparisons between the materials in Figure 2.3 depend not only on the PCM properties, but also on our assumptions for the rest of the system. The sensitivity of LCOE to these assumptions is discussed in Section 2.7.

2.6.2 LCOE results for all candidate PCMs

In Figure 2.4, we show LCOE results that reveal some of the most attractive materials for further development in CSP systems. In this figure, materials are separated into temperature ranges that correspond to the limitations of storage tank, heat exchanger, and receiver materials: carbon steel up to 400 °C, stainless steel up to 650 °C, specialty containment materials such as inconel or ceramics for higher temperatures, and a final separation for PCMs at very high temperatures (over 1,000 °C). The range of LCOE results for each material are based on 1,000 simulations using the range of model inputs given in Table 2.1. Because we use a range of performance and cost inputs, these results give an estimate of the range of possible LCOE values that may be achieved with use of each PCM, based on potential developments in CSP technology. The middle line in each box indicates the median LCOE, while the bottom and top of the boxes represent the 25th and 75th percentiles, and the whiskers extend to the full extent of the range of our LCOE results.

Considering differences in LCOE results for materials with different melting temperatures, Figure 2.4 shows that the intermediate temperature range of 400-650 °C results in the lowest LCOE values. In this range, the system realizes a high combination of power block and receiver efficiencies, and is able to take advantage of affordable structural materials. At lower temperatures, the size of the storage system, receiver, and field must be very large to compensate for low power block efficiency. At higher temperatures, transitioning to more expensive materials for the storage tank, heat exchanger, and receiver is not worth the increase in power block efficiency. However, these trade-offs lead to only minor differences
Figure 2.4: LCOE results based on the top candidate PCMs in each temperature range that we consider. The range of LCOE results for each material is based on the range of model inputs in Table 2.1. The middle line in each box indicates the median LCOE for the material, while the bottoms and tops of the boxes represent the 25th and 75th percentiles, and the whiskers extend to the full extent of the range of LCOE results. Green boxes indicate materials with thermal conductivity over 10 W/mK (metals) and blue boxes indicate materials with lower thermal conductivity (salts). The even distribution of both material classes shows that each has potential for future application in CSP systems.
in LCOE results. This means that a wide array of materials in different temperature ranges have potential for successful development in CSP systems.

Figure 2.4 also shows that both material classes (metals in green and salts in blue) could be used in CSP systems with a competitive LCOE. Metals have the advantage of high thermal conductivity, and some salts have exceptionally low material cost; the net result is that both material classes still present opportunities for development. Within metals, we find that alloys of aluminum, magnesium, and silicon result in the lowest LCOE. Within salts, the best materials appear to be chlorides or fluorides, which generally have lower material cost and/or higher energy density than other salts.

For additional insight into the relationship between PCM properties and LCOE for real materials, we consider Figure 2.5. Here, there is a general trend that higher energy density and lower material cost lead to lower LCOE. Also, similar to in Figure 2.4, the best materials appear to be aluminum, magnesium, and silicon metal alloys and chloride and fluoride salts. However, PCMs with a wide range of energy densities and material costs result in comparable LCOEs, due to differences in melting temperature and thermal conductivity that impact the performance and cost of the rest of the system. This illustrates the importance of considering all material properties, including their influence on the full system design and cost. By doing so, our holistic system model provides a valuable method for comparing and selecting PCMs for CSP.

2.7 Influence of heat exchanger design and other factors

The results given in Sections 2.5-2.6 are highly dependent on the assumptions in our model. We have made an effort to tailor these assumptions towards performance and cost targets that are expected to be met by the CSP community by 2030, though significant advancements in certain areas may shift the paradigm that we have presented.

As one example, our results depend greatly on development of heat exchanger designs. A simple shell-and-tube heat exchanger may have an area density (or surface area to volume ratio), $\alpha_{HX}$, of near 100-200 m$^2$/m$^3$, while “compact” heat exchangers are typically defined
Figure 2.5: LCOE results for candidate PCMs against their energy density and material cost. While high energy density and low material cost generally lead to low LCOE, materials with wide ranges of these properties produce very similar LCOE results due to differences in thermal conductivity and melting temperature. This illustrates the value of our holistic model that considers all material properties and their influence on the overall system.
as those with $\alpha_{HX}$ over 400 m$^2$/m$^3$, and bio-inspired heat exchangers have been designed with $\alpha_{HX}$ as high as 10,000 m$^2$/m$^3$ (one of the highest $\alpha_{HX}$ that has been observed in nature is 17,500 m$^2$/m$^3$ for human lungs [79]). Porosities for heat exchangers typically range from 0.6-0.9, but high porosity values may be difficult to achieve for heat exchanger designs with high $\alpha_{HX}$. Figure 2.6 shows the impact of this potential range of heat exchanger development on storage cost and LCOE of a CSP system using one candidate PCM, NaCl. Salts such as this one, due to their low thermal conductivity, are particularly sensitive to heat exchanger design because heat exchanger cost makes up a significant fraction of the overall system cost. With NaCl as a PCM, a simple heat exchanger design with a low $\alpha_{HX}$ results in a high storage system cost because it uses a lot of material (leading to high heat exchanger cost) and displaces a large amount of volume (leading to high containment cost). A heat exchanger with a low $\alpha_{HX}$ also leads to a larger temperature difference through the PCM during charging and discharging, which reduces the efficiency of the power block and receiver. However, development of advanced heat exchanger designs or creative ways to increase effective thermal conductivity of salt materials could drastically reduce these costs. Figure 2.6 shows that this would result in much lower storage cost and LCOE that would out-compete most metal materials, whose LCOE is much more weakly influenced by heat exchanger design.

As another example of the influence of our assumptions on the relative LCOE of CSP systems that use different PCMs, high temperature structural material development could enable low cost receiver, containment, and heat exchanger designs at higher temperatures. This would allow CSP systems to take advantage of higher power block efficiency, and would make higher temperature materials more attractive compared to those that we have presented as the best materials thus far.

For a more complete sensitivity analysis, Figure 2.7 presents standardized regression coefficients [80] for the nine parameters that we varied in our analysis that have the strongest influence on LCOE. These coefficients represent the number of standard deviations that
Figure 2.6: Influence of heat exchanger design on (a) storage cost and (b) LCOE for a CSP system that uses NaCl as a PCM. Transitioning from a simple to an advanced heat exchanger design can significantly reduce storage cost and the resulting LCOE for a CSP system that uses a salt material with low thermal conductivity.
LCOE would change for a standard deviation increase in the respective input parameter. Negative values mean that if the input parameter increases, then LCOE decreases. From top to bottom, these parameters are sorted by their average magnitude across all materials we simulated. This analysis reveals that the most important assumption in our model is the percentage of Carnot efficiency that the power block achieves, $f_{PB}$. This power block efficiency has a large influence because it impacts the size, and therefore cost, of all system components. We also find that the specific costs of the power block, $c_{PB}$, and the concentrator field, $c_{field}$, have a large influence on LCOE because they make up significant portions of the overall cost. In addition, several aspects of the heat exchanger influence LCOE strongly, including the area density ($a_{HX}$), porosity ($p_{HX}$), manufacturing cost factor ($c_{HXmfg}$), and material cost ($c_{HXmat}$). This is especially true for NaCl and other materials with low thermal conductivity. In these cases, the heat exchanger represents a larger portion of the overall system cost, and more strongly impacts the temperatures and resulting efficiencies of the power block and receiver. Compared to metals, the future potential success of salt materials in CSP systems then more strongly depends on advancements in heat exchanger design.

2.8 Conclusion

In this paper, we have developed a new holistic system model of a CSP plant with latent heat storage. This model takes inputs of plant design, performance assumptions, and cost assumptions, then calculates LCOE based on PCM properties. We first applied this model to hypothetical variations in PCM properties, and quantified the relative value of each property. Then, we applied the model to specific candidate PCMs, which revealed the materials that are most attractive for CSP development, and provided insight into the trade-offs between different types of materials.

Our model involves several high-level assumptions, and the results depend on the specific system design. Thus, we completed sensitivity analysis to determine the influence of the most important input parameters on the overall system LCOE. We also note that alternative configurations for the receiver, storage, and power block subsystems could be considered. For
Figure 2.7: Standardized regression coefficients for the top nine input parameters that influenced our modeled LCOE outcomes. Power block fraction of Carnot efficiency, $f_{PB}$, is the input with the strongest impact on LCOE because it determines the size and resulting cost of all system components. Several aspects of heat exchanger design are also critical, especially for salt materials with low thermal conductivity such as NaCl.
example, storage could be located on top of the tower, which would modify tower requirements and cost. Additionally, latent heat storage is best matched to isothermal heat engines (such as Stirling engines), though other power cycles that operate with variable temperature (steam Rankine, super-critical \( \text{CO}_2 \) Brayton) could employ a cascaded latent heat storage system with multiple PCMs. These alternative designs would require a modified system model to evaluate the influence of PCM properties on overall system performance and cost.

This work shows the value of holistic system modeling when selecting a PCM for a CSP application. Several PCM properties (melting temperature, energy density, thermal conductivity, and material cost) all influence storage cost, but importantly, also impact the cost of other subsystems which can have an even stronger influence on the overall success of the system. By considering these effects, our model and results can be used by engineers and plant designers to improve selection of PCMs for CSP.
CHAPTER 3
PERFORMANCE MODELING AND TECHNO-ECONOMIC ANALYSIS OF A
MODULAR CONCENTRATED SOLAR POWER TOWER WITH
LATENT HEAT STORAGE

Modified from a paper published in Applied Energy [73].

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In this paper, we established a new concept for a modular, dispatchable solar power tower.
This concept was our answer to: “what is the best way to combine latent heat storage with
CSP at a small scale?” This paper also provided a fairly comprehensive summary of my
performance modeling and techno-economic analysis of this concept.

The standard limitation of small scale power systems (<10 MWₑ) is low power block
efficiency: at our temperatures, we predicted 30% efficiency with a Stirling engine compared
to the conventional 45% efficiency for a large steam Rankine cycle. However, we found
that our design made up for this deficiency in a few ways. First, the optical efficiency
of our 0.76 MWₑ heliostat field was 30% higher than large fields, due to lower cosine and
attenuation losses. Next, by combining our receiver and thermal storage tank into a single
unit, we effectively eliminated an entire subsystem that traditionally makes up ~20% of
total system cost. Our valved thermosyphon design also enabled dispatchability with passive
controls, which reduced operation and maintenance requirements. From these results, we
demonstrated the potential of our concept to match performance and cost of large CSP, but
at a small scale, where it could be cost-competitive with alternatives of PV with batteries
and natural gas combustion turbine plants (if we could build it - see Chapters 4 and 5).

3Concept development
4Designed model
5Performed calculations
6Wrote paper
3.1 Abstract

In this paper, we present performance simulations and techno-economic analysis of a modular dispatchable solar power tower. Using a heliostat field and power block three orders of magnitude smaller than conventional solar power towers, our unique configuration locates thermal storage and a power block directly on a tower receiver. To make the system dispatchable, a valved thermosyphon controls heat flow from a latent heat thermal storage tank to a Stirling engine. The modular design results in minimal balance of system costs and enables high deployment rates with a rapid realization of economies of scale. In this new analysis, we combine performance simulations with techno-economic analysis to evaluate levelized cost of electricity, and find that the system has potential for cost-competitiveness with natural gas peaking plants and alternative dispatchable renewables.

3.2 Background

The decreasing cost of wind turbines and photovoltaic panels (PV) is driving rapid deployment of renewables on electric grids around the world. Notably, renewable electricity in the United States is predicted to double from 2013 to 2040 [81]. However, the inherent variability (intermittency and diurnal cycle) of wind and solar presents a significant challenge. Such variable generation strains the grid by requiring other electricity sources to adjust output to match demand. This adjustment is difficult for today’s United States grid that is largely made up of inflexible base load power generators (e.g. nuclear, coal) with just a small subset of generation coming from peaking power plants (mostly natural gas). Figure 3.1 highlights this difficulty, and a study by the National Renewable Energy Laboratory [5] quantifies its potential future impact: when solar PV provides 22% of total electricity generation to the western United States grid, 50% of annual electricity generation from each additional unit of PV must be curtailed because large nonflexible power sources cannot decrease output. Thus, if renewables are to continue growth, the current grid system must change.
Figure 3.1: (a) Daily peak solar PV output on the California Independent System Operator grid has been growing at $\sim$1.6 GW/yr since 2012. (b) As a result, net demand (after taking into account solar and wind electricity production) is projected to drop in the middle of the day, leading to overgeneration and high ramp rates. Hourly net demand for 2011-2016 is from publicly available data (Supplementary Figures 1-4). Curves for 2018 and 2020 are extrapolated from 2016 data based on projected growth in demand and installation of wind and solar.
The current most economic solution to the intermittency of renewables would be a reduction of existing base load generation and addition of natural-gas peaking plants to increase grid flexibility and allow for fluctuations in PV output. Alternatively, energy storage could allow grid operators to time-shift production from renewable resources to times of high electricity demand. Existing technologies include pumped hydroelectric (93.6% of current United States grid storage capacity [82]), thermal (3.4%), electro-chemical (2.4%), and compressed air (0.5%) energy storage. However, these existing grid storage options account for only 2.1% of total electricity capacity in the United States, in contrast to the 43% from natural gas plants [81, 82]. Further, the mature technologies in this group (pumped hydroelectric, compressed air) have barriers to future growth; common challenges include low energy density, geological and environmental concerns, high installation cost, and slow manufacturing learning curves [82–84]. This means that, in the pursuit of a 100% renewable energy grid, there is a tremendous opportunity for new, innovative energy storage technologies to grow and make an impact on the future grid.

Emerging technologies that may provide grid flexibility include electrochemical batteries, electrolyzers coupled to fuel cells, and thermal energy storage with concentrated solar power. In 2016, electrochemical batteries represented ~0.05% of grid capacity in the United States[81, 82]. Several battery technologies are currently receiving significant investment, and the United States Department of Energy aims to reduce levelized cost of electricity (LCOE) from >20 down to 10 ¢/kWh [85–88], or to 14 ¢/kWh for the combination of batteries with PV [89]. While this cost goal is competitive for many applications, it may be difficult to reduce capital costs from near 300 to below 150 $/kWh within the next few years [15, 16, 86–89], with raw material costs that add up to over 100 $/kWh [15]. Similar to electrochemical batteries, renewable electricity may be coupled to electrolyzers that produce chemical fuels that can later be used for electricity generation by fuel cells. However, fuel cell storage systems have levelized costs ranging from 18-50 ¢/kWh [17, 90], and high capital costs will have to be reduced in order to reach competitive prices.
Significant efforts to develop thermal energy storage (TES) with concentrated solar power (CSP) are also underway. Fundamentally, thermal energy storage appears to be a low cost method: current TES capital costs are 20-25 $/kWh, and the United States Sunshot program has set a target of 15 $/kWh by 2020 [18, 91]. These values are well below the cost of battery storage even after accounting for the efficiency of converting heat to electricity. Integration into CSP plants has already led to full system costs near the 2020 Sunshot goal of 6 ¢/kWh [92], and several pathways for further development have been proposed [22].

Current state-of-the-art CSP tower storage systems use molten salt for sensible heat storage. The next generation of this technology is being developed to operate at elevated temperatures, using high temperature salts and a supercritical CO\textsubscript{2} power cycle to increase power block efficiency [18, 22]. Alternatively, particle-based storage and receivers have potential to enable high power block temperatures with lower cost materials, and may use particles both as a storage media and receiver heat transfer fluid [59, 93]. A third direction that future CSP systems may move towards is in the use of phase change materials (PCMs) for latent heat thermal energy storage. This method has advantages of increased energy density, isothermal operation, and potential to reduce cost [36, 37, 39]. Some latent heat storage designs use salt PCMs, and research efforts are working to improve their effective thermal conductivity with heat transfer enhancements [38, 43]. Other designs use metal PCMs, which have higher thermal performance, but require solutions to corrosion issues [28, 44, 94]. Both salts and metals have been considered for encapsulation within hollow spheres [49, 95], and many heat exchanger designs involve heat pipes [96]. Most development of latent heat TES has been limited to laboratories, but includes system-level demonstrations that include both PCMs and heat pipes [25, 51, 97, 98]. Even more nascent, thermochemical energy storage also has the potential for low cost storage [99].

Successful advancements in CSP and TES could provide a solution to the intermittency of PV and wind and their subsequent need for complementary energy storage. However, a major challenge for existing CSP is that traditional designs require plant sizes of over 100
MW_{electric} to reach high efficiency and low cost, which leads to initial capital investments of over $1B \[18, 100\]. This makes it difficult to secure financing, and limits the technology to utility-scale markets. In recent years, several companies (e.g. eSolar, Graphite Energy, Vast Solar, Aora Solar) have developed creative designs that employ smaller scale towers and non-traditional receiver and storage systems. However, there has been limited commercial success or development beyond initial pilot plants for these small systems. This means that significant opportunities remain for new CSP system designs to incorporate TES at a small scale and low cost.

In this paper, we present new analysis of a CSP design that could act as a renewable peaking plant with high modularity and low cost. Figure 3.2 shows our proposed configuration with latent heat thermal energy storage and a power block integrated in the same location on top of a small solar power tower. This configuration leads to a compact design with minimal parasitic losses and operation and maintenance requirements. We have previously presented preliminary versions of this concept \[101\] with initial techno-economic analysis \[102\] and subsystem thermal modeling \[103\]. In this paper, we build upon these efforts with a far more detailed model of performance and cost. We begin by describing the design choices and modeling that led to a refined low-cost design of this system. We present thermal and techno-economic analysis that evaluates the performance of the system. We conclude that the system has the potential to produce renewable dispatchable electricity at prices comparable to natural gas peaking plants and alternative renewable dispatchable technologies.

### 3.3 Design configuration

The design configuration that we consider for this analysis is depicted in Figure 3.2 and is hereafter referred to as STEALS (Solar Thermal to Electricity via Advanced Latent heat Storage).

A heliostat field reflects sunlight onto a solar receiver that is elevated by a steel truss tower. The entire thermal system is located on top of the tower. The receiver design is
Figure 3.2: (a) The Solar Thermal to Electricity via Advanced Latent heat Storage (STEALS) design integrates latent heat thermal energy storage, a thermal valve, and a Stirling engine on top of a small scale solar power tower. (b) Heliostats reflect sunlight through a cavity receiver to the bottom of a thermal storage tank that is filled with a phase change material (PCM). Sodium heat pipes are embedded in the PCM, and extend vertically from the receiver to the top of the storage tank to passively distribute heat through the storage system. The top of the heat pipes interface with the evaporator of a thermosyphon-based thermal valve. In this thermal valve, liquid sodium evaporates, travels up through vapor tubes (red arrows), and condenses on the heater head of the Stirling engine. Liquid sodium then returns to the evaporator through a return funnel (blue arrows). To regulate electricity output, opening or closing a valve throttles the flow of sodium and controls the heat flow from the storage to the power block. Here we consider systems ranging from 0.1 to 1 MW_{electric}, which would involve multiple Stirling engines in parallel on a single tower.
an optical cavity with an aperture that allows concentrated sunlight to be incident on the solar absorber (the bottom of the thermal energy storage (TES) tank). To transfer heat into and out of TES, sodium heat pipes are attached to the top of the tank and extend down through the phase change material (PCM) to the solar absorber. These heat pipes spread into a flattened conical shape at the top and bottom of the tank to maximize surface area for efficient heat transfer with adjacent subsystems and are mechanically connected only at the top of the tank to minimize risk of failure due to stresses that occur during thermal cycling. Here we use the aluminum-silicion (Al-Si) eutectic alloy as a PCM because of its melting temperature, high energy density, high thermal conductivity, and low cost. To protect stainless steel construction materials from corrosion, all surfaces of the storage tank and heat pipes that are exposed to PCM are coated with a ceramic coating that is deposited by a plasma-spray method [78]. Our initial laboratory testing demonstrates that several ceramics (yttria-stabilized zirconia, magnesia-stabilized zirconia, alumina) can reliably prevent corrosion with Al-Si at temperatures above 600 °C. The storage tank is insulated using a standard kaowool insulation material.

The key to dispatchability of this system is the ability to control when heat flows from the TES to the Stirling engine. Alternative to pumping a heat transfer fluid through the TES tank, our method of controlling heat flow uses a valved thermosyphon (“Thermosyphon thermal valve” in Figure 3.2(b)). In the “on” state, a working fluid (liquid sodium) evaporates at the thermosyphon evaporator (which is the same as the top surface of the TES tank). It then travels to the Stirling engine at the top of the thermosyphon because of the pressure difference created by evaporation. At the Stirling engine, the working fluid condenses as it releases its latent heat, and returns to the evaporator by gravity through a liquid return pipe in the center of the thermosyphon. In the “off” state, the liquid return is blocked by a shut-off valve, and once all working fluid has collected above this shut-off valve, the working fluid ceases to transfer heat. This thermal valve design is able to transfer high heat fluxes with very small temperature drops in the “on” state, and is highly insulating in the “off”
By tightly integrating system components in the same location on top of the tower, pathways for heat are very short which minimizes exergetic losses (both heat losses and temperature drops) from the solar receiver to the power block. Tower-based storage with passive heat transfer via heat pipes also eliminates the capital and operational costs of piping and pumps that are required in conventional solar power tower designs that locate their energy storage and power block on the ground. The penalty for suspending much of the system on top of the tower in our design is that structural tower cost increases because of the large mass that must be supported. In our analysis, this sets a limit on the relevant scale of this system to 1 MW_{electric}, based on existing towers built for wind turbines (see Supplementary Text).

The cavity receiver in this design minimizes radiative and convective losses from the solar absorber, and the aperture can be closed to minimize storage losses when not receiving sunlight. Use of Al-Si as a storage material, with a melting temperature of 577 °C, allows use of stainless steel as a structural material, leading to relatively inexpensive construction. Higher temperature PCMs would give increased Stirling engine efficiency, but also increase receiver and storage losses, and would potentially result in higher containment material cost. Stirling engines are attractive for this system because they can achieve relatively high efficiency at a small scale.

