

Solar Thermocline Storage Systems

Preliminary Design Study

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Final Report, June 2010

EPRI Project Manager
C. Libby

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Black & Veatch

National Renewable Energy Laboratory

Sandia National Laboratories

Purdue University

Electric Power Research Institute (EPRI)

NOTE

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The following organizations prepared this report:

Black & Veatch
650 California, Fifth Floor
San Francisco, CA 94108

Project Manager
J. Pietruszkiewicz, PE

Principal Investigators
B. Brandon, Estimation
R. Hollenbach, Process Design
M. Lamar, Process Design
J. Smith, Mechanical Design

National Renewable Energy Laboratory
1617 Cole Blvd, MS 5202
Golden, CO 80401

Principal Investigators
C. Turchi
D. Bharathan
G. Glatzmaier
M. Wagner

Sandia National Laboratories
P.O. Box 5800, MS 1127
Albuquerque, NM 87185-1127

Principal Investigator
G. Kolb

Purdue University
585 Purdue Mall
West Lafayette, IN 47907-2088

Principal Investigator
S. Garimella
S. Flueckiger
Z. Yang

Electric Power Research Institute (EPRI)
3420 Hillview Avenue
Palo Alto, CA 94304

Principal Investigators
C. Libby
L. Cerezo
R. Bedilion

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PRODUCT DESCRIPTION

Solar thermal energy storage (TES) has the potential to significantly increase the operating flexibility of solar power. TES allows solar power plant operators to adjust electricity production to match consumer demand, enabling the sale of electricity during peak demand periods and boosting plant revenues. To date, TES systems have been prohibitively expensive except in certain markets. Two of the most significant capital costs in a TES system are the storage medium (typically molten salt) and the storage tanks. Thermocline storage is a relatively unproven TES method that has the potential to significantly reduce these costs. In a thermocline system, approximately 75% of the required storage medium is replaced with an inert quartzite rock, and only one storage tank is required instead of the two typically needed for high-temperature TES. This report includes preliminary designs and cost estimates for molten salt thermocline systems with capacities ranging from pilot scale to commercial scale. Thermal and system level modeling was conducted to determine the performance of these systems.

Results and Findings

The study determined the application areas in which thermoclines might be economically competitive. Cost estimates were developed for the construction of thermocline systems, and similar estimates were developed for the corresponding state-of-the-art two-tank storage systems for comparison. Both parabolic trough indirect storage and central receiver direct storage systems were evaluated, ranging from 100 to 3500 thermal megawatt-hours (MWh_t) in size. The results confirm that the thermocline offers the lowest installed capital cost over the two-tank system at each design capacity.

Challenges and Objectives

The potential benefits of TES are significant; however, experience is limited and costs and performance of the various technology options remain uncertain. The intent of this study was to develop a basic design for the thermocline technology as well as detailed process flow diagrams, heat and material balances, and detailed equipment lists that provide a starting point to develop this technology at pilot scale and later at full commercial scale. One key objective was to provide a meaningful comparison to the current state-of-the-art two-tank TES technology to determine whether the thermocline cost and performance might be competitive.

Applications, Value, and Use

Although the cost of solar energy is still high compared to traditional generation options, this cost is expected to decrease as technologies mature and deployment increases. Thermal energy storage presents a unique opportunity to reduce the levelized cost of electricity while providing increased plant operating flexibility and energy value. This report shows that thermocline systems might offer a lower cost option for a wide range of solar technologies and storage applications. The results of this study will be beneficial to any energy company or project developer considering a solar thermal project.

EPRI Perspective

The first utility-scale concentrating solar thermal power plants in the world were built in southern California in the 1980s, and several new large-scale plants are currently under development in the United States, Spain, and other locations throughout the world. Although a two-tank molten salt system is now operational in Spain, there has been limited RD&D in the United States to implement the multi-tank molten salt technology and develop next-generation TES technologies. There is also a need to determine the ideal operation and integration of those technologies into electric grid operations. The thermocline process has the potential to significantly reduce the costs of thermal energy storage without compromising performance, which could in turn greatly increase the utility of solar power and lead to wide-scale adoption of the technology. Continued research along with the operation of the first utility-scale thermal energy storage units in the next few years will provide more concrete data by which to compare thermal energy storage systems. This work supports a long-term vision for a broad generation portfolio that includes renewable energy as a cost-competitive option.

Approach

The approach was to define and optimize parameters for several design cases and develop AACE Class 4 project estimates for each. The main components of the Class 4 estimate are the design basis, process flow diagrams, and equipment lists. These design details allowed the project team to work with vendors and obtain quotes for necessary equipment. These data were used to complete an EPC estimate for the construction of the thermocline systems. In parallel, thermodynamic and system models were developed to determine the thermal stability of the system under different operating scenarios and examine the performance of thermoclines.

Keywords

Thermal energy storage
Solar thermal energy
Thermocline
Parabolic trough
Central receiver
Power tower
Renewable energy
Molten salt

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1

EXECUTIVE SUMMARY

Introduction

A broad portfolio of cost-competitive supply technologies will be needed to satisfy the world's rising demands for energy while meeting climate policy and other societal objectives. EPRI is interested in the near-term development and deployment of concentrating solar power (CSP) technologies, that can serve growing electricity demand and offer energy companies an economical, zero emissions generation option. The highest intensity solar energy is typically within a few hours of peak summer loads, making it a particularly attractive renewable option. The advancement of solar thermal energy storage (TES) systems is critical to lowering the cost of electricity, firming capacity, and providing operating flexibility to utility scale CSP plants. Successful implementation of TES is expected to increase the value proposition for CSP plants and accelerate deployment. In order for TES to be widely adopted, lower system capital costs must be developed and demonstrated. This study shows that thermocline systems are potentially lower cost than the current state-of-the-art technology.

A molten salt thermocline system is a single tank storage system that uses a thermal gradient to separate the heated salt arriving from the solar field from the cold return salt. It uses a low-cost filler material to reduce the amount of more expensive molten nitrate salt required. The tank stores energy as cold salt is pumped from the cold base of the tank, either directly through the solar field or indirectly through a salt-to-oil heat exchanger to absorb heat, and is then returned to the hot top of the tank. The flow is reversed when the steam generator requires additional heat from the stored salt. The use of a single tank instead of a two-tank storage system, along with the use of an inexpensive filler material to reduce the amount of molten salt required is expected to result in a lower cost TES option.

Background

The first major demonstration of solar TES technology was a 182 MWh_t single-tank indirect mineral oil thermocline at Solar One in 1982. A two-tank direct mineral oil system later followed at the SEGS I trough plant in California in 1985. Both TES demonstrations ended prematurely due to fires and were not rebuilt. In the mid-1990s, the Solar Two central receiver project successfully demonstrated two-tank direct molten salt storage. This R&D project was decommissioned in 1999. Although a two-tank molten salt system is now operational in Spain, there has been limited RD&D in the U.S. to implement the multi-tank molten salt technology and develop next generation TES technologies. Molten salt thermocline has only been demonstrated at small scale. Figure 1-1 shows the schematic for a 2.3-MWh_t packed-bed thermocline storage demonstration conducted by Sandia National Laboratories in 2001 using binary molten-salt fluid and a mix of quartzite rock and silica sand filler material.

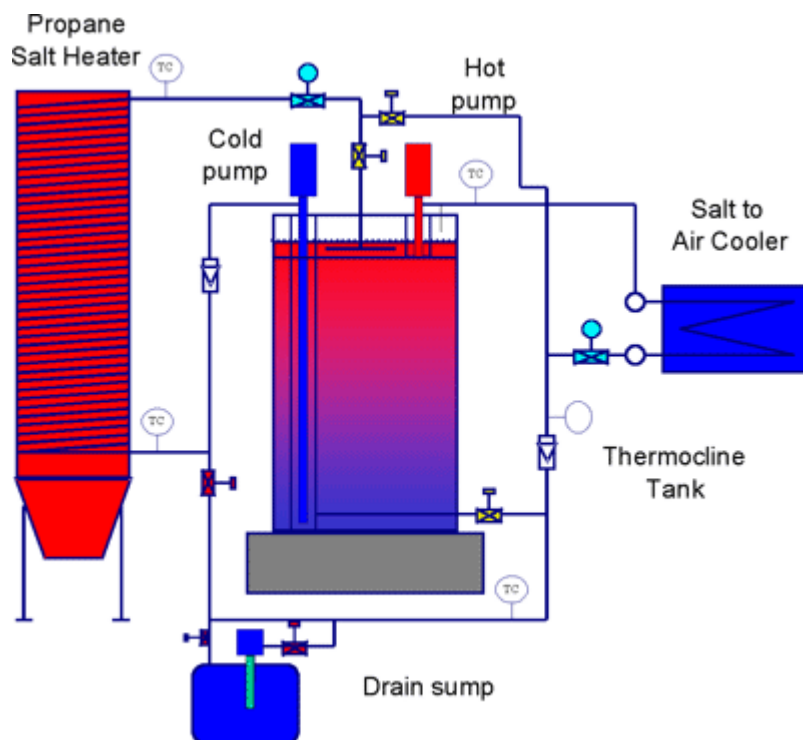


Figure 1-1
Thermocline Test at Sandia National Laboratories (Source: Sandia National Laboratories)

Many new CSP plants are expected to be developed in the Southwestern U.S. in the next few years. Arizona Public Service has plans for a 280 MW parabolic trough plant with indirect, two-tank molten salt storage. Recently SolarReserve announced central receiver projects with two-tank direct molten salt storage in Nevada and California. California has over 6 GW of announced CSP projects that may be candidates for TES systems. These new plants and others yet to be announced offer opportunities to conduct applied TES technology research at utility-scale installations. There is also a need to determine the ideal operation and integration of these technologies into electric grid operations.

Project Overview

Under its Generation Technology Industry Demonstration Program, EPRI is researching next generation TES systems. EPRI has chosen to focus on a molten salt thermocline TES system based on a recent review of TES technologies¹. The thermocline preliminary design work conducted under this project includes utilizing existing thermodynamic and operational thermal storage modeling capability and determining system design parameters. It is expected that the work will ultimately transition into a demonstration project if the results are favorable. Further pilot-scale analysis and testing may be prudent before developing a full-scale demonstration.

¹ *Program on Technology Innovation: Evaluation of Solar Thermal Energy Storage Systems*. EPRI, Palo Alto, CA: 2008. 1018464.

Basic structural and engineering designs of a molten salt thermocline TES system were developed by Black & Veatch for a range of storage capacities and molten salt compositions to create a meaningful cost comparison with conventional two-tank systems and determine the applications areas where thermocline systems may be more competitive. Detailed thermal and operational models were used by Sandia National Laboratories, the National Renewable Energy Laboratory, and Purdue University to determine thermal stability and evaluate performance under different operating conditions. Prior studies have shown single tank thermoclines to be economical for parabolic trough systems. The current study evaluates parabolic trough applications as well as higher temperature central receiver systems.

Project Objectives

The potential benefits of TES are significant; however, experience is limited and costs and performance of the various technology options remain uncertain. The intent of the study was to develop basic structural and engineering designs for the thermocline technology, as well as detailed process flow diagrams, heat and material balances, and detailed equipment lists to provide a starting point to develop this technology at pilot scale and later at full commercial scale. One key objective was to provide a meaningful comparison to the current state-of-the-art two-tank TES technology to determine if the thermocline cost and performance may be competitive. Cost estimates for thermocline and two-tank, direct and indirect systems were developed for the range of storage temperatures and capacities.

Results

Cost estimate results are summarized in Table 1-1 and Figure 1-2. The costs include both direct and indirect costs, which may make them appear higher than historically reported values (see Chapter 4 for more details). The thermocline system offers the lowest installed capital cost over the two-tank system at each design capacity. The main capital cost difference between the thermocline design and the two-tank design is the amount of molten salt required for each. The thermocline requires roughly half as much salt as the traditional two-tank design, greatly reducing the cost of expensive molten salt. The main capital cost difference between the indirect parabolic trough (synthetic oil) design and the direct central receiver (molten salt) design is the oil/molten salt heat exchanger, which is not required for direct storage designs. The installed cost per kWh_t decreases as the capacity of the TES increases. This is expected, as most process equipment benefits from economies of scale. It was determined that the largest single-tank thermocline capacities for the selected design conditions is approximately 1500 MWh_t for the indirect parabolic trough design and 3500 MWh_t for the direct central receiver design. These represent the lowest cost designs for the two types of systems.

Table 1-1
Total Capital Cost Summary (Direct & Indirect Costs)

Thermal Energy Storage Method	Total Capital Cost (\$/kWh _t)					
	100 MWh _t	500 MWh _t	1000 MWh _t	1500 MWh _t	3000 MWh _t	3500 MWh _t
Direct Thermocline Central Receiver	132	61	46	44	37	34
Direct Two-Tank Central Receiver	181	78	57	55	50	50
Indirect Thermocline Parabolic Trough	246	106	84	70	72	73
Indirect Two-Tank Parabolic Trough	275	143	116	111	95	89

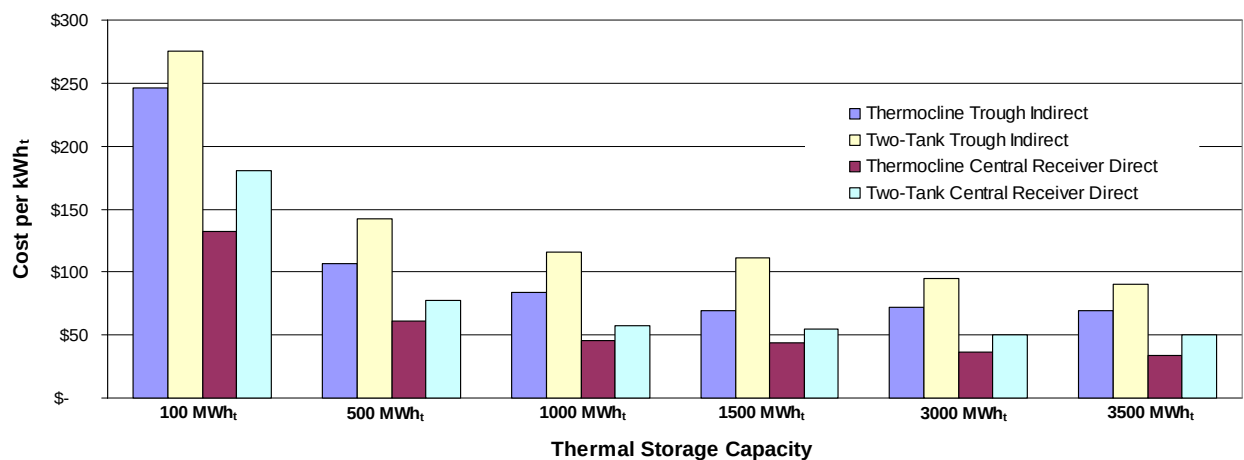


Figure 1-2
Total Cost Estimates (Direct & Indirect Costs) per kWh_t Storage

Performance

The performance analyses investigated the annual performance of a thermocline system compared to a two-tank system, the thermal performance of the thermocline with regards to thermal gradients and natural convection during dwell periods, and the distributor manifold performance. The system-level modeling analysis showed that if sliding pressure operation² is employed, the annual performance of a thermocline storage system should be comparable to a two-tank system. The thermal analysis determined that the use of filler materials, which provide cost benefits, unfortunately promote diffusion in the tank and spread of the thermocline region.

² The term sliding pressure refers to steam turbine operation when the Rankine cycle temperature and pressure is dropped to maintain the minimum amount of superheat (typically about 50°C).

During tank discharge, the thermal model showed some heat loss occurring over the course of a 6-hour discharge cycle, particularly close to the tank wall, but the salt remained within two degrees of the hot operation limit. During dwell conditions, there was significant cooling of the molten salt at the bottom of the fillbed, but the salt at the top of the tank was largely insensitive to external tank losses.

