Technical and economic feasibility of molten chloride salt thermal energy storage systems

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A techno-economic study is performed to assess the feasibility of molten chloride salt thermal energy storage (TES) systems for next generation concentrating solar power. Refractory liners internally insulate tanks to allow tank shells to be constructed from carbon steel. The liner must not be wetted by salt to maintain predictable thermal properties and manageable heat loss out of the tank. The commercial scale tank liner is an anchored brick and mortar design with expansion joints to accommodate thermal expansion. Finite element analysis is performed to optimize the thermal and mechanical profile of the tank. Equalizing the shell temperature between the water-cooled foundation and the shell wall is necessary to minimize differential thermomechanical stress and lower overall stress values below industrial allowable limits. The cost of the TES system is estimated to be $60/ kWh\textsubscript{th}, which is four times greater than Department of Energy targets. Solutions to reduce system cost and overall risk are proposed.

1. Introduction

The U.S. Department of Energy Generation 3 (DOE Gen3) program seeks to develop higher efficiency concentrating solar power (CSP) plants that can provide cost-competitive, flexible power in the U.S. electric grid \cite{1}. The Gen3 CSP plant proposed herein closely resembles the configuration of current molten salt power towers with two-tank sensible heat thermal energy storage (TES).

A solar plant with a central receiver and molten salt heat transfer fluid generally allows for the highest operating temperatures and greatest efficiencies \cite{2}. When equipped with a TES system (hot and cold salt storage tanks), the tank volume may contain sufficient hot liquid to provide heat energy for electrical power generation for up to 16 h during off-sun periods \cite{3}. The current state-of-the-art (Gen2 CSP) utilizes molten nitrate salt at the heat transfer and storage fluid. This salt however has a maximum operating temperature of approximately 565 °C, and the net thermal to electric conversion efficiency is limited to about 35% due in part to the use of Rankine steam systems for power generation. The enabling innovation in the proposed Gen3 CSP design is the use of a ternary-chloride salt (MgCl\textsubscript{2}/NaCl/KCl) that is stable to temperatures well above the proposed operating point of 720 °C. This enables the use of a more efficient supercritical CO\textsubscript{2} (sCO\textsubscript{2}) closed-loop Brayton cycle, which predicts net-cycle efficiencies of ≥50% \cite{4}.

Chloride salts are favored over other salt blends due to their advantageous thermophysical properties, low-cost, and high natural abundance \cite{5}. Research into molten chloride salts includes multiple power sector applications, including Gen4 nuclear reactor systems, which has investigated and characterized binary chloride salts such as MgCl\textsubscript{2}/KCl and MgCl\textsubscript{2}/NaCl \cite{6}. Previous techno-economic studies have confirmed that the MgCl\textsubscript{2}/NaCl/KCl blend is more suitable for Gen3 CSP systems, as it avoids the high costs and the high vapor pressure of LiCl and ZnCl\textsubscript{2}, respectively \cite{7,8}. Moreover, the ternary chloride is readily available as an industrial grade material, with established protocols to purify the material and reduce the overall corrosivity of the salt \cite{9,10}.

A challenge in realizing next generation molten salt TES is providing mechanical containment of the salt. Current CSP plants operating with nitrate-based salts at 290 °C to 565 °C use carbon and stainless steel for piping and tanks. The strength of containment alloys depends on the temperature of the system. Affordable stainless steels exhibit a rapid
decline in allowable stress as temperatures exceed about 550 °C [11].
While high nickel alloys retain their strength at and above 550 °C, studies have shown that fabricating the tanks from these materials would be prohibitively expensive [5]. Therefore, thermally insulating tanks with an internal liner is necessary to lower the tank shell temperature and enable the use of lower cost steels in tank fabrication.