Cold start-ups occur in CSP plants when they are first turned on, after extended periods of low solar resource, or after downtime due to maintenance. This is an important consideration for most CSP designs that must use heat tracing to avoid freezing of a pumped heat transfer fluid, but is less critical for STEALS because heat pipes are used. Sodium heat pipes begin to become active at hot side temperatures near 500 °C, which means that heat transfer into the thermal storage tank will be slower during initial start-up until the bottom of the storage tank reaches near-operational temperature. This causes a slight efficiency loss, but is a minor concern because with proper insulation the system remains above 500 °C for
several days without heat input.

3.4 Methods

3.4.1 System level performance

To analyze the system-level performance of STEALS, we integrated three different software packages. First, we developed an in-house MATLAB code to model the time-dependent thermal performance of a complete STEALS system throughout a daily cycle of operation. This MATLAB code also used material and manufacturing cost factors to generate estimates of subsystem costs. We ran an initial MATLAB simulation to find the critical inputs to run SolarPILOT, a program developed by the National Renewable Energy Laboratory (NREL) that optimizes heliostat field and receiver geometries for solar thermal tower systems. Optimization simulations in SolarPILOT yielded heliostat field efficiencies and receiver and tower geometries that we input back into the MATLAB model for a more refined simulation. Finally, we simulated the full lifetime performance and cost of a STEALS plant using the CSP Generic Model within the System Advisor Model (SAM). The SAM simulation used the optical efficiencies generated by SolarPILOT and the thermal performance and cost outputs from our refined MATLAB simulation for a fully integrated, self-consistent analysis.

To consider system-level performance and cost under different possible scenarios, we used the process above as we varied system size, power block efficiency, and subsystem cost inputs. For each scenario, we ran parametric studies on insulation thickness and number of heat pipes to improve the TES system in terms of levelized cost of electricity.

3.4.2 MATLAB model

The time-dependent MATLAB model we developed to analyze our design uses a numerical finite-difference method that discretizes all system components into nodes of thermal masses that are connected via a thermal resistance network. The nodes considered in the model include the absorber, PCM, evaporator and condenser of the thermal valve, and the
power block. The full thermal resistance network that the model uses, and heat flow cal-
culations, are given in the Supplementary Figures 5-7 and Supplementary Tables 2-4. Our
custom code is also given as Supplementary Information. This code can be copied and pasted
into a MATLAB workspace in order to run a representative simulation of a 400 kW_{electric}
STEALS system with a 35% efficient Stirling engine as the power block. In addition, a
comprehensive list of inputs to typical SolarPILOT and SAM simulations are given in the
Supplementary Tables 7-8, and can be used with each of these publicly available programs
to duplicate our results.

To run a simulation, we first input nominal power rating and expected subsystem ef-
ficiencies so that the full geometry of the system can be calculated, then assign material
properties to all system components. We also input initial and external conditions, includ-
ing the initial temperatures of each subsystem component, as well as hourly direct normal
irradiation (using TMY data in Daggett, California on March 20) and the operation of the
thermal valve and heat rejection system, which may be specified as “on” or “off” at all times
of day. We then run a time-dependent simulation for a 24 hour cycle, with time steps of 0.02
seconds.

Our MATLAB model calculates heat transfer rates throughout the system at a given
time step based on temperatures and thermal resistances between nodes that are in thermal
contact with each other. It then calculates temperatures at the next time step by apply-
ing energy balances to each node. To include phase change of the thermal energy storage
material, the model uses an enthalpy-tracking method for PCM nodes (enthalpy results are
provided in Supplementary Figure 18), similar to the method used by Kotze et al [44]. The
PCM nodes are defined such that only heat transfer in the radial direction from heat pipes
is considered; this model is conservative as it neglects vertical heat transfer pathways that
would improve heat transfer, but was verified as a reasonable assumption with finite element
analysis (Supplementary Figure 19). The effective thermal resistance of the vapor space in
the valved thermosyphon changes depending on whether the valve is in the “on” or “off”
state, and the thermal resistance of the Stirling engine also changes depending on whether it is “on” or “off”. Efficiency of the Stirling engine is temperature-dependent and is calculated as a percentage of Carnot efficiency. After a simulation is finished, the model calculates performance results, including exergy flows (Supplementary Figure 20), daily thermal efficiencies, and design point thermal losses. These thermal performance results can then be input to SolarPILOT for heliostat field optimization and SAM for a lifetime performance analysis.

Our MATLAB model evaluates capital costs by applying material and manufacturing cost factors on a per-kg, per-m², or per-quantity basis to all system components. This yields system-level capital cost values that can be used as inputs for SolarPILOT and SAM simulations. Material cost inputs come from current metal stock exchange prices, quotes, and manufacturer websites. Manufacturing and labor cost estimates assume mass production. Stirling engine cost estimates are based on literature review and discussions with Stirling engine companies. Our tower cost model is given in Supplementary Figures 8-14. Our operation and maintenance cost model is given in Supplementary Figures 15-17 and Supplementary Table 6. A full list of cost factors is given in Supplementary Table 5.

3.4.3 SolarPILOT

To optimize heliostat field layout, tower height, and receiver dimensions for each full system simulation, we used SolarPILOT, a program developed by the National Renewable Energy Laboratory specifically for this purpose [104]. We used the default location of Daggett, California for all simulations, and based heliostat geometry on the eSolar design [106] by using mirrors that are 2.05 m wide and 1.1 m tall. We used a “flat plate” receiver type with an absorptivity and emissivity of 0.99 to represent the aperture of our cavity receiver [107]. Other key cost and thermal performance inputs came from our MATLAB simulations of the system. All SolarPILOT inputs for a typical simulation can be found in Supplementary Table 7.
For a wide range of nominal power ratings and power block efficiencies, we performed initial parametric studies in SolarPILOT to find optimal receiver height to within 0.05 meters, receiver elevation angle to within 5 degrees, and tower height to within 0.1 meters to minimize total plant cost. Because receiver elevation angle had a minor impact on cost (<1% over our full range of system sizes), we set its value to -35 degrees for all subsequent simulations. From initial parametric studies, we also found a general dependence of optimal receiver size on design point thermal power, and used a logarithmic fit to initial data to set receiver size for all subsequent simulations. These two simplifications reduced the number of free variables in SolarPILOT simulations, leading to reduced computation time with no significant impact on final results. The final free variable in SolarPILOT was tower height, which we optimized to within 0.1 meters using a parametric study for each different system configuration.

Final outputs from SolarPILOT included the tower height, receiver size, and optical efficiencies required for MATLAB and SAM simulations. Supplementary Figure 21 also provides results of a parametric study for a specific configuration with a design point thermal power of $0.91\, \text{MW}_{\text{thermal}}$ that considers efficiency and cost of non-optimal tower heights and receiver elevation angles.

### 3.4.4 System Advisor Model (SAM)

For final evaluation of performance and cost of STEALS, we used the "Generic CSP model" of SAM (version 2016.3.14) [105]. We used Daggett, California as the location for analysis, consistent with MATLAB and SolarPILOT simulations. Solar field optical efficiency data came from the final optimized configuration simulation in SolarPILOT. All other thermal performance inputs came from our MATLAB simulation. Cost inputs came from either our MATLAB simulation or were based on Sunshot goals [18, 91]. We assumed a plant lifetime of 25 years. Detailed input values for a typical SAM simulation can be found in Supplementary Table 8.

SAM simulations gave outputs of hourly electricity generation, heat transfer rates, subsystem efficiencies, and costs. These final results allowed us to investigate trends for different
system designs and make comparisons to other technologies.

3.4.5 Grid demand

To generate Figure 3.1(a), we downloaded peak PV power output data from the “Daily Renewables Watch” archive on the California Independent System Operator (CAISO) website for each day from April 20, 2010 to September 30, 2016. To generate the data curves from 2011 to 2016 in Figure 3.1(b), we compiled hourly solar and wind data from the CAISO website for seven days from March 1 to April 6 (March 1, 7, 13, 19, 25, 31, and April 6) from each year, such that each day of the week is included exactly one time. We then extrapolated from 2016 data with assumptions that total electricity demand grows at a rate of 1.28% [108], peak solar electricity generation grows at a rate of 1.64 GW/year, and wind electricity generation grows at a rate of 0.15 GW/year. Supplementary Figures 1-4 provide detailed solar and wind data.

3.4.6 LCOE comparison

We generated the LCOE data points for STEALS in Figure 3.6 using the methods described above for system-level modeling. To compare to other technologies, we used several other sources of data. For natural gas, we used the Annual Technology Baseline published by the National Renewable Energy Laboratory [109], and adjusted the calculations for a 30% capacity factor combustion turbine natural gas plant to vary natural gas fuel price between 2.50-4.50 $/MMBtu, and efficiency from 35-45%. This resulted in four data points that we surrounded with a more general figure for natural gas LCOE in Figure 3.6. For CSP, we compiled data from the SolarPACES database [110] and from https://www.csp.guru [100] to find estimates of LCOE for plants that exist today, and added these data points to Figure 3.6. We also calculated the LCOE of a modified “On the Path to Sunshot” [18] case to project future cost of a conventional CSP system that is operated in the same way as STEALS. For this case, we kept all inputs from an original “On the Path to Sunshot” SAM file, but modified the hours of storage and restricted dispatch in the middle of the day to reach a 30%
capacity factor (which is below the Sunshot design of 66.7% capacity factor, and increases cost relative to Sunshot goals). We then extended the CSP cost range in Figure 3.6 down to the modified Sunshot case of 10.5 ¢/kWh at a power block efficiency of 55% and a power rating of 100 MW_{electric}. For PV with batteries, we started with the default system model in SAM. We then considered a range of PV module costs from 0.6-0.9 $/W, PV balance of system costs from 0.9 to 1.0 $/W, battery costs from 200-400 $/kWh with replacements after either 10 or 15 years, and battery balance of system costs from 0.4-0.7 $/W [111]. For consistency with our STEALS and modified Sunshot CSP analysis, we used the same financial parameters for all SAM simulations. Because PV and batteries are scalable, we used horizontal lines for LCOE values in Figure 3.6(b). All of the data points used to generate the shapes used in Figure 3.6 are given in Supplementary Table 10.

3.5 Subsystem performance and cost

For each subsystem we consider here, results are based on a full system that has a nominal power rating of 200 kW_{electric}, a 30% efficient Stirling engine, and 5 hours (3.3 MWh) of thermal energy storage. The system operates with a ∼30% capacity factor as a complement to PV. Performance and cost are evaluated using the methods described in the Methods section.

3.5.1 Heliostat field and solar receiver

In design of the heliostat field and solar receiver for STEALS, we made several qualitative decisions to address technical and economic challenges. Small 2.2 m² focused mirrors (based on the eSolar design) minimize on-site construction cost [106]. A steel strut tower, similar to ones used for wind turbines, high tension power lines and simple water towers, can provide the support needed for a thermal storage tank and leverages existing high-volume manufacturing processes. For the small-scale STEALS design, and location in the northern hemisphere, a primarily north-facing heliostat field is most efficient because it minimizes cosine losses [112]. The orientation of a cavity receiver can be adjusted to match a north-facing field and gives
added benefits of reduced convective and radiative losses from the receiver surface relative to an external receiver. Finally, locating the receiver below the thermal storage tank enhances the performance of heat pipes within the tank by providing gravity-assisted working fluid return, avoiding issues encountered by previous heat pipe thermal storage systems [25].

Figure 3.3: (a) The heliostat field for a 760 kW\textsubscript{thermal} STEALS system uses \(\sim 1000 \text{ m}^2\) of reflective heliostat area in a north-facing configuration. The small size of STEALS, compared to (b) a conventional CSP solar field, improves design point optical efficiency from 65.6 to 83.7\% and reduces combined field, tower, and receiver cost by 50\% (based on the thermal power at the receiver).

Based on the qualitative design decisions above, and optimization in SolarPILOT, Figure 3.3(a) shows a reference case 765 kW\textsubscript{thermal} STEALS solar field. For scale, Figure 3.3(b) also provides a comparison to the 545 MW\textsubscript{thermal} “On the Path to Sunshot” solar field design [18, 113]. Because of its reduced size, the STEALS field and tower exhibit a much higher combined optical and receiver efficiency than conventional CSP designs (83.7\% for STEALS compared to 65.6\% for conventional designs, at design point, evaluated by SolarPILOT and SAM simulations). Further, the smaller tower height (20 m compared to 170 m) permits use of a simple steel strut tower and leads to a 30\% decrease in tower cost on a \$/kW\textsubscript{thermal} basis. In sum, the heliostat field, tower, and receiver for STEALS cost a total of 149 \$/kW\textsubscript{thermal}, compared to the Sunshot target cost of 291 \$/kW\textsubscript{thermal}. 
3.5.2 Thermal storage

The ability to control heat transfer into and out of a thermal storage tank is central to STEALS. Near-isothermal charging (heat transfer into the PCM) and discharging (heat transfer out of the PCM) are particularly important, as temperature gradients in the storage system result in significant exergetic losses. Energy density, thermal cyclic stability, material compatibility, and material cost are also critical to the overall success of the design. With this in mind, we considered \( \sim 100 \) salt and metal materials with melting temperature near \( 600 ^\circ C \) that had significant material property data established (Supplementary Table 1) \([36, 37, 39, 94]\), and down-selected to three candidates (K-Ca-Cl, Al, and Al-Si). We then ran full system simulations with these three materials, which revealed that a system using Al-Si resulted in the lowest overall system cost, primarily because of its high energy density and high thermal conductivity.

With Al-Si as the PCM, we ran parametric studies on size and spacing of heat pipes to obtain a thermal storage design that is compatible with the solar insolation and heat flow required by STEALS over a typical 24 hour cycle. In Figure 3.4(a), it is clear that solar energy is not available when net electricity demand is highest. To account for this discrepancy, STEALS must store energy as shown in Figure 3.4(b). Resulting details of phase and temperature throughout the PCM due to heat input and extraction are presented in Figure 3.4(c) and Figure 3.4(d), respectively. Following Figure 3.4 from bottom to top, latent heat is stored in the PCM overnight, until early morning discharge at 5 a.m. when net demand begins to rise. The PCM completely solidifies during this discharge, and cools as far as \( 11 ^\circ C \) below its melting temperature. Electricity generation stops at 7 a.m., while the PCM begins to receive solar thermal energy at 6 a.m. The PCM melts during the day, reaching a fully melted state between 2 and 3 p.m. and heating as high as \( 47 ^\circ C \) above its melting temperature. Evening dispatch begins at 3 p.m. when net demand rises for a second time, and continues until 10 p.m., after which some latent energy is stored in the liquid PCM overnight, until the cycle repeats. As is typical with thermal storage systems, the thermal
Figure 3.4: The STEALS thermal storage system time-shifts solar electricity production while minimizing superheating and subcooling of the PCM to achieve high efficiency. Panel (a) highlights the mismatch between solar resource (direct normal irradiation from TMY data in Daggett, California on March 30) and projected net electricity demand (demand minus generation from PV and wind in March 2020). (b) To overcome this temporal mismatch, the system requires four modes of operation. (i) Charging (red in panel (a)) occurs during peak solar resource in the middle of the day, while (ii) discharging (green in panel (b)) occurs in the morning and evenings. (iii) Some overlap of charging and discharging occurs during times of low solar resource when PV production is significantly lower than rated capacity. (iv) Separate from these times, the system is in a standby mode overnight to store energy for morning dispatch. To consider spatial variations within the PCM, panels (c) and (d) track phase and temperature as a function of distance from each heat pipe. Morning discharge beginning at 5 a.m. leads to solidification of the liquid PCM and additional cooling of the solid PCM. Sunlight beginning at 6 a.m. contributes to this discharge and limits the cooling to 11°C below the melting point. By 7 a.m., morning discharge is complete. The valve switches “off”, and all incoming heat is directed to the PCM. The PCM melts around heat pipes first (far left in (c) and (d)), and a liquid front then propagates outward. Between 2-3 p.m., the PCM fully melts and reaches temperatures 47°C above the melting point. Evening discharge begins at 3 p.m., during which the PCM solidifies and cools to a minimum PCM temperature (at 10 p.m.) of 10°C below the melting point. Overnight, the liquid outer region of PCM remains largely static, with minimal heat loss (98.6% annual storage efficiency) and highly uniform temperature.
profile remains constant in this overnight standby mode, and minimal heat loss results in an annual storage efficiency of 98.6%. Specific to STEALS, temperature gradients are minimal at all times because of the high thermal conductivity of Al-Si, which leads to high exergetic efficiency (Supplementary Figure 20). The system is able to turn on and off within minutes, resulting in the potential to improve overall grid flexibility akin to spinning reserves.

### 3.5.3 Thermal Valve

We have published detailed modeling of the thermosyphon thermal valve separately [103] and incorporated the results of this modeling into our full system MATLAB model. In the “on” state, the heat flow efficiency of the valved thermosyphon is over 99% and the temperature drop across the valved thermosyphon is 26°C. Both the high heat flow efficiency and small temperature drop contribute to high exergetic efficiency. The temperature drop is dominated by solid state conduction through the evaporator and condenser plates, which contributes ∼80% of the 26°C temperature drop.

In the “off” state, the temperature drop across the valved thermosyphon is ∼525°C. Such a large temperature drop is achieved by fully sequestering the sodium working fluid above the mechanical valve to prevent liquid sodium from reaching the evaporator. This leaves the primary heat transfer paths as conduction through vapor tube walls and insulation (∼80%) and radiation through the vapor tubes (∼20%). Comparing “on” and “off” states, the STEALS thermal valve has an on/off thermal conductance ratio of ∼10^4. Additionally, this valved thermosyphon approach is attractive due to its simple, scalable design coupled with minimal O&M cost and parasitic electrical power demands.

### 3.5.4 Risks

The STEALS technology is not fully proven and our projected subsystem performance and cost should be considered preliminary. Several technical advances must be experimentally demonstrated before our model is validated. Risks exist for each subsystem, and until solutions to these risks are developed, our results represent the possible performance of an
idealized future system rather than a mature existing technology.

One risk is our assumed heliostat field cost of 75 $/m². This value is lower than costs of existing plants (for example, the Ivanpah Solar Power Facility at 170 $/m²) [18] and current heliostat manufacturer claims (Stellio, Vast Solar at 100 $/m²) [114], and our assumption relies on significant progress in heliostat development. Another risk is that our Al-Si PCM has not been demonstrated for extended periods of time under the conditions that we model. We assume that the material is stable with cycling, and has no thermal degradation or segregation of individual elements over time. These effects have not appeared in other experiments [44], though could develop over time. An additional risk for our thermal storage system is the reliability of ceramic coatings. The coatings could react with the PCM, fracture during cycling (due to thermal gradients or volume change of the PCM during phase change), or otherwise allow the PCM to corrode containment materials. Extended life cycle tests are necessary to prove that the system will not fail within 25 years of operation. Finally, our model of the thermosyphon thermal valve predicts a thermal efficiency of 99.6%, though this efficiency value and the reliability of the design over time has not been experimentally validated.

In addition to the risks discussed above, there is some uncertainty in our analysis methods. For example, our MATLAB model uses a simplistic thermal resistance network and does not have the accuracy of finite-element analysis. We assume material properties to be constant rather than temperature-dependent. Our cavity receiver design and performance analysis in SolarPILOT is only an initial estimate that could be refined with ray-tracing analysis to improve accuracy. There is also inherent uncertainty in predicting future weather, and our use of TMY data (with hourly resolution) in SAM is not a perfect representation. However, the purpose of this study is only to determine whether or not the system has potential for cost-competitiveness with alternative technologies. Even though these uncertainties are present, they do not affect the general conclusions of this paper.
3.6 System-level performance

To determine LCOE for STEALS and to make an even comparison to alternative technologies, we consider 1) an annual electricity production schedule that complements PV, 2) subsystem contributions to overall cost, and 3) critical design parameters that influence cost and performance.

3.6.1 Hourly electricity production and annual efficiency

One potential market for STEALS is to provide load shifting to complement the growth of PV. Such dispatch control leads to a low capacity factor (∼30%), which means that the power block is utilized much less than in a baseload plant. This results in higher LCOE, but also provides higher value electricity, because of the added benefit of dispatchability. For example, natural gas peaking plants that will be installed on the grid in 2022 have a projected LCOE of 10.5 ¢/kWh, compared to baseload natural gas plants at 5.6 ¢/kWh [115].

Figure 3.5 demonstrates seasonal variation in electricity production from an annual SAM simulation. With 5 hours of thermal energy storage, summer electricity production reaches about 10 hours per day during non-peak sunlight hours. Winter electricity production is lower than summer, but still averages near 5 hours of full-load capacity per day. For locations with high solar resource, this seasonal variation mirrors changes in demand, which peaks in the summer when cooling loads are high [116].

Figure 3.5 also shows that there are also specific days during each season in which the solar resource is particularly high or low, creating corresponding changes in electricity production. On days of high solar resource, restricting dispatch in the middle of the day when storage is full but the sun is still shining forces the system to de-focus heliostats to dump thermal energy from the field. This results in a 7% loss in total available thermal energy on an annual basis. During periods of low solar resource, the energy storage system does not fully charge which results in a shorter duration of production at the end of the day. This is
Figure 3.5: In annual modeling of STEALS, we restricted electricity production to periods of high net demand in the morning and early evening (totalling 8-11 hr daily). Because of variation in solar insolation (from TMY data), actual electricity output varied both seasonally and daily, and did not always meet desired electricity output. In our model of a system with 5 hours of storage, 80% of desired electricity production was annually achieved. Reliability could be improved by increasing heliostat field or thermal storage size, but would come as a tradeoff with cost of these components; alternatively, reliability could be achieved through hybridization with an alternative fuel.
an inherent limitation of CSP, and demonstrates the difference between dispatchability and firm supply of plants that can absolutely guarantee production throughout the year. To reduce the number of low-production days, the size of the solar field and the thermal storage system may be expanded, but this would come at increased cost. In practice, the relative sizes of the solar field, thermal storage, and power block lead to a tradeoff between cost and reliability that can be customized for specific applications. Another alternative is to hybridize the system with an additional source of heat input (e.g. biofuels, propane, natural gas).