Thermal ratcheting was identified as a potential concern for the thermocline technology, and it should be examined as part of a detailed design process before large scale systems are developed. Thermal ratcheting may occur over time as the packed bed is thermally cycled. As the quartzite rock filler is cooled it contracts and compacts in the bottom of the tank. When the tank is reheated the quartzite cannot return to its original position. The quartzite then expands and places pressure on the walls of the tank. Over time this process could potentially damage the tank. In the current proposed design, there are no provisions in place to manage thermal ratcheting. Past operating experience has not shown ratcheting to be an issue; however, with the higher temperature systems proposed in this report, the study group concluded that further examination is necessary.

Organization of Report

The conceptual design study consists of five main areas:

- Technology overview (Chapter 2)
- Thermocline design (Chapters 3)
- Cost estimates (Chapter 4)
- Thermal stability and performance modeling (Chapter 5)
- Conclusions (Chapter 6)

The table of contents further guides the reader to specific discussion areas. There are many appendices that contain detailed information about the designs, costs and materials.

2

TECHNOLOGY OVERVIEW

Solar Technologies

Solar thermal electric technologies, or concentrating solar thermal power (CSP) plants, produce electricity by collecting solar radiation using various mirror or lens configurations. The concentrated energy from the sun is focused on a receiver that contains a heat transfer fluid, which is used to transfer heat energy to a power block with a turbine or engine that converts the heat to electricity. Four main types of CSP plants are currently in use or in development:

- Parabolic trough
- Central receiver (power tower)
- Compact linear Fresnel reflector (CLFR)
- Dish/engine

This study includes discussions of parabolic trough and central receiver technologies, which use a centralized power block to generate electricity; this configuration makes large scale power plants of 50 MW or greater the most economically viable option for these systems. These two technologies are currently the most mature CSP technologies and could be coupled with thermocline storage systems.

In a CSP system only the direct normal insolation (DNI) component of solar radiation contributes to the thermal energy absorbed by the plant. As a result, a single-axis or two-axis sun tracking system can be an important component of a CSP plant, allowing the mirrors to maximize the amount of DNI that is reflected onto the receiver and achieve high working temperatures for the heat transfer fluid.

Parabolic Trough

Parabolic trough plants use a field of linear parabolic collectors, shown in Figure 2-1, to redirect and concentrate sunlight onto a tube receiver located at the focal line of the mirrors. Each collector tracks the sun by rotation about a horizontal axis. The heat transfer fluid is typically a synthetic oil mixture with a maximum operating temperature of 390°C (735°F). With a synthetic oil HTF, the steam generator produces live steam at nominal conditions of 377°C (711°F) and 100 bar (1465 lbf/in²), and reheat steam also at a temperature of 377°C (711°F). An important aspect of parabolic trough technology is that the electrical energy production is separated from the solar energy collection, creating a natural insertion point between these two elements for a thermal energy storage system. Most thermal storage technologies are compatible with parabolic

trough CSP plants. For a parabolic trough system storage capacities up to 16 hours of full load turbine operation are feasible. Thermal storage is also inherent in a parabolic trough system, in that the high fluid volume in the collector field provides over 15 minutes of thermal storage, or thermal inertia, which can be used to provide a form of buffer storage.



Figure 2-1
Parabolic Trough Collector Field

Parabolic trough is a mature commercial technology that has generated electricity reliably for over two decades. The most recent CSP plant installations have utilized trough technology, and the financing for trough plants without TES is comparable to other mature, commercial generation technologies. There is ample design and performance data available for trough plants.

Central Receiver

Central receiver, or power tower, plants use a collector field array of several thousand sun-tracking heliostats to redirect and concentrate solar radiation onto a tower mounted single receiver. The heat transfer fluid is typically water/steam or a sodium/potassium nitrate salt mixture. For either fluid, receiver outlet temperatures up to 650°C (1200°F) are feasible. If molten salt is used, a conventional steam generator can produce live steam at nominal conditions of 125 bar (1800 lbf/in²) and 540°C (1005°F), and reheat steam at a temperature of 540°C. As in a parabolic trough system, the collector array and electrical generation equipment are separate, offering a natural point for including a thermal storage system, and either molten salt or steam can serve as a storage medium if used in the receiver. A molten salt storage system was used at the Solar Two demonstration facility, and steam accumulators are currently used at Abengoa's central receiver CSP plants in Spain for short-term buffer storage. Other types of thermal storage also could prove successful for a central receiver system. Figure 2-2 shows the heliostats and towers at Abengoa's facility in Spain.



Figure 2-2
Central Towers and Heliostats at Abengoa's Central Receiver Plants in Spain (2008)

The ultimate performance and cost estimates for mature, molten salt central receiver technology are very attractive, but the main challenges for the technology at this time lie in scaling up the receiver assembly for larger plant sizes and in operating the molten salt system in a consistent and reliable manner. Commercial operation of central receiver technology is currently being demonstrated at the 11 MW PS-10 and the 20 MW PS-20 plants in Spain; both plants use direct steam in their operations, rather than molten salt. The Solar Tres power tower currently under development in Spain will be a molten salt system with 17 MW capacity.

Thermal Energy Storage Operation

The primary purpose of thermal energy storage is to compensate for the sometimes variable nature of solar energy, as well as enabling operation of a solar energy system during times of the day when the sun is no longer available. The energy contained in the storage system can be dispatched in any number of ways according to the desired output profile, but the strategies for using energy from storage fall into three main categories:

- Buffering power delivery
- Extending delivery period
- Displacing delivery period

Both two-tank TES and single-tank thermocline have the ability to extend or displace the delivery of energy from the solar facility. Electrical utilities designate time-of-use (TOU) periods for electricity demand from customers; during periods of high demand, or on-peak times, the market price of electricity is higher than during off-peak, or low demand periods. While solar thermal power plants appear capable of providing energy for a large part of the on-peak TOU period, in some regions the on-peak period can extend well into the evening hours when solar facilities without storage are no longer capable of delivering energy. Two main options exist for altering the output of a solar thermal power plant to better match the load profile in locations that have significant demand into the evening hours. The first operations strategy requires diverting a portion of the output from the collector field over the course of daily operation to charge the storage system, which is then discharged once the sun has set to extend the delivery period. Alternatively, all of the energy output from the collector field can be used to charge the storage system at the beginning of daily operation, delaying plant startup until the storage system has been fully charged. The plant will then have sufficient energy in storage to continue operating through the peak and into the evening use period. This strategy simply shifts the delivery period a couple hours later into the day. Both of these storage options are depicted in Figure 2-3.

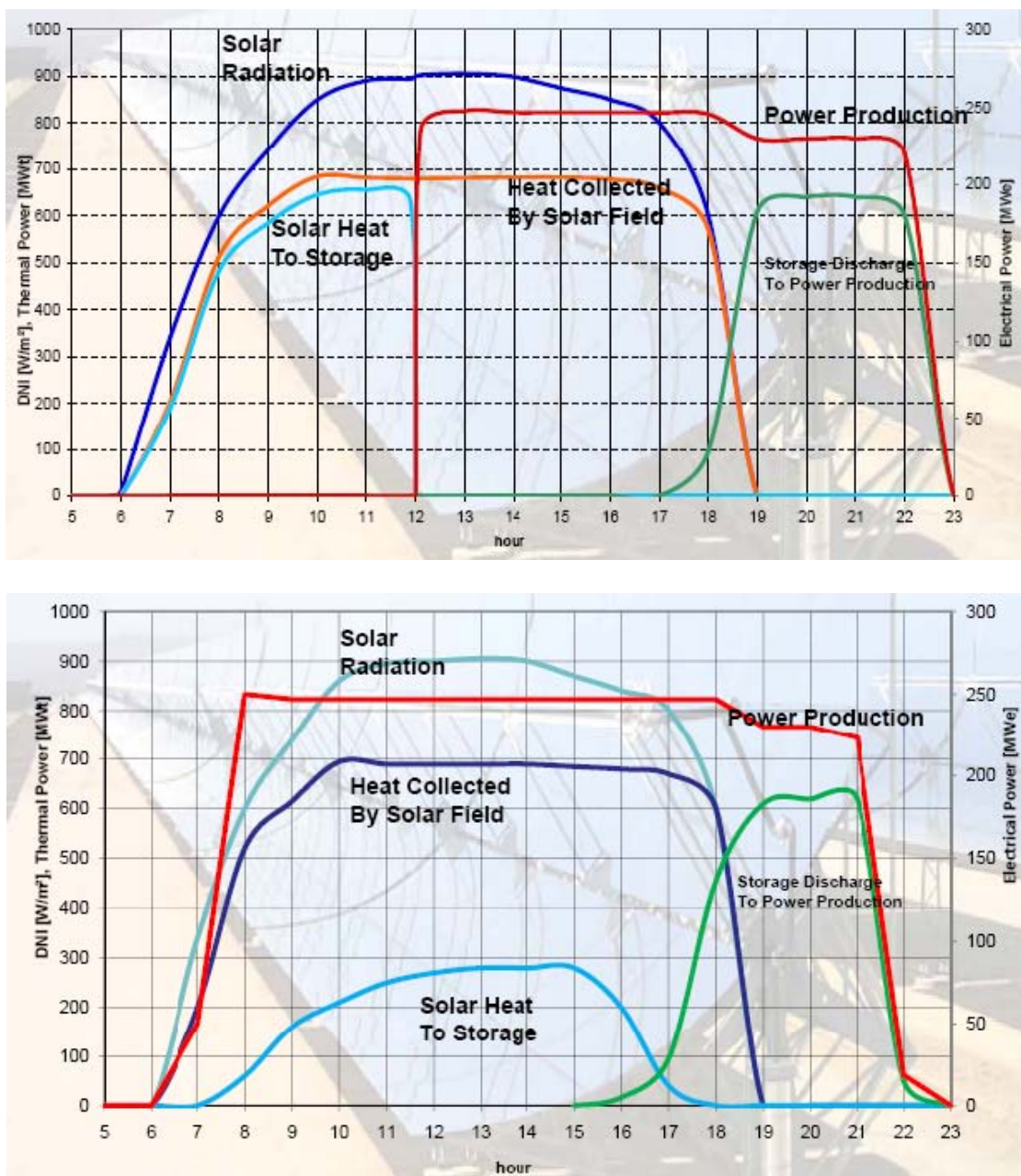


Figure 2-3
Displacement and Extension of Power Production using Thermal Energy Storage³

Thermal Energy Storage Technologies

Two-tank and thermocline systems use sensible heat storage, in which the temperature of a liquid is raised without the material changing phase. They are considered active storage technologies, characterized by a storage medium that circulates through the storage system, and relies on forced convective heat transfer to move energy into and out of the storage medium. Nitrate salts are the most common liquid storage media used in solar thermal energy systems.

³ Solar Millennium

Two-Tank Indirect

The distinguishing feature of the two-tank indirect system is that the HTF that circulates through the collector field remains separate from the storage medium kept in the tanks. The HTF is typically a synthetic oil such as Therminol VP-1 (currently in use at the California SEGS plants⁴) or Dowtherm A, and the storage medium is likely to be molten salt. The system consists of a cold tank, normally operating at 290°C (554°F) or less, a hot tank, operating at temperatures up to 390°C (703°F), the storage medium, the heat exchangers for transferring energy from the heat transfer fluid to the storage medium (and back), the storage medium pumps, and the associated balance of system equipment, such as an ullage gas system and electric heat tracing for all molten salt components. Electric heat tracing is required to maintain inventory temperature in the event of an extended plant outage, while the ullage gas system prevents oxidation of the storage medium. A schematic diagram of a two-tank indirect system is shown in Figure 2-4.

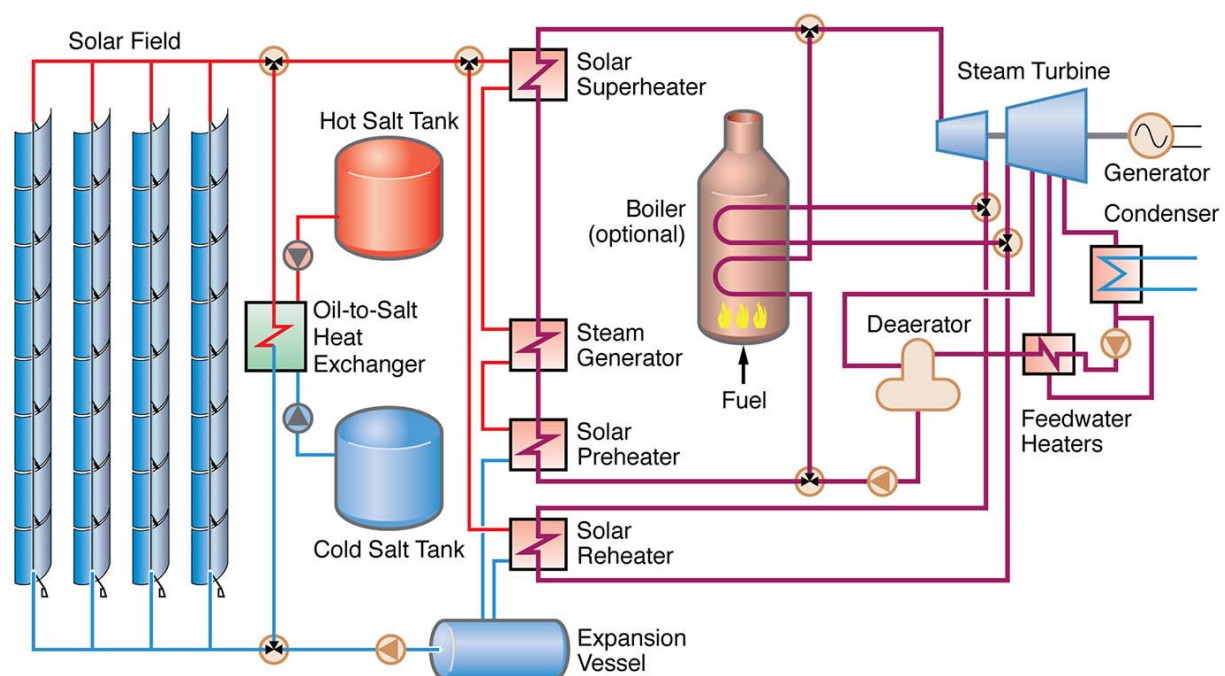


Figure 2-4
Two-Tank Indirect Thermal Storage System

The thermal energy storage system is charged by taking hot HTF from the solar field and running it through the oil-to-salt heat exchangers. Simultaneously, cold molten salt is pumped from the cold storage tank, and delivered to the heat exchangers. In the heat exchangers, the salt and the heat transfer fluid flow in a countercurrent arrangement. Heat is transferred from the HTF to the cold salt flowing through the heat exchanger, which leaves as hot salt that is then stored in the hot salt tank. When the energy in storage is needed, the flows of both the HTF and the salt are

⁴ S.D. Odeh, G.L. Morrison, and M. Behnia, "Thermal Analysis of Parabolic Trough Solar Collectors for Electric Power Generation", Darwin: ANZES Annual Conference, 1996.

reversed in the oil-to-salt heat exchangers in order to reheat the HTF. Countercurrent flows in the heat exchangers are necessary in order to maximize heat transfer between the two fluids. The reheated HTF is then used in the power block to generate steam to run the power plant.

The feasibility of the indirect system is proven and at present the concept is associated with the lowest technological risk. However, the transfer of energy from the heat transfer fluid to the salt during charging, and the transfer of heat from the salt to the heat transfer fluid during discharging, both require a temperature drop across the oil-to-salt heat exchanger. As such, the temperature of the heat transfer fluid delivered to the steam generator when operating from thermal storage is 10 to 20 °C lower than when operating directly from the collector field. Due to temperature and efficiency reductions associated with the heat exchangers, both the output and the efficiency of the Rankine cycle are unavoidably lower when operating from thermal storage. The round trip efficiency of a storage system is defined as the net electricity delivered from the storage system divided by the amount of electricity that would have been generated from the solar field thermal energy had it been directly converted to electricity. A typical efficiency for an indirect two-tank trough storage system is about 93 percent, whereas a future trough with a direct two-tank molten salt storage system might be 98 percent efficient.