Molten salt tanks with internal liners have been investigated for decades to provide corrosion protection and thermal management. Early studies involving molten nitrate salt containment at Sandia National Laboratories considered the use of a thin, liquid-tight Incoloy sheet at the salt interface with underlying insulating firebrick to insulate carbon steel shells [12,13]. Similar liner designs have also been proposed, consisting of a stainless steel barrier layer protecting underlying firebrick insulation [14]. These studies have shown that with optimal design an internal liner may not only reduce the cost of the overall system, but also dampen the stress induced on the tank shell [15,16]. More importantly, they act as a design basis for containment of higher temperature chloride salts, for which internal tank insulation is required.

The handling and containment of molten chloride salts was investigated in a previous study [17]. The tank liner used refractory brick to provide internal insulation and protection against a zinc chloride salt by maintaining the wall temperature at 550 °C, allowing the use of stainless steel for the tank wall, thereby substantially reducing the cost of the hot tank. A lesson learned from the study was that porous refractories are readily permeated by the salt, which degrades the insulating ability of the liner. The tank was sized to contain 400 kg of molten salt. However, due to the porosity of the brickwork, the amount of salt that was eventually loaded into the hot tank was close to 2400 kg, of which 2000 kg was absorbed by the brick. This amount is consistent with the 57% porosity of the insulating firebrick (IFB), which corresponds to an empty volume of 1000 L (2200 kg of the molten salt) [17].

Despite the saturation of the liner, the tank did not fail, but did exhibit greater heat loss and necessitated more insulation on the tank exterior, resulting in a hotter shell temperature. However, fully soaked internal insulation is not a viable path forward. Saturation of the bricks causes an increase in the thermal conductivity of the IFB’s, which increases heat flux out of the tank [17]. Such a design requires a much thicker inner liner to reduce heat loss, which adds cost and weight and removes salt from the available working inventory of TES media. A better solution is to protect the IFB from salt intrusion.

To address these concerns, a low-porosity refractory material has been identified, which forms a passivating layer at the refractory/salt interface and inhibits salt permeation to the underlying IFB layer [18, 19]. This study evaluates the techno-economic feasibility of a liquid-tight refractory internal tank liner. The TES system should ideally target the SunShot goal of less than $15/kWhth, using the financial assumptions defined within the On the Path to SunShot study [20].

This work discusses refractory liner design, refractory mounting, expansion joints, roof insulation, and foundation cooling. Heat loss from the salt tank is first estimated by 1-dimensional heat transfer through the multilayer wall, which serves as an initial design point for more advanced tank modelling. Finite element analysis (FEA) further refines the design to predict 3-D thermal profiles and mechanical stress of a commercial-scale, internally lined salt tank.

The refractory liner in this study is a three-component design, composed of a dense, chemically resistant brick at the salt interface (hot face), an IFB layer, and a microporous insulating board adhered to the inner tank shell. "Hot wall" and "cold wall" designs are studied. In the hot wall design the refractory liner is wetted and a stainless-steel tank shell is kept at approximately 500 °C. In the cold wall design the refractory liner is kept dry and a carbon steel tank shell is kept at less than 100 °C. Finally, the overall cost of the full-scale TES system is discussed, in which the material and labor cost are subdivided and reported in $/kWhth.

2. Materials and methods

The salt storage tanks were designed following the procedures of American Petroleum Institute (API) Standard 650, Welded Tanks for Oil Storage. However, API 650 is limited to a maximum design temperature of 93 °C (260 °C if following the additional requirements of Annex M). With the minimum operating salt temperature expected to be around 500 °C, it was decided that additional design requirements were needed for safety. Therefore, allowable stress limits from American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section IID were applied to the tank steel, and FEA methods were used to identify and mitigate high stresses in the tank steel and refractory lining.

It should also be noted that the molten salt storage tanks were designed to have a small internal nitrogen pressure to help with transient events expected to occur in the plant and minimize ingress of ambient atmosphere. While API 650 allows internal pressures as high as 18 kPa when the additional requirements of Annex F are met, the size of the tank limited the internal pressure capability to a maximum of 10 kPa. The tanks were not designed to accommodate external pressure (internal vacuum).