Total annual energy efficiency of STEALS includes subsystem efficiencies of the heliostat field (71.2%), solar receiver (95.2%), thermal storage (98.6%), and thermal valve (99.6%). We assume a Stirling engine efficiency of 30%, which includes parasitic power for engine startup and heat rejection. This results in an 18.3% annual solar to electric efficiency (defined as the ratio of net electricity produced to the sunlight energy incident on heliostats). For comparison to conventional designs, the Sunshot CSP power tower design has a lower heliostat field efficiency (50%), lower solar receiver efficiency (92%), similar thermal storage efficiency (99.5%), and higher power block efficiency (55%) than STEALS, resulting in an overall annual system efficiency of 23.6% [113]. The main limitation of STEALS is its low power block efficiency, but it is able to improve on heliostat field efficiency by employing a smaller field, and improves upon receiver efficiency by using a cavity rather than external receiver. In addition, STEALS nearly eliminates parasitic power requirements for engine start-up, heat rejection, and pumped heat transfer fluids, which consume 8.3% of electricity generated by conventional CSP plants.

3.6.2 Levelized cost of electricity

STEALS has the potential to be cost-competitive with current technologies because of its integrated design. Our bottom-up LCOE analysis (Supplementary Tables 5-6 and Supplementary Figures 8-17 and 22-24) results are given in Figure 3.6, and consider a range of power block efficiencies, power block costs, and system sizes. This allows our analysis to
be extended from Stirling engines to other power blocks that have different performance or cost (Supplementary Table 9), which is useful because several other modular heat engines are currently under development. We also note that the cost results in Figure 3.6 are based on large-scale production with low manufacturing cost, and represent idealized future costs rather than the cost of a STEALS plant that could be built today.

In Figure 3.6(a), we quantify the impact of Stirling engine efficiency on the LCOE of STEALS. We use efficiencies from 23-35% to provide a range from slightly below current Stirling engine performance to possible future performance with a minor improvement in technology. The lower LCOE data points for STEALS result from a power block cost (including heat rejection) of 1 $/W, while the upper LCOE data points result from a power block cost of 1.77 $/W, based on a capital cost of 1 $/W and replacements after 10 and 20 years. Over this range of power block efficiency and cost, the LCOE of STEALS is 8.1-12.8 ¢/kWh. The primary effect of improving power block efficiency is a reduction in the thermal power rating of the storage system and heliostat field. This leads to fewer heliostats, less PCM, and less mass held on top of the tower, all of which reduce cost.

Figure 3.6(a) also compares the LCOE of STEALS to other technologies: natural gas combustion turbine plants, conventional CSP towers and troughs, and PV with batteries. A detailed description of how we generated data points (which we then surrounded with approximate shapes to give an idea of cost ranges) is given in the Methods section and in Supplementary Table 10. For natural gas and PV with batteries, we used established cost models to evaluate each technology with a 30% capacity factor under scenarios that range from plants that exist today (darker shading) to future projections of performance and cost that may be achieved within the next few years (lighter shading). For CSP, the data points in Figure 3.6(a) are from existing plants with capacity factor between 20-60% [100, 110]. The bottom right corner of the CSP cost range represents potential future costs and comes from a model of a system that meets Sunshot goals but is operated with 30% capacity factor (instead of the Sunshot goal of 67%) to reach an LCOE of 10.5 ¢/kWh.
Figure 3.6: STEALS has the potential to be cost-competitive with both renewable technologies and natural gas peaking plants at a capacity factor of 30%. In this figure, we present LCOE of both STEALS and alternative technologies, based on the methods described in the Methods section and Supplementary Table 10. For each technology, we calculated LCOE and then surrounded our calculated values with a general cost range to give an idea of how costs compare. Color gradients represent the transition from what could be built today (dark colors) to what might exist in a few years (light colors). (a) The LCOE of STEALS falls with increasing power block efficiency, and depends significantly on power block cost. The low cost STEALS data points assume a power block capital cost of 1 $/W. The high cost data points assume 1.77 $/W, based on a capital cost of 1 $/W and replacements after 10 and 20 years. Both low and high data points correspond to a 400 kW electric system. (b) The LCOE of STEALS is nearly independent of system size. This is due to the modularity of Stirling engines as the power block. High and low LCOE curves are generated with Stirling engines efficiencies of 23 and 35%, respectively. Stirling engine cost is 1.36 $/W, corresponding to a 1 $/W capital cost and replacement after 15 years.
Our calculated LCOE of STEALS is close to this value, but at a much lower power block efficiency. STEALS must make up for this lower power block efficiency with higher optical efficiency, lower parasitic power requirements, and lower balance of system costs. In a more broad comparison between the various technologies in Figure 3.6(a), we see that STEALS has future potential to being cost-competitive with both renewable dispatchable technologies and natural gas peaking plants.

In Figure 3.6(b), we examine the impact of system size on the LCOE of STEALS. In this figure, the lower STEALS LCOE curve corresponds to a Stirling engine efficiency of 35%, and the upper LCOE curve corresponds to a Stirling engine efficiency of 23%. We find an optimum LCOE of 9.5 ¢/kWh at a size of 400 kW_{electric}, though we also observe LCOE to be nearly independent of system size: there is just a 3% variation in LCOE over the full range of 100 kW_{electric} to 1 MW_{electric}. This is due to the modular nature of the Stirling engine as a power block, and the scalability of the thermal storage system.

Figure 3.6(b) shows that STEALS has potential to be competitive with natural gas peaking plants and the future of CSP technology at a much smaller scale. This small scale and the modularity of STEALS is attractive for five reasons. 1) Because the cost of STEALS is nearly independent of system size, it may be specifically designed for small scale market applications that are not available to conventional CSP. Half of microgrids in the United States require less than 1 MW_{electric} [23], and a large CSP plant would be under-utilized and excessively costly for these applications. 2) Small scale results in low capital cost (in the range of $300k for a 100 kW_{electric} STEALS system), which makes crowd-sourced community-based solar farms or other funding options attainable. 3) Low capital cost reduces the risk of financing new projects compared to the billions of dollars of capital required for conventional CSP. Finally, 4) rapid technology progress and 5) rapid realization of economies of scale are possible through higher throughput manufacturing of small plants.

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3.7 Conclusion

There are many ongoing efforts to reduce cost and/or increase the value of electricity from CSP. These include increasing temperature to improve efficiency, lowering cost through use of new materials, and using the thermal storage system for multiple grid services, among others. The technology herein, STEALS, proposes to compete with peaking plants by introducing modularity without sacrificing system efficiency. Locating a cavity receiver, high energy density latent heat storage, and power block all on top of a small scale tower results in high optical and thermal efficiency to balance the lower conversion efficiency of a Stirling engine. It also leads to low balance of system costs and allows for passive heat transfer that reduces operation and maintenance requirements. The design is capable of scaling up or down in size (10-1,000 kW_{electric}) with minimal design changes, and incorporates a novel thermosyphon-based thermal valve to make the system dispatchable.

Building on our preliminary development of this concept, this new detailed analysis shows that the system has a possibility of low capital cost, as well as low O&M cost, making this design appealing for further development. Additional experimental investigation is required to validate the performance and cost results of our model, and advances in heliostats (cost reduction) and Stirling engines (reliability and cost reduction) are necessary to allow STEALS to reach the performance and cost values that we present here. While our results represent an idealized future system, we are optimistic that with future work STEALS may become cost-competitive with natural gas peaking plants, the expected future of CSP, and of PV with batteries. Large arrays of STEALS plants could be installed as a complement to PV and wind for grid-scale storage, or individual STEALS plants could provide electricity for remote locations or microgrids. With the addition of STEALS into the generation mix, the technology could help enable deeper penetration of renewable electricity from wind and PV, and could accelerate progress in the pursuit of a 100% renewable electricity grid.
3.8 Acknowledgment

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CHAPTER 4
EXPERIMENTAL DEMONSTRATION OF A DISPATCHABLE LATENT HEAT STORAGE SYSTEM WITH ALUMINUM-SILICON AS A PHASE CHANGE MATERIAL

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This paper presents the design and experimental results of our first prototype. Here, our primary goal was to prove that our concept could function at all in practice, and that our overall system design was feasible. Our results demonstrated just what we intended.

1) Small temperatures gradients within the PCM illustrated the value of Al-Si as a PCM for its high thermal conductivity. (2) We showed effective on/off control with the valved thermosyphon when integrated into a full system. (3) We integrated subsystems together with minimal exergetic losses.

These results showed that our design was physically possible to build, and demonstrated strong performance in the thermal storage and heat exchanger subsystems. However, the primary limitation of this prototype was low thermal-to-electric conversion efficiency of the thermoelectric generators. Also, our heaters were under-sized, so each “simulated day” took about 72 hours, leading to lower than desired thermal efficiency. In summary, this prototype provided positive feedback on our thermal storage system design, but also identified a few ways that we could improve our second prototype (see Chapter 5).

7Concept development
8Design and construction of prototype and test system
9Performed experiments
10Data analysis
11Wrote paper

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4.1 Abstract

In this work, we present the design, construction, and experimental results of a prototype latent heat thermal energy storage system. The prototype consists of a thermal storage tank with 100 kg of the aluminum-silicon eutectic as a phase change material, a valved thermosyphon that controls heat flow from the thermal storage tank to the power block, and thermoelectric generators for conversion of heat to electricity. We tested the prototype over four simulated days, where each day consisted of four phases of operation: charging, discharging, simultaneous charging and discharging, and storage. Our results show three major conclusions. First, the thermal energy storage system was able to receive and distribute heat with small temperature gradients - less than 5°C throughout the thermal storage tank. Second, the valved thermosyphon was able to effectively control heat transfer, demonstrating an on/off thermal conductance ratio of 430. Third, the interfaces between subsystems had small temperature drops: of the ∼ 560°C temperature drop from the thermal storage tank to the heat rejection system, ∼ 525°C occurred across the power block. This work overcomes the challenges of integrating previously-developed subsystems together, providing a proof-of-concept of this system.

4.2 Introduction

Energy storage is likely to play a prominent role in future electric grids that will generate a significant portion of their supply from renewable sources. By compensating for the variability of generation from photovoltaics and wind power, energy storage can reduce the cost of grid operation and resolve the inefficiencies of turning generators on and off [11]. In doing so, energy storage may also create the added benefit of reducing carbon emissions from the electricity sector.

Of the potential options for energy storage, thermal energy storage (TES) combined with concentrated solar power (CSP) appears to be one of the lowest-cost solutions. Current costs for TES are 20-25 $/kWh, and many pathways are being investigated that have potential to
reduce cost to 15 $/kWh [18]. This cost becomes larger when the efficiency of converting heat to electricity is considered, but still outperforms the most likely alternative technology to provide grid-scale energy storage: electrochemical batteries. Battery costs are \( \sim 300 \) $/kWh today, and are likely to remain above 150 $/kWh based both on experience rates [16] and on limits of material costs [15].

Sensible heat storage with molten salt as a storage material is the state-of-the-art for TES with CSP, and has been implemented in commercial plants that are operating today, such as the 110 MW Crescent Dunes plant in Tonopah, Nevada. This technology takes advantage of inexpensive materials and mature processes, but is limited to relatively low temperatures (\( \sim 550^\circ C \)) and large plant sizes (>100 MW) [22]. Further development for improved efficiency and reduced cost would require new salt storage materials and new components throughout the system that are compatible with high temperatures.

Latent heat storage (LHS) is a potential alternative that can achieve higher energy densities than sensible heat storage and can operate nearly isothermally, which is appealing for both cost reduction and for high power block efficiency. No commercial CSP plants currently use LHS, but significant research-level progress been made in both phase change materials (PCMs) and heat transfer designs for LHS systems. Within materials development, both salts and metal alloys have received attention [36–39]. Salts can be low cost materials, but have low thermal conductivity, which increases the cost of heat exchangers and has encouraged research on thermal conductivity enhancements [38, 39, 41–43]. Metals solve this issue because they have high thermal conductivity, but are very corrosive, and no systems have been demonstrated as stable for extended periods of time. Within heat transfer design development, difficulty is presented by the solid-liquid phase change, but several solutions have been experimentally explored: traditional heat exchangers with pumped fluids [44–46], encapsulated PCMs [48, 49], pool boilers [50, 51], and heat pipes [24, 25, 27, 28, 53–56, 118]. The first two of these approaches are actively driven systems, whereas pool boilers and heat pipes are passive. Here we focus on the latter, because they avoid parasitic power usage and
reduce maintenance requirements.

A comprehensive review of the combination of heat pipes with LHS is provided by [24]. This review shows that the first experimental endeavors for combining solar power, high temperature heat pipes, and latent heat storage were completed for space applications. These efforts included salt PCMs with melting temperatures from 766-980°C, heat pipes with sodium and potassium as heat transfer fluids, and organic Rankine, Brayton, and Stirling engines for the power block [53–56, 118]. Notably, the National Aerospace Laboratory in Japan demonstrated a system efficiency of 20% using LiF as a PCM, sodium heat pipes, and a Stirling engine [56]. Since the review provided by [24], several new projects have also added to this field of work. In particular, Temple University and Infinia Corporation built a latent heat storage system with a Stirling engine, designed for a terrestrial parabolic dish solar concentrator. They demonstrated some initial success, but experienced dry-out issues with their heat pipe design [25]. Sandia National Laboratories also investigated latent heat storage with Dish Stirling [27, 28]. They found potential for cost-competitiveness, but required further work on flexible high temperature heat transfer pipes and material compatibility between their Cu-Mg-Si PCM and containment materials.

In the designs mentioned above, heat pipes were useful for their very high effective thermal conductivity, but did not allow for complete control of heat flow and did not result in entirely dispatchable power generation. However, recent advances have allowed heat pipes to be more controllable, with designs such as variable conductance heat pipes, pressure controlled heat pipes, and diode heat pipes, among others [119]. In previous work, we also demonstrated a valved thermosyphon that could control the flow of the internal heat transfer fluid and thereby act as a “thermal valve” [103, 120].

Building upon the progress in latent heat storage and heat pipes discussed above, we developed and investigated a concept called Solar Thermal Electricity via Advanced Latent heat Storage (STEALS) [73, 101, 102, 117, 121, 122]. This concept is depicted in Figure 4.1. In an effort to overcome previous issues encountered when integrating latent heat storage
Figure 4.1: Solar Thermal Electricity via Advanced Latent Heat Storage (STEALS) integrates a thermal storage system (including a tank of phase change material (PCM) and a valved thermosyphon for heat flow control) with a power block and a cavity receiver on a solar power tower. STEALS uses a heliostat field 1,000× smaller than conventional solar power towers, and passive heat transfer mechanisms via heat pipes and thermosyphons.
with a dish concentrator, STEALS locates its thermal storage on top of a small scale solar power tower. A heliostat field reflects sunlight to a cavity receiver, where heat is absorbed at the bottom of a thermal storage tank. Heat is distributed to a PCM by heat pipes within the thermal storage tank. Above the PCM is a valved thermosyphon, which acts as a “thermal valve” between the thermal storage tank and the power block. This valved thermosyphon allows complete de-coupling of sunlight collection and power production, unlike other storage systems that keep running after the sun goes down but are not able to turn on and off on demand. The STEALS design is attractive because it has low operation and maintenance requirements, and our techno-economic analysis has demonstrated that it has the potential for low cost, dispatchable electricity generation at a modular scale (< 1 MW) [73].

In this paper, we present the design and construction process of a prototype of the STEALS system, and give experimental results from 4 consecutive simulated days of operation. Heat was input from a resistive heater, delivered to an aluminum-silicon phase change material by sodium heat pipes, and controllably dispatched by a valved thermosyphon to thermoelectric generators which converted the heat to electricity. Our results provide a proof-of-concept of the STEALS technology: the thermal storage system was able to receive and distribute heat with small temperature gradients, the valved thermosyphon was able to effectively control heat transfer, and we observed small temperature drops at interfaces between subsystems.

### 4.3 Prototype design and construction

Our prototype design involves a thermal storage tank with 100 kg of aluminum-silicon alloy and sodium heat pipes for even heat distribution. This thermal storage tank is connected to a valved thermosyphon for heat flow control, and thermoelectric generators (TEGs) for conversion of heat to electricity. Figure 4.2 shows the computer-aided design of the prototype within our experimental test chamber. In the following section, we describe each subsystem of the prototype in detail: the thermal storage system, valved thermosyphon, TEGs, and heat exchanger. We then explain our construction process for integrating these subsystems.
together into the final prototype.

Figure 4.2: Computer-aided design of the STEALS prototype and test chamber. The prototype thermal storage tank is 16 inches in diameter and height, and is filled with 100 kg of the aluminum-silicon eutectic as a phase change material. Three heat pipes extend down from the top of the thermal storage tank and spread into a conical shape at the top and bottom for increased area for heat transfer. Above the thermal storage tank is a valved thermosyphon, which is used to control heat flow from the PCM to thermoelectric generators (TEGs). The TEGs are used to convert heat to electricity in this prototype, but also provide a thermal resistance that may represent alternative heat engines that could be used. Within the heat pipes and valved thermosyphon, red arrows represent sodium vapor flow and blue arrows represent liquid sodium flow.

4.3.1 Thermal storage system

Our prototype thermal storage system is based on the STEALS concept and includes a storage tank, PCM, and heat pipes. To maximize the size of the system within laboratory constraints, we designed the storage tank to have a diameter and height of 16 inches. Because of its high energy density, high thermal conductivity, and low cost, and based on previous
techno-economic analysis of our full system [73], we selected the aluminum-silicon eutectic alloy (88% aluminum, 12% silicon) as our PCM. We used the 4047 aluminum alloy (11-13% silicon) because it was inexpensive and readily available in the form of welding rod, which we could easily cut into small pieces before adding to our storage tank. Our own differential scanning calorimetry experiments on this alloy (Figure 4.3) showed its reliability with melt/freeze cycling and determined its heat of fusion to be near 470 J/g, which is within the range of published values for eutectic aluminum-silicon (462-560 J/g) [36, 44, 123, 124]. Based on the size of the storage tank, we used 100 kg (0.376 m³) of PCM to reach a capacity of 13 kWh of thermal energy storage.

Figure 4.3: Differential scanning calorimetry data collected over ~50 melt/freeze cycles of the 4047 aluminum alloy (11-13% silicon) used as the PCM for this prototype. The phase change temperatures of 563°C and 582°C for melting and freezing, respectively, are near the phase change temperature of 577°C for the aluminum-silicon eutectic (12% silicon) established in the literature. This data also indicates the heat of fusion of this alloy stabilizes near 470 J/g after cycling.
Rather than using a pumped fluid for transferring heat into and out of the PCM, we selected a more passive method of heat transfer using heat pipes. In previous modeling, we determined that the optimal spacing of heat pipes to minimize cost of a full scale STEALS system was 13-22 inches [73]. Applied to our prototype scale, this would result in a design with only one heat pipe. However, to add redundancy and to more accurately mimic the assembly process of a full scale system, our prototype design used three heat pipes arranged symmetrically within the thermal storage tank. The heat pipes were attached to the top of the storage tank and extended down into the PCM, as shown in Figure 4.2. Because the heat pipes were not connected to the bottom of the thermal storage tank, they were allowed some movement during cycling which reduced the risk of mechanical failure due to thermal expansion. Attachment to the top of the tank also allowed the heat pipes and thermal storage tank to be coated individually with a protective ceramic coating, before the heat pipes were placed inside of the tank. The bottom of the heat pipes was 1/4” from the bottom of the thermal storage tank, leaving a small gap that was later filled by PCM. The heat pipes spread into a conical shape at the bottom of the tank to increase surface area for improved heat transfer from the absorber surface, and at the top of the tank to increase surface area for heat transfer to the valved thermosyphon. Photographs of the heat pipes used for this prototype are shown in Figure 4.4; they were 16 inches long overall, had a diameter of 1 inch in the middle section (the straight tube length was 12 inches long), and increased to a diameter of 6.2 inches at the largest section of the conical end pieces.

The materials we selected for the thermal storage system were a critical component of the design. To match the operation temperature of the aluminum-silicon PCM (melting temperature of 577°C), we selected sodium as the working fluid for the heat pipes [125]. Both the heat pipes and thermal storage tank were constructed of 304 stainless steel, and two layers of #100 mesh (316 stainless steel) lined the inside of the heat pipes to improve sodium distribution and provide nucleation sites for evaporation. We had both the exterior of the heat pipes and interior of the storage tank (all surfaces that would be exposed to PCM)
Figure 4.4: Heat pipe design and construction process. (a) We welded the outer shell of the heat pipes together in-house before coating with (b) a plasma-sprayed ceramic coating (magnesia-stabilized zirconia, Metco 210NS-1). (c) After coating, we welded the heat pipes to the evaporator of valved thermosyphon. The heat pipes were 16 inches long overall, had a diameter of 1 inch in the middle section (the straight tube length was 12 inches long), and increased to a diameter of 6.2 inches at the end.
coated with a plasma-sprayed ceramic coating to prevent corrosion between the steel and the PCM. For this coating, we selected a high density magnesia-stabilized zirconia material (Metco 210NS-1). A nickel-chromium-aluminum bond coat (0.03-0.04” thick) was deposited on the surface of the steel parts, then the ceramic coating (0.012-0.015” thick) was deposited by several passes of a robotic spray gun over the surface as the part rotated on a turntable. Previous corrosion tests with this coating showed stability with the PCM in a molten state for >1000 hours (no visual change before and after exposure of this coating on a 3/8” diameter stainless steel rod in molten Al-Si at a temperature of 650°C), but the coating was unproven on the specific geometry and exact thermal cycling conditions of this prototype.