Two-Tank Direct

In a two-tank direct system, the fluid which circulates through the receiver of a power tower is also used as the storage medium. Like the indirect system, the direct system consists of a cold tank and a hot tank, the storage medium and the associated balance of system equipment, such as the electric heaters for inventory maintenance during plant outages. However, unlike the indirect system, this design uses the same fluid in both the storage system and the receiver, which eliminates the need for a second set of heat exchangers used to transfer thermal energy between the heat transfer fluid and the storage medium in the indirect system. When molten salt is used as the storage medium, the cold and hot tanks can operate at temperatures up to 293°C (559°F) and 560°C (1040°F), respectively. A schematic diagram of the system is shown in Figure 2-5.

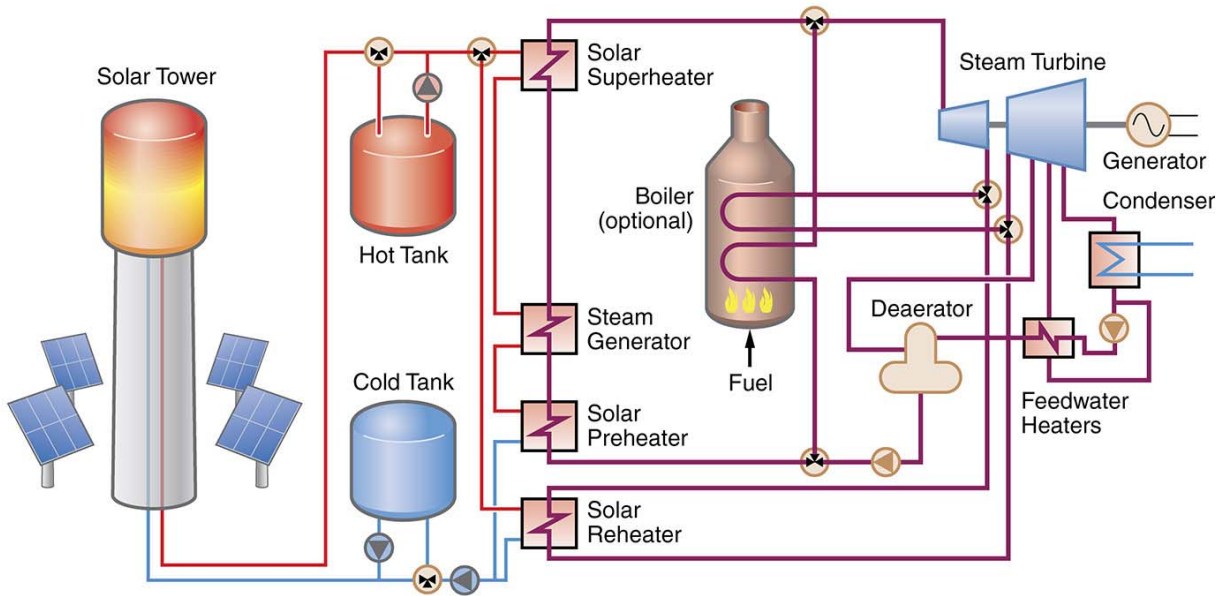


Figure 2-5
Two-Tank Direct Thermal Storage System

To charge the system, fluid from the cold tank is circulated through the receiver, and returned to the hot tank. To discharge the system, fluid from the hot tank is circulated through the steam generator, and returned to the cold tank. All of the fluid from the receiver passes through the hot storage tank. Depending on the residence time in the tank, the temperature of the fluid leaving the tank is 0 to perhaps 1.5°C lower than the temperature entering the tank. As a result, the performance of the Rankine cycle in a plant with a two-tank direct storage system is essentially the same as a plant without thermal storage.

It may seem that only a single tank would be needed for the charged storage medium, but a cold tank is required to contain the volume of storage material that has already discharged its energy to the steam generator. During storage system discharging, the collector field will likely not be receiving solar energy, although it is possible to charge and discharge simultaneously, as was demonstrated at Solar Two.

Two-tank direct TES systems will likely use a molten salt as the storage medium. Nitrate salts are relatively inexpensive compared to synthetic oils and can provide tank storage capacity ranging between 3 and 16 hours of full load turbine operation. The primary disadvantage to a molten salt storage system is the relatively high freeze point of typical nitrate salts. As such, considerable care must be taken to ensure that the salt does not freeze in the solar field or elsewhere in the storage system. This includes installing an electric heat trace system on all equipment that comes in contact with the salt.

Single-Tank Thermocline (Indirect or Direct)

Like the two-tank systems, a thermocline can operate either directly, with the storage medium also serving as the HTF in the collector field, or indirectly, with a separate storage media and HTF. A thermocline system involves a single tank that is used to store both the hot and cold fluid, further reducing the cost of the TES system. This single-tank configuration features the hot fluid on top and the cold fluid at the bottom of the tank. The zone between the hot and cold fluids is called the thermocline. While a thermocline can simply combine the hot and cold storage fluids into a single tank, the primary advantage of the thermocline storage system is that most of the storage fluid can be replaced with a low-cost filler material. This filler displaces the majority of the molten salt that would be used in a comparable two-tank system, and provides a robust and inexpensive storage medium. A thermocline with a packed bed would actually be considered a dual-media storage system, as it utilizes both a liquid and solid medium for storing energy.

To charge the system, hot fluid is introduced at the top of the tank, flows down through the porous bed, and leaves from the bottom of the tank; in the process, heat is transferred from the hot fluid to the porous filler material. To discharge the system, the flow is reversed; cold fluid enters at the bottom of the tank, and is heated as the fluid flows up through the porous bed. A schematic diagram of a direct thermocline system is shown in Figure 2-6. On the cold fluid side of the tank, the system includes a bypass line to return cold fluid to the suction side of the pump during storage charging.

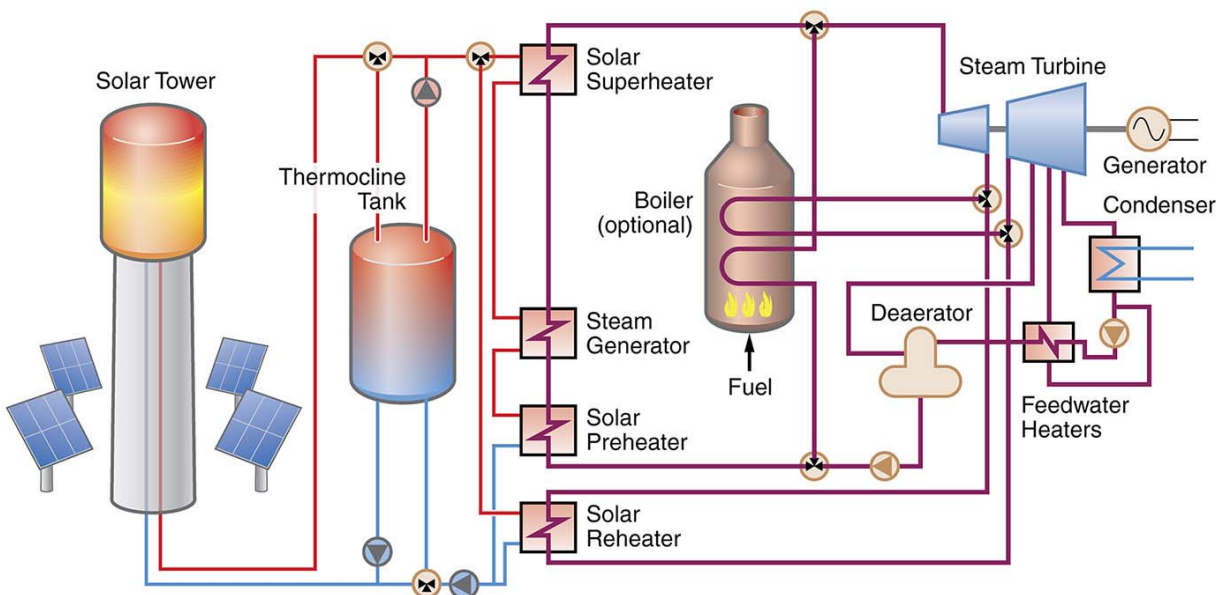


Figure 2-6
Single Tank Direct Thermocline System

The principal liability for a thermocline is a fluid-to-solid media heat transfer coefficient which is necessarily less than infinite. As a result, a thermal gradient is established in the storage media, and the gradient can grow to occupy the entire height of the tank. To prevent the gradient from

increasing to the full tank height, the temperature of the fluid leaving the tank at the end of a discharge cycle must be allowed to fall below the design collector field outlet temperature, and the temperature of the fluid leaving the tank at the end of a charge cycle must be allowed to rise above the design collector field inlet temperature. Figure 2-7 shows the performance of both a direct and indirect thermocline system at the end of a 3-hour charge cycle and at the end of a discharge cycle. As predicted, the temperature at the tank outlet decays below the collector field outlet temperature throughout the discharge cycle, while the tank inlet temperature gradually rises above the collector field inlet throughout the charge cycle.

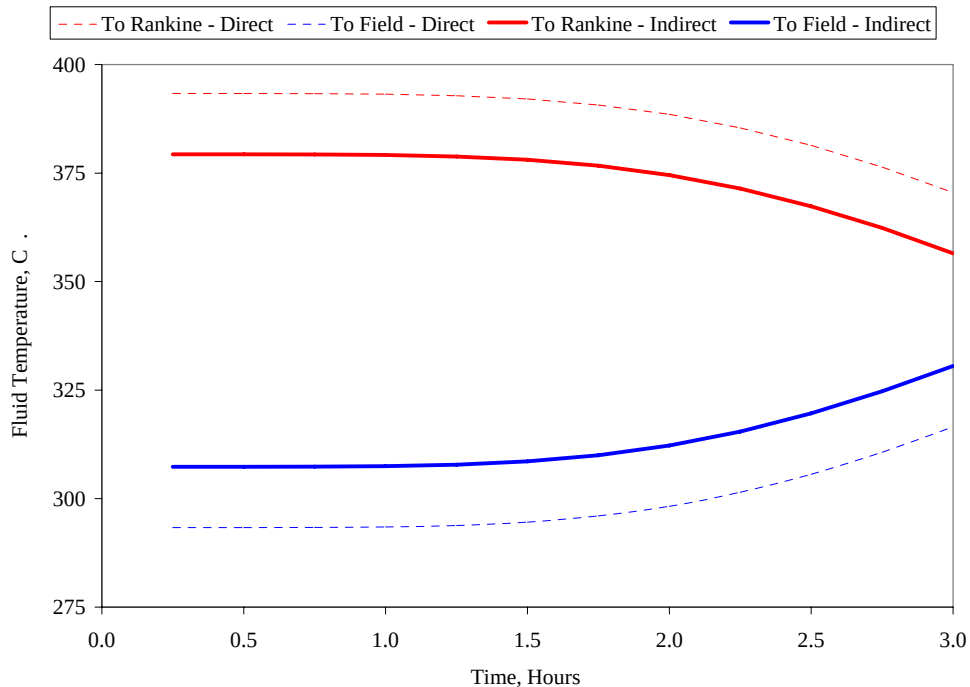


Figure 2-7
Indirect and Direct Thermocline Fluid Temperatures during Storage System Charging and Discharging

Thus, as the temperatures entering and leaving the thermocline tank diverge from the design temperatures, the operation of the thermocline tank will influence the performance of both the Rankine cycle and the collector field. Further, the magnitude of the effect will depend on the coincident output of the Rankine cycle, and the coincident thermal output of the collector field. In addition, the degree to which the thermocline is subjected to a full or partial charge cycle during the day, and a full or partial discharge cycle at the end of a day, will influence the shape and the size of the thermal gradient the following day. Further discussion of thermocline performance is included in Chapter 5.

Operating Experience

The two-tank direct and indirect storage systems are the only thermal energy storage systems to have seen commercial operation at a grid-connected CSP plant. The two-tank indirect system is

the closest to achieving commercial status, with one unit now operational at the Andasol 1 plant in Spain, and more either under construction or planned for operation in the next few years. The rest of the storage system concepts have undergone testing as prototypes and are poised to become commercial ventures with further research and development. For most of these systems in the demonstration stage, a pilot plant presents the logical next step required for achieving technological maturity.

Several early storage systems used oil as the storage fluid. Oils are not practical for future commercial projects for several reasons. The maximum operating temperatures are limited to 300-400°C (570-750°F) due to thermal degradation. The lower operating temperatures relative to molten salt central receiver systems means the steam cycles have lower Rankine conversion efficiencies. Even at these operating temperatures oils have a fairly high vapor pressure. A pressurized vessel would be required if oil were used as the thermal storage medium. Pressurized tanks can significantly increase the cost of the TES system, and the large tank sizes required to store thermal energy in this temperature range could make the TES prohibitively expensive. Pressurized tanks of oil also pose a fire hazard. The commodity prices of synthetic oils are too high for them to be considered as storage fluids. Molten salt and water are lower cost fluid options, and molten salt has the benefit of atmospheric pressure operation.

National laboratories, particularly in Europe and the U.S., are investing heavily in thermal energy storage research. The U.S. Department of Energy has awarded 15 grants totaling up to \$67.6 million dollars for FY 2009. The projects cover a broad range of TES technologies, including the proposed construction of a prototype thermocline system at the Arizona Public Service CSP plant in Red Rock, AZ, solid media storage, thermochemical storage and phase change materials. Through these grants and other renewable research, the DOE intends to spur the commercialization and deployment of solar technologies and to reduce the levelized cost of electricity generated at CSP facilities. The DOE goals include reducing the LCOE from 13-16 cents/kWh today with no storage to 8-11 cents/kWh with 6 hours of thermal storage capacity by 2015, and to less than 7 cents/kWh with 12-17 hours of thermal storage by 2020.⁵

Two-Tank Indirect

The two-tank indirect system has recently been proven at large scale. The first large scale commercial system was commissioned in early 2009 at the Andasol 1 plant in Spain. The technology is expected to be commercially viable and is considered the current state-of-the-art in thermal energy storage systems. Andasol 2 was commissioned in late 2009. Both plants have 1010 MWh_t storage capacity, providing roughly 7.5 hours of full power output at 50 MWe. The Andasol storage systems represent an important step towards incorporating thermal energy storage into CSP plants – the performance data from these plants will provide a valuable source of information for future thermal storage systems, and the knowledge and experiences gained will help pave the way for future two-tank TES systems. Andasol 3 is currently under construction with an identical TES system, and other plants with the same design are in development. Figure 2-8 shows the Andasol 1 storage tanks during construction.

⁵ Department of Energy, “DOE Funds 15 New Projects to Develop Solar Power Storage and Heat Transfer Projects For Up to \$67.6 Million”, <http://www.energy.gov/news/6562.htm>, 2008.



Figure 2-8
Thermal Storage Tanks under Construction at Andasol 1 (2007)

In addition to the Andasol storage systems, Arizona Public Service has contracted Abengoa to design and build a plant that will include an indirect thermal storage system using molten salt as the storage medium. As of this publication, ground breaking was scheduled for late 2010. The Solana plant will be located near Gila Bend, Arizona and will provide 280 MW of electricity to APS customers. The design uses a molten salt storage system consisting of six to eight storage tanks (three to four pairs of hot and cold tanks) with the capability for six hours of full load operation. Figure 2-9 shows the proposed layout for the Solana plant; the molten salt storage tanks are labeled “5” in the figure.



Figure 2-9
Artist's Rendering of APS Solana Power Plant; Molten Salt Storage Tanks are Labeled “5”⁶

⁶ Arizona Power Service, “About Solana Generating Station”,
<http://www.aps.com/main/green/Solana/Technology.html>, © 1999-2009

With the multiple TES systems at Andasol in operation and the APS Solana plant planned for 2009, the two-tank indirect storage system will likely be the first TES technology to achieve full commercial status, and can be considered the state of the art for thermal energy storage systems.