The tank was modelled by dividing the computation into three distinct sub models: ANSYS Fluent (computational fluid dynamic (CFD) software) was used to model the internal fluids; ANSYS Mechanical Thermal was used to model the temperature profiles through the solid surfaces (refractory, tank shell, insulation, etc.); and ANSYS Mechanical Static Structural was used to model the stresses in the tank under thermal and static loads. In addition to the software programs mentioned above, a fourth program, ANSYS Workbench, was eventually used to organize this multiphysics project and map results from one model to another. Further details regarding the ANSYS methodologies are found in the Supplemental Information.

The tank liner must perform two critical functions: (1) protect the tank shell from excessive heat (thermally weakening the shell) and (2) protect the tank shell from chemical attack by the salt (corroding the shell). The two objectives are related because corrosion rates increase with increasing temperature. To estimate this performance, a generic 1-D model was created to predict tank heat loss as a function of liner material properties. The model calculates the insulation thickness necessary to achieve a given heat loss and finds the optimal thickness of internal insulation. The target heat lost from the tank envelope is < 276 W/m^2 to maintain the target TES efficiency of no more than 2% heat loss from the full-scale tank over a 24-h period. Reference Equation S(1) for the full calculation procedure.

The 1-D heat loss calculations used an Excel-based model to estimate TEST costs and compare the cost of different TES configurations. The Excel solver automatically calculates the volume of salt required to meet the desired hours of storage and number of tank pairs using the molten chloride salt thermochemical properties. The tank foundation, shell, and refractory liner are re-sized accordingly. For a given tank configuration, the volume of each type of refractory (material and brick shape) is calculated and costed using material and labor rates provided by solicited quotes. The tank shell and foundation were modelled separately for each configuration under the guidance of an industry consultant. These costs were combined with salt cost estimates from suppliers and estimates for other tank elements, such as internal tank piping, to generate total and $/kWhth costs.

2.1. Theory/calculation

The thermal conductivity of refractory materials is inversely proportional to their porosity. Hence, high-porosity refractories are needed for good insulation. Considering the hydrostatic pressure within the salt tank and the low viscosity and wetting ability of the chloride salts, keeping such a liner dry is a major challenge. Thus, it is important to 1) create an impermeable layer in contact with the salt to protect underlying insulating layers. 2) understand how thermal conductivity of the
liner changes in the event of salt wetting.

Mathematical models have been developed to predict the thermal conductivity of porous materials. These models often consider a two-phase system, composed of a dense solid skeleton and a void space, typically occupied by static air. In such a system at room temperature, the effective thermal conductivity will approach the thermal conductivity of air as porosity of the material increases, i.e. as the physical makeup of the material is increasingly dominated by air-filled pores.

The effective thermal conductivity of porous materials can be calculated using Equation (1) [21], where the porosity of the solid (ε) and the thermal conductivities of the solid skeleton (κs) and air (κa) are known. The parameter X is the proportionality coefficient, which is a function of porosity (ε) and varies depending on the material in question. Equation (2) describes the X value for mullite-based refractory materials [21], and Equation (3) describes the X value for anorthite-based refractory materials [22].

\[
(1 - \varepsilon) \frac{\kappa_s - \kappa_a}{\kappa_s} + \varepsilon \frac{\kappa_a - \kappa_s}{\kappa_s + 2\kappa_a} = 0
\]

\[
X = 0.984 + 0.016 \varepsilon^{0.1912}
\]

\[
X = e^{1 - 1.467 \varepsilon^{0.112}} + 1
\]

The effective thermal conductivity of the overall system changes if the pores are filled with a material with a different thermal conductivity than air, e.g. molten salt. If molten salt infiltrates the refractory and completely wets the pores, the thermal conductivity of air (κa = 0.07 W/mK at 800 ◦C) is replaced by that of salt (κsalt ≈ 0.44 W/mK at 800 ◦C). Using Equations (1)-(3), the thermal conductivity of the refractory skeleton was calculated using materials data from manufacturers and the thermal conductivity of air at 800 ◦C. The thermal conductivities of the material skeletons are then used to calculate the effective thermal conductivity of the refractories wetted in molten salt (Table 1). The dense bricks and reference brick are mullite-base materials, and the insulating firebrick is anorthite-based.