4.3.2 Valved thermosyphon

The key to heat flow control in the STEALS design is the valved thermosyphon. The valved thermosyphon allows the system to turn power on and off regardless of when sunlight is collected, making STEALS fully dispatchable. Rather than pumping a heat transfer fluid through a heat exchanger to extract thermal energy from the PCM, this design regulates heat flow by a more passive method shown by the arrows in Figure 4.2. Liquid sodium on the evaporator plate evaporates from heat input at the interface with the thermal storage tank. It then travels through the three vapor tubes and condenses on the top interior surface (condenser) that interfaces with the power block. The liquid sodium is collected by a funnel and directed through a liquid return tube to travel by gravity back to the evaporator. To turn heat flow from the PCM tank to the power block off, the only action required is to close the valve in the liquid return tube, blocking sodium from returning to the evaporator and preventing the sodium cycle from continuing. We previously modeled [103] and experimentally validated [120] this design, though this prototype is our first attempt to integrate this valved thermosyphon with a thermal storage system and power generation.

Our prototype valved thermosyphon (Figure 4.5) was constructed of 304 stainless steel, and like the heat pipes, used sodium as a working fluid. Three layers of #100 stainless steel mesh were tack-welded to the evaporator surface to encourage sodium to spread evenly
Figure 4.5: The valved thermosyphon is the key to heat flow control in the STEALS design. (a) Interior view of the valved thermosyphon evaporator space, showing three symmetric vapor tubes and one liquid return tube in the middle. (b) Interior view of valved thermosyphon condenser space, showing three vapor tubes that extend into the space and the condensate return in the center of the funnel. (c) Full view of the valved thermosyphon with support structure and TEG/heat exchanger array. To transfer heat from the PCM to the power block, sodium evaporates at the evaporator plate, travels upward through the three vapor tubes, and condenses at the condenser plate. Liquid sodium is collected by a funnel that directs flow back to the evaporator through a liquid return tube. To turn this heat transfer pathway off, the valve in the liquid return tube is closed, causing sodium to collect above the valve and preventing the cycle from continuing.
across the surface and provide nucleation sites for sodium evaporation. The valve in the liquid return tube was an off-the-shelf component (Swagelok part number SS-12UW-TW-HT) made of stainless steel. The overall height of the valved thermosyphon was 14.3 inches. The overall diameter was 16 inches. The vapor tubes had a diameter of 1 inch, located 4 inches from the center of the valved thermosyphon. The liquid return tube also had a diameter of 1 inch.

4.3.3 Thermoelectric generators and heat exchanger

In this prototype, we used thermoelectric generators (TEGs) as the power block for conversion of heat to electricity. TEGs are solid-state heat engines that have historically had low efficiency, but have improved significantly in recent years [126] and have potential for low-cost power generation [127–129]. Further, because they have no moving parts, TEGs inherently require very little maintenance, which is attractive for applications in space or other remote locations. Significant improvements have recently been made in laboratories [130], though TEGs are largely unproven for commercial applications at high temperatures.

To obtain TEGs for this prototype demonstration, we partnered with Marlow Industries who provided us with custom modules that could operate with a hot side temperature of near 600°C.

We arranged 14 TEGs on top of the condenser plate of the valved thermosyphon, as shown in Figure 4.6(a). All 14 TEGs had a Bi$_2$Te$_3$ module on the cold side. On the hot side, 5 of these were cascaded with a high temperature skutterudite module, 2 were cascaded with a non-functional skutterudite module, and 7 used an insulation dummy. The cascaded TEGs used a 4-terminal configuration. The insulation dummies were two layers of 1/16” thick alumina-silica fiber insulation sheets, and were meant to mimic the thermal resistance of the active skutterudite TEG modules which were limited in availability. For the skutterudite TEG modules, we placed a 1/16” thick sheet of graphoil between the TEG and the condenser of the valved thermosyphon for improved thermal contact. Similarly, we used a thermal paste (Techspray 1977-DP) between the skutterudite and Bi$_2$Te$_3$ modules, and between the Bi$_2$Te$_3$
Figure 4.6: The prototype TEG array consisted of 14 TEGs, each of which used a Bi$_2$Te$_3$ TEG module on the cold side. (a) Five of these TEGs (numbers 2, 7, 11, 12, and 14) were cascaded with a high temperature skutterudite module on the hot side. For the other TEGs, we used either non-functional skutterudite TEGs (numbers 8 and 13) or insulation (numbers 1, 3, 4, 5, 6, 9, and 10) to simulate the thermal resistance of the skutterudite TEGs. (b) The cascaded TEGs used a skutterudite TEG module on the hot side and a Bi$_2$Te$_3$ TEG module on the cold side, with a thermal paste (Techspray 1977-DP) between the two TEG modules. (c) We clamped each TEG to a pressure of 15 psi using four compression springs and a clamping plate that pressed a copper-water heat exchanger (HEX) directly into contact with the cold side of the Bi$_2$Te$_3$ TEG. To determine heat flow through the TEGs, we measured water flow rate through the HEX with a flow meter, and the temperatures at the inlet and outlet of each TEG with thermocouples ($TC_{in}$ and $TC_{out}$, respectively).
modules and the cold side heat exchangers. The overall dimensions of each TEG was 4cm wide x 4cm long x \( \sim 1 \) cm tall. Detailed photos are given in Figure 4.6(b).

Each TEG was clamped in place by four compression springs that applied a pressure of 15 psi to the stack of TEGs, heat exchanger, insulation, and clamping plate (Figure 4.6(c)). Each of the 14 copper-water heat exchangers had an inlet with flow perpendicular to the TEG at one corner and a serpentine flow path for water to reach the outlet at the other corner. Each TEG was also individually instrumented with hot and cold side temperature measurements, and wiring for current and voltage measurement. This allowed for performance evaluation of each TEG independently, in addition to the entire array together.

4.3.4 Prototype construction process

Construction of this prototype began with fabrication of the thermal storage tank and heat pipes. The thermal storage tank was made of two pieces (a 16” diameter, 4” tall deep drawn cup and a 16” diameter, 8” tall rolled cylinder) welded together to reach the final dimensions of 16” diameter and 16” height. To construct the heat pipes, we first had the conical shapes at the ends of the heat pipes formed by metal spinning. We then inserted two layers of #100 stainless steel mesh into the straight tube and the cone at the heat pipe evaporator before welding the parts together into the shape shown in Figure 4.4(a). We then sent the thermal storage tank and heat pipes to a coating company (Oerlikon Metco) to deposit the magnesia-stabilized-zirconia coating (M210NS-1) on the surfaces that would be exposed to PCM.

The next step in our construction process was to form and weld together the top components of the valved thermosyphon. Then, once they were coated, we welded the heat pipes to the evaporator plate of the valved thermosyphon (Figure 4.4(c)), tack welded #100 stainless steel mesh on the interior surface of the valved thermosyphon evaporator plate, and welded this evaporator plate to the rest of the valved thermosyphon. With the shell of the heat pipe/valved thermosyphon assembly complete, we then prepared the interior of these components. We cleaned all surfaces with hot water and soap, rinsed with a series of...
solvents (acetone, isopropyl alcohol, methanol, and water), and passivated the evaporator surface with a 30% nitric acid solution [125]. We then added approximately 80 g of sodium to each heat pipe and 120 g of sodium to the valved thermosyphon. We filled sodium one at a time by evacuating each component, melting sodium, and injecting it in with a custom apparatus [122].

After moving the prototype into our test chamber, we added PCM to the storage tank by cutting 4047 aluminum rods (3/16” diameter) into 4-8 inch long pieces and inserting these pieces into the tank by hand. We filled the interior spaces of the valved thermosyphon with loose fill insulation (Figure 4.7(b-c)) and arranged guard heaters and radiation shields around both the thermal storage tank and valved thermosyphon.

With the prototype in position, we performed an initial system heat-up to melt the PCM, then added more PCM into the storage tank, and repeated two more times until the tank was filled with 100 kg of PCM. During these initial heat-ups, we ran diagnostic tests on our safety and control systems, and obtained initial experimental results [117]. We also de-gassed
the heat pipes and valved thermosyphon, by pulling vacuum on each component while at operating temperature, to remove impurities and non-condensable gases. Before running the final experiment to obtain the results we present here, we added more sodium so that approximately 180 g of sodium were inside the heat pipes and 360 g of sodium were inside the valved thermosyphon.

4.4 Experimental design and analysis methods

Here we describe our test chamber and heaters, measurement system, experimental design, and data analysis methods.

4.4.1 Test chamber and heaters

To test the performance of our prototype, we ran experiments in a vacuum chamber, primarily for safety concerns in the event of sodium leaks from failure of the heat pipes or valved thermosyphon. The interior dimensions of this chamber were 36” wide x 36” long x 48” tall, leaving space for the prototype, heaters, radiation shields, and water and wire feedthroughs. Before heating up the system to run experiments, we evacuated the chamber to 100 mTorr using two Edwards E2M28 rotary vane diffusion pumps, then to below 10 mTorr using a Brooks CT-8 on-board cryogenic pump.

The thermal storage tank was supported by a steel stand with 82% of the area directly underneath the tank cut out by water-jetting (Supplementary Figure S1). We placed a resistive heater (Thermcraft PH-C-12.5OD) beneath this tank stand, with a 1/2” space between the heater and the bottom of the tank, to simulate radiative heat input from the sun. We painted the heater and the bottom of the tank with Pyromark 2500 paint to increase emissivity and subsequent heat transfer from the heater to the thermal storage tank.

One concern we had in designing this experiment was that because of its small scale, this prototype has a higher surface area to volume ratio than a commercial scale system would have. To replicate the modeled thermal loss of a 200 kW system, which could reach a storage efficiency of 99% with ~0.5 m of standard kaowool insulation [73], would require
the prototype to use 1000 km of the same insulation. A more practical 0.1 m of insulation on the prototype would result in a storage efficiency of 74%, with heat losses from the sides of the tank 25 times higher than would occur on a commercial scale system (see calculations in Supplementary Information).

To overcome this difficulty, and make the prototype behave more closely to how a larger scale system would, we placed resistive guard heaters around the thermal storage tank (Thermcraft VFR-180-18-14-V-S) and the valved thermosyphon (Thermcraft VFR-180-18-13.7-V-S). We planned to control the temperatures of the guard heaters to be slightly below the temperatures on the outside of the thermal storage tank and valved thermosyphon, and thereby simulate different amounts of insulation to represent different designs that could exist for future larger scale systems. However, early on in our experiment, we measured the temperature of the top of the PCM guard heater to be hotter than the top of the thermal storage tank, even without any electrical power input to the guard heater. To avoid actively putting heat into the system with the guard heaters, which would not be representative of operation of a real system, we shut off the PCM guard heaters after initial heat-up. Similarly, because of the temperature gradient between the evaporator and condenser of the valved thermosyphon, we found that the valve guard heaters were putting heat into the valved thermosyphon. To prevent this from continuing, we shut off the valve guard heaters part-way through the experiment. Then for the remainder of the experiment, both sets of guard heaters simply acted as passive insulation, which led to high thermal losses and reduced efficiency relative to a full scale commercial system. In Section 4.5.2, the effect of turning off the guard heater around the valved thermosyphon can be seen in Figure 4.8(d); the temperature of “Valve Guard” distinctly drops after it was turned off, between the “Simultaneous” and “Discharge” phases of simulated “Day 2.”

4.4.2 Measurement system

To measure temperatures throughout the prototype and test system, we built a LabView program to read data from an array of 105 thermocouples. All thermocouples were
type K, with some exposed junction and some ungrounded sheathed thermocouples. These thermocouples were connected to NI 9213 thermocouple modules in a National Instruments CompactDAQ chassis which directed measurements to a desktop computer for data storage.

To measure electrical power output from the TEGs, we used a separate LabView measurement system that read current and voltage data from an individual TEG at a time, and continuously switched between the array of 19 TEGs (14 Bi₂Te₃ TEGs and 5 skutterudite TEGs). This system used Keithley multimeters, a low-current multiplexer for voltage, a high-current multiplexer for current, and a National Instruments CompactDAQ chassis with NI 9207 modules for current and voltage measurement. We measured both current and voltage output from a single TEG at a time, and used the multiplexers to switch measurement from one TEG to the next. For the high temperature skutterudite TEG modules, we swept voltage from -0.8 to 1.3 and back to -0.8 V with steps of 0.3 V. For the low temperature Bi₂Te₃ TEG modules, we swept voltage from -2.2 to 4.1 and back to -2.2 V with steps of 0.9 V. Each voltage sweep took about one minute, so each module was measured once every 20 minutes for the duration of the experiment. To convert current (I) and voltage (V) data to TEG power output, we used a linear fit to the I-V curve from each measurement to estimate the maximum power point, and used this as the power output of the individual TEG module at that time.

4.4.3 Experimental design

Our primary goal in this experiment was to expose the prototype to conditions similar to those that would be experienced in commercial operation, and determine whether or not the STEALS concept could, in practice, function as intended. Specifically, we wanted to assess (1) whether the thermal storage system could receive and distribute heat with small temperature gradients, (2) if the valved thermosyphon could effectively control heat flow, and (3) if interfaces between subsystems would lead to significant temperature drops or other losses. To test the performance of our prototype based on these metrics, we ran an experiment of four simulated days. We defined the beginning of a simulated day as the time
at which the PCM was completely solid and at a temperature of 560°C, and had just begun heating up. Each day then consisted of four phases:

- **Charging.** We input heat from the primary heater to the bottom of the thermal storage tank. This represents solar heat input during the day.

- **Simultaneous** electricity production while charging. With the primary heater still on, we opened the valved thermosyphon to transfer heat to the TEGs for electricity production. This represents electricity generation at the end of the day while the sun is still shining.

- **Discharging** from stored energy. We turned the primary heater off and left the valved thermosyphon open to transfer the latent heat of the PCM to the TEGs for electricity production. This represents electricity production at night when the sun is not shining.

- **Storage.** At the end of discharge, we closed the valved thermosyphon and left the primary heater off so the only heat flows were losses from the storage system. This represents an overnight period when electricity demand is low, before the sun starts shining to repeat the cycle.

During the simultaneous phase, we found that the heat transferred out of the PCM (including useful heat transferred to the TEGs as well as losses to the sides of the storage tank and valved thermosyphon) was higher than the heat input to the PCM from the primary heater. Because of this net heat flux out of the PCM, some of the PCM solidified. We then added an additional re-heating phase between the simultaneous and discharging phases, so that at the beginning of discharge, all of the PCM would be liquid. This meant that our experiment had two distinct periods of electricity generation, and two on/off cycles of the valved thermosyphon, for each simulated day.

To further illustrate this experimental design, temperatures of the heaters on the outside of the prototype are provided in Figure 4.8(d). Primary heater input to the bottom of the
thermal storage tank began just before the beginning of the day and lasted through charging, simultaneous, and re-heating phases until the beginning of the discharging phase. When on, the primary heater operated at a temperature of near 780°C. When off, it dropped to a temperature closer to that of the thermal storage tank, near 540°C. As discussed in Section 4.4.1, the guard heaters were not actively powered and simply acted as passive insulation for a majority of the experiment. The PCM guard heater temperature was then slightly lower than the temperature of the outside of the thermal storage tank, near 570°C for most of the experiment but dropping to 500-510°C at the end of the storage phase. Valve guard heater temperatures were higher when the valve was open (550°C for the first day and a half before the guard heater was shut off, and 370°C for the rest of the experiment) compared to when the valve was closed (220-230°C for simulated days 2-4).

4.4.4 Data analysis methods

To calculate the heat flows in Section 4.5.2 and Figure 4.9, we used temperature measurement data and one-dimensional analytical heat transfer models. Full details of our heat flow calculation methods, including uncertainty in the measurements and calculations (which we performed using the root sum squared method), can be found in Supplementary Information, but we provide a brief summary here.

To calculate thermal power delivered to the power block, we used the heat exchanger on the cold side of the TEGs as a calorimeter and calculated heat flow as the product of the water flow rate (measured with an Omega FLR1012-D flowmeter), density and heat capacity of water, and temperature difference between the inlet and outlet of the heat exchanger. To calculate PCM loss and thermosyphon loss, we took temperature measurements on the inside and outside of the guard heaters and modeled the thermal resistance between these locations based on published material properties. We then calculated the heat flow rate through each guard heater as the ratio of the temperature difference to the thermal resistance. To calculate heater power input to the bottom of the thermal storage tank, we used a radiation calculation method based on temperature measurements on the top surface of the primary heater and
the bottom surface of the thermal storage tank. We verified this calculation with an estimate based on the latent heat of the PCM and other calculated heat flows.

4.5 Experimental results and discussion

In the following three subsections, we show temperatures, heat flows, and electrical power output of our prototype during an experiment of four simulated days. Our results show that this system performed well for three key criteria: (1) the thermal storage system was able to receive and evenly distribute heat with small temperature gradients, (2) the valved thermosyphon effectively controlled heat flow, and (3) we found small temperature drops at interfaces between subsystems.

4.5.1 Temperature measurements

Figure 4.8 shows temperatures measured at key locations of the prototype during the four simulated days of operation. Here we describe temperature measurements from bottom to top, following the general direction of heat flow, while considering prototype performance based on the three criteria discussed above.

For our first criterion of prototype performance, we found that PCM temperatures were very uniform throughout the experiment. Figure 4.8(c) shows temperature measurements from five locations inside the thermal storage tank, represented by their radial and vertical positions, respectively: (0", 6"), (3",1"), (6", 6"), (3",12") and (6",12"). These temperatures in the PCM remained within ∼5°C for the duration of the experiment, which is primarily a result of the high thermal conductivity of the aluminum-silicon PCM. This occurred while we observed temperature differences of 10-12°C on the outside of the tank, which we attribute to imprecise thermocouple locations and higher contact resistances for these measurements.

For our second criterion of prototype performance, we found that the valved thermosyphon was able to effectively and repeatably control the temperature delivered from the thermal storage tank to the TEGs. In Figure 4.8(b), we observe that when the valve was closed (during the charging phase), the temperature drop across the interior of the valved ther-
Figure 4.8: Temperatures measured during four simulated days of prototype operation. Each day consisted of 4 phases. Heat was input during the charging and simultaneous phases, and the valved thermosyphon was open during simultaneous and discharge phases. These temperature results demonstrate strong performance in several aspects of the prototype. (a) TEG temperature responded quickly to operation of the valved thermosyphon. (b) The temperature drop from the evaporator to the condenser of the valved thermosyphon repeatably dropped from 310-325 to \(<6^\circ C\) when the valve was opened. (c) All PCM temperatures remained within \(\sim 5^\circ C\) for the duration of the experiment, and were relatively constant while the PCM was changing phase. (d) shows that heat was input from the primary heater at a temperature of 780 °C, and because they were not powered, guard heater temperatures followed the temperatures of the subsystems they surrounded. Throughout the prototype, interfaces between subsystems demonstrated small temperature drops, such that 525 of the 565°C temperature drop from the PCM to the heat exchanger occurred across the TEGs.
mosyphon from the evaporator to the condenser was 310-325°C, and when the valve was open (during simultaneous and discharging phases), this temperature drop was < 6°C.

For our third performance criterion, we consider the temperature drops at interfaces between subsystems. When the system transferred heat from the storage tank to the TEGs to produce electricity, Figure 4.8 shows that the temperature difference between the PCM and the evaporator of the valved thermosyphon was 15-20°C, the temperature difference across the valved thermosyphon was < 6°C, and the temperature difference from the condenser of the valved thermosyphon to the hot side of the TEGs was ~ 15°C. Thus, a majority of the temperature drop, ~ 525°C, occurred across the TEGs, which is desired for high potential Carnot efficiency at the power block.

TEG temperatures in Figure 4.8(a) closely tracked those of the valved thermosyphon in response to valve operation. When the valve was open, TEG hot side temperatures ranged from 500-550°C, and cold side temperatures ranged from 35-95°C. When the valve was closed, TEG hot side temperatures ranged from 110-150°C, and cold side temperatures typically ranged from 25-30°C. These temperature drops show that the thermal resistance of the TEGs was well matched to the heat flows of this prototype, and that the TEGs give an accurate representation of the thermal performance of a commercial system design that may use an alternative power block.

4.5.2 Heat flows and thermal efficiency

Figure 4.9 shows heat flows throughout the prototype during this experiment, calculated as described in Section 4.4.4. Here, we discuss these heat flows and use them to evaluate the ability of the prototype to control heat flow with the valved thermosyphon and to determine subsystem thermal efficiencies.

During the charging phase, thermal power input from the main heater to the thermal storage tank was 900-1000 W. Of this, 280-285 W were lost to the sides of the PCM, 30 W were lost to the sides of the valved thermosyphon, and 50-55 W were lost through the valved thermosyphon to the heat exchanger. Other miscellaneous losses also occurred from
Figure 4.9: Prototype heat flows. During charging, \(~25\text{-}30\%\) of the 900-1000 W of thermal power input from the main heater to the storage tank went into the PCM, while the rest was lost to the sides of the PCM, the sides of the valved thermosyphon, into the heat exchanger, and other miscellaneous losses. During simultaneous and discharge phases, thermal power delivered through the valved thermosyphon increased dramatically as the valve was opened, leading to an associated increase in electricity output from the TEGs.
the tank stand, evaporator plate, and condenser of the thermal valve, but were not directly measured. Based on the heat of fusion of the PCM, and the time it took to melt, we estimate that the remaining 250-300 W went into the PCM as it melted.

Because a commercial STEALS system would receive heat input from concentrated sunlight, we are less concerned with the miscellaneous losses that were not directly measured because they are specific to our test setup rather than providing an accurate assessment of a real system. However, these losses did reduce heat input to the PCM, which increased charging time and reduced prototype efficiencies.

When the valve was opened for the simultaneous and discharge phases, the thermal power delivered to the power block increased from 50-55 W to 380-400 W. Correspondingly, thermosyphon loss increased from 30 W to 135-140 W during these periods. However, PCM loss remained nearly constant at 280-285 W because of the nearly constant temperature of the PCM even during discharge. During the simultaneous phase, heat input from the primary heater was the same as during the charging phase, 900-1000 W, and when the heater was turned off for discharge, the heater cooled down but remained at a slightly higher temperature than the bottom of the storage tank, such that about 60 W of heat was still input from the main heater to the storage tank. Miscellaneous losses from the tank stand, evaporator plate, and condenser of the thermal valve not depicted in Figure 4.9 decreased from 200-350 W to 80-160 W when the heater was turned off, due to the reduced temperature of the tank stand during that time.