Two-Tank Direct

The SEGS I system included a 110-MWh_t two-tank direct TES system that was used in plant operation from 1985-1999. SEGS 1 used Caloria, a type of mineral oil, as both the HTF in the collector field and the storage medium. In 1999, the storage tanks were completely destroyed when the flammable oil caught on fire. Like other thermal energy storage systems that are either in the pre-commercial prototype or demonstration stages, the SEGS I TES was a one-of-a-kind storage system, and was not included in any of the later SEGS facilities. In SEGS II-IX Caloria was replaced by higher-temperature Therminol oil in the collector field, but since Therminol is difficult to store due to its higher vapor pressure at the operating temperatures in the plant, the later SEGS systems did not include thermal energy storage systems.

A two-tank molten salt storage system was first used at the Themis central receiver plant in France in the late 1980s. It used circular-horizontal hot and cold tanks with a total capacity of 40 MWh_t. Hitec salt was used in the receiver and storage system. In 1996 another central receiver project, Solar Two, demonstrated the same direct molten salt concept except with two vertical cylindrical tanks. This design provided the foundation for current two-tank molten salt thermal storage systems. Figure 2-10 shows the two-tank direct molten salt TES system at Solar Two, which has since been dismantled.

There are several direct central receiver projects in various stages of development. Gemasolar (also known as Solar Tres) is currently under construction in Spain and is expected to be running in mid-2011. The plant is 17 MW and will have 15 hours of storage capacity using the two-tank direct molten salt approach. SolarReserve recently announced power purchase agreements for three central receiver projects that are considerably larger, but will employ similar two-tank direct storage technology. The SolarReserve plant generation capacities range from 50 MW to 150 MW and will have a minimum of 7 hours of storage capacity. The 50 MW plant in Spain will have sufficient capacity to operate up to 24 hours a day with an annual capacity factor of about 80%.



Figure 2-10
Thermal Energy Storage System at Solar Two, Barstow, CA

The Italian National Agency for New Technologies, Energy and the Environment (ENEA) has been testing molten salt in a Solar Collector Test Loop facility since 2004 to study the effects of molten salt on the valves and other process components of a parabolic trough installation. After more than 2000 hours of operation and approximately 200 fill and drain cycles, ENEA reported that no major obstacles to molten salt operation in the test collector loop were encountered. According to ENEA, further research is needed to fully characterize such items as the sealing and gasket materials and any rotating joints that come into contact with the molten salt.

While conducting the collector loop tests, ENEA simultaneously developed a design for a pilot project, dubbed Archimede, which will integrate a parabolic trough and a direct two-tank TES system with a combined-cycle plant, using molten salt as the HTF. The project stalled due to a delay in receiving national subsidies for solar thermal power plants, but is expected to come online in 2010. Like Andasol for indirect systems, Archimede will be an important source of operating and performance data for a direct storage system.

For indirect trough TES applications, the cost of the oil-to-salt heat exchanger is high due to the large surface area needed for the low approach temperature of the oil. Significant cost savings are anticipated for direct trough storage systems if the collector field can be operated using molten salt as the HTF. Circulating molten salt through the collector field presents distinct challenges in contrast to a central receiver plant. Salt freeze recovery and heat loss in the collector field loops is a concern, as are the capabilities of the collector field process components, such as the piping, valves, and pumps.

The high freezing point of molten salt remains an issue for both direct and indirect systems, in troughs especially, and is the subject of current R&D efforts. Both the DLR and the DOE are developing salts with much lower melting temperatures than those associated with the common

binary nitrate salts, having achieved melting points as low as 100°C (212°F)⁷, and they are performing experiments with these salts to study freeze recovery⁸. The Department of Energy has also awarded grants for near-term development of advanced heat transfer fluids, including funding for projects that will attempt to develop low melting point eutectic salt mixtures.

Single-Tank Thermocline (Indirect or Direct)

The Solar One central receiver project included a 182 MWh_t indirect mineral oil thermocline system which began operating in 1982. It was shut down when a steam explosion caused a small rupture of the thermocline tank in 1988. Although the resulting fire was extinguished relatively quickly and with minimal damage, the storage tank was not reused. A single-tank thermocline using thermal oil has also been successfully tested at the Plataforma Solar de Almería, one of Europe's most prominent CSP research facilities. The PSA thermocline is a component of the Small Solar Power Systems – Distributed Control System (SSPS – DCS), which includes a parabolic trough collector field and a Multi-Effect Distillation (MED) desalination plant in addition to the thermal energy storage system. The direct thermocline systems uses dual-media storage; Therminol 55 from the collector field is the storage fluid and a metal filler inside the tank stores the heat. The system provides up to 5 MWh_t storage capacity. Although the thermocline testbed has been operating successfully since the early 1980's, no plans for further commercial development of this system are proposed at this time.

In 2001, Sandia National Laboratories successfully demonstrated a 2.3-MWh_t packed-bed thermocline storage system with binary molten salt fluid and quartzite rock and sand filler material.⁹ In developing the design of the thermocline testbed, Sandia evaluated various filler materials and found that a quartzite and sand mixture was an economical and practical choice and that both materials were able to withstand immersion in an isothermal molten salt bath as well as repeated thermal cycling tests with molten salt. Relatively simple cost analyses were conducted to evaluate the costs of materials, molten salt and filler materials for a larger commercial-scale thermocline TES system. The analyses showed that thermocline-based energy storage configurations may offer the least-cost energy storage option, being about 35 percent cheaper than a similar-sized two-tank TES system. The system studied by Sandia sought to improve on the indirect two-tank storage system concept, but a thermocline concept could also be applied to a direct system, and, like the two-tank system, will benefit from molten salt HTF research.

A single-tank direct thermocline storage system was proposed and designed for the 1 MW Saguaro parabolic trough power plant owned by Arizona Power Service. The plant began

⁷ Brosseau, D. and Kolb, G., "Sandia Thermal Storage Activities", Presented at Trough Workshop, Golden, CO. 2007.

⁸ G. Kolb, C. Ho, B. Iverson, T. Moss, and N. Siegel, "Free-thaw Tests of Trough Receivers Employing a Molten Salt Working Fluid", Proceedings of 2010 ASME Energy Sustainability Conference, ES2010-90040, Sandia National Laboratories, Albuquerque, NM, May 2010.

⁹ D.A. Brosseau, P.F. Hlava, and M.J. Kelly, "Testing Thermocline Filler Materials and Molten Salt Heat Transfer Fluids for Thermal Energy Storage Systems Used in Parabolic Trough Solar Power Plants", Albuquerque, NM 2004.

operation in December 2005 without a TES system, but APS and the national labs began studying options for retrofitting the plant to include six hours of thermal storage in order to provide electricity during the evening peak period. A design and cost estimate for the system was developed by Nexant, Inc. and Sandia National Laboratories provided a performance analysis of the proposed system. Both studies indicated that the thermocline was a viable option for the Saguaro plant, and a recent grant from the DOE will be used to build and test the thermocline storage system. The project is one of three thermal storage systems that will be incorporated into an operational power plant as part of this research phase. The results of the Saguaro project will provide much-needed performance data and operations experience for a thermocline system.

Development Status

Only the two-tank technologies have been demonstrated in commercial operation, and although a two-tank direct system using Caloria oil was successfully operated for fourteen years at SEGS I, systems using molten salt as the storage medium have yet to see continuous long-term commercial operation. Both the compatibility of the raw materials and the long-term durability, reliability, and performance of critical components of thermal storage systems using molten salt remain uncertain. Extended operation of molten salt systems in addition to further testing and research will be required to complete commercial development and to fully characterize the technical constraints on molten salt TES.

Thermocline technology would benefit greatly from operation and testing at a non-commercial pilot plant, much like Solar Two, or by simply incorporating a storage system into an operational CSP plant, as Arizona Public Service intends to do at the Saguaro plant. Although the oil thermocline at the PSA has performed well, the molten salt thermocline has yet to see any testing beyond the prototype studied at Sandia National Laboratories, and a full-scale pilot system integrated with a CSP plant will permit further observation and validation of this concept. Table 2-1 has a summary of current development needs for two-tank and thermocline technologies.

Table 2-1
Current Development Needs for Thermal Storage Technology

Storage Technology	Development Needs
Two-Tank Indirect	• Operating experience from the planned commercial facilities (Andasol, Solana) • More units in commercial operation • An established supply-chain for system components • Evaluation of long-term durability and reliability of materials in use with molten salt
Two-Tank Direct	• Validation of molten salt operation in a full-sized collector field • Operation and evaluation of a commercially operated unit • Evaluation of long-term durability and reliability of materials in use with molten salt
Single-Tank Thermocline (Direct or Indirect)	• Improved modeling techniques, including an effective method for sizing the thermocline tank • A pilot plant or commercially operated unit • Thermal ratcheting assessment

3

THERMOCLINE DESIGN AND OPERATION

Design Basis

Initially four storage capacities were modeled for both indirect trough and direct central receiver systems for a total of eight design cases. Over the course of the analysis, the largest possible single-tank sizes were identified as roughly 1500 MWh_t for the indirect trough and 3500 MWh_t for the direct central receiver. Two additional design cases were developed to show the full range of tank sizes that may be feasible for this technology. Although cost estimates were not specifically developed for the 1500 MWh_t direct central receiver and the 3500 MWh_t indirect parabolic trough, cost estimates can reasonably be extrapolated or interpolated for these corresponding cases. Table 3-1 contains a summary of all the design cases considered.

Table 3-1
EPRI Thermocline Molten Salt Energy Storage Design Cases

Design Case	MWh _t	Maximum TES Temperature	Technology
1	100	400°C (752°F)	Indirect Trough
2		560°C (1040 °F)	Direct Central Receiver
3	500	400°C (752°F)	Indirect Trough
4		560°C (1040 °F)	Direct Central Receiver
5	1000	400°C (752°F)	Indirect Trough
6		560°C (1040 °F)	Direct Central Receiver
7	3000	400°C (752°F)	Indirect Trough
8		560°C (1040 °F)	Direct Central Receiver
9	1500	400°C (752°F)	Indirect Trough
10*		560°C (1040 °F)	Direct Central Receiver
11*	3500	400°C (752°F)	Indirect Trough
12		560°C (1040 °F)	Direct Central Receiver

* Values were interpolated or extrapolated from other design case results

The selected storage fluid for all cases was a binary sodium/potassium nitrate salt mixture (60% NaNO_3 /40% KNO_3). It is likely that any future system incorporating a liquid storage media will use molten salt, with the exception of steam storage, which can be used for short-term buffer storage. Therminol is the most common heat transfer fluid in parabolic trough systems, and its maximum operating temperature is limited due to thermal degradation. Consequently the maximum inlet temperature for the molten salt storage fluid was set as 400°C (752°F) for the indirect trough storage systems in this study. For central receiver systems, the maximum temperature was set by the thermal degradation temperature of the molten salt; the inlet storage temperature was set as 560°C (1040°F).

Design Conditions

The complete thermocline storage system design requirements are presented in Appendix A. The major assumptions include the following:

- Molten salt temperature at solar receiver inlet (direct and indirect systems): 293°C (559°F)
- Six hours storage (solar multiple is approximately 1.8)
- Void fraction is 25%
- Barstow, California is assumed as the basis for cost and weather conditions

High level design considerations include:

- Structural integrity of tank and foundation
- Thermal analysis of tank and connected auxiliary hot piping systems
- Tank height/diameter ratio optimization (multiple modular thermocline tanks possible)
- Wall material
- Thermal insulation
- Heat tracing

Design details considered:

- Design of tank pressure boundary as per API standards
- Basic foundation design
- Upper, auxiliary and lower manifold designs to minimize entrainment
- Ullage space
- Pumps
- Piping
- Valves
- Heat exchangers

- Auxiliary systems including heat tracing
- Piping and tank thermal insulation
- Auxiliary salt melting equipment

Drawing and process diagrams are provided for Cases 1-8. Cost estimates for these design cases include tanks, pumps, valves, piping, filler material, TES fluid material, heat exchangers, and auxiliary systems. AACE Class 4 estimates are provided in Chapter 4.

Process Designs

Two CSP technologies are considered in this study: parabolic trough with indirect TES and central receiver with direct TES. In both systems molten salt is the storage fluid. For the parabolic trough, Therminol is the heat transfer fluid in the solar field. It exchanges heat with molten salt to charge and discharge the storage system. The heated Therminol is used to generate steam to produce electricity. The main disadvantage of the trough system is that energy storage is limited to temperatures of 400°C (752°F). A process flow diagram of this option is presented in Figure 3-1.

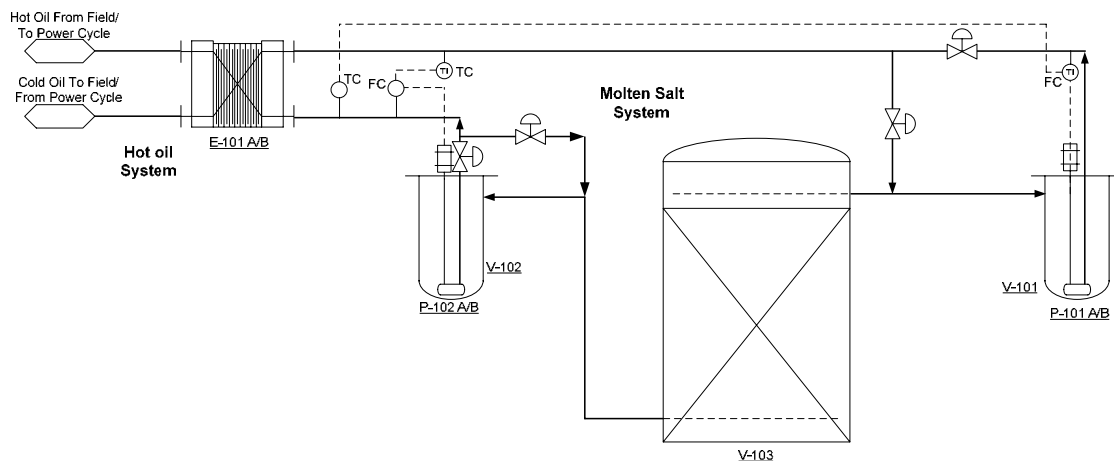


Figure 3-1
Design Option 1, Indirect Molten Salt Thermocline Storage for Parabolic Trough Design

In the central receiver design the salt is heated directly in the field before it enters the TES system. The salt is cooled by generating steam to produce electricity. Since salt is being used as the primary heat transfer fluid, the system can store heat at temperatures up to 560°C (1040°F), the maximum working temperature of the salt.¹⁰ The advantage of this system is that the higher temperatures reduce equipment sizes and flow rates; the storage volume is directly proportional to the difference between the charge and discharge temperatures. The main disadvantage of this

¹⁰ Bradshaw, R. W. and S. H. Goods, *Accelerated Corrosion Testing of a Nickel-Base Alloy in a Molten Salt*, SAND2001-8758, Sandia National Laboratories, Livermore CA, Printed November 2009. Sandia recently released R&D results from 2001 that suggest that the maximum operating temperature of salt can be safely raised to 650°C if oxygen is used as a cover gas in the storage tanks.

process is that it requires more expensive materials due to the higher temperatures. A process flow diagram of this option is presented in Figure 3-2. Complete process flow diagrams for both the indirect and direct systems are given in Appendix B.