The change in effectively thermal conductivity is directly proportional to the porosity of the refractory. The effective thermal conductivity of highly porous insulating firebrick increases by 285%, whereas the thermal conductivity of denser hot face refractory increases by only by only 2–10%. Accounting for changes in thermal conductivity from salt wetting is essential for estimation of the heat loss from a salt-wetted liner.

The calculated values were validated against the measured increase in thermal conductivity of the wetted IFBs in a reported study [17]. The IFB thermal conductivity increased from 0.43 W/mK to 0.73 W/mK after being wetted by molten salt [17]. The model predicts an effective thermal conductivity of 0.75 W/mK, which is in good agreement with the reported value (0.73 W/mK).

3. Results and discussion

3.1. Down-selection of “cold wall” tank design and thermal modelling

The favored 3-layered liner is based off magnesium industry refractory liners, consisting of a hot face brick at the salt interface, IFB’s, and microporous insulating boards at the refractory/carbon-steel interface [23].

The liner is assumed to be of equal thickness on the floor and sidewall of the tank, and the same liner structure will be installed in both the hot and cold tanks to simplify construction. The hot face of the liner is designed to prevent permeation of molten salt into the insulating layers. With dry insulation, the liner lowers the temperature of the tank shell to allowable levels for carbon steel. The target shell temperature is below 60 ◦C, thereby allowing for an uninsulated outer surface.

Fig. 1 shows a rendering of a lined hot tank (salt at 720 ◦C). This tank is internally lined with the 3-layered refractory liner mentioned above, with roof insulation composed of ceramic fiber blanket.

Both the circle and triangle profiles adhere to the heat loss targets for a tank, which calls for a heat loss less than 2% over a 24-h period (<276 W/m²). However, the wetted “hot wall” liner design is nearly 1.2 m thick (triangles), which is neither cost effective nor practical to construct. Moreover, the salt is predicted to freeze in the microporous insulating layer. This material would likely fail if contacted by salt, resulting in potential failure of the tank. A “cold wall” liner design that becomes wetted (squares) will see heat flux increase to 436 W/m², which dramatically lowers the thermal efficiency of the hot tank. The model predicts that this configuration will also result in a freeze layer developing in the microporous insulating layer. Only a dry “cold wall” liner configuration that is kept dry is deemed as a feasible solution for an internally lined molten chloride tank. Hence the integrity of the hot face material is critical to prevent ingress of salt into the primary insulation regions of the liner.

3.2. Critical components of the salt tank liner

Given the down-selected materials, a wholistic assessment of liner design, thermal performance, durability, and cost was performed. Fig. 2a shows the preliminary engineering drawings for a refractory-lined salt tank. The tank is proposed as butt-welded construction with a bottom-to-shell junction of conventional construction, as shown in industry standards such as API Standard 650 – (Welded Tank Steel for Oil Storage). API 650 requires weld preheating for welds thicker than 1-1/4 inches. Therefore, if the tank could be kept at or below 1-1/4” plate then the cost and labor would be much less. As such, the final tank design utilizes 1-1/4” plate for both the tank bottom and shell.

Fig. 2b and c shows highlighted regions of interest within the tank refractory liner. Fig. 2b shows the wall-to-floor detail, which highlights a radiused brick profile. The radiused brick base will add stability to the brick wall and prevent a hinge from forming where the floor meets the wall. The floor and wall use special tapered key bricks. These are used where the back side is wider than the front side and provide further stability to the wall. Initial thermal modelling of the hot face/IFB/microporous insulation liner showed cold spots in the corners of the tank, which induced mechanical stresses exceeding API 650 limits (Fig. S1). Accordingly, the IFBs and microporous insulation in the tank corner were substituted with lower cost, higher thermal conductivity fireclay bricks (orange colored bricks) and optimized to homogenize the thermal profile between the floor and wall of the tank.