This heat flow rate data provides several metrics for the ability of the valved thermosyphon to control heat flow. Considering the increase in heat flow rate and corresponding decrease in temperature drop across the valved thermosyphon when opened, we calculate an on/off thermal conductance ratio of 430 (where thermal conductance is the ratio of heat flow rate to temperature drop). This demonstrates that the valved thermosyphon is able to effectively transfer heat when desired, but limit losses when closed. Another important metric for heat flow control of the valved thermosyphon is “on”/“off” response time. When
the valve was opened for discharge, it took 13-14 minutes for the system to ramp up to 50% of full power, and 47-57 minutes to ramp up to 90% of full power. After the valve was closed to enter the storage phase, it took 55-83 minutes for the system to ramp down to 50% of full power, and 210-253 minutes to ramp down to 10% of full power. The longer shut-down time is due to the process of collecting all sodium above the valve in the condensate funnel, and because of the relatively large thermal mass of the insulation surrounding the valved thermosyphon. However, a turn-on time of 15 minutes opens the STEALS technology to grid services that typically require spinning reserves.

As our final consideration of the ability of the valved thermosyphon to control heat flow, we define its exergy efficiency $\eta_{X,TS}$ as the ratio of exergy out of the valved thermosyphon to exergy into the valved thermosyphon:

$$\eta_{X,TS} = \frac{X_{out}}{X_{in}}$$

where

$$X_{out} = Q_{\text{delivered}}(1 - \frac{T_{\text{amb}}}{T_{\text{condenser}}})$$

and

$$X_{in} = (Q_{\text{delivered}} + Q_{TSloss})(1 - \frac{T_{\text{amb}}}{T_{\text{evaporator}}}).$$

$Q_{\text{delivered}}$ is the energy delivered through the valved thermosyphon to the heat exchanger, $Q_{TSloss}$ is heat lost to the sides of the valved thermosyphon, and $T_{\text{amb}}, T_{\text{evaporator}},$ and $T_{\text{condenser}}$ are the temperatures of the water heat exchanger, the evaporator of the valved thermosyphon, and the condenser of the valved thermosyphon, respectively.

Based on the temperature and heat flow data in Figure 4.8 and Figure 4.9, we calculate exergetic efficiency during the discharging phase ($\eta_{X,TS,\text{dis}}$) to be 70-73%. Over 99% of the exergy loss came from heat loss to the sides of the valved thermosyphon, and less than 1% of exergy loss was due to temperature drop across the valved thermosyphon. This indicates that a commercial scale system with sufficient insulation to limit side heat loss.
would perform with an exergetic efficiency of near 99% [73]. On a daily basis, accounting for losses that occurred during the extended charging period of our simulated day, the daily exergetic efficiency \( \eta_{X,TS,\text{day}} \) of the valved thermosyphon was 27-30%. See Section 4.4.1 for a discussion of insulation losses.

The heat flow data in Figure 4.9 also allow us to determine a variety of thermal efficiencies of the prototype (Table Table 4.1). One such efficiency is an “instantaneous discharge efficiency,” \( \eta_{\text{inst.discharge}} \), which we define as the ratio of thermal power delivered to the heat exchanger, \( \dot{Q}_{\text{delivered}} \), to the total thermal power out of the thermal storage tank, \( \dot{Q}_{\text{PCM tank}} \):

\[
\eta_{\text{inst.discharge}} = \frac{\dot{Q}_{\text{delivered}}}{\dot{Q}_{\text{PCM tank}}}
\]

where the thermal power out of the thermal storage tank can be expanded into components of thermal power delivered, thermosyphon loss \( \dot{Q}_{TS\text{loss}} \), PCM loss \( \dot{Q}_{\text{PCM loss}} \), and heater thermal power input \( \dot{Q}_{\text{heater}} \).

\[
\dot{Q}_{\text{PCM tank}} = \dot{Q}_{\text{delivered}} + \dot{Q}_{TS\text{loss}} + \dot{Q}_{\text{PCM loss}} - \dot{Q}_{\text{heater}}
\]

During the discharging phases of the experiment, \( \eta_{\text{inst.discharge}} \) was 50-53%. However, this efficiency does not account for losses from the tank stand and thermosyphon condenser. A more accurate representation of the ability of the prototype to transfer stored energy to the power block may be the “discharge efficiency,” \( \eta_{\text{discharge}} \), which we define as the ratio of energy delivered to the heat exchanger during an entire discharge period \( (Q_{\text{delivered, discharge}}) \) to the latent heat stored in the PCM (based on the mass, \( m_{\text{PCM}} \), and heat of fusion, \( \Delta h_f \), of the PCM). For all four simulated days, we find that \( \eta_{\text{discharge}} \) was 13-17%.

\[
\eta_{\text{discharge}} = \frac{Q_{\text{delivered, discharge}}}{m_{\text{PCM}} \Delta h_f}
\]

A third efficiency we can consider is the overall system “heat flow efficiency”, or the ratio of all useful heat delivered \( Q_{\text{delivered}} \) (during simultaneous and discharge phases) to heater input for a full simulated day:

\[
\eta_{\text{heatflow}} = \frac{Q_{\text{delivered}}}{Q_{\text{heater}}}
\]
For simulated days 2-4, we calculate $\eta_{heat\text{-}flow}$ to be 4.3-5.6%. However, this is an artificially low representation of the thermal efficiency of the system, partially because it includes losses over a much longer period of time during charging and re-heating than would be experienced by a real system (each simulated day took 74-87 hours, because the primary heater was undersized). Further, the heat flow efficiency of this prototype is lower than a commercial scale system because of its small size and subsequent high surface area to volume ratio. We also note that we could not accurately estimate the overall system heat flow efficiency for simulated day 1, because during the beginning charging period some power was input from the guard heaters but the magnitude of this power could not be estimated.

### 4.5.3 Electrical power output

In Figure 4.9, electricity production from the TEGs when “on” was near 5 W. In the “off” state, electricity production was 1.5-2 W. The high off-state power production is partially due to losses through the valved thermosyphon, but is also a limitation of using TEGs as the power block, because their power output depends only on their hot and cold side temperatures and cannot be further controlled. Other heat engines such as Stirling engines could be actively turned off once the hot side temperature is sufficiently reduced. This would reduce heat flow through the valved thermosyphon when it is not desired and thus increase storage efficiency. We also note that in previous preliminary experiments, TEG power output reached as high as 14 W. The TEGs degraded over time, and had much lower efficiency than expected.

We define “daily TEG efficiency,” $\eta_{TEG}$, as the ratio of TEG electricity output ($W_{TEG}$) to heat delivered to the heat exchanger ($Q_{delivered}$) during simultaneous and discharge phases:

$$\eta_{TEG} = \frac{W_{TEG}}{Q_{delivered}}$$

For all simulated days, $\eta_{TEG}$ was 1.2-1.3%. While they were useful in this experiment for their simplicity, TEGs are not yet commercially available with high efficiency and reliability in the temperature range of our thermal storage system (above 500°C). As a result, the
overall “daily system efficiency,” $\eta_{\text{system}}$, (ratio of electricity output to heat input from the primary heater) was 0.04-0.05%.

\[ \eta_{\text{system}} = \frac{W_{\text{TEG}}}{Q_{\text{heater}}} \]

However, a full scale commercial scale system may use a higher efficiency power block (such as a Stirling engine at 30% efficiency), which would also result in a higher system efficiency.

Table 4.1: Summary of prototype heat flows and efficiencies. In the Charging state, the primary heater was powered, the valved thermosyphon was closed, and losses were meant to be minimized. In the Discharging state, the primary heater was not powered, the valved thermosyphon was open, and heat was intentionally transferred to the TEGs for electricity generation. All efficiencies are defined in Sections 4.5.2 and 4.5.3.

<table>
<thead>
<tr>
<th>Power [W]</th>
<th>Charging</th>
<th>Discharging</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary Heater Input</td>
<td>900-1000</td>
<td>60</td>
</tr>
<tr>
<td>PCM Loss</td>
<td>280-285</td>
<td>280-285</td>
</tr>
<tr>
<td>Thermosyphon Loss</td>
<td>30</td>
<td>135-140</td>
</tr>
<tr>
<td>Thermal Power Delivered to HEX</td>
<td>50-55</td>
<td>380-400</td>
</tr>
<tr>
<td>Miscellaneous loss</td>
<td>200-350</td>
<td>80-160</td>
</tr>
<tr>
<td>TEG Electricity Output</td>
<td>1.5-2</td>
<td>5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Efficiency [%]</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Valved thermosyphon exergy efficiency (discharge)</td>
<td>$\eta_{\text{T},\text{S},\text{dis}}$</td>
<td>70-73</td>
</tr>
<tr>
<td>Valved thermosyphon exergy efficiency (daily)</td>
<td>$\eta_{\text{T},\text{S},\text{day}}$</td>
<td>27-30</td>
</tr>
<tr>
<td>Instantaneous discharge efficiency</td>
<td>$\eta_{\text{inst.discharge}}$</td>
<td>50-53</td>
</tr>
<tr>
<td>Discharge efficiency</td>
<td>$\eta_{\text{discharge}}$</td>
<td>13-17</td>
</tr>
<tr>
<td>Heat flow efficiency</td>
<td>$\eta_{\text{heatflow}}$</td>
<td>4.3-5.6</td>
</tr>
<tr>
<td>Daily TEG efficiency</td>
<td>$\eta_{\text{TEG}}$</td>
<td>1.2-1.3</td>
</tr>
<tr>
<td>Daily system efficiency</td>
<td>$\eta_{\text{system}}$</td>
<td>0.04-0.05</td>
</tr>
</tbody>
</table>

4.5.4 Corrosion results

One of the issues encountered in this experiment was material compatibility between the Al-Si PCM and the steel tank used for containment. If steel is exposed to molten Al-Si, it will dissolve into the melt in a matter of hours. To prevent this from happening in our prototype, we used a MgO-ZrO$_2$ coating (Metco 210NS-1) deposited by a plasma-spray method on the heat pipes and thermal storage tank. However, post-experimental analysis
(see Supplemental Figures S19-S21) showed that in some locations, the coating failed and allowed PCM to come into contact with the steel tank. This failure may have been caused by pinholes left after the coating process, or could have been a result of cracks formed because of thermo-mechanical stresses during cycling. The coating failure led to dissolution of the steel tank and heat pipe walls into the PCM. While we observed no significant impact on the PCM thermal performance and there was no complete tank failure, we found that in some locations the thickness of the steel wall reduced from 0.095 inches to as low as 0.014 inches. It is likely that if we continued the experiment for a few additional cycles, more of the steel wall would have dissolved into the PCM and would have led to a tank failure that would allow PCM to leak out of the thermal storage tank. While the MgO-ZrO$_2$ coating provided a temporary solution for the duration of our laboratory experiments, an alternative design or coating method is likely necessary for long-term operation.

4.6 Conclusion

In this work, we designed, built, and tested a prototype latent heat storage system using the aluminum-silicon eutectic as a PCM and a valved thermosyphon for heat flow control. Over four consecutive simulated days, this prototype demonstrated strong performance for all key metrics that we evaluated. First, the thermal storage tank was able to receive and distribute heat with temperature gradients of $< 6 ^\circ$C throughout the PCM. Second, the valved thermosyphon was able to control heat flow with an on/off thermal conductance ratio of 430 and an exergetic efficiency of 70-73%. Third, temperature drops at interfaces were minimal, such that a majority of the temperature drop across the system occurred across the power block ($525$ of $560 ^\circ$C), which is desired for high efficiency.

Both thermal and electrical efficiency of the prototype were lower than could be achieved by a commercial full scale system. This is primarily due to the small scale of the prototype and lower-than-expected performance of the TEGs. Thermal performance could be improved by scaling up and improving solar heat input, and electrical conversion efficiency could be improved by switching from TEGs to a higher efficiency power block (such as a Stirling...
engine).

Due to corrosion challenges, future success of this technology requires development of a containment system that is compatible with the Al-Si PCM, or use of an alternative PCM. A breakthrough in this area, and further work to demonstrate larger scale systems, may enable STEALS to provide low cost, modular, dispatchable electricity for microgrids or larger scale energy storage.

4.7 Acknowledgements

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4.8 Appendix A. Supplementary Material

Supplementary data associated with this article can be found, in the online version, at https://dx.doi.org/10.1016/j.apenergy.2018.02.067.
CHAPTER 5
PROTOTYPE LATENT HEAT STORAGE SYSTEM WITH ALUMINUM-SILICON AS
A PHASE CHANGE MATERIAL AND A STIRLING ENGINE FOR ELECTRICITY
GENERATION

A paper under review in Energy Conversion and Management.

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This is the key experimental paper of my thesis. Building upon what we learned in the
previous 3 years, we set out to meet our final ARPA-E project goals with this prototype:
power generation for >5 hours at >100 W_e and 9% thermal to electric efficiency. However,
after initial diagnostic experiments, we found that we could vary the engine set-point to
reach higher power levels (over 1 kW_e), and much higher efficiency (18.5% conversion of
stored thermal energy to electricity, and 8.6% efficiency on a daily basis). This was ∼200×
higher performance than in our first prototype, and exceeded the reported efficiency of any
previous experimental demonstrations that combined latent heat storage with a heat engine.

This demonstration was a major advancement in developing our proposed system, and
made our previous prediction of 18% annual efficiency appear to be reasonable. Reviewing
our previous cost modeling from Chapter 3, we still believe that this system has some
potential for commercial development, as long as the technical hurdle of Al-Si containment
(Chapter 6) can be overcome.

12Concept development
13Design and construction of prototype and test system
14Performed experiments
15Data analysis
16Wrote paper
5.1 Abstract

In this work, we present the design and experimental results of a prototype latent heat thermal energy storage system. This prototype used 100 kg of Al-Si as a phase change material with embedded heat pipes for effective heat transfer, a valved thermosyphon to control heat flow out of the thermal storage system, and a Stirling engine to convert heat to electricity. We tested this system for 11 simulated days of operation; each day included charging of the thermal storage tank, simultaneous electricity generation with heat input, and electricity generation from stored heat alone. On each simulated day, we set the engine to a different power level, allowing us to investigate the response of heat pipes and our valved thermosyphon to part-load conditions. The prototype demonstrated a maximum efficiency of 18.5% in converting stored heat to electricity, at a maximum power output of just over 1 kW_e. Extending these results to a commercial scale system with solar heat input, our modeling indicates that a discharge efficiency of over 30% and an annual efficiency of 18% could be achieved. This work confirms previous performance and cost modeling, and demonstrates that this system has potential for future commercial development.

5.2 Background

Energy storage is an emerging component of electric grids, because it could enable high levels of deployment of inexpensive renewable electricity. As photovoltaics and wind power grow, so will the need to compensate for their inherent variability, and the value of energy storage will continue to increase [11]. While many storage technologies are currently being developed, one option with potential for very low cost is thermal energy storage. With an incremental energy storage cost of 15-20 $/kWh [18], thermal energy storage alone or in combination with concentrating solar power (CSP) has a competitive advantage over other technologies for long duration (>5 hour) storage [10, 111]. This means that CSP could contribute significantly to reducing global carbon emissions, because it provides clean energy itself while also complementing growth of photovoltaics and wind power [5].
Though CSP has not been a historically prominent technology (it currently accounts for
≈0.2% of global electricity generation capacity [2]), its value is being increasingly recognized,
and commercial development is expanding [100, 131]. Solar power towers are currently either
planned or under construction in Australia, Chile, China, Greece, Morocco, South Africa,
and the United Arab Emirates. While this is encouraging for CSP generally, these plants
are relatively expensive (>10¢/kWh of electricity generation), and very large (100 MWₑ or
more) [2, 20, 21]. Further, their storage systems are all based on sensible heat storage with
molten salt materials. These salts are stable only in a narrow temperature range (290-560°C
[20]), leading to relatively low energy density and limited potential for high power block
efficiency. This leaves opportunities for further technology development via (i) improving
efficiency by increasing temperatures, (ii) reducing cost with alternative methods of storage
and heat transfer, or (iii) creating modular designs to open CSP to new markets.

One alternative storage method that has received significant attention is latent heat
thermal energy storage. In this method, energy is stored in the phase change of a material,
which is attractive for its higher energy density than traditional sensible heat storage. This
reduces the amount of storage material that must be used, which reduces the size and
resulting cost of the thermal storage system. Conceptually, CSP with phase change materials
(PCMs) has been proposed in combination with many different heat engines for electricity
generation, including organic Rankine, Brayton, and Stirling engines, and thermoelectric
generators [24, 53, 56, 61, 118, 132]. However, only a few prototype systems have integrated
both PCM storage and power generation into a complete system demonstration [26, 50, 51,
54, 133–135]. Further, most of this previous work has used salt materials as a phase change
material (PCM). These salts are generally chosen because they have very low material cost
(sometimes below 1 $/kWh) and reasonably high energy density. However, the critical
limitation of salt materials is their low thermal conductivity, which necessitates advanced
heat exchanger designs in order to achieve high exergetic efficiency and high charge/discharge
rates [27, 38, 39, 41, 60].
As an alternative class of materials that could be effectively used as PCMs, metals have much higher thermal conductivities. This leads to simpler integration into storage systems, with uncomplicated low-cost heat exchanger designs. This advantage has encouraged several studies to focus on metal alloys as PCMs [28, 44, 124, 136, 137]. These studies have generally found that aluminum alloys are especially compelling for their combination of low material cost, high energy density, and high thermal conductivity. Aluminum also has lower embodied energy than other metals, making it a particularly sustainable material [71]. An issue with aluminum alloys, however, is corrosion with conventional metal containment systems; this has led to efforts to find reliable containment systems, including those using ceramics [66, 138, 139], or encapsulation methods [49, 140].

In previous work, we designed a novel modular solar power tower that integrated an aluminum-silicon alloy (Al-Si) into a complete system held on top of the tower. This design is distinctly different from traditional CSP systems because of its small scale (∼100 kWₑ rather than ∼100 MWₑ) and its passive heat transfer with heat pipes rather than pumped fluids. This provides advantages in several areas: ability to target market applications with low power requirements, low capital cost resulting in low risk, rapid technology development through iteration, and low operation and maintenance requirements. We modeled the performance and cost of a commercial system based on this design, and found that it had potential for market competitiveness when operated as a peaking plant to complement electricity generation from photovoltaics [73]. We also built an initial prototype of this system, using thermoelectric generators for converting heat to electricity. This first prototype demonstrated that the system design would function, but had relatively low conversion efficiency from the thermoelectric generators and suffered high thermal losses [117].

In this paper, we present an improved prototype latent heat thermal storage system, with a Stirling engine for higher thermal-to-electric conversion efficiency and improved heat transfer design for higher thermal efficiency. We begin by describing our prototype design, construction process, and measurement system. Then, we present experimental results for
11 simulated days of operation, including temperatures, heat flows, and efficiencies. Finally, we discuss how these results relate to previous work, consider potential future improvements, and extend our results to a commercial scale system. Together, our results and analysis show that latent heat storage in general, and this system in particular, has potential for future commercial success.

5.3 Prototype design and construction

In this section, we describe our prototype design, construction process, and measurement system. The prototype, shown in Figure 5.1 (with a photograph provided in Figure 5.2), included the components of our design that would be located on top of the tower: a thermal storage system, a valved thermosyphon, and a Stirling engine.

5.3.1 Thermal storage system

Our thermal storage system design (PCM, tank, and heat pipes) was centered around our selection of Al-Si as a PCM. We chose the eutectic composition of Al-Si (12% silicon) for isothermal phase change and congruent melting, and used the 4047 aluminum alloy (11-13% silicon) because it was readily available and inexpensive. Based on the energy density of this alloy (470 kJ/kg [117]) and the physical constraints of our test chamber, our prototype included 100 kg of PCM, which corresponds to 13 kWh of thermal energy storage. Including the volume displaced by heat pipes, additional space for the volume change of the PCM upon phase transformation from solid to liquid (5%), and extra head space for ease of construction, our thermal storage tank had both an internal diameter and height of 0.4 m. The tank walls were 1.5 mm thick, and were made of 304 stainless steel.

To transfer heat into and out of the PCM, our heat exchanger design used an array of sodium heat pipes. Previous modeling found the cost-optimized distance between these heat pipes to be 40 cm [73]. For this prototype, that would have resulted in a design with just one heat pipe. However, to demonstrate that the system could function with multiple heat pipes in parallel, we used three heat pipes arranged symmetrically within the storage tank,
Figure 5.1: Our prototype included all components of our design that would be held on top of a solar power tower. The thermal storage tank, made of ceramic-coated stainless steel, contained 100 kg of Al-Si as a PCM to store 13 kWh of thermal energy. Three sodium heat pipes are located inside the thermal storage tank to evenly distribute heat into and out of the PCM. The valved thermosyphon could be opened or closed to control sodium flow (liquid represented by blue arrows, vapor by red arrows) and subsequent heat transfer from the storage tank to the Stirling engine, where the stored thermal energy was converted to electricity. Heaters underneath and to the sides of the thermal storage tank simulated heat input from concentrated sunlight, and all components were contained inside a vacuum chamber during experiments for safety precaution.
18.5 cm apart. Each heat pipe had an overall length of 0.4 m, including a 0.3 m length of straight tubing (2.5 cm outside diameter) in the middle, with a 5 cm tall, 15 cm wide conical shape at either end. The intent of this shape was to minimize the PCM volume displaced while maximizing surface area for heat transfer at the bottom and top of the thermal storage tank, but still using simple geometry that could be easily manufactured. Each heat pipe was made of 304 stainless steel, and was lined on the interior with 2 layers of #100 stainless steel mesh to wick sodium and enhance sodium nucleation. Support ribs were welded to the interior of the bottom surface to limit deformation due to thermal and mechanical stresses. Each heat pipe was filled with 120 g of sodium.

To create a corrosion barrier between the PCM and stainless steel surfaces, both the tank and heat pipes were coated with a plasma-sprayed ceramic coating. The coating material was a magnesia-stabilized zirconia (Metco 210NS-1), with a Ni-Cr-Al alloy (Metco 443) as a bond coat.