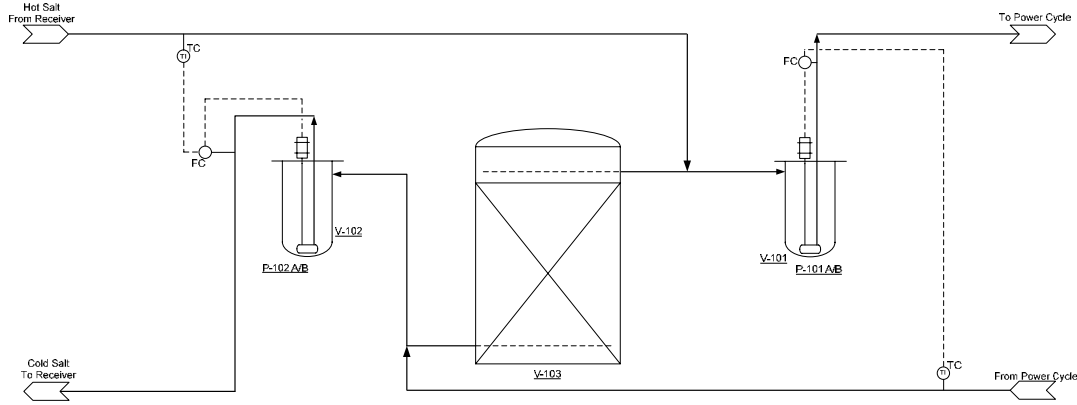


Figure 3-2
Design Option 2, Direct Molten Salt Thermocline Storage for Central Receiver Design

Tank Sizing

Thermocline tanks were sized using the method designed by Zhen Yang and Suresh Garimella¹¹. In this method, a tank diameter is assumed and the energy density of the tank is determined by dividing the desired power generation by the area of the cross section. This can then be used to solve for the Reynolds number using Appendix C.

$$\frac{P}{A} = k_{l,c} \frac{T_h - T_c}{d_s} \text{Re Pr} \quad \text{Equation 3-1}$$

In this equation P is the power generation, A is the horizontal cross-sectional area of the tank, $k_{l,c}$ is the thermal conductivity of the salt, T_h is the temperature of the hot salt, T_c is the temperature of the cold salt, d_s is the diameter of the tank, Re is the Reynolds number, and Pr is the Prandtl number. Since the Prandtl number is a known constant, it is possible to solve this equation for the Reynolds number.

The next step is to calculate the stored thermal energy of the tank per unit area. This makes it possible to solve for the non-dimensional useful energy ($H\eta_{95}$) as shown in Equation 3-2.

$$\frac{Q}{A} = [\varepsilon \rho_l C_{p,s} + (1 - \varepsilon) \rho_s C_{p,s}] T_h (T_h - T_c) d_s H \eta_{95} \quad \text{Equation 3-2}$$

¹¹ Yang, Z., Garimella, S.V. "Thermal Analysis of Solar Thermal Energy Storage in a Molten-salt Thermocline," Solar Energy, Vol. 84, 2010, pp. 974-985.

In this equation Q is the useful energy extracted from the tank, ε is the void fraction, ρ is density, and C_p is the specific heat capacity at constant pressure, H is the tank height divided by the filler size (nondimensional variable), and η_{95} is the efficiency. The subscripts l and s refer to the molten salt liquid and the solid filler, respectively. The efficiency is based on a 95% allowable threshold between the lower and upper temperature limits of the molten salt stored. Under this thermal constraint, any salt outflow with temperature below the 95% threshold is considered not useful for steam generation (just as one example of a cut-off). This value is somewhat arbitrary, chosen to reflect the inherent thermal degradation of the discharge process with time. If colder temperatures are viable for steam generation, the threshold should be lowered to reflect greater flexibility of the thermocline tank operation. This in turn would increase the discharge efficiency of a given tank as more of the energy extracted is considered useful. Figure 3-3 shows the change in efficiency for a tank of fixed height and Reynolds number as a function of minimum allowable discharge temperature. As the allowable discharge temperature is lowered, the discharge efficiency increases (going to the left of the x-axis).

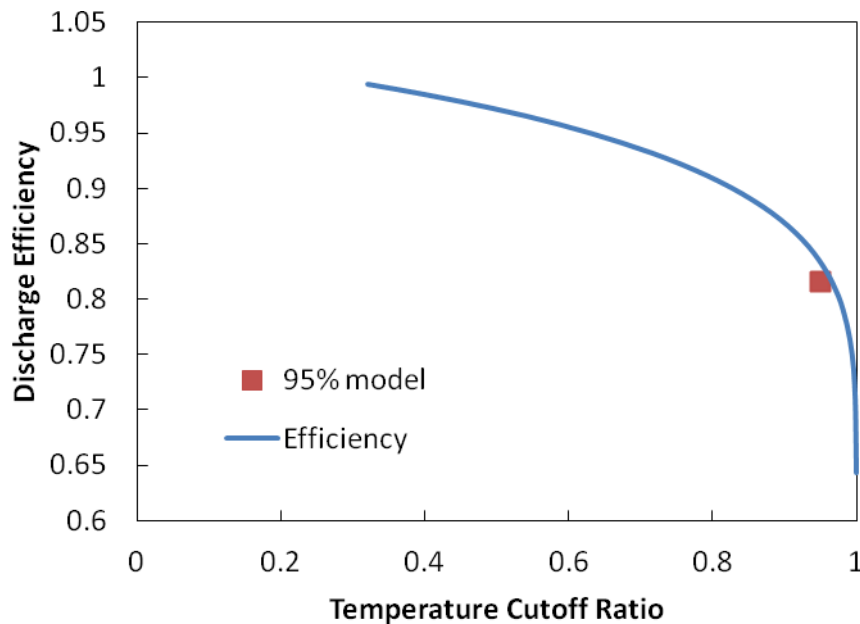


Figure 3-3
Efficiency Curve as a Function of Allowable Discharge Temperature

Once $H\eta_{95}$ is known the efficiency is assumed to be 100% and H is calculated. Then the efficiency is calculated from Equation 3-3.

$$\eta_{95} = 1 - 0.1807 \text{Re}^{0.1801} \left(\frac{H}{100} \right)^{0.00234 \text{Re}^{-0.6151} + 0.00055 \text{Re} - 0.485} \quad \text{Equation 3-3}$$

This efficiency is substituted into the previous calculation, and the final result is determined through multiple iterations until the height of the tank converges on a single value. This height is the minimum height required to provide the design power generation while also maintaining a stable thermocline. It can be seen from repeated calculations that taller tanks with smaller diameters have the highest efficiencies if the tanks are of equal volume. For a fixed tank height tanks of different diameters will perform identically under adiabatic conditions.

This approach implies that taller tanks are preferred. However, there are many design considerations that limit the height of the tank such as the maximum bearing capacity of the soil and earthquake code requirements. A civil engineering study was performed using soil information for Barstow, California.¹² Due to the density of the salt and quartzite-sand mixture, it was determined that the maximum liquid level in the tank should be limited to 39 ft. The details of this study are included in Appendix D. This is the maximum height that can be supported by a typical foundation, and it was used for the basis of the thermocline design. If a 10% oversized foundation is designed (above 150' in diameter), it is possible to support a molten salt level up to 49 ft (50 ft tank height). A figure showing the maximum thermocline capacity for standard and oversized foundations is presented in Appendix K.

No active convection cooling is required in the assumed civil foundation design. The insulation bricks and lower temperature salt at the bottom of the tank limit the concrete temperature well below 100°C (212°F) for both the parabolic trough and the central receiver thermocline designs.

Material Selection

The binary molten salt used in this project was assumed to be a mixture of sodium nitrate and potassium nitrate. This compound is corrosive in the presence of liquid water. However there is little to no water present at the process conditions. As a result, material selection is driven only by the temperature and pressure requirements of the process.

All equipment and tanks that operate below 800°F (427°C) were assumed to use high temperature carbon steel SA-285-C. This includes all piping and surge tanks, as well as the head and base plate of the tanks. This material is well suited to the service, and is very cost effective.

All equipment and tanks operating above 800°F (427°C) were assumed to be composed of Stainless Steel 347 (SA-240-347). This material has been historically used in this service; however, there is a slight risk of material sensitization. Black & Veatch recommends a materials testing program be completed to further qualify the material for use in the final design. A discussion of sensitization and its potential impact is presented in Appendix E.

Tank Design

The final height of the tank was assumed to be 40 feet for all of the original design cases except Case 8, in which the larger foundation permitted constructing a tank 47 feet tall. All of the tanks

¹² "Environmental Conditions Barstow, California"

are designed with 1 foot of ullage space available at the top to accommodate thermal expansion, as well as 3 feet of molten salt at the top to provide space for the distribution manifold. The remainder of the tank is filled with quartzite packing and molten salt. A summary of results is presented in Table 3-2. Results for the additional design cases (9-12), representing the maximum feasible single-tank sizes for indirect and direct systems, can be found in Appendix J. Based on industry experience, a maximum practical tank diameter of 160 feet was assumed. Complete sizing information for all of the remaining process equipment (surge tanks, heat exchangers, pumps) is provided in Appendix F.

Due to the heating and cooling of the salt, the liquid level in the tank will vary over time. At 560°C (1040°F) the salt will have a liquid level of 39 feet, providing 3 feet of heel in the tank above the hot molten salt nozzle. When the salt is cooled to 293°C (559°F), it will lose approximately 10% of its volume. If the entire vessel were cooled to 293°C, the salt would lose 1 foot 3 inches of height.

Table 3-2
Summary of Molten Salt Thermocline Storage Tank Sizes

Design Case	MWh _t	Number of Tanks	Tank Height (ft)	Tank Diameter (ft)
1 Indirect Trough	100	1	40	48
2 Direct Central Receiver		1	40	31
3 Indirect Trough	500	1	40	107
4 Direct Central Receiver		1	40	69
5 Indirect Trough	1000	1	40	152
6 Direct Central Receiver		1	40	98
7 Indirect Trough	3000	2	40	160
8 Direct Central Receiver		1	47	155

For cost comparison purposes, a two-tank thermal energy storage system was designed as well as the thermocline system. The two-tank TES system is similar to the thermocline system, except the single thermocline tank and the surge tanks are replaced with two molten salt tanks. One molten salt tank holds cold salt at 293°C (559°F), and the other molten salt tank holds salt at either 400°C (752°F) or 560°C (1040°F) depending on the design case. The sizes for the equivalent capacity two-tank systems are provided in Table 3-3.

Table 3-3
Summary of Molten Salt Two-Tank Storage Tank Sizes

Design Case	MWh _t	Number of Two-Tank Sets	Tank Height (ft)	Cold Tank Diameter (ft)	Hot Tank Diameter (ft)
1a Indirect Trough	100	1	46	35.5	36.5
2a Direct Central Receiver		1	46	22.5	23
3a Indirect Trough	500	1	46	78.5	81.5
4a Direct Central Receiver		1	46	50	51.5
5a Indirect Trough	1000	1	46	111	115
6a Direct Central Receiver		1	46	70.5	73
7a Indirect Trough	3000	2	46	136	141
8a Direct Central Receiver		1	46	122	126

Heat and Material Balance

Molten salt flow rates are presented in Table 3-4. The charging salt flow is the amount of salt heated by the solar collector field. The discharging salt flow is the amount of salt used to produce electricity. For the purposes of this design, a solar multiple of 1.8 was assumed. A detailed heat and material balance containing physical properties and the different operating modes is presented in Appendix C.

Line sizes for the system are presented in Table 3-5. Line sizes were calculated to provide pressure drops below 1 psi per 100 feet of pipe. For the purposes of line sizing, all pipes are assumed to be schedule 40.

Table 3-4
Molten Salt Flow Rates

Design Case	Thermal Storage (MWh)	Charging Molten Salt Flow (lb/hr)	Discharging Molten Salt Flow (lb/hr)
1 Indirect	100	1,488,444	826,913
2 Direct	100	596,493	331,385
3 Indirect	500	7,442,220	4,134,567
4 Direct	500	2,982,463	1,656,924
5 Indirect	1000	14,884,441	8,269,134
6 Direct	1000	5,964,926	3,313,848
7 Indirect	3000	44,653,323	24,807,402
8 Direct	3000	17,894,777	9,941,543

Table 3-5
System Line Sizes

Design Case	Charging Line Size (inches)	Discharging Line Size (inches)
1 Indirect	10	8
2 Direct	8	6
3 Indirect	20	16
4 Direct	14	12
5 Indirect	24	20
6 Direct	18	16
7 Indirect	36	30
8 Direct	26	20

Insulation

A data sheet for the thermocline tank is attached in Appendix K. This data sheet shows the mechanical details considered in the design of the vessel and the fundamental design of the shell and vessel head. Insulation is required both for personal protection and to minimize the operating costs of the facility. Mineral wool was selected as insulation because it is suitable for the high temperatures of the tank and because it offers the best process economics. The thickness of the insulation was optimized to minimize project costs. These costs were estimated assuming insulation prices of \$1.71/ft² for a 2-inch-thick board of insulation. Labor costs were assumed to be \$38.25/hr, which is typical for California. Power prices were assumed to be \$0.35 per kWh and prices were assumed to escalate at 3% per year. Capital costs were depreciated over a period of 5 years, which is typical for renewable energy projects. The final insulation thicknesses are presented in Table 3-6.

Table 3-6
Summary of Insulation Thickness

Service	Temperature	Thickness (in)
Cold Salt	288°C (550°F)	15
Hot Salt (Trough Design)	400°C (752°F)	17
Hot Salt (Power Tower Design)	560°C (1050°F)	23

Impoundment Wall Design

In order to ensure molten salt containment in the event of a tank failure, an impoundment wall must be constructed around the thermocline tank. Figure 3-4 shows a general arrangement of this design. The impoundment wall would be constructed from compacted soil and is designed to contain the complete volume of the tank, including molten salt and quartzite rock.

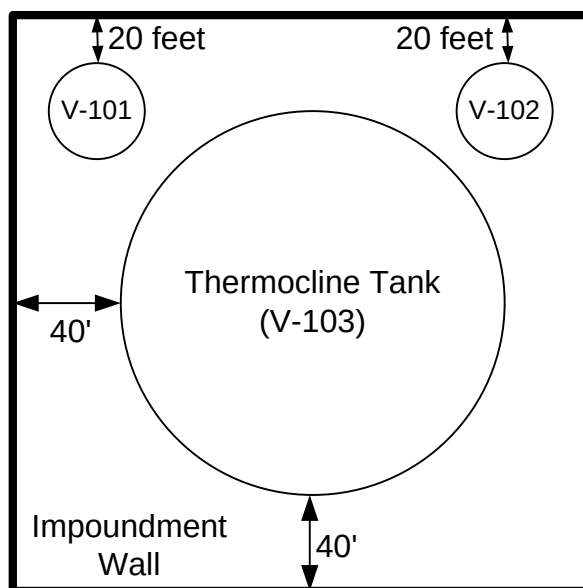


Figure 3-4
General Arrangement of the Impoundment Wall

The minimum distance from the thermocline tank to the impoundment wall is 40'. This distance was selected following the guidelines provided by NFPA 59A. Impoundment wall dimensions are provided in Table 3-7. This information is to be taken as a general guideline, and should be confirmed in a detailed design.

Table 3-7
Impoundment Wall Design Information

Case	Wall Height (ft)	Distance From Tank (ft)
1 Indirect	4.5	40
2 Direct	2.5	40
3 Indirect	10	40
4 Direct	7	40
5 Indirect	13	40
6 Direct	10	40
7 Indirect	13	40
8 Direct	10	45

Thermocline Distributors Design

The molten salt distributors are critical to the operation of the thermocline. The intended role of the distributors is to provide a uniform distribution of the molten salt flow over the entire cross-sectional area of the tank. This is necessary to maintain the separate temperature regions within the tank. A sketch of the proposed upper and lower distributor designs is given in Figure 3-5.