Fig. 2c shows the top of the refractory wall where the wall transitions to the roof of the tank. Important here are the use of marine anchors to fasten the upper refractory wall to the tank shell (Fig. S2). Given the size of the refractory liner, these anchors are required to keep the liner attached to the wall. The marine anchors can be manufactured out of any alloy, such that ensures material compatibility with the salt, and they are not welded to the tank. Rather there will be a small clip welded to the tank wall where the 50 mm (2-inch) handle of the anchor is set into from the top (Fig. S3). This clip can be short piece of steel angle or small pipe. The anchor allows for some movement of the brick upwards while restraining the brick in the horizontal direction. The anchor is positioned such that it remains on the backside of the brick to prevent

<table>
<thead>
<tr>
<th>Refractory</th>
<th>Porosity, ε</th>
<th>κa (W/mK)</th>
<th>κs (W/mK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dense Brick A</td>
<td>6%</td>
<td>1.37</td>
<td>1.43</td>
</tr>
<tr>
<td>Dense Brick B</td>
<td>8%</td>
<td>1.95</td>
<td>2.00</td>
</tr>
<tr>
<td>Dense Brick C</td>
<td>17%</td>
<td>1.34</td>
<td>1.47</td>
</tr>
<tr>
<td>Insulating Firebrick</td>
<td>80%</td>
<td>0.20</td>
<td>0.77</td>
</tr>
<tr>
<td>Reference Brick</td>
<td>57%</td>
<td>0.43</td>
<td>0.75</td>
</tr>
<tr>
<td>Reference Brick (measured) [17]</td>
<td>57%</td>
<td>0.43</td>
<td>0.73</td>
</tr>
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</table>
exposure to the hot gases and liquids. In the presented design, there are two rows of anchors, with each anchor penetrating about 1/3 into the hot face brick.

Fig. 3 shows greater detail of thermal expansion joints. These are required in the design, in order to accommodate the thermal expansion expected in the liner during initial heat up. The mortar joints and thermal-expansion joints are considered major risks for salt penetration into the IFB layer. Corrugated metal expansion joints with flexible graphite have been selected for this application (Fig. 3a). The corrugated metal joints are available in different sizes and alloys. High nickel alloys are preferred, due to their chemical compatibility in molten chloride salt. The expansion joints will be manufactured to meet the specific mechanical requirements of the liner, such as the overall thickness of the joint and the amount of compression need to seal the joint upon liner heat up. When calculating expansion joint size, compression in the mortar joints was assumed to reduce overall lining expansion. In addition, directly under each expansion joint is a thin layer of graphite foil to prevent the brick layers close to the expansion joint from bonding together. These have been proven highly effective as high temperature gasket in containment of at sealing are common use as flexible, high temperature gaskets in petrochemical plants and refineries [24]. The liner floor consists of a series of 12-sided expansion rings that avoid intersections between expansion joints. The expansion joints are placed between bricks periodically throughout the liner in vertical and horizontal orientations (Fig. 3b).

A mortar joint thickness of 2–4 mm is recommended for the hot face bricks. The thin joints will be employed for this application for corrosion mitigation. Similarly, 2 mm mortar joints will be used for backup insulation layers. Inorganic potassium silicate-based mortars are favored for all layers due to their high thermal stability and use in corrosive environments. In addition, this inorganic-based approach avoids the use of aqueous-based binders, which may unintentionally introduce moisture into the system.

3.3. Thermomechanical modelling of salt tank

The mechanical stress on the tank induced by thermal expansion of the refractory liner was modelled using FEA. The early thermal and stress models showed significant overstress areas in the tank wall and floor due to the differential expansion between the refractory and the steel tank (Fig. S1). Using the original three-layered liner (hot face/IFB/microporous insulation), the radiused refractory corner created a cold spot in the floor/wall of the tank. The stresses induced from this temperature gradient exceeded the ultimate strength of the refractory materials.