5.3.2 Valved thermosyphon

A key to the value of our system design is its ability to turn on and off on demand. To accomplish this, our design included a valved thermosyphon as depicted in Figure 5.1. The red arrows in this figure represent sodium vapor, and blue arrows represent liquid sodium. Starting at the bottom of the valved thermosyphon, a layer of liquid sodium on the evaporator surface receives heat input from the heat pipes of the thermal storage tank. This sodium evaporates and travels up the vapor tubes, then condenses on the heater head of the Stirling engine. The condensed liquid sodium is collected by a funnel, and travels down through a liquid return tube to the evaporator, where it continues the cycle. The liquid return tube has a valve in the middle (Swagelok SS-12UW-TW-HT), which can be closed to prevent sodium from returning to the evaporator, effectively turning “off” heat transfer from the thermal storage tank to the Stirling engine.

We previously presented detailed design and thermal analysis of this valved thermosyphon concept [73, 103], and experimentally demonstrated the idea alone [120] and integrated with
Figure 5.2: Photograph of prototype before addition of side heaters and insulation. The thermal storage tank in the middle of the photograph contained 100 kg of the Al-Si eutectic as a PCM. Above this, a valved thermosyphon provided “on”/“off” control of heat transfer to the Stirling engine at the top of the test chamber which converted the heat to electricity. Underneath the storage tank is a resistive heater; beneath the heater is insulation. For scale, the interior of the test chamber was 0.9 m wide and 1.2 m tall.
an initial prototype thermal storage system [117]. Here, we slightly modified the design to allow incorporation of the Stirling engine into the system. Our primary concern was the engine vibrates during normal operation, and if the vibrations were translated down to the heat pipes, the coating on the exterior of the heat pipes would be at increased risk for mechanical failure. To dampen these vibrations, we integrated flexible tubing (Kurt J Lesker MHT-TE-B03) into the vapor and liquid return tubes. Another design aspect unique to this version of the valved thermosyphon was that we included Swagelok VCR fittings in the middle of the vapor and liquid tubes so that we could construct the valved thermosyphon in two parts, which improved our assembly process. As a final modification from previous designs, we reduced the condenser diameter to match the side of the Stirling engine heater head, to minimize nonessential area where heat losses could occur.

The overall height of the valved thermosyphon was 0.3 m. The vapor tubes and liquid return tube had an outside diameter of 2.5 cm. The evaporator diameter was 0.4 m (matching the thermal storage tank), and the condenser diameter was 0.25 m (just larger than the Stirling engine heater head). The evaporator surface was lined with three layers of #100 stainless steel mesh, and all other surfaces were bare. The interior was filled with 400 g of sodium. All components of the valved thermosyphon were made of 1.6-3.2 mm thick 304 stainless steel.

5.3.3 Stirling engine

For this prototype, our power block was a 3 kW\textsubscript{e} Stirling engine from American Superconductor. This Stirling engine used a gamma-type free piston design for a high reliability, with helium as a working fluid and water cooling to maintain a low temperature at the cold side of the engine. The electrical power output of the engine matched the frequency of the piston movement, $\sim$60 Hz. At full power, the electrical output of the engine was 3 kW\textsubscript{e} (200 VAC\textsubscript{RMS} at 15 A\textsubscript{RMS}), with a design hot side temperature of 700°C. In these experiments, we ran the engine at 0.1-1 kW\textsubscript{e} (40-120 VAC\textsubscript{RMS} and 2.5-8 A\textsubscript{RMS}) with a hot side temperature of 400-550°C. This off-design operation reduced efficiency relative to potential commercial
system performance, in which case the Stirling engine would be custom built and optimized for the power level and temperature.

5.3.4 Prototype construction process

As a first step in our prototype construction process, we fabricated the structural components of the tank and heat pipes by standard drawing, metal spinning, machining, and welding processes. We then polished all interior surfaces of the tank and exterior surfaces of the heat pipes, and used a plasma-spray process to coat these surfaces with a magnesia-stabilized zirconia coating (METCO 210NS-1), including an intermediate Ni-Cr-Al alloy bond coat (METCO 443) between the steel and the magnesia-stabilized zirconia. Next, we constructed all components of the valved thermosyphon, again using standard machining and welding processes. Our design included Swagelok VCR connections in the vapor and liquid tubes so that we could separate the valved thermosyphon into two parts. We then welded the bottom half of the valved thermosyphon to the heat pipes, and the top half to the Stirling engine.

With all structural components of the prototype fabricated, we prepared the interiors of the heat pipes and valved thermosyphon by cleaning and de-greasing these surfaces. We rinsed the components with water, filled them with acetone for ~2 hours, then rinsed again with water and dried overnight with a nitrogen purge. We then passivated the inside surfaces to enhance sodium wetting and improve corrosion resistance by filling the components with 50% nitric acid for ~45 minutes, before rinsing with water and again drying overnight with a nitrogen purge. Next, we filled each heat pipe individually with sodium using a custom sodium charging apparatus. This involved evacuating the heat pipe, melting sodium inside of a separate charging vessel, then opening a valve to allow the liquid sodium to flow into the heat pipes.

To install the prototype inside our test chamber, we began by mounting the Stirling engine, with the top half of the valved thermosyphon attached, onto support blocks at the top of the chamber. We then moved the bottom part of the prototype into the chamber,
and made the connection between the bottom and top parts of the valved thermosyphon. With the prototype in place, we filled the valved thermosyphon with 400 g of sodium using the same process as for the heat pipes. Then, we installed thermocouples throughout the system, and put insulation and radiation shields on the outsides of the heaters and other prototype components.

To fill the thermal storage tank with PCM, we placed rods of solid PCM at room temperature into the tank by hand. However, this only filled the tank with a 40% packing factor, so we used the heaters to melt all of the PCM, cooled the system down, and repeated the process three times until we had the desired amount of PCM inside the tank. This also allowed us to run diagnostic tests to ensure that the measurement system and Stirling engine were running properly.

5.3.5 Test chamber, heaters, and insulation

We tested our prototype inside of a stainless steel vacuum chamber as shown in Figure 5.1. The purpose of this chamber was to reduce safety risk from potential failure of the sodium heat pipes, the water cooling system, or the storage tank. During our experiment, the chamber pressure was maintained near 25 mTorr by two Edwards EM28 rotary vacuum pumps. This operation in vacuum reduced thermal losses compared to a commercial system, but this only had a minor impact on results because the primary thermal resistance to heat losses was still insulation. In a commercial application, the interior of the storage tank would likely be an inert environment to prevent oxidation of the Al-Si PCM, though the system would not be under vacuum.

To simulate solar heat input from a heliostat field through a cavity receiver, we arranged heaters underneath and to the sides of the thermal storage tank. We placed the bottom heater (Thermcraft PH-C-16-OD) in direct contact with the thermal storage tank, using a refractory oxide mud (EA198) to smooth the heater surface and improve thermal contact. This heater was the same diameter as the storage tank, and had a maximum power output of 4 kW at 208 V, though during the experiment we typically operated it near 1 kW. The
two semi-cylindrical heaters on the sides of the tank (Thermcraft VFR-180-18-14-V-S) were spaced 2.5 cm from the outside of the tank, and used exposed heating elements to provide radiative heat input at a maximum power of 1 kW at 110 V.

Because our prototype was $\sim100\times$ smaller than a commercial scale design, it had a higher surface area to volume ratio and therefore would have experienced significantly higher thermal losses than a commercial scale system if we had used standard inexpensive insulation. To partially overcome this challenge, we placed high-performance insulation and radiation shields on the outsides of the heaters and the valved thermosyphon. Our arrangement underneath the bottom heater included a 1 cm thick board of calcium silicate insulation, a gridded steel plate, four sheets of 1 mm thick stainless steel as radiation shields, and a 0.6 m square, 5 cm thick piece of microporous silica insulation (Promalight 1000X). On the outside of the side heaters, we used two wraps of 1 cm thick aerogel insulation (Pyrogel XTE). We also used this same aerogel insulation to wrap the vapor tubes, liquid return, and all interior spaces of the valved thermosyphon. On the exterior, we also placed two sheets of 1 mm thick stainless steel that spanned from the bottom of the chamber to the top of the valved thermosyphon to act as radiation shields.

5.3.6 Measurement system and data analysis methods

The key data to capture from our experiments included temperatures, heat flows, and electrical power input and output. For temperature measurements, we used 105 type K thermocouples, connected to a data acquisition system (National Instruments cDAQ-9178 chassis with NI-9213 modules) that transferred data to a computer via LabView software. These thermocouples were placed inside the PCM (inside of ceramic protection tubes), on the sides of the PCM tank, on the insides and outsides of the heaters, on the insides and outsides of insulation, inside each heat pipe, at 10 locations inside the valved thermosyphon, at the inlet and outlet of the Stirling engine water cooling system, on the exterior of the Stirling engine, and on the test chamber walls. In some locations, we used sheathed and ungrounded thermocouples so that they could be secured in place or would be protected.
from sodium, and in other locations, we used exposed junction thermocouples for improved response.

To measure the heat rejected from the Stirling engine, we used its water cooling system as a calorimeter. We placed thermocouples within 15 cm of the inlet and outlet of the cold side heat exchanger, and measured the water flow rate with a flowmeter (Omega FLR6305D). We then estimated heat flow as the product of the volumetric flow rate, water density, water heat capacity, and temperature difference across the heat exchanger.

To determine electrical power input to each heater, we multiplied the supply voltage by current input, which we measured using current transducers (CR Magnetics CR4120-30). Then, to estimate thermal power input from the heaters to the thermal storage tank, we subtracted heat loss from the outside of the heaters. For this heat loss, we measured temperatures on either side of the insulation surrounding each heater, calculated a 1-dimensional thermal resistance of this insulation, and estimated heat loss as the ratio of the temperature difference to the thermal resistance. We used a similar method with temperature measurements and a thermal resistance calculation to estimate heat loss from the sides of the valved thermosyphon, and from the side of the thermal storage tank when the heaters were off during discharge.

To control and measure electrical power output from the Stirling engine, we used a separate system developed by American Superconductor-ITC. The engine controller used an active rectifier operating as a boost converter to rectify the AC power coming from the linear alternator windings, creating a 360 V DC rail. Then, electrical output voltage and power data from the engine was measured with a National Instruments CompactDAQ data acquisition system.

5.4 Experimental results

Here we discuss our experimental results, including temperatures, heat flows, and efficiencies. Our experiment lasted for 11 simulated days, in which each day lasted 32-42 hours and included four modes of operation: (i) charging, (ii) simultaneous electricity generation
with heat input, (iii) re-heating to melt all of the PCM, and (iv) discharge with electricity
generation from stored heat only. For each simulated day, we varied the Stirling engine
set-point, allowing us to investigate performance of the system at different thermal power
levels. This variation was intentionally randomized to avoid any aging effects on trends in
performance. We ended the experiment not due to failure of any prototype components, but
due to malfunction of the rotary feed-through that we used to open and close the valved
thermosyphon.

5.4.1 Temperatures and heat flows

Figure 5.3 provides key temperatures and heat flows measured throughout the prototype
for the duration of our experiment. Figure 5.4 more closely examines this data for simulated
day 4, which demonstrated the highest daily system efficiency.

In the red curves of Figure 5.3(a) and Figure 5.4, we show temperatures measured at
12 different locations within the PCM, at heights ranging from 1-30 cm from the bottom
of the tank, and at radii 0-18 cm from the center of the tank. For the duration of the
experiment, all of these temperatures typically remained within 25°C of the PCM melting
temperature of 577°C (with exceptions during the extended discharges on days 1, 7, and
9). This included super-heating at the end of the charging period up to 590-600°C, and
sub-cooling at the end of the discharge period to 500-570°C. We also observed very uniform
temperatures spatially throughout the PCM. The twelve lines that represent each location
are difficult to distinguish, as they were all measured to be within 10°C for all parts of the
day except the sensible part of the discharge period. This is close to expected performance
from previous modeling, which predicted a temperature drop of ~10°C during discharge for
a heat pipe spacing of 40 cm [73]. These uniform PCM temperatures indicate high exergetic
efficiency of the storage system, and are a result of the high thermal conductivity of Al-Si
as a PCM and the high effective thermal conductivity of the sodium heat pipes.

Figure 5.3(b) shows five temperatures inside the valved thermosyphon: two in the vapor
space near the evaporator, and three in the vapor space near the condenser (i.e. the Stirling
Figure 5.3: Experimental data collected for 11 simulated days of operation. (a) PCM temperatures at 12 different locations typically remained within 25°C of the PCM melting temperature of 577°C, with less than 10°C temperature gradients throughout the thermal storage tank. (b) The evaporator maintained a relatively high temperature for the duration of the experiment, while the condenser temperature fluctuated depending on whether the valved thermosyphon was in an “on” or “off” state. Beyond “on”/“off” control, the thermal power delivered to the engine in (c) was primarily controlled by the Stirling engine, whose effective thermal resistance was modified as we ran it at power levels from 0.1-1 kW as shown in (d).
engine heater head). The evaporator temperature remained above 500°C for most of the experiment, while the condenser temperature fluctuated depending on whether the valved thermosyphon was in an “on” or “off” state. In the “off” state, this condenser temperature was typically 240-260°C, but when turned “on,” we observed a sharp rise up to 450-550°C. This rise corresponded to a similarly sharp increase in thermal power transferred to the Stirling engine, demonstrating effective and repeatable heat transfer control.

Figure 5.3(c) provides data for the thermal power delivered to the Stirling engine. This is the sum of the thermal power measured by the calorimeter at the cold side of the engine and the electrical power output from the engine (Figure 5.3(d)). Each day, we varied the thermal power to the engine using the engine controller; by varying the engine output voltage, we modified the amplitude of the piston stroke, which changed the compression ratio of the Stirling engine and therefore its effective thermal resistance. The resulting amount of heat pulled through the engine in these experiments ranged from 1.1 kW\(_t\) at the lowest power level, up to 4.3 kW\(_t\) at the maximum power level. These thermal power rate extrema corresponded to 90 and 1000 W\(_e\) of electrical power output from the engine.

Building upon these general temperatures and heat flows, in Figure 5.4 we consider the transient response of the system to changes in operation on the fourth simulated day. When the valve was opened at the transition from (i) the charging phase to (ii) the simultaneous phase, the condenser temperature in Figure 5.4(a) and the thermal power delivered to the engine in Figure 5.4(b) rapidly increased. Engine power rose to 50% of maximum power output in \(~6\) minutes, and 90% of maximum power output in \(~18\) minutes. When the valve was closed to turn electricity generation off, we observed a similar response: power output fell by 50% in \(~18\) minutes. However, this included a delay of \(~10\) minutes before there was any noticeable change in power output. This delay was due to the fact that once sodium was collected above the valve, it took several minutes for its temperature to fall below 500°C, the point where the vapor pressure of the sodium sufficiently reduced to cause heat piping to stop.
Figure 5.4: Experimental data collected for the fourth simulated day, which demonstrated the highest daily system efficiency. Temperatures in panel (a) and thermal/electrical power in panel (b) responded quickly to transitions between modes of operation: (i) charging, (ii) simultaneous electricity generation with heat input, (iii) reheating to melt all PCM, and (iv) discharge with electricity generation from only stored thermal energy.
In Figure 5.4, we also observe a reduction in condenser temperature and power output over time during both (ii) simultaneous and (iv) discharge periods. During the simultaneous period on simulated day 4, power output reached a maximum of 770 W and fell to 751 W over 4 hours; during the discharge period, power output reached a maximum of 758 W and fell to 676 W over 3.2 hours. This is an expected response of any latent heat storage system, where the temperature delivered to the power block and resulting power output depends on the state of charge. When fully charged, power output is highest, but this reduces over time as PCM solidifies around heat exchanger surfaces, reducing the temperature delivered to the power block. The effect is relatively small in our system, with \( \sim 10\% \) reduction in power over the course of a full discharge cycle. However, if we had used a PCM with low thermal conductivity (1 W/mK) and a design with the same distance between heat pipes, the thermal resistance of the PCM would have been 0.7 K/W by the end of the discharge (calculated based on 1-D conduction through the PCM). This could have led to as much as a 90\% reduction in power output during a discharge cycle. This illustrates the value of our high thermal conductivity PCM, and the importance of heat exchanger design for successful latent heat storage systems.

As a final note on the temperature and heat flow data shown in Figure 5.3, the extended power generation at low power levels at the end of day 1, 7, and 11 were due to accidental incomplete closing of the valve in the valved thermosyphon. During this time, all of the PCM had solidified, and the system reached a near-steady state in which heat input matched heat losses and a reduced amount of power delivered to the engine.

### 5.4.2 Valved thermosyphon performance

Here, we consider performance specific to the valved thermosyphon in our prototype. First, Figure 5.5(a) shows the average temperature drop through the vapor space of the valved thermosyphon during each discharge period. This temperature drop increased linearly with thermal power at low power levels, as expected for any thermal system. At higher power levels, we observe larger temperature drops that indicate that we are approaching the sonic...
heat pipe limit [125]. For a commercial scale system, the design would have to account for this limit by appropriately sizing the diameter of the heat pipes and valved thermosyphon.

Figure 5.5: Performance of our valved thermosyphon. (a) At low power levels, the average temperature difference across the valved thermosyphon during discharge was as low as 10°C. At high power levels, this increased non-linearly, indicating that we reached the sonic limit, which could be overcome in a future design by increasing vapor tube diameter. (b) The exergetic efficiency of the valved thermosyphon ranged from 90% at a thermal power of 3.7 kW$_t$ to 98% at a thermal power of 1 kW$_t$.

Next, this temperature drop data can be combined with the thermal power during discharge to evaluate the effective thermal conductance of the valved thermosyphon. During discharge, the temperature drop of 10-120°C and thermal power of 1-4 kW$_t$ demonstrate an “on” state thermal conductance of $\sim$50-100 W/K. This is more than two orders of magnitude higher than in the “off” state ($\sim$0.5 W/K). As the system increases in size to a commercial scale, losses in the off state would fall due to the change in geometry, and this “on”/“off”
thermal conductance ratio could increase to as high as $10^4$ \[73\].

Figure 5.5(b) considers the exergetic efficiency of the valved thermosyphon. Here, exergy represents the potential for electricity production from the Stirling engine, which depends on both the amount of heat input and the temperature at which the heat is delivered. Exergy efficiency of the valved thermosyphon is then the ratio of exergy output to exergy input, $X_{\text{out}}/X_{\text{in}}$, where $X_{\text{out}} = Q_{\text{out}}(1 - T_{\text{cold}}/T_{\text{cond}})$ and $X_{\text{in}} = Q_{\text{in}}(1 - T_{\text{cold}}/T_{\text{evap}})$. $Q_{\text{out}}$ is the thermal energy delivered to the engine, and $Q_{\text{in}}$ is the thermal energy input to the valved thermosyphon (the sum of power delivered to the engine and thermal losses to the sides of the valved thermosyphon based on a 1D conduction calculation through insulation). $T_{\text{cold}}$ is the temperature of the cold side of the engine, and $T_{\text{cond}}$ and $T_{\text{evap}}$ are the temperatures of the condenser and evaporator of the valved thermosyphon, respectively. This exergetic efficiency ranged from 90\% at high thermal power levels, to 98\% at lower thermal power levels. Our estimated thermal losses to the side of the valved thermosyphon were quite low ($<10$ W), which means that nearly all of the exergy loss is due to the temperature drop across the valved thermosyphon. For an optimized commercial scale system, this temperature drop would be lower, and the exergy efficiency would be maintained closer to the 98\% value that we present here at low thermal power levels \[73\].

5.4.3 System efficiency

We consider the thermal efficiency of our prototype in Figure 5.6(a). Here, thermal efficiency (discharge) is the ratio of thermal energy delivered to the Stirling engine during the discharge period to the total energy stored in the latent heat of the PCM. Thermal efficiency (daily) is the ratio of the total thermal energy delivered to the engine to the total thermal power input to the thermal storage tank by heaters throughout an entire day. With these results, we observe a maximum thermal efficiency (discharge) of 84\% and thermal efficiency (daily) of 36\%, and see that thermal efficiency increases nearly linearly with power. This is primarily because at higher power levels, the discharge duration was shorter, which reduced the amount of time for thermal losses to occur from the sides of the thermal storage
tank and valved thermosyphon. In a larger system with lower surface area to volume ratio and effective insulation, both of these thermal efficiencies could approach near 99% [73].

![Figure 5.6: Experimental efficiencies. (a) A maximum of 84% of the latent heat stored in the PCM was successfully transferred to the Stirling engine during discharge, and a maximum of 36% of the total thermal power input was successfully transferred to the Stirling engine on a daily basis. (b) Stirling engine efficiency increased with power up to 24%, until very high power levels led to lower hot side temperatures and reduced efficiency. Combined thermal and engine efficiency led to a maximum of 18.5% of stored latent heat being converted to electricity, and a maximum of 8.6% of total heat input over the entire day being converted to electricity.](image)

In Figure 5.6(b), we provide results of both the Stirling engine efficiency and overall system efficiency. Stirling engine efficiency is the ratio of electrical power output to thermal power delivered to the engine, and system efficiency is the product of thermal efficiency and Stirling engine efficiency. System efficiency (discharge) then represents the percentage of latent heat stored in the PCM that was converted to electricity during the discharge period, and system efficiency (daily) represents the percentage of thermal power input from the heaters that was converted to electricity over an entire day.

As power increased in Figure 5.6(b), Stirling engine efficiency initially increased, but then later decreased. This was due to two competing effects: first, efficiency increased at
higher levels of part load (this engine was designed for 3 kW_e), as fixed thermal conduction losses through the outer walls of the engine remained nearly constant while engine power output increased. Second, higher power levels led to a larger temperature drop through the PCM, heat pipes, and valved thermosyphon. This resulted in a reduced hot side engine temperature, which reduced engine efficiency. Combined, these two effects led to a maximum Stirling engine efficiency of 24%, at a power level of 770 W_e and with an average hot side temperature of 499°C during discharge. Considering both thermal and engine efficiencies, the overall system efficiency (discharge) then reached a maximum of 18.5% at an engine power level of 950 W_e, and system efficiency (daily) reached a maximum of 8.6% at an engine power level of 770 W_e.