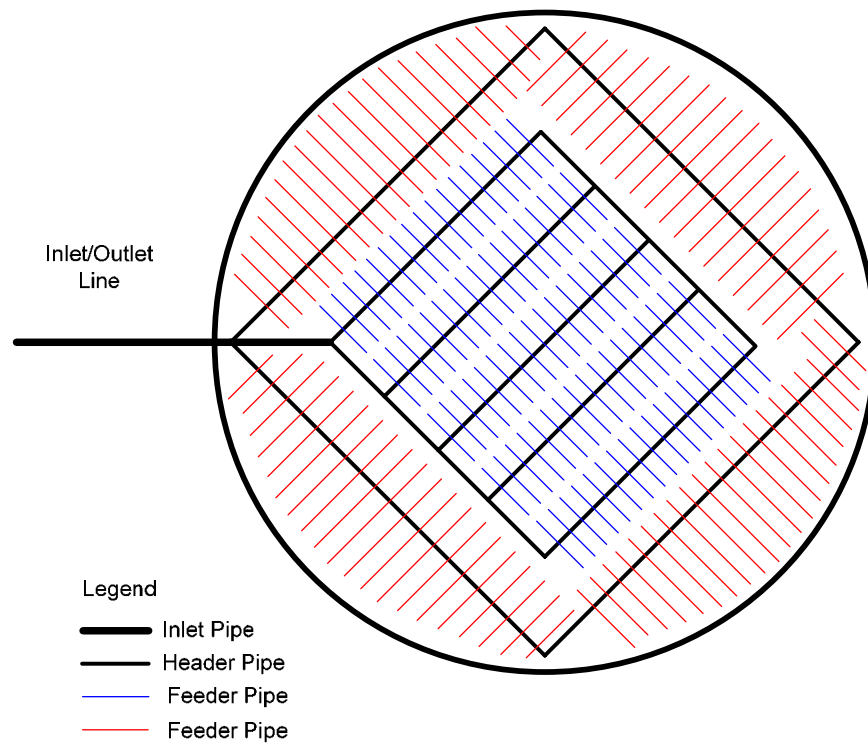


Figure 3-5
Thermocline Distributor Design

Additional details about the feeder pipe are provided in Figure 3-6. The bottom feeder pipe uses a larger guard pipe to support the weight of the quartzite material. The larger pipe is screened to remove quartzite from the molten salt. The top feeder pipes do not require a supporting external pipe, and simply rest on top of the quartzite rock. A sketch showing the manifold arrangement, as well as the nozzle placement and thermocouple placement is provided in Appendix M.

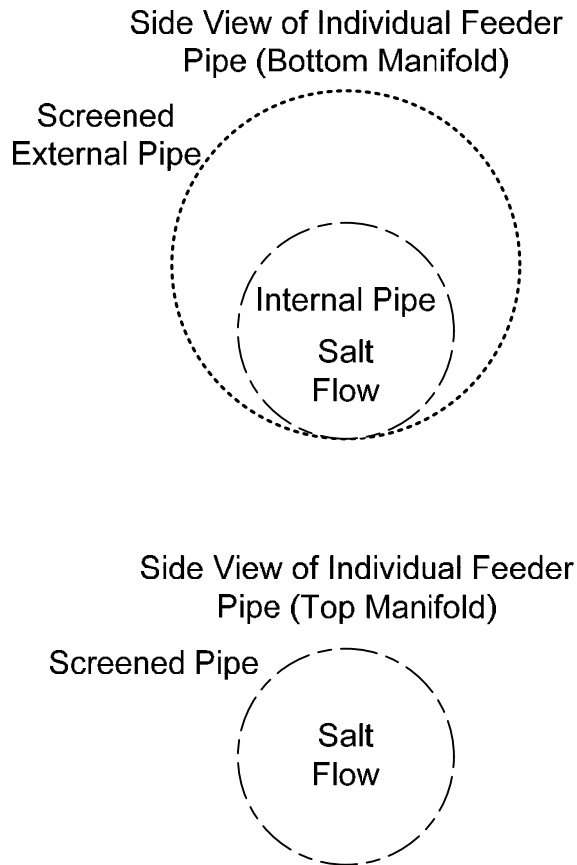


Figure 3-6
Thermocline Distributor Feeder Pipe Design

An orifice density of 3 holes per square meter was assumed to ensure adequate circulation of the molten salt. The orifice diameter is 0.5 inches, which was selected to minimize plugging of the distributor due to the presence of quartzite fines. A hydraulic analysis was completed on the distributor and the recommended pipe sizes are given in Table 3-8. It was assumed that screening material could be placed over the orifice holes to prevent sand from entering the distributor. This information is preliminary, and a detailed analysis of the distributor will need to be completed in the future in order to ensure proper separation and support of the quartzite material in the thermocline environment. This design is only a preliminary concept of the distributor, and further details will need to be determined through a detailed design. The distribution of the orifice holes will need to be optimized for uniform flow across the thermocline.

Table 3-8
Distributor Pipe Size

Design Case	1 Indirect	2 Direct	3 Indirect	4 Direct	5 Indirect	6 Direct	7 Indirect	8 Direct
Inlet Pipe Diameter (in)	12	10	20	14	24	18	36	26
Header Pipe Diameter (in)	8	6	12	8	16	12	24	18
Feeder Pipe Diameter (in)	3	2	6	4	8	5	8	6
Number of distribution points	420	200	2,100	900	4,200	1,800	11,000	5,100
Total Pressure Drop (psi)	0.37	0.10	0.99	0.42	1.07	0.72	1.10	0.63

Purdue University also performed a CFD investigation of the distributor manifold design proposed for the thermocline system. The manifold model was applied to the eight original proposed design cases to assess the uniformity of the outflow from the manifold into the fillerbed region. GAMBIT was used to mesh the distributor and the adjacent filler regions, as shown in Figure 3-7.

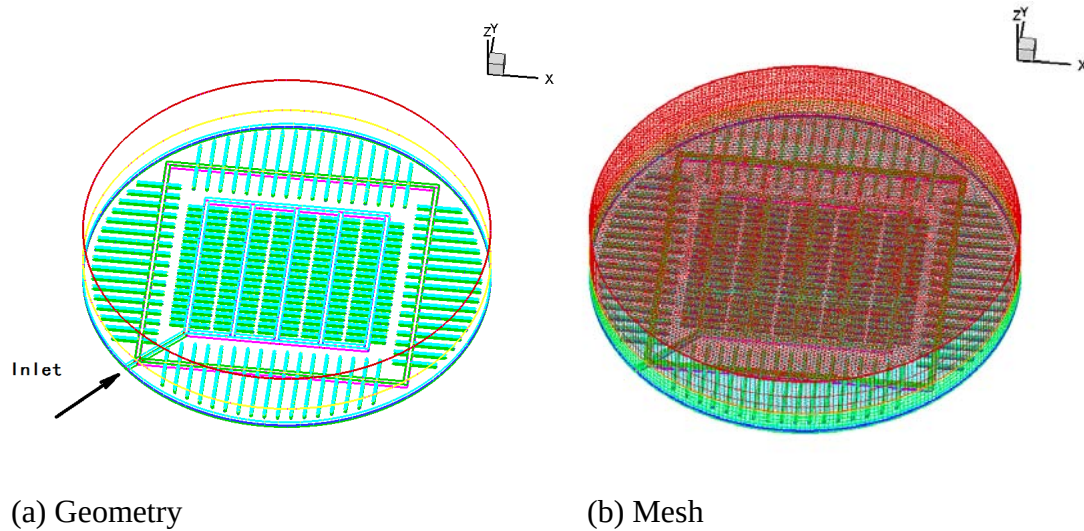


Figure 3-7
(a) Distributor and its Adjacent Regions, and (b) the Corresponding Mesh

The following equations govern the computation of distributor flow:

Flow in the tube region: The relation between pressure p and mean flow velocity u_m is expressed below, where D is tube diameter, l the flow distance and λ the flow friction factor which is determined by Figure 3-8:

$$0.5\rho u_m^2 = -\frac{D}{\lambda} \frac{dp}{dl}$$

Equation 3-4

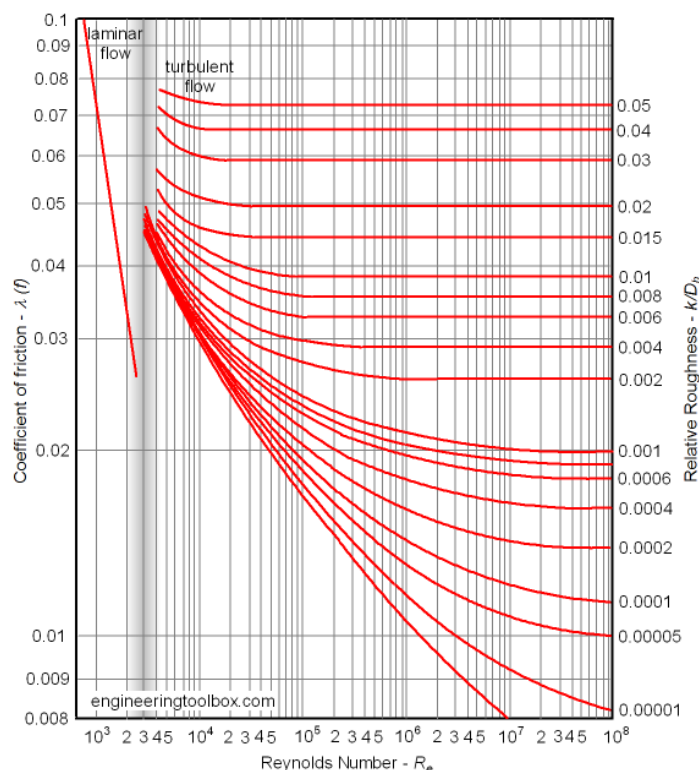


Figure 3-8
Flow Friction Coefficients from Moody's Chart¹³

It is noted that the flow resistance at connections between tubes is neglected in the calculation due to its small value relative to that in the straight tubes. The flow resistance in the interim region (as shown in the inset in Figure 3-6) between the external and interior tubes is neglected relative to the larger resistance in the filler region.

Flow through perforated tubes: The perforated tube wall is assumed to have a uniform distribution of 1/2 inch holes and is treated as a flow resistance film in the model. The flow resistance through an orifice can be estimated as illustrated in Figure 3-9.

¹³ <http://www.chem.mtu.edu/~fmorriso/cm310/MoodyLFpaper1944.pdf>

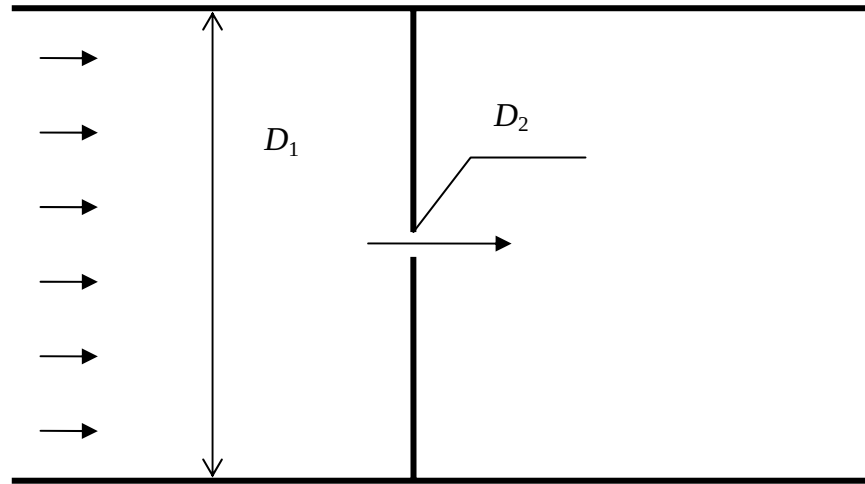


Figure 3-9
Flow through an Orifice

The relationship between flow rate and pressure difference across the orifice is given as¹⁴:

$$Q = \frac{\pi D_2^2}{4} \sqrt{\frac{1}{1 - \left(\frac{D_2}{D_1}\right)^4}} \sqrt{\frac{2(p_1 - p_2)}{\rho}} \quad \text{Equation 3-5}$$

In the case of perforated feeder tubes, D_2 is equivalent to the diameter of the holes in the feeder tubes and D_1 can be represented by the pitch between the holes. With the given geometries in the 8 cases, the ratio between D_2 and D_1 is very small (≈ 0.02), and therefore, the above equation reduces to:

$$Q \approx \frac{\pi D_2^2}{4} \sqrt{\frac{2(p_1 - p_2)}{\rho}} \quad \text{Equation 3-6}$$

Flow in the filler region: Flow in the filler region can be modeled by Darcy's Law which is applicable for flows in porous media at low Reynolds numbers. The typical Reynolds number in the eight cases considered is very low (~ 1), and thus may be readily modeled as follows:

$$\bar{u} = \frac{K}{\mu} \nabla p \quad \text{Equation 3-7}$$

¹⁴ <http://www.pipeflowcalculations.com/orifice/theory.htm>

$$K = \frac{d^2 \varepsilon^3}{175(1 - \varepsilon)^2}$$

Equation 3-8

In Equation 3-7, $\vec{u}$ is velocity vector, K is permeability and μ is liquid viscosity. K is determined using Equation 3-8¹⁵. The diameter of the filler particles is fixed at 1 cm in this study.

The effectiveness of the distributor is assessed based on the distribution of fluid flux over the cross section at a distance of 0.2D (D is the tank diameter) away from the distributor. Figure 3-10 through Figure 3-17 show the normalized fluid flux distribution for the different cases. In the figures, the flux is normalized by the average value over the cross section. As shown, the maximum flux is about 2 times the minimum, indicating that the distributor design could be improved. Note that the velocity scales are different for each plot, as indicated by each color legend.

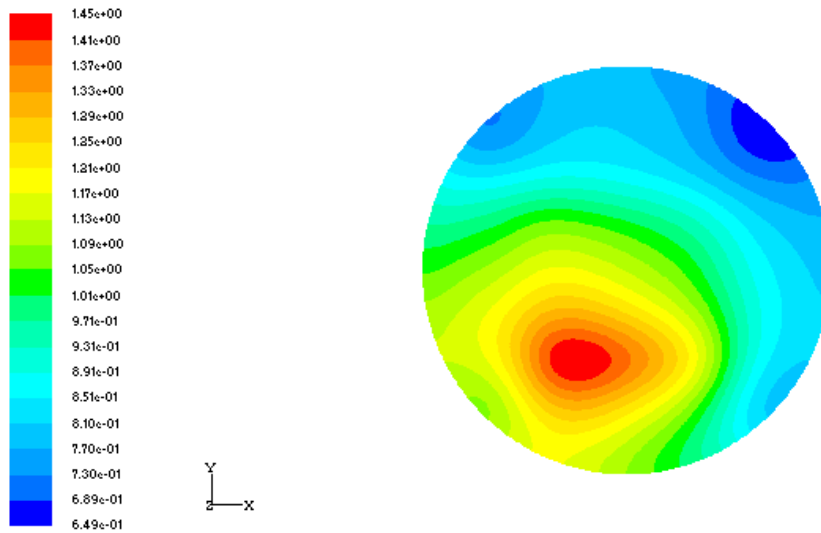


Figure 3-10
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 1

¹⁵ Beckermann, C., and Viskanta, R., 1988, "Natural Convection Solid/Liquid Phase Change in Porous Media," Int. J. Heat Mass Transfer, 31, pp. 35–46.