Total displacement due to thermal expansion was shown to close all expansion joints (Fig. S1a) and yet still be insufficient to reduce stress on the tank wall. The lateral force exerted on the tank wall by the expanded refractory liner was predicted to induce a mechanical stress on the tank wall.
wall greater than 345 MPa (Fig. S1b), which exceeds API 650 limits. Should the tank shell itself not fail immediately, the liner on the tank bottom would heave from the center in the vertical direction by about 400 mm to relieve stress (Fig. S1c).

To address these issues, the IFBs in the tank corner were substituted with superduty firebricks, which evened the thermal profile between the wall and floor. These superduty firebricks are represented in orange in Fig. 2b. Fig. 4a shows ANSYS heat transfer shell temperature results which demonstrate that this configuration results in compatible temperature readings throughout the tank corner (51.7 °C ± 13.4 °C). The ANSYS thermal model provides a shell temperature at specified coordinates (e.g. floor, wall, roof). Knowing thermophysical properties of the salt and the size of the tank, the heat flux is estimated to be in the range of 232 and 260 W/m², which is below the heat flux target set by the 2% heat loss criteria (≤276 W/m²).

The quantity and required compression of the expansion joints was also updated, based on mechanical models. This increases the number of expansion joints by approximately 250%. Expansion joints are now found in the hot face and IFB layers in the floor and wall of the tank, where previously they were only located in the hot face layer. The updates made to both the tank corner and the expansion joints yielded a more favorable tank stress analysis (Fig. 4b). In contrast to the initial model shown in Fig. S1c, the new model predicts maximum stresses ranging between 135 MPa and 161 MPa, which is within the allowable stress limits of SA-516 Grade 70 steel under API 650 (<173 MPa).

3.4. Tank foundation design

Two types of foundations were considered to support the salt tank; a soil-bearing mat foundation and a mat foundation supported on auger-cast concrete piles. Because a site-specific geotechnical report has not been performed, the foundation design followed a conservative approach, consisting of a mat foundation supported by piles to keep overall and differential settlements to a minimum. Differential settlements increase the risk of cracking in the refractory, which is likely to lead to tank failures. The following geotechnical criteria were used to
design the preliminary foundation (a geotechnical report would be needed for a specific site to validate the assumed criteria):

- Allowable Pile Capacity: 1112.5 kN (250 kips) with a 15% increase for short-term loads
- Pile Diameter: 0.61 m (24-in)
- Pile Depth: 12.2 m (40-ft)
- Pile Stiffness: 152,421 kN/m (870 kips/in)
- Minimum Pile Spacing: 2.3 m (7-ft, 6-in)

Visual Foundation (3D foundation analysis and design software by Integrated Engineering Software) was used to design the foundation and piles. Fig. 5a is a rendering of the mat piles and loading.

In order to prevent significant thermal stress in the tank shell, the tank floor needs to be cooled to roughly the same temperature as the side walls. Heat loss from the tank floor, which is insulated by the foundation, is significantly slower compared to the tank walls that are exposed to the atmosphere. Thus the foundation needs to be actively cooled.

The first design of the cooling portion of the foundation consisted of a continuous bank of steel tubes, touching and welded together at the sidewalls. The tubes would be welded below a large steel plate. The tank steel floor would sit directly on top of this steel plate with a 50 mm (2-inch) gap between them.

This foundation was simulated in 3-D using ANSYS Fluent. In order to reduce the overall domain size, a detailed, meshed representation of this foundation was not included in this first model of the cold wall design; however, the Fluent model was configured with a thermal representation of this foundation within the boundary condition specification. An appropriate convective heat transfer coefficient was calculated for air flow through pipes and applied to the boundary condition (details of calculation in Equation S(2)). It assumes a heat flux of 276 W/m² and the Dittus-Boelter Correlation to estimate the convective heat transfer coefficient.