5.5 Discussion
5.5.1 Comparison to prior experimental demonstrations

Only a few experimental efforts to combine latent heat storage with power generation have previously been completed [26, 50, 51, 54, 117, 133–135]. These systems typically either used thermoelectric generators, Rankine cycles, or Stirling engines for power conversion, at power levels of 1-10 W_e or 1-3 kW_e. They included salt, sugar, or metal PCMs with enough thermal storage capacity for 0.5-5 hours of generation. They also employed a variety of different heat exchanger designs, including systems with pumped oils, heat pipes, and pool boilers.

Compared to these prior systems, our prototype is unique in its dispatchability, with the ability to effectively turn on and off heat transfer out of the thermal storage tank using a valved thermosyphon. In addition, our use of Al-Si as a PCM provides a distinct advantage for its combination of high thermal conductivity, high energy density, and low material cost. Our concept with tower-based storage is also novel, in contrast with previous designs for dish or parabolic trough concentrators.

In our heat pipe design, we learned from and improved upon a previous demonstration that used an NaF-NaCl salt as a PCM [26]. In this previous design, an extensive network of
heat pipes shared a common vapor space. Due to the geometry of the design, sodium was not equally distributed to all regions of the thermal storage system. This led several of the heat pipes to experience dry-out, which limited heat transfer from the PCM in these regions. Learning from this work, we isolated each of our heat pipes individually, so that each had its own internal vapor space and would not dry out of sodium.

Considering the overall performance of previous lab-scale demonstrations, efficiencies have been significantly lower than they potentially could be for commercial systems. Instantaneous efficiency for systems with Stirling engines has reached 15-25% [51, 133], but full system efficiencies (for all previous demonstrations that we are aware of) have been limited to 3% or less [117, 133, 134]. Thus, our performance of 8.6% daily efficiency and a 18.5% discharge efficiency (converting stored heat to electricity) represents a significant improvement for systems that combine latent heat storage with power generation.

5.5.2 Limitations of this prototype

While our prototype achieved strong results, we have also identified a few areas for further improvement. Our PCM, heat pipes, and valved thermosyphon demonstrated small temperature gradients and high exergetic efficiency, but nuances in our design could be optimized to both reduce off-state losses and improve on-state performance. This could lead to enhanced thermal efficiency, higher maximum power output, and improved overall system efficiency.

A significant source of heat loss while charging and storing energy in this prototype occurred through the valved thermosyphon in the “off” state. By using thinner vapor tube walls, improved insulation, or increasing the overall height of the valved thermosyphon, heat losses from conduction could be reduced. In addition, bent vapor tubes would block direct radiation from the evaporator to the condenser, further limiting thermal losses in the valved thermosyphon. Thermal losses from the storage tank were also inherently high due to the small size of our prototype, and the amount of insulation we could use was constrained by the size of our test chamber. In a future application, additional insulation could be added.
at a low cost for improved thermal performance.

Another limitation of this prototype was in its maximum thermal power capacity. We originally designed the prototype for 1 kW, but in actual operation, we transferred up to 4.3 kW from the thermal storage tank to the Stirling engine. At these higher thermal powers, we observed a non-linear increase in temperature drop through the valved thermosyphon and heat pipes, due to sodium vapor velocities approaching the sonic limit. To prevent this from happening in a future system, the diameter of the vapor tubes could be increased to reduce sodium vapor velocity.

Finally, the Stirling engine that we used for these experiments was designed for 3 kW, though due to limitations discussed above, we only ran the engine at part loads of 0.1-1 kW. This resulted in a maximum thermal-to-electric conversion efficiency of 24%, at an average hot side temperature of 499°C (~40% of potential Carnot efficiency). However, optimized Stirling engines have been demonstrated at over 40% efficiency (over 50% of Carnot efficiency) [35], and applying this performance to our operation temperatures would result in a conversion efficiency of over 30%.

5.5.3 Implications for a commercial system

The primary difference between this prototype and a commercial system is size. A commercial system with a storage tank 200× bigger than our prototype would have a 6× lower surface area to volume ratio, naturally leading to a much higher storage efficiency. With additional low cost insulation, improvement to 99% annual thermal storage efficiency would be reasonable, and would match performance of previously demonstrated thermal storage systems [141]. Adding in a Stirling engine efficiency of 30%, a heliostat field efficiency near 70%, and a receiver efficiency of 95%, our previous prediction of 18% annual efficiency also still appears to be reasonable [73].

In this prototype, we showed that the system could quickly respond to desired changes in operation, and could turn on and off in ~18 minutes. A commercial system, with reduced surface area to volume ratio, would increase its thermal mass less than linearly with power
rating. This means that its response time would be even faster: our previous modeling predicted an on/off response time of just a couple of minutes [73]. This would allow the technology to effectively act as a peaking plant for integration of variable generation from renewable electricity.

Generally, our experimental results matched our previous expectations and confirm the potential performance of a commercial system of this design. Reviewing our previous cost analysis inputs, we still project that levelized cost of electricity could reach < 10 ¢/kWh at a 30% capacity factor. That would make this system competitive with alternatives of photovoltaics with batteries and natural gas combustion turbines [73], and provides continued motivation for development of this technology.

5.6 Conclusion

In this paper, we presented the design and experimental results of a prototype latent heat storage system. This prototype included a thermal storage tank with Al-Si as a PCM, a valved thermosyphon for heat transfer control, and a Stirling engine for converting heat to electricity. We tested performance for 11 simulated days of operation, under conditions similar to those that would be experienced in a commercial system. To the best of our knowledge, this is the first published demonstration of a dispatchable thermal storage system that integrates a metal PCM and a Stirling engine. We also provide the first report of system efficiency for multi-day experiments on any system that combines a PCM with a Stirling engine.

Our results demonstrated strong performance and a significant improvement over our previous prototype results [117]. Both prototypes demonstrated effective heat transfer, with minimal temperature gradients throughout the PCM and high exergetic efficiency in the valved thermosyphon. However, this second prototype operated at higher power levels (over 1 kWₑ) and with more effective insulation, improving thermal efficiency from near 15% up to 80%. The use of a Stirling engine in this second prototype also greatly improved thermal-to-electric conversion efficiency. This increased overall system efficiency (daily) from 0.05%
to 8.6%, and system efficiency (discharge) from 0.2% to 18.5%. Considering future scaleup and optimization, it is reasonable to expect further improvement to 18% annual system efficiency. This confirms our previous performance and cost modeling, and shows that this system has potential to be cost-competitive with alternative technologies.

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CHAPTER 6
EXPERIMENTAL STUDIES ON CONTAINMENT OF ALUMINUM ALLOYS FOR THERMAL STORAGE APPLICATIONS

This chapter summarizes experimental work that we completed towards finding a reliable method of containment for Al-Si as a PCM for CSP. Thank you to Corey L Hardin, Robert E Bell, Steven T Christensen, Judith C Gomez-Vidal, Philip A Parilla, Nathan P Siegel, and Eric S Toberer for their help with coating method development, designing and performing experiments, and analyzing results.

The motivation for selecting Al-Si came from the techno-economic and performance modeling that I completed in Chapters 2 and 3. There, I found that the combination of high thermal conductivity, high energy density, moderate material cost, and appropriate melting temperature made Al-Si the best candidate material among all options we considered. However, that analysis assumed that the CSP plant could reliably operate for 25 years, with coatings or other methods employed to contain Al-Si with minimal corrosion over time. This is a significant technical hurdle, and an open problem that is still being investigated.

As a contribution towards solving containment for Al-Si, we carried out a series of tests on several potential materials and coating methods. This included steady-state experiments on monolithic materials to establish thermodynamic stability, and cycling experiments in conditions similar to those that would be experienced in a commercial system. Our results provided some encouraging progress, identified important considerations and points of failure, and found a short-term containment solution that allowed us to build and test a prototype system. However, development of a long-term solution for reliable containment of Al-Si is still on-going, and further work is necessary for future success of this technology.
6.1 Introduction

In developing PCMs for CSP, there are two primary material classes that are generally considered: salts and metals. Salt PCMs have received the most attention, due to their low material cost, which gives them a potential for low thermal storage system cost. As one example, the “Concentrating Solar Power Generation 3 Demonstration Roadmap” exclusively considers chloride salts as leading candidate PCMs [22]. However, salt PCMs typically have thermal conductivity below 1 W/mK, which means that they require advanced heat exchanger designs to achieve sufficient exergetic efficiency for successful overall system performance [97]. Further, salt PCMs often have large volume change when they change phase between solid and liquid (∼25% for some chloride and fluoride salts [39]), which can cause significant mechanical stresses during melting or void formation and reduced thermal performance during solidification. Finally, salts also pose difficulty because they can be quite corrosive to containment materials [68].

As an alternative class of materials, metals may also be considered for use as PCMs in CSP. In comparison to salts, metals generally have similar gravimetric energy density, higher volumetric energy density, higher material cost, and higher thermal conductivity. The appeal of these properties (primarily high thermal conductivity) has led many researchers to identify metal PCMs as an attractive option for latent heat storage [28, 44, 124, 136, 137]. My own analysis in Chapter 2 found the same result, though also importantly found that the trade-offs between the two material classes depend on many assumptions about the entire CSP system. In addition, one outstanding issue with metals as PCMs is corrosivity. For the specific application of thermal energy storage, many metals do not have established containment systems that can reliably last for the duration that a power plant would need to operate.

From the above discussion, it is clear that both salt and metal PCMs pose risks for development. In this thesis, I selected Al-Si as a PCM based on techno-economic analysis, but with the knowledge this was a high-risk, high-reward decision. The high thermal conductivity
of Al-Si created a very strong potential for a simple overall system design with high exergetic efficiency, but also required that we identify a reliable method of containment. As a first step in investigating potential solutions, we reviewed previous efforts towards containment of Al-Si. While I do not provide a comprehensive review here, the following discussion provides context for our subsequent experimental efforts.

Generally, most other metals will naturally form alloys with aluminum, due to the fact that when these metals are exposed to aluminum, it is energetically favorable for the two to mix rather than remain separate. This includes the standard containment material used for CSP thermal storage, steel, which will dissolve into aluminum alloys quite rapidly. Thus, traditional heat exchanger and storage tank designs made from metal materials are not a feasible option for use with Al-Si. Alternatively, most ceramic materials are inert in molten aluminum [66, 138, 142]. This is why in most applications that require processing of aluminum, large ceramic vessels are used for containment. These vessels are typically heated from the exterior by gas-fired burners, and the molten aluminum inside can be poured into molds to solidify before being further processed. The gas-fired burners provide sufficient thermal energy to melt the aluminum, but this design leads to large temperature gradients which degrades thermal efficiency (a minor concern in most applications because energy is just one component of overall system cost). Further, this application does not require efficient heat removal from the aluminum, so internal heat transfer surfaces are not required.

Contrary to the traditional aluminum processing industry, in our solar power tower concept, thermal efficiency during both heat input and extraction have a very strong influence on overall system performance and cost. Therefore, the same methods used in previous industry applications would not be effective for our system. Even with the high thermal conductivity of Al-Si, we must use a heat exchanger design with significant surface area in contact with the PCM to minimize temperature gradients during charging and discharging. During charging, our aim is to minimize receiver temperature and resulting radiative and convective losses; during discharging, our goal is to maximize the temperature and resulting
efficiency of the power block. However, the challenge is that this must also be done with containment and heat exchanger materials that are thermodynamically inert with Al-Si and can withstand significant thermo-mechanical stresses.

Previous research has shown that Al-Si can be contained by methods of micro- or macro-encapsulation [49, 140]. In micro-encapsulation, small particles (36 micron diameter) can be treated by a boehmite treatment (boiling in water followed by drying) to form an AlOOH shell, then subjected to a heat-oxidation treatment (930°C for 6 hours in an oxygen-rich environment) to form an alumina coating on the outside of the particle. Notably, in this process, cracks that form from expansion of Al-Si can be self-healed by exposure to the oxygen atmosphere. The resulting small coated particles could be used in a packed-bed heat exchanger design, or potentially even with heat pipe working fluids. In the second method, macro-encapsulation, Al-Si can be simply placed inside an alumina crucible, and a seal between the crucible and a lid can be made by using an Al film and heating in an environment with oxygen. Similar to micro-encapsulation, the resulting containers of Al-Si could be used with packed-bed or potentially heat pipe heat exchanger designs.

While the above processes could result in reliable containment systems, they each use a significant amount of ceramic material, which decreases the energy density of Al-Si as a PCM, and require further development of potentially complex heat exchangers. As our goal was to avoid these limitations and achieve the highest possible thermal performance for our design, we hoped to identify a containment system that used thin-walled, high thermal conductivity heat exchanger components. One way to accomplish this would be to use metal containment materials with thin ceramic coatings as a corrosion barrier. Such a system had not been previously developed for thermal energy storage with Al-Si as a PCM, and so we put significant effort towards experimentally evaluating potential containment materials and ceramic coating methods. In the rest of this chapter, I summarize this effort, though I also note that this is still a work in progress.
6.2 Experimental results

Here I describe a series of experiments that we performed to evaluate potential materials and coatings for containment of Al-Si. These experiments followed a progression from initial screening tests (short duration, simple geometry) to more rigorous experiments (longer duration, with phase change cycling, on more complex geometries). In chronological order, the samples that we tested included monolithic materials, paintable ceramic coatings, plasma-sprayed ceramic coatings, a chromium nitride coating deposited by physical vapor deposition, and an atomic layer-deposited alumina coating.

6.2.1 Monolithic material screening

To validate compatibility of potential containment materials in aluminum, we ran initial tests in which we placed a variety of samples into baths of molten aluminum alloys and qualitatively observed whether or not corrosion occurred on a pass/fail basis. Our samples included both metals (steel, nickel, and titanium alloys) and ceramics (alumina, zirconia, silicon carbide) in baths of multiple aluminum alloys (pure aluminum, aluminum-silicon, aluminum-nickel, aluminum-titanium). Most of the samples were in the form of cylindrical rods, 15 cm in length and 1 cm in diameter, though some of the ceramic samples had a rectangular cross-section. In our tests, we put each sample into a separate molten aluminum bath at a temperature of near 700°C for ~100 hours in a nitrogen atmosphere.

The results of these tests generally confirmed previous literature. We found that all metal alloys corroded severely, with most samples dissolving almost completely into the aluminum bath in this relatively short duration. However, all ceramic samples demonstrated thermodynamic stability, with no noticeable signs of corrosion or other sample degradation. This encouraged us to perform further experiments on a variety of different coating methods for applying these ceramic materials to steel surfaces.
6.2.2 Paintable coatings

Solution processes can be a very inexpensive method for depositing materials onto surfaces at a large scale. In a variety of ways, a solution that contains the material to be deposited can be applied to a surface then dried, leaving behind the desired material as a coating. One method of applying these coatings is to make a paint solution with the ceramic coating, and apply the paint with a non-marking brush to make a relatively even surface. We tested this method with several paintable ceramic coatings including silicon carbide, boron nitride, yttria, alumina, and zirconia. We mixed these coatings with manufacturer-prescribed ratios of water, applied them to both stainless steel and ceramic samples, and allowed them to cure in air for 2 hours at room temperature followed by 9 hours each at 93 and 427°C. Unfortunately, we found that the coatings did not adhere well either to stainless steel or to ceramic surfaces. Therefore, we did not pursue further testing for compatibility with aluminum alloys. In the future, paintable or other solution processed coatings may still be an effective method for depositing ceramic coatings, but in our work we were not able to identify any effective commercially available coatings, and thereafter turned to other potential coating methods.

6.2.3 Plasma-sprayed ceramic coatings

The next method that we investigated for applying ceramic coatings to steel surfaces was plasma spraying. In this process, the part to be coated was rotated while a robotic spray gun moved vertically along the length of the part and deposits the coating. The spray gun made many passes, depositing several layers of coating, so that any porosity in the coating (5-20%) was staggered and there was no direct pathway from one side of the coating to the other. Though the temperature of the plasma was quite high (16,000°C), once it reached the surface of the part, it rapidly solidified and the part remained at a modest temperature of <250°C. For the coatings that we tested, the final coating thickness was about 0.3 mm. In addition, all coatings were preceded by deposition of a ~0.6 mm thick nickel-chromium-aluminum
bond coat (Metco 443) by the same plasma-spray process.

6.2.3.1 Steady-state experiments

As a first test of plasma-sprayed coatings, we investigated thermodynamic stability during steady-state exposure to molten aluminum. We applied alumina (Metco 105), magnesia-stabilized zirconia (Metco 210 and 210NS-1), and yttria-stabilized zirconia (Metco 204F) coatings to stainless steel rod samples that were 15 cm long with a diameter of 1 cm. Similar to our process for monolithic material testing, we immersed these samples in molten Al-Si at 700°C, with a nitrogen atmosphere, for durations of \( \sim 1,000 \) hours. When we pulled the samples out of the Al-Si baths, we found positive results for all samples. While we observed some wetting of the Al-Si to the outside of the rods, we found no signs of corrosion or failure of the coatings. This verified our previous tests on monolithic samples that the ceramic coating materials are thermodynamically stable with Al-Si. In addition, this proved that the plasma-sprayed coatings could be applied to stainless steel surfaces with good adhesion and sufficient uniformity to prevent the Al-Si from reaching the stainless steel surface.

6.2.3.2 Cycling experiments

In our next step of testing plasma-sprayed coatings, we evaluated mechanical stability upon exposure to phase change cycles of Al-Si. Though we had demonstrated success under steady state operation, there was still significant risk of coating failure due to two sources of mechanical stress. The first of these is the mechanical stress imposed by the phase change of the PCM, which results in a volume change of \( \sim 5\% \) [36]. When the phase changes from solid to liquid during charging, the PCM would naturally expand into available space, but depending on the geometry of the system, some regions of PCM may not have space to expand into. This leads to compressive forces on coating surfaces. Then, when the phase changes from liquid to solid during discharging, the PCM naturally contracts around heat pipe and container wall surfaces, leading to formation of voids and tensile stresses on coating surfaces. In addition to these stresses from phase change of the storage material, a second
source of mechanical stress is due to temperature changes and mismatch of thermal expansion coefficients between the coating material and the steel container and heat exchanger. During thermal cycles, the temperatures of the thermal storage tank, heat exchanger, and coating vary, but because they have different thermal expansion coefficients, they will each expand or contract different amounts, leading to corresponding stresses on each material. For example, steel may have a thermal expansion coefficient of 12 m/mK, while alumina may have a thermal expansion coefficient of 8 m/mK. Then if temperature varies from the original design point, the steel container will expand 50% more than an alumina coating, causing tensile stresses on the alumina coating which could crack rather than remaining in contact with the steel.

For our first mechanical stability tests, we used identical samples as in our initial steady-state tests. We applied the same alumina, magnesia-stabilized zirconia, and yttria-stabilized zirconia coatings to stainless steel rod samples that were 15 cm long with a diameter of 1 cm. We then immersed these samples into molten Al-Si, with a nitrogen environment, and cycled the Al-Si between solid and liquid phases (between 525-625°C) 50 times over ∼1,000 hours. From these tests, we found that samples coated with alumina exhibited significant cracking, with the outer layers of the coating falling off the steel rod. In this case, Al-Si was able to penetrate through the coating, and began to corrode the steel underneath. However, both the yttria-stabilized zirconia and magnesia-stabilized zirconia coatings showed no evidence of failure, and appeared to withstand the experimental cycling conditions. This positive result was likely due to higher mechanical strength of these coatings, and encouraged us to pursue further testing. However, this was a limited sample size, and it is true that optimized plasma-sprayed alumina coatings may have produced a different result.

6.2.3.3 Larger scale experiments

Having identified yttria-stabilized zirconia and magnesia-stabilized zirconia as leading candidate coating materials, we then progressed to tests on physical components more similar in size and shape to those of a commercial system. For these experiments, our samples
included larger buckets with diameters and heights of 20 cm, and a 20 cm long pipe with a 2.5 cm diameter that expanded to 15 cm at the end, mimicking the dumbbell shape of our heat pipe design. To these components, we applied the same magnesia-stabilized zirconia and yttria-stabilized zirconia coatings as in previous testing. We filled the buckets with solid pieces of Al-Si, and in the case of yttria-stabilized zirconia, placed the heat pipe inside of the bucket. We then heated the aluminum until it melted, and cycled the Al-Si through 50 solid/liquid phase changes over the course of \(~\)20 days, with temperatures ranging slightly above and below the Al-Si melting temperature of \(577^\circ\text{C}\) (between \(525-625^\circ\text{C}\)).

The results of our larger scale tests on buckets and a heat pipe are shown in Figure 6.1. In the bottom right corner of Figure 6.1(a), and in Figure 6.1(b)-(c), we see that the yttria-stabilized zirconia coatings failed most severely. In both the bucket and heat pipe, we observed formation of layered cones at many different locations. Due to the geometry of the cones, it appears that growth starts from a single point, then expands with one additional ring layer for each phase change cycle. This failure at a single point may be caused by a defect from the coating process, or due to thermo-mechanical stresses, though the absolute cause of failure is still unknown. However, as will be discussed in Section 6.2.3.4, it is clear that once the Al-Si penetrates through the coating to the steel surface, the steel dissolves into the Al-Si, and this new alloy pushes up the coating from below. Given enough time with the Al-Si in the molten state, this would lead to enough dissolution of the steel wall into the Al-Si that the tank would completely fail and allow the PCM to pour out of the tank, or the heat pipe would fail and the PCM would fill the heat pipe causing it to become non-functional.