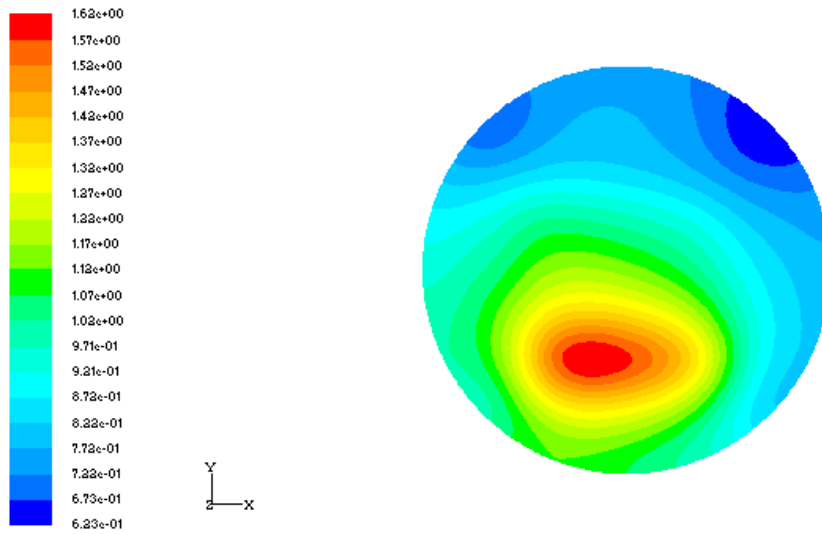


Figure 3-11
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 2

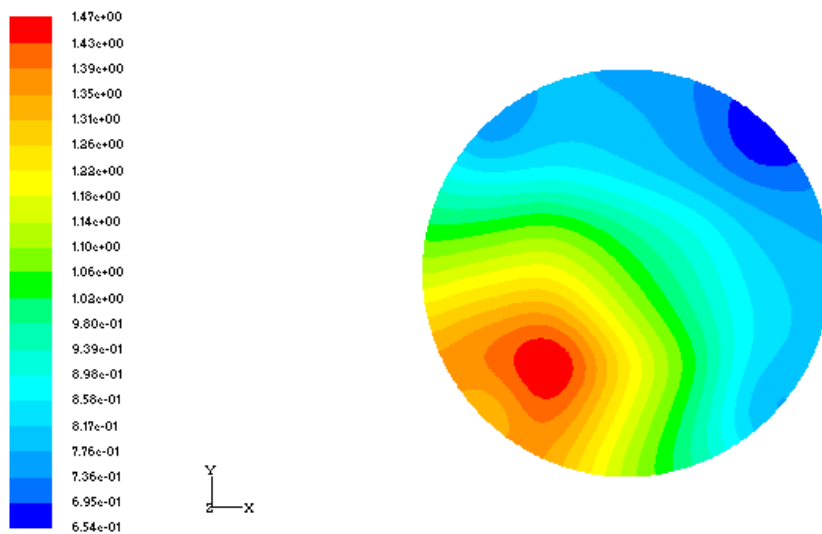


Figure 3-12
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 3

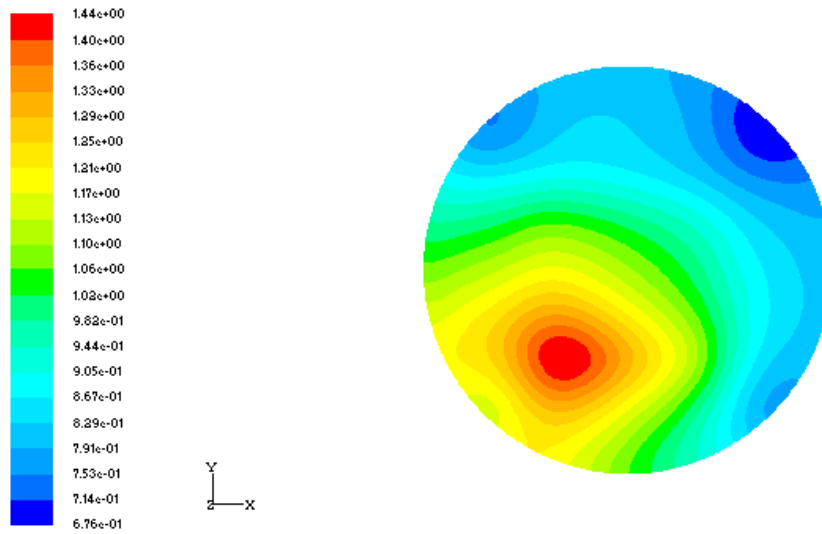


Figure 3-13
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 4

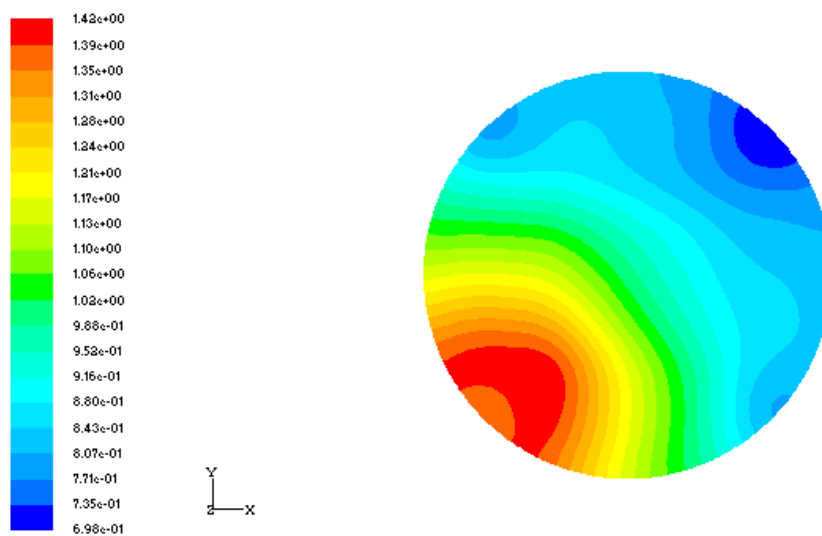


Figure 3-14
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 5

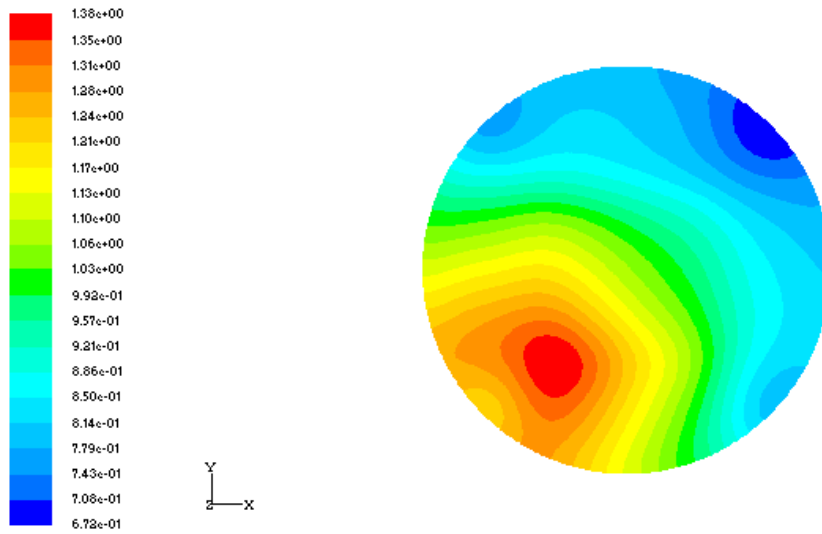


Figure 3-15
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 6

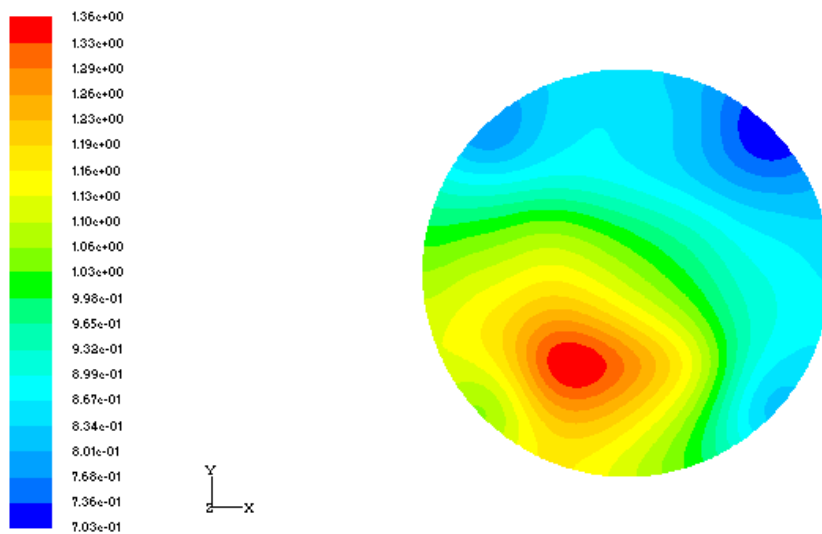


Figure 3-16
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 7

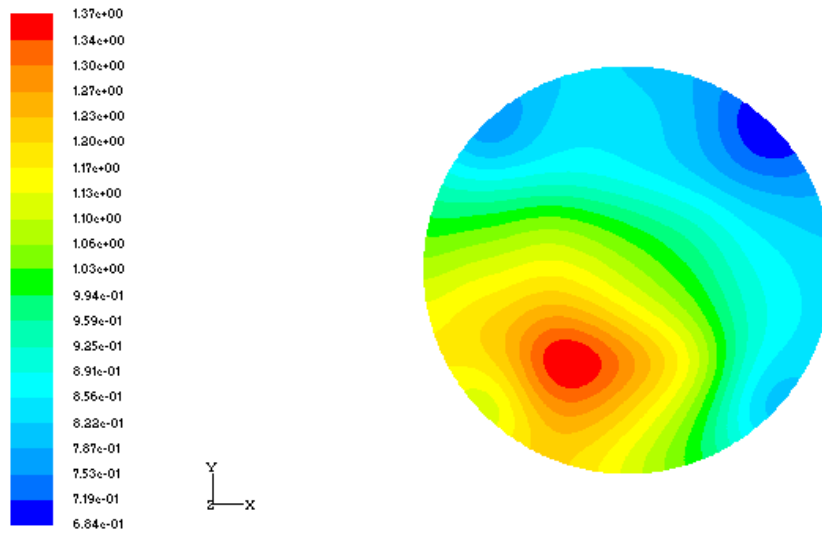


Figure 3-17
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor – Case 8

To improve flow uniformity of the distributor, two potential options are:

1. Reduce the diameter and increase the number of orifices.
2. Use header and inlet tubes of larger diameter.

Case 8 is considered as an example: increasing the feeder diameter to 26 inches generates the flux distribution shown in Figure 3-18 below. The ratio of the maximum and minimum fluxes is now reduced from 2 to 1.86.

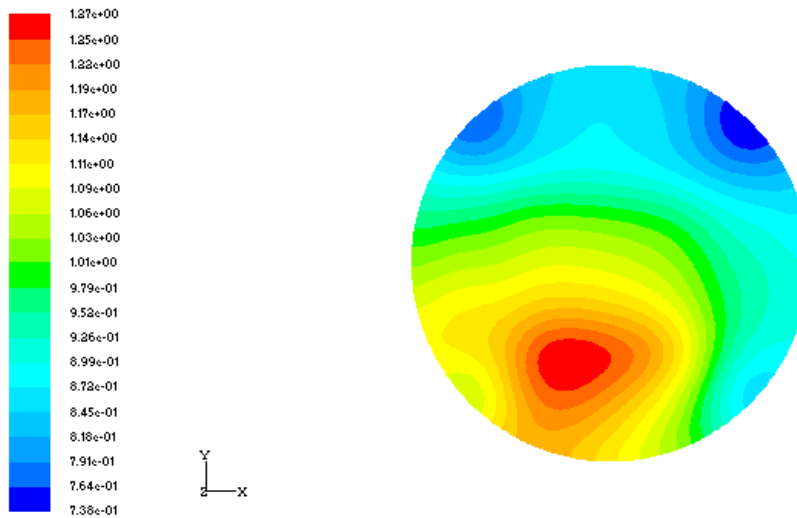


Figure 3-18
Normalized Flow Flux Distribution for Case 8 with Increased Feeder Diameter

Further investigation of the thermocline distributor manifolds with different structures was considered in order to achieve a more uniform flow flux distribution inside the thermocline tank, relative to the original manifold structure.

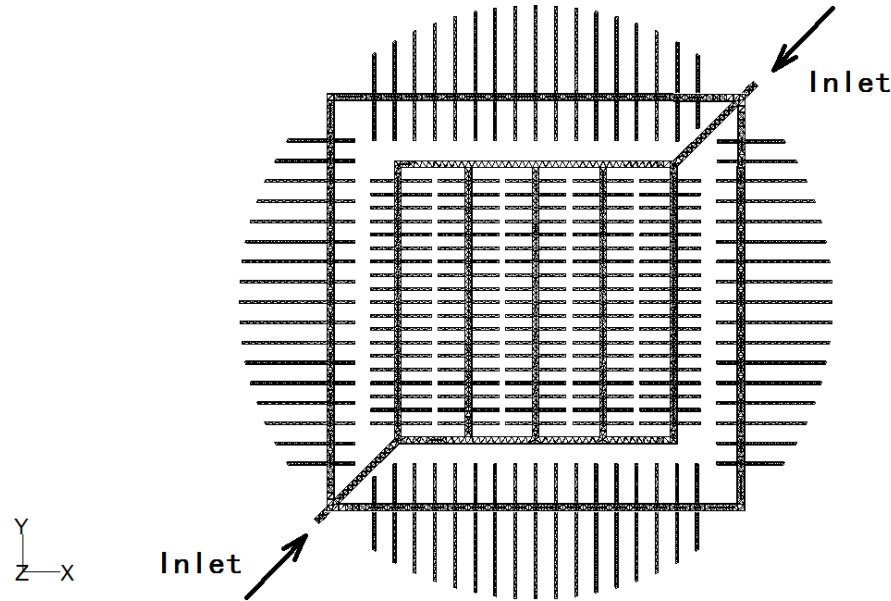


Figure 3-19
Distributor Manifold with Two Inlets

Figure 3-19 illustrates a modification of the original manifold. Two inlets are installed at the two opposite corners in this case. Figure 3-20 shows the normalized flow flux for Case 1 over a cross-sectional plane that is at a distance of $0.2D$ from the distributor. The flow flux distribution appears symmetric, unlike that obtained earlier with only one inlet. The maximum/minimum flow flux ratio in Figure 3-20 is 1.47, which represents a significantly improved distribution compared to the ratio of 2.23 obtained with only one inlet. This indicates that the flow flux distribution is more uniform with a two-inlet distributor.

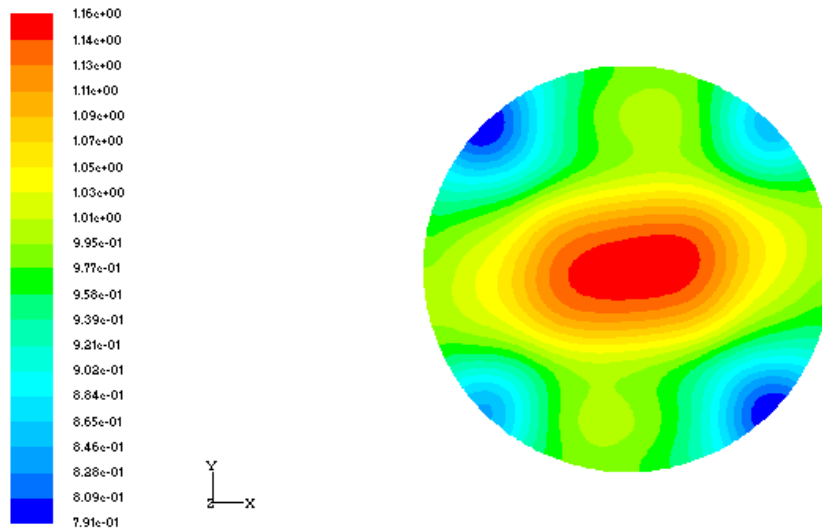


Figure 3-20
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor (Case 1, Two Inlets)

Another approach for improving the uniformity of flow flux distribution is to use a non-uniform distribution of hole density (hole number per unit area) at the feeder walls. The distributor can be segregated into different sections with different hole densities, as shown in Figure 3-21. The ratio of hole number density among areas A, B, C, D, and E is 15:35:35:21:21. Areas C and B, which are the farthest away from the inlet, have the higher densities of holes.

The resulting flow flux distribution is shown in Figure 3-22. It is clear from the figure that the maximum/minimum flow flux ratio is reduced further to 1.33 (compared to 1.47 in the two-inlet distributor case above), which indicates that the flow flux distribution is more uniform. It is possible to achieve further improvements in flow distribution uniformity if the pore number density in each area in Figure 3-21 is better-optimized.