The cooling system must be variable enough to match the heat flux range from the tank shell, which is 232 W/m² to 260 W/m² with a shell temperature range from 37 °C to 57 °C. The initial approach was to investigate air and water cooling options.

The investigation showed that the thermal conductivity variation in the sand due to moisture content was so great that any storm event would immediately cause the heat transfer rate in any exposed sand (around the outside edge) to increase dramatically, leading to significant temperature variation (and stresses) across the bottom of the tank particularly at the tank chine. It was determined that air cooling could not accommodate even slight environmental changes without causing unacceptable temperature variations though the tank bottom. Therefore, both the sand and air cooling were eliminated from the design in favor of a water-cooled foundation.

The modified foundation design is shown in Fig. 5b. The sand was replaced by non-shrink grout with roofing felt between the tank bottom and grout layer. The 1” schedule 80 cooling tubes will be placed on the top of the concrete foundation with the grout poured to a thickness of 76 mm (3-inch) above the top of the tubes. Grout was utilized instead of embedding tubes in the foundation, so that any cracks that may develop due to the tubes and thermal stresses will be contained to the grout and not compromise the foundation. Even if the grout develops cracks it will continue to transfer the tank load to the foundation.

The tubes will be set with a 50 mm (2-inch) gap between them and will run straight from one side of the foundation to the other with bends to miss anchor bolts as required. A supply and return header will be required on each end of the foundation, allowing tubes to flow in alternating directions, maintaining a uniform temperature across the bottom of the tank. A cooling tower or chiller will be required to maintain the appropriate tank bottom temperature as environmental conditions change. It is expected that temperature measurement will be required on the tank shell and bottom to provide control input to the cooling water system.

3.5. Installed cost $/kWhₕₜₖ

Given the TES capacity requirements specified by the DOE Gen3 program, the two tank pairs are recommended (two hot tanks, two cold tanks) to reduce tank size, provide operational redundancy, and reduce risk of plant shutdown due to a single tank failure. Such a design allows the plant to operate at ≈50% storage capacity should a tank need to be down for service or repairs. The estimated tank diameter of a single-pair tank design exceeds that of current CSP plants and increases the risk associated with the tank. In the two-pair tank scenario with an overall thermal-to-electric plant efficiency of 50% (net), 110 MWₑ power rating, and a total of 12 h of storage, the predicted hot and cold tank heights are 11 m and diameters are 41.8 m and 40.2 m, respectively. CSP construction consultants have stated that the risk of tank failure significantly increases if the tank diameter is greater than 40 m, so opting for a two-tank scenario would far exceed the 40 m target. Reducing storage hours to 10 would keep the tank diameters below 40 m with minimal impact on levelized cost of energy (LCOE).

Based on historic use in industry, it is not expected that a refractory liner will last for a 30-year plant life without the need for periodic system shutdown and maintenance. The four-tank scenario allows inspection and repair of the tank liners without a complete shutdown of the plant. The tanks are sized to hold the heel of the duplicate tank, thus during a service shutdown of one tank, the plant could operate at approximately 55% capacity, accounting for transfer of the other tank’s salt heel. As operating experience is gained, the quality and durability of the refractory liner is expected to improve over time.

Fig. 5. 3D model of concrete mat of piles (a). Cross section view, showing foundation and cooling details (b).
A cost model for the Gen3 lined-tank design was developed using information from industry consultants related to refractory material cost, tank shell cost, and construction labor cost. Cost of salt and salt transportation were obtained from the bulk salt supplier. The projected TES costs consist of two pair of internally insulated tanks. Both the hot and cold TES storage tanks have identical internal refractory liners inside carbon-steel shells. A breakdown of the costs of each component is shown Fig. 6, which yields an overall TES cost of $60/kWhth.

3.6. Potential cost reduction strategies

To lower material costs, alternative suppliers of refractories have been identified that could provide generic materials with nominally the same properties but at a lower cost. Initial estimates indicate that this could save up to 7% in material costs, with some added risk associated with material quality. Nevertheless, even with generic refractories, the raw material costs exceed the $3/kWhth target.