Moving to results of the first magnesia-stabilized zirconia coating that we tested, two buckets that were coated with this coating (Metco 210) are shown in Figure 6.1(a). The difference between these two buckets was that they were made by different manufacturers. In the first bucket (top right in Figure 6.1(a)), we observed de-lamination of the coating from the side wall, near the bottom, where the bucket bowed outward due to the manufacturing
Figure 6.1: Larger scale plasma spray coating experimental results. (a) Clockwise from top left, we tested two buckets with a magnesia-stabilized zirconia coating (Metco 210), a bucket of yttria-stabilized zirconia (Metco 204F), and a denser magnesia-stabilized zirconia coating (Metco 210NS-1). We observed mixed results with these buckets and with a heat pipe that was coated with the yttria-stabilized zirconia, shown in panels (b) and (c). In negative test results, layered cones grew from failure points, with one layer per phase change cycle. However, the denser magnesia-stabilized zirconia coating (Metco 210NS-1) appeared to successfully contain Al-Si with no failure.
process before coating. This de-lamination allowed the Al-Si PCM to come into contact with the steel wall, which rapidly dissolved into the PCM and created a hole in the side of the bucket where the liquid PCM eventually leaked out. After this result, we tested a second bucket (top left in Figure 6.1(a)) that was made with more uniformly straight side walls. Unfortunately, this bucket also experienced coating failure, with formation of layered cones (similar to those that formed on the bucket and heat pipe that were coated with yttria-stabilized zirconia). In this case, we found layered cones at several locations, all on the bottom surface of the bucket. This may have been due to higher thermo-mechanical stresses in these locations, which could have caused cracks that allowed Al-Si to reach the steel surface and initiate corrosion. However, the apparently random locations of the layered cones also support the hypothesis that the failure may have been due to defects in the coating process. The true cause of the failure is still under investigation, and further optimization of the coating process may have produced different results.

In our final large-scale test of plasma-sprayed ceramic coatings, we applied a high density magnesia-stabilized zirconia coating, Metco 210NS-1 (~10% higher density than the first magnesia-stabilized zirconia coating), to the bucket shown in the bottom left of Figure 6.1(a). In contrast with the previous negative results, this coating showed no change in morphology or failure at any location after exposure to phase change cycles of Al-Si. While we do not have a definitive explanation for why this coating succeeded under the same conditions as the previous failures, two hypotheses still remain. The first is that the low porosity (5-8% compared to 8-20% for the yttria-stabilized zirconia coating) and high density of this particular coating may have helped reduce the number of coating defects, preventing Al-Si from reaching the steel at any location. A second possibility is that the tank that we tested with this coating was manufactured to higher tolerance, leading to fewer non-uniformities which would have either improved the coating process or reduced thermo-mechanical stresses. Regardless, this coating appeared to effectively act as a barrier between the Al-Si and the steel, and we then used it for our full prototype demonstrations (Chapters 4 and 5).
6.2.3.4 Prototype experiments

In our prototype demonstrations, we coated both the interior of a stainless steel thermal storage tank (0.4 m in diameter and height) and three heat pipes (2.5 cm diameter in a 0.3 m long straight tube region and a 5 cm tall, 15 cm diameter cone) with the high density magnesia-stabilized zirconia coating (Metco 210NS-1). In each experiment, we melted and froze the Al-Si ∼10 times, and held the Al-Si at operating temperature for a total of ∼30 days. During the experiments, we did not notice any degradation in performance or failure from the perspective of the outside of the tank. However, after each experiment, we cut a cross section of the tank and heat pipes to analyze the results from the interior perspective. From the cross section of the first prototype tank, we found a negative result that in some locations the coating had failed and the tank wall had begun to dissolve into the Al-Si PCM. There was no immediate consequence to this failure, but enough dissolution occurred that some locations at the bottom of the tank had reduced in thickness from 2.4 mm to 0.4 mm. If we had continued the experiment for a few more cycles, we likely would have observed a complete failure, allowing the Al-Si to leak out of the tank.

To analyze the failure of the coating in our prototype experiments in more detail, Figure 6.2 shows a cross-section of one of the heat pipes from the first prototype, with the heat pipe still embedded inside the solid Al-Si PCM. In Figure 6.2(a) we can see that the coating failed in multiple locations on the exterior of the heat pipe. Taking a closer look at one of these failed locations in Figure 6.2(b), we find that the coating has been separated from the steel surface, with the PCM between the coating and the steel of the heat pipe. Analysis of this region with energy dispersive spectroscopy is given in Figure 6.2(c)-(i). Here, the elements of the coating (zirconium, magnesium, and oxygen) appear to remain in contact with the steel surface until we reach an area where failure has occurred. In this failure region, the PCM elements (Al and Si) have penetrated underneath the coating, and have reacted with the steel. Both iron (Fe) and chromium (Cr) have dissolved into the Al-Si, though it appears that more chromium is present in the corrosion zone, indicating that its rate of dissolution
Figure 6.2: Prototype coating results. Panel (a) shows a cross-section of one of the heat pipes embedded in PCM after our first prototype experiment. In several locations, including the one shown in panel (b), we observe failure with Al-Si penetrating under the coating, reacting with the steel surface, and pushing the coating away from the steel. Energy-dispersive spectroscopy of this location in panels (c)-(i) reveals the locations where each element is present. The coating (zirconium (Zr), magnesium (Mg), and oxygen (O)) remains in contact with the steel heat pipe until we reach an area where the PCM elements (Al and Si) have come between the coating and the steel. Both primary elements of the steel, iron (Fe) and chromium (Cr), have dissolved into the Al-Si.
into the PCM was higher than that of iron.

The coating failure that we observed in our prototype experiment occurred with the same coating (a high density magnesia-stabilized zirconia, Metco 210NS-1) and under similar conditions as our final bucket test that showed no signs of failure. We have not definitively determined the cause of this difference, but there are several potential explanations. One is that, for the prototype, the coating process led to defects that existed before our experiment started. The geometry was slightly different for the prototype compared to the previously tested bucket, and because the process has not been optimized, the coating may have had defects that led to the failure that occurred. Another possible reason for failure of the prototype coating compared to previous success is a difference in thermo-mechanical stresses on the coating. In the previous successful bucket cycling test, heat was input from the sides and top of the bucket, whereas in the prototype, heat was input from the bottom of the tank. This, in combination with differences in geometry, could have led to differences in mechanical stress on the coating both due to temperature changes and phase changes of the Al-Si PCM. Stress concentration could have caused the coating to crack, allowing the Al-Si to penetrate through the coating to steel surfaces. It is not evident, from the tests that we have done thus far, which of these (or other) theories is the correct explanation of the coating failure, and more work is still required to determine if the failure mechanism can be overcome in the future by modification of the coating process or changes to the system design.

6.2.4 Chromium nitride coatings

In another set of experiments, we tested chromium nitride coatings applied to steel surfaces by a plasma-enhanced physical vapor deposition process. In this coating process, parts are placed into a vacuum chamber with a small amount of gas that is either inert or reactive. A positive electric charge is supplied to the coating material, while the part to be coated is controlled to a negative electric charge. Ions of the coating material then flow to the part, and deposit on the surface. This process is similar to that previously discussed for the
plasma-sprayed oxide coatings, but rather than using a spray gun for deposition, the process involves fixtures and fewer moving parts. In addition, the chromium nitride coatings that we considered were significantly thinner (2.5 microns) than the plasma-sprayed oxide coatings (300 microns).

We were attracted to this process and chromium nitride coatings because of their proven success in aluminum die casting applications. In die casting, molten aluminum is injected into a steel die at high pressure, then allowed to cool and solidify before being removed. In this industry, one issue is “soldering” of the aluminum to the steel die, meaning that the aluminum casting may alloy with and remain stuck to the die. As a solution, the interior of the mold may be coated with a barrier that prevents corrosion and allows the casting to be removed so that the mold may be re-used. The specific coating that we used has been demonstrated to be able to sustain 16,000 cycles of this process [143].

6.2.4.1 Experimental design

To prepare samples for this experiment, I began with straight rods of four different steel materials (alloys 304, 316, 1060, and 440C), and rounded and polished the ends of the rods by hand using buffing wheels and polishing compound (~1 micron grit size). I selected a variety of different steel samples so that I could investigate whether the steel chemistry and properties would have an impact on the effectiveness of the coating. I then had the samples coated with a 2.5 micron thick chromium nitride coating (FortiPhy by Phygen Coatings).

I placed each sample rod (and one control 304 stainless steel sample that was polished but not coated) into an alumina crucible, and filled each crucible with 4047 aluminum-silicon. I used a separate crucible for each rod so that if one of the rods failed, it would not affect the PCM composition that the other rods were exposed to. To measure the temperature of the PCM, I placed a thermocouple inside an alumina tube in the crucible with the uncoated 304 steel sample. I then exposed the samples to 20 melt/freeze cycles in a nitrogen atmosphere-controlled furnace. Each cycle took slightly more than 2 hours, in which I heated and cooled the PCM between 550-640°C - slightly above and below the melting temperature of 577°C.
After this cycling, I turned the experimental set-up upside-down, and re-heated the PCM until it melted out into a catch pan, so that I could remove the samples from the PCM for post-experimental analysis.

6.2.4.2 Results

Figure 6.3 shows initial results from visual inspection of the rods, as well as images of the rod surfaces from scanning electron microscopy, before and after our experiment on the chromium nitride coating. Qualitatively, we observed the formation of a green-blue oxy-nitride film (determined by energy dispersive spectroscopy) on the outside of all rods, as seen in the photographs in Figure 6.3(b) and Figure 6.3(d). We attribute this to the presence of oxygen in the furnace during the experiment (we have seen evidence of oxygen in previous experiments that have used this furnace). More significantly, we observed significant corrosion in the case of the un-coated 304 stainless steel rod (Figure 6.3(a)-(b)), but no change to the surface of the coated 304 stainless steel rod (Figure 6.3(c)-(d)). In many locations of the un-coated sample rod, we observed the formation of layered cones, where the number of layers matches the number of Al-Si phase change cycles (as seen in previous testing of plasma-sprayed oxide coatings). However, no such cones or other corrosion occurred on the coated sample rod. This provides evidence that the coating is both thermo-dynamically stable in the presence of Al-Si and mechanically robust, as the coating was able to withstand exposure to 20 phase change cycles with no effect.

This test provided an encouraging result, and demonstrated that chromium nitride coatings have some potential for future application in containment of Al-Si. However, in addition to this positive result on the coated 304 stainless steel rod sample, we did also observe failure at the rounded corner of two of the rods that we tested (316 and 1060 steel). This failure occurred at a single location for each of the rods, and could have been due to imperfections in the polishing or coating processes, or due to stress concentration at the rounded corner that could have cause the coating to fracture. The exact cause of the failure is uncertain, but it is clear that additional testing, including experiments on larger scale components, is still
necessary before the coating can be fully developed for the application of Al-Si containment.

![SEM images of chromium nitride coating](image)

Figure 6.3: SEM images of chromium nitride coating on the un-coated sample (a) before and (b) after the experiment, and on the coated 304 steel sample (c) before and (d) after the experiment. The coating provided corrosion resistance at all locations on this rod. However failure of other rod samples shows that further testing is necessary to develop this coating for the application of Al-Si containment.

### 6.2.5 Atomic layer deposition coatings

Another potential method for creating conformal coatings of ceramic materials is atomic layer deposition (ALD). In this method, a sample is placed inside of a chamber, and a series of reacting gases are passed through the chamber. The gas reacts with the surface of the sample, forming a coating layer that is exactly 1 atom thick. Then a different gas is passed through the chamber, reacting with the first coating layer to produce a second layer of
coating that is one atom thick. The procedure repeats until the desired coating thickness is reached.

ALD is attractive because it can be applied to very large parts with complex geometries, but still produce a conformal, defect-free coating. This gives it potential for low-cost, scalable coatings. Thus, we performed an initial test of an ALD alumina coating on stainless steel rods to investigate whether or not the coating could withstand exposure to melt/freeze cycles in an Al-Si PCM.

6.2.5.1 Experimental design

We tested four sample rods coated with an alumina coating by atomic layer deposition. For these samples, we used 15 cm long, 1 cm diameter tubes made of 316 stainless steel with closed, rounded ends (Omega SS-38-6CLOSED). We electro-polished two of these tubes for 3 minutes, and left the other two rods un-polished. We then placed all four rods into an apparatus that held the open end of the rod but left the rest of the rod exposed inside an ALD chamber. We performed 500 gas flow cycles at 200°C, which deposited a 77 nm thick alumina coating (measured by ellipsometry).

My method of testing these samples was nearly identical to my method previously described for evaluating the chromium nitride coatings. I placed each sample rod (and one control sample that was not coated) into an alumina crucible, and filled each crucible with Al-Si. I then exposed the samples to 20 melt/freeze cycles in a nitrogen atmosphere-controlled furnace. Each cycle took ~3 hours, in which I heated and cooled the PCM between 545-610°C - slightly above and below the melting temperature of 577°C. After this cycling, I turned the crucibles upside-down, and re-heated the PCM until it melted out into a catch pan, so that I could retrieve the samples for post-experimental analysis.

6.2.5.2 Results

Figure 6.4 presents results from our cycling experiments on the ALD-coated rods. From left to right in Figure 6.4(a), we show a photograph of (i) an as-received rod, (ii) the control
test of an un-coated rod exposed to melt/freeze cycles in Al-Si, (iii) a coated rod exposed to the same conditions, and (iv) a coated rod that was also exposed to the same conditions, but then placed in a bath of 3 molar hydrochloric acid to remove all residual aluminum from the surface. We see a significant amount of aluminum stuck to the exposed rods, but this has been removed from the sample that was placed in the hydrochloric acid bath. Unfortunately, we observe no difference in the results from the un-coated to the coated rod samples. The atomic-layer-deposited alumina coating does not appear to have provided any barrier to corrosion.

While this experiment provided a negative result, it is interesting that as in previous experiments, we observe formation of layered cones on both the un-coated and coated samples. In Figure 6.4(b), we provide scanning electron microscope (SEM) images of these layered cones, and see that there are approximately 20 layers - one layer for each melt/freeze cycle of the experiment. As a different view of this result, in Figure 6.4(c) we show an SEM image of the coated sample that, after the experiment, was placed in a bath of hydrochloric acid to remove all residual aluminum. As the hydrochloric acid reacted with the aluminum, it penetrated underneath the cones and removed material from beneath the surface of the steel rod. This clearly demonstrates that the steel rod had begun to dissolve into the Al-Si, which confirms our previous work discussed above. Finally, I also note that this test was done on a single sample, with one attempt at the ALD coating process. It is still possible that with improved sample preparation and an optimized coating process, this coating and method could still result in a reliable solution for Al-Si containment in the future.

6.3 Discussion

In this chapter, I have summarized several experimental efforts to find a reliable containment system for Al-Si. Initial testing of monolithic materials confirmed that several different ceramics were compatible with Al-Si. With this result, we then investigated several different methods for applying ceramic coatings to steel surfaces, which had potential for both high reliability and high thermal performance. In testing plasma-sprayed oxide coatings, we
Figure 6.4: ALD coating test results. The coating did not appear to have provided any corrosion resistance, as we observe similar results on the control uncoated sample to the coated sample in panel (a). In panel (b), we observe formation of layered cones, with approximately one layer per thermal cycle. After immersion in an hydrochloric acid (HCl) bath to remove aluminum that remained on the outside of the rod, in panel (c) we see dimples that indicate that the steel rod is dissolving into the PCM. Note: the boxes in panel (a) are approximate locations of panels (b) and (c), but are not to scale.
performed steady-state and cycling experiments on rod samples, and larger scale tests on buckets, a heat pipe, and our prototype system. From these experiments, we found mixed results, with success in some tests but also many failures that may have been due to coating defects or thermo-mechanical stresses. Next, we tested chromium nitride coatings deposited by physical vapor deposition, which mostly showed successful prevention of corrosion, but also failed in some locations. Finally, we tested an alumina coating deposited by atomic layer deposition, but this coating did not appear to provide any corrosion resistance.

Throughout our experimental campaign, we did not demonstrate a coating that could reliably contain Al-Si for a long duration, and thus future work is still required for development of this technology. However, multiple methods that we investigated (plasma-spray coatings, chromium nitride coatings) showed some successful results, and other methods (solution processing, atomic layer deposition, and others) could still be further developed and optimized to provide a solution to Al-Si containment. This future work could identify the operating parameters necessary to deposit reliable coatings, or could inform system design changes that could reduce thermo-mechanical stresses. If successful, a low-cost, high thermal performance containment system would enable our solar power tower concept to be further developed with scale-up to a pilot plant and beyond.
CHAPTER 7
SUMMARY AND BROADER IMPACTS

Energy storage could allow deep de-carbonization of the electricity sector, by enabling growth of clean electricity from variable generators such as photovoltaics and wind power. Among energy storage technology options, thermal energy storage is particularly attractive due to its low incremental storage cost. Further, TES can be combined with concentrating solar power as a low-emission and efficient source of heat input. To leverage these advantages, TES for CSP is currently being investigated by universities and in industry, with several competing system designs and fundamental methods of storage. Latent heat storage, which stores energy in the phase change of a material, has significant potential because of its high energy density. This high energy density means that only a small volume of material must be used to store a large amount of energy, leading to compact storage systems with low overall cost potential. However, development of materials and successful system designs for latent heat storage is challenging, and requires significant progress before it can be deployed commercially. With this motivation, my thesis has focused on advancing material development and successfully integrating PCMs into thermal storage systems for CSP.

In the first paper of this thesis, I presented a new method to evaluate the impact of material selection on the performance and cost of concentrating solar power. My holistic system model took all critical PCM properties as inputs, and estimated the resulting LCOE for a system using this PCM. My modeling method used a very general treatment of other system components (field, tower, receiver, heat exchangers, power block) so that it was widely applicable to a wide range of potential CSP system designs. The results of this model quantified the value of each PCM property, and provided other insights that will be useful to the entire field of latent heat storage for CSP. I identified several promising candidate materials that should be considered for further development, and discussed the
explicit impacts that these materials have on the rest of the system. Other researchers will also be able to apply my model to their specific designs, and evaluate different materials to identify the best option for their design.

In the second major contribution of this thesis, I worked with several collaborators to develop a new concept of a solar power tower that could leverage the advantages of Al-Si as a PCM. Our proposed system was distinct from previous work for several reasons. We designed the entire system to be held on top of a tower, to minimize the length of pathways for heat flow, thereby minimizing heat losses and also cost. We invented a new valved thermosyphon design, making the system dispatchable but while still maintaining simplicity and low operation and maintenance requirements. Our design was also unique in our method of solar concentration and system size. Previous work has either attempted to integrate thermal energy storage with a dish concentrator, or has considered designs for large scale CSP. Our system was much smaller (∼100 kW_e rather than ∼100 MW_e, but still used a heliostat field to concentrate sunlight. This allowed us to reach high optical efficiency (70% instead of 50% on an annual basis), but also allowed us to use a simple tower support rather than a moving mechanical arm.

To evaluate this novel concept, I built an integrated performance and cost model with original calculations in MATLAB. My performance model characterized several important aspects of the design, with predictions of small temperature gradients (high exergetic efficiency) in the PCM and valved thermosyphon. My model also provided optimization of heat pipe spacing and insulation thickness. By considering transient responses of the system, my model also confirmed that our design could be effectively used as a peaking plant, with rapid response to desired changes in power output. I also used SolarPILOT to model performance of the heliostat field, and SAM to simulate performance over the entire lifetime of the plant. Finally, my techno-economic analysis evaluated the potential LCOE of the system. The results showed that, if commercialized, the system has the potential to be cost-competitive with both conventional (natural gas) and renewable (PV with batteries) technologies for
dispatchable electricity generation, but at a flexible small size.

With encouraging results from model predictions and subsystem demonstrations, the next step in my thesis was to demonstrate a fully integrated prototype. For our first prototype, we built the key components of the system that would be held on top of a tower. Our thermal storage system included 100 kg of Al-Si to store 13 kWh of thermal energy, making it one of the larger lab demonstrations that has combined latent heat storage with electricity generation. Our experimental results showed that our subsystems (thermal storage tank, valved thermosyphon, and heat engine) could be effectively integrated together, and demonstrated successful thermal control and uniform temperatures in the thermal storage system. However, the primary limitation of this first prototype was the low performance of the thermoelectric generators we used as a heat engine.

For our second prototype, I led efforts to improve design and experimental methods, with a goal of significantly increasing system efficiency. Due to technology readiness level, we shifted away from thermoelectric generators and instead used a Stirling engine for electricity generation. We also changed our heater design, increasing the rate of heat input, and improved our selection of insulation to reduce thermal losses. The resulting improvement from our changes was significant: we increased power output by two orders of magnitude up to 1 kW, and increased thermal efficiency by more than 5×. We successfully converted 18.5% of stored latent heat to electricity, and demonstrated 8.6% efficiency on a daily basis. These efficiencies were higher than any other previously demonstrated systems that combined latent heat storage with electricity generation, and provided significant progress towards our prediction that we could reach near 30% conversion efficiency and 18% annual efficiency in a commercial system.

To build upon this thesis with future work, a key breakthrough is still required in reliable containment of Al-Si. My experimental efforts investigated several potential solutions for this challenge, and helped identify potential issues and areas for improvement. In initial tests, we showed that standard metal construction materials would rapidly corrode with Al-Si. We
showed that multiple ceramic materials have thermodynamic stability with exposure to Al-Si. We also tested multiple types of coatings, including plasma-sprayed oxides, physical-vapor-deposited chromium nitride, and atomic-layer-deposited alumina. Some coatings appeared to be successful, but we also identified through cycling tests that mechanical strength of the coated material is an important consideration. Broadly, our efforts provided mixed results, and our system still lacks demonstration that it could reliably operate for 25 years. Several different methods have been proposed by other researchers, and could be leveraged for future success. The possibility of finding a solution is certainly feasible, but these ideas still need to be evaluated for our specific concept, with experimental proof of reliability.

The lasting effects of this thesis will be in thermal and cost modeling methods, insights regarding PCMs and their impacts on CSP systems, and experimental demonstration of a new solar power tower concept. My published papers will help inform others about how to select materials and design latent heat storage systems, and provide tools for others to effectively evaluate alternative TES concepts. My performance and cost analysis of our novel solar power tower design will provide motivation for future commercial development, with much lower risk due to successful experimental demonstrations. My work will also inspire development of similar systems that learn from and build upon my progress. Broadly, I hope that the work I have done for this thesis provides a contribution to global efforts to provide clean, reliable electricity and reduce the effects of climate change.
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