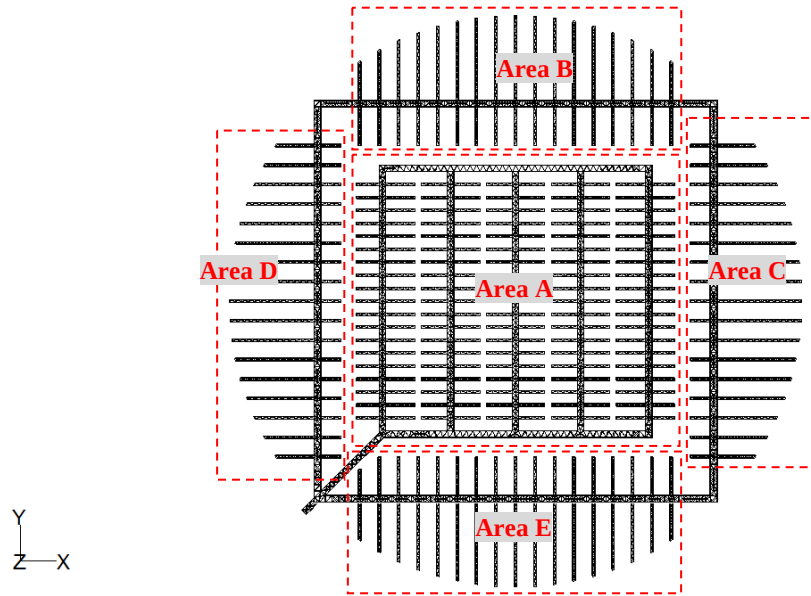


Figure 3-21
Non-Uniform Distribution of the 1/2-inch Holes on the Distributor Manifold

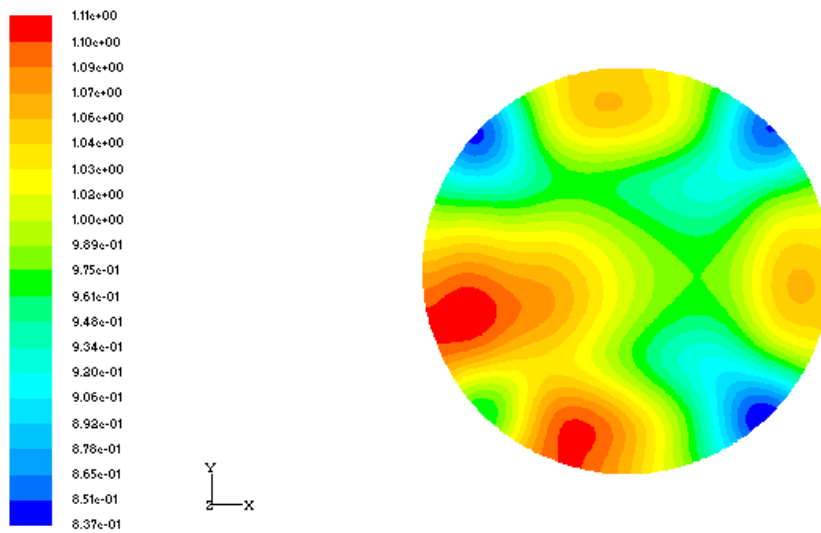


Figure 3-22
Normalized Flux Distribution over the Cross Section at a Distance 0.2D Away from the Distributor (Case 1, Distributor in Figure 3-21)

Surge Tanks and Molten Salt Pumps

The main function of the surge tanks (V-101 and V-102) is to provide a platform for the hot and cold molten salt pumps. The pumps are mounted in the surge tanks because the main thermocline tank is filled with quartzite rock and sand. Although it was recommended that the molten salt pumps be placed directly inside the tank following the Solar Two project, the current project team chose to mount the pumps in the gravity-fed surge tanks adjacent to the main tank. By removing the pumps from the main thermocline tank it will not be necessary to install wells for the pumps, which will simplify the design of the thermocline tank. In addition, the surge tanks will allow the pump shafts to be significantly shorter which will have a considerable impact on the final cost of the pumps. Next, if the pumps were housed in the thermocline tank the cold salt pipe would pass through the hot salt zone, which could significantly impact pump reliability and operation. Finally, this design simplifies pump maintenance and will allow for easier access. It is noted that the alternative approach to embed pumps with long shafts and sleeves in the tanks may also be acceptable and potentially reduce costs. This may be explored further in a detailed design phase.

The surge vessels were designed with a diameter large enough to accommodate the number of pumps required for each design scenario in addition to an installed spare. The height of the vessel was selected to allow for approximately 30 seconds of residence time in the tank. The surge tanks are gravity fed, which means salt will flow between them only when there is a difference in liquid level. This difference is measured in liquid head (units: feet). For the flow rates reported in the heat and material balance, a 3 foot difference in head is required for the salt to flow from the surge tanks to the thermocline tank. This height difference is accomplished by raising the surge tank skirt height such that it can support a liquid level at least 3 feet above the top of the thermocline tank. A mechanical sketch of the surge tank is presented in Appendix H.

The pumps for this process were limited to 4000 gallons per minute of throughput at any design pressure. Based on the largest commercially available pump, capacities above this limit would require multiple pumps. Using a standard design for the pump lowers the overall cost of the equipment and makes it easier to obtain replacement parts. Each pump was assumed to have a variable speed drive, and one installed spare was included in the estimates for both the hot service and cold service to ensure reliable operation.

Heat Exchanger

For the trough design, a welded plate and frame heat exchanger was selected to accomplish the heat transfer between the salt and the circulating oil. This was chosen over a shell and tube heat exchanger due to the high duties and close temperature approaches required in each design case. A welded plate and frame exchanger has been successfully used before in this service and is a proven design. The plate and frame heat exchanger is based on sizing from Alfa Laval, which has experience with heat exchangers in this service.

Additional Design Considerations

There are a number of assumptions made in this study that should be confirmed for a final design.

1. The tank heights were selected based on a soil study performed in Barstow, California. Different locations may permit larger tanks to be constructed, which would increase the efficiency of the system and reduce costs. Conversely, building a thermocline tank in a location with poor soil may reduce the height of the tank or increase the foundation size, either of which would increase costs.
2. Based on the operating experience at Solar One, which did not experience increased stresses on the tank from thermal ratcheting¹⁶, the current process design does not make any allowances for the structural impact of thermal ratcheting. Thermal ratcheting may occur over time as the packed bed is thermally cycled. As the quartzite is cooled it contracts and compacts in the bottom of the tank. When the tank is reheated the quartzite cannot return to its original position. The quartzite then expands and places pressure on the walls of the tank. Over time this could potentially damage the tank. To eliminate risk associated with thermal ratcheting there is an R&D effort at Sandia to analyze this issue for different temperature spreads, size and type of aggregates used as filler, and packing density. Design considerations to minimize the impact of thermal ratcheting will be provided, if needed, to minimize the impact on the structure.
3. Depending on the exact application of the thermocline system, some of the equipment in the scope of the estimate may be unnecessary. For example, if an existing solar plant were retrofitted to use thermal energy storage, the main circulation pumps may already be installed as part of the existing facility. Therefore some cost savings may be achieved in the actual application of the thermocline system compared to the estimates presented in this study.
4. The rock and sand filler design is an important consideration. It is important to provide adequate screening and support material to separate the quartzite from the distribution manifold. This is necessary to prevent rock and sand from exiting the thermocline tank and potentially damaging other pieces of process equipment. In addition, the quartzite has the potential to generate a significant amount of fines that may need to be filtered from the system periodically. It may be necessary to install a filter that can be purged outside the thermocline tank. A rock only region around the distribution manifolds may help filter quartzite fines from the distribution system.
5. It may be necessary to inspect the welding on the thermocline tank periodically to check for potential corrosion damage. For this reason, provisions should be made to remove some insulation from the outside walls as necessary to inspect the bare metal surface.
6. Instead of relying on the thermocline to keep the hot and cold fluids separate, the use of insulating floaters as separators has been suggested in the literature. A floater would consist of a horizontal disk made of insulating material of an appropriate density that falls between the densities of the hot and cold fluids. Active mechanisms can be used to locate and traverse the floater within the tank and position it to yield the maximum returned energy from the storage tank. Use of a floater, however, will rule out the use of fillers in the tank, simultaneously. NREL believes the floater option is a potentially viable, practical solution to TES that would be worth further investigation.

¹⁶ S. E. Faas, L. R. Thorne, E. A. Fuchsm and N. D. Gilbertsen, "10 MWe Solar Thermal Central Receiver Pilot Plant: Thermal Storage Subsystem Evaluation Final Report." Sandia National Laboratories: June 1986.

Operating Modes

Four TES operating modes are examined in this study.

In Operating Mode 1, Thermal Charging (TC), cold molten salt is removed from the bottom of the tank and heated with solar energy. The heated molten salt is then returned to the top of the tank. There is no electricity generation. In this scenario thermal energy is stored in the tank.

In Operating Mode 2, Thermal Discharging (TD), hot molten salt is removed from the top of the tank and used to generate electricity. The cooled molten salt is then returned to the bottom of the tank. There is no thermal input in this scenario. As a result, thermal energy is being removed from the thermocline tank.

In Operating Mode 3, Thermal Regulation (TR), the system is simultaneously charging and discharging; the molten salt is heated by the solar collector field at the same time salt is being used to generate electricity. If the solar input exceeds the power generation, thermal energy is stored. If the solar input is less than the power generation, there is a net removal of thermal energy from the tank. For the purposes of equipment sizing it is assumed that the system has a solar multiple of 1.8, i.e., at the design point, the solar field generates 1.8 times more thermal energy than the energy required to operate the Rankine cycle at full load. This operating mode does not apply to the indirect parabolic trough cases because the oil-to-salt heat exchangers only allow energy to be transferred in one direction at any given time. Only the direct central receiver design cases, in which the charging and discharging operations are decoupled, reference this operating mode.

Finally, in Operating Mode 4, Stand-by (TS), which occurs nightly or whenever the solar input drops below a certain level, no electricity generation takes place. In this mode a small amount of molten salt is circulated throughout the system to maintain the temperature above the freezing point of the salt. This slowly reduces the energy stored in the tank. If the molten salt in the tank is in danger of dropping below the freezing point, all lines will be drained and the heat tracing will be activated. Note that the lines between the thermocline tank and the surge tanks cannot be drained. If necessary, salt can be drained to the salt melting sump.

Process Description

This section describes the general scope and operation of the TES system. The information is intended to serve as a general description of the process for both design options (trough and central power tower). Please refer to the process flow diagrams (PFDs) in Appendix B and the heat and material balances (H&MBs) in Appendix C while reading through the process description. Plant streams and components marked as outside battery limits (OSBL), such as the solar field and the power block, were considered to be outside the scope of this project. The functions of each component will be explained further in the sections below.

For all of the thermocline systems, cold molten salt is added or removed from the bottom layer of the Thermocline Storage Tank (V-103), and hot molten salt is added or removed from the top layer. Heat tracing protects the lines from freezing in the event of a no-flow scenario.

Indirect Parabolic Trough Design

Operating Mode 1: TC

Cold molten salt is gravity fed from the Thermocline Storage Tank (V-103) to the Cold Molten Salt Surge Tank (V-102) at 293°C (559°F). The gravity feed allows salt to flow in either direction between the tanks, and allows cold molten salt to be added or removed from the Thermocline Storage Tank (V-103) through a single nozzle. Cold molten salt is then pumped from the Cold Molten Salt Surge Tank (V-102) to either the Oil/Molten Salt Exchanger (E-101) or the Thermocline Storage Tank (V-103) using the Cold Molten Salt Pump (P-102).

The Oil/Molten Salt Exchanger (E-101) is a plate heat exchanger that transfers energy between the molten salt and hot circulating oil. The flow rate of molten salt through the exchanger is adjusted by flow control to maintain the outlet salt temperature at 400°C (752°F). Flow control is accomplished by manipulating the variable speed drive of the Cold Molten Salt Pump.

Hot oil is supplied to the heat exchanger by the parabolic trough field (OSBL) to heat the cold molten salt. The cooled oil is then returned to the trough field (OSBL) for reheating. The hot salt is then returned to the Thermocline Storage Tank (V-103).

Operating Mode 2: TD

Hot molten salt is gravity fed from the Thermocline Storage Tank (V-103) to the Hot Molten Salt Surge Tank (V-101) at 400°C (752°F). The surge tank and thermocline liquid levels are maintained near the same elevation because the tanks are connected with an open pipe. The gravity feed allows salt to flow in either direction between the tanks. The hot molten salt is then pumped from the Hot Molten Salt Surge Tank (V-101) to either the Oil/Molten Salt Exchanger (E-101) or the Thermocline Storage Tank (V-103).

In the heat exchanger, the hot molten salt is cooled by the Therminol oil returning from the Power Generation Facility (OSBL). The flow rate of molten salt through the exchanger is adjusted by flow control to maintain the outlet salt temperature at 293°C (559°F). Flow control is accomplished by a variable speed drive set on the Hot Molten Salt Pump (P-101).

Cool molten salt from the Oil/Molten Salt Exchanger (E-101) is returned to the Thermocline Storage Tank (V-103). The oil heated in the heat exchanger is then returned to the Power Generation Facility to produce electricity (OSBL).

Operating Mode 4: TS

In Operating Mode 4, a small amount of molten salt will circulate throughout the TES system using both the hot and cold molten salt pumps. Thermal oil circulating pumps will be maintained at low flow. This is done to maintain the temperatures above the minimum for both the TES system and the HTF oil loop to prevent freezing. If the salt temperature approaches the freezing point, flow will be stopped and all lines will be allowed to drain back to the thermocline tank.

Heat tracing will then be used to maintain the temperatures of both the molten salt and Therminol oil systems.

Direct Central Receiver Design

Operating Mode 1: TC

In this mode cold molten salt is gravity fed from the Thermocline Storage Tank (V-103) to the Cold Molten Salt Surge Tank (V-102) at 293°C (559°F). The gravity feed allows salt to flow in either direction between the tanks, and allows cold molten salt to be added or removed from the Thermocline Storage Tank (V-103) through a single nozzle. Cold molten salt is then pumped from the Cold Molten Salt Surge Tank (V-102) to the Central Power Tower (OSBL) with the Cold Molten Salt Pump (P-101).

In the central power tower, solar energy is used to heat the molten salt from 293°C (559°F) to 560°C (1040°F). This temperature is maintained through flow control of the molten salt. Flow control is accomplished using a variable speed drive on the Cold Molten Salt Pump (P-102). Hot salt from the Central Power Tower (OSBL) is returned to the Thermocline Storage Tank (V-103) by gravity flow.

Operating Mode 2: TD

In this mode hot molten salt is gravity fed from the Thermocline Storage Tank (V-103) to the Hot Molten Salt Surge Tank (V-101) at 560°C (1040°F). The gravity feed allows salt to flow in either direction between the tanks, and allows salt to be added or removed from the Thermocline Storage Tank (V-103) through a single nozzle. Hot molten salt is then pumped from the Hot Molten Salt Surge Tank (V-102) to the Power Generation Facility (OSBL) using the Hot Molten Salt Pump (P-101).

At the Power Generation Facility thermal energy will be removed from the molten salt, cooling it down to the minimum defined TES operating cut-off temperature (See Chapter 5). This temperature is maintained through flow control of the salt. This flow control is accomplished using a variable speed drive on the Hot Molten Salt Pump (P-101). Cold salt from the Power Generation Facility (OSBL) is returned to the Thermocline Storage Tank (V-103) by gravity flow.

Operating Mode 3: TR

In Operating Mode 3 there are two molten salt circulation loops. In the first loop cold salt is removed from the bottom of the Thermocline Storage Tank (V-103) at 293°C (559°F) and pumped to the Central Power Tower (OSBL). The salt is then heated to 560°C (1040°F) and returned to the top of the thermocline tank. This loop is the same as Operating Mode 1, and is described in detail above.

In the second molten salt loop, hot salt is removed from the top of the tank at 560°C (1040°F) and pumped to the Power Generation Facility (OSBL). At the Power Generation Facility the salt is cooled to 293°C (559°F) and returned to the bottom of the thermocline tank. This loop is the same as Operating Mode 2, which is described above.

If the flow rate of molten salt through the receiver is greater than the flow rate of molten salt through the power cycle loop, the thermocline boundary layer in the tank will descend as hot molten salt occupies a larger fraction of the tank. If the flow rate of the power cycle