Fig. 7 shows a breakdown of the foundation and tank (shell) cost pie slices shown in Fig. 6. The design, which uses piles and grout, drastically increases the TES cost. Although this is concerning, the design may provide opportunities for an overall cost reduction. For example, the use of metal piers in their foundation design may allow the tanks to be built taller than previously thought, leading to a more even aspect ratio and an overall smaller amount of refractory liner and tank shell. Past molten salt TES systems have used much simpler foundation designs, consisting of lubricating sand and a steel skid plate on a concrete slab. Applying such a design to the Gen3 system would eliminate the piers, grout system, and felt paper sections of the cost breakdown and drastically reduce cost and complexity.

Furthermore, current commercial TES systems use air-cooled foundations, rather than water-cooled foundations, due to cost concerns. The water-cooled foundation design will likely be eliminated in favor of active air cooling. This will be the subject of future studies, to balance the increased risk of insufficient foundation cooling with the added benefits of significant cost reduction. The overall TES system design present herein follows a very conservative approach, which is reflected in its high overall cost. However, there are certainly simplifications that can be applied with further analysis of the tank site, thermal expansion behavior, and design trade studies.

4. Conclusions

This study assesses the techno-economic viability of molten chloride salt tank construction. The design uses a refractory-ceramic internal liner, which thermal insulates the tank shell and allows for carbon steel to be used in tank fabrication. Based on thermal modelling and industry experience, a “cold wall” design is proposed that utilizes non-wetted insulation in the liner and lowers the tank shell temperature <100 °C. It is imperative that the material at the interface with the salt (the hot face) inhibit salt permeation through the liner. Should the insulation be wetted, experimentally verified calculations predict the effective thermal conductivity of the liner material will increase, leading to an increase in heat flux and greater risk of tank failure. Development of insulating materials that maintain their physical integrity and insulating ability if wetted by salt would significantly advance the prospects of the lined-tank design.

The current liner is constructed with a radiused brick profile to support the liner’s vertical wall, which is mechanically anchored to the tank shell at the top of the tank. Thermal expansion of the tank liner must be accommodated to avoid unacceptable stresses on the tank shell. The expansion joints are constructed with salt-compatible materials and placed in both the hot face and insulating layers. These expansion joints are designed to close during heat up, leaving the liner under compression without inducing excessive stress on the shell. With the aid of iterative FEA models, the thermal and mechanical profile of the salt tank was optimized to reduce the differential thermostatic stress of the tank shell to levels allowable under API 650 and ASME Boiler and Pressure Vessel Code, Section IID codes (173 MPa).

We propose a 4-tank (two tank pairs) TES system to provide operational redundancy and reduce risk of plant shutdown due to tank maintenance and repair. However, the projected TES cost of $60/kWhth greatly exceeds the SunShot 2030 target of $15/kWhth. The largest cost drivers are the refractory materials and the complex water-cooled foundation design.

Simplifications are proposed to reduce the overall cost and risk of the system. These include fail-safe, wettable insulating materials, methods to inhibit salt permeation through mortar joints, and methods to reduce the number of required expansion joints by increasing the native compressibility of the liner. This may also be accommodated by raising

![TES Costs ($/Kwth)](image)

*Fig. 6. TES cost in $/Kwth. Cost of the overall 4-tank system is $60/kWth. The cost of the liner is subdivided into raw material, labor, and contingency costs. The raw materials costs of the system are over $7/kWth and exceed the $3/kWth target by a factor of about 2.5.*
the tank shell temperature from to design point of 60 °C to a value between 200 and 400 °C. While this would necessitate the use of external tank shell insulation, it would allow the shell expansion to move in concert with the liner expansion, thereby eliminating or greatly reducing the need for expansion joints in the liner.

CRediT authorship contribution statement


Declaration of competing interest

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

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Appendix A. Supplementary data

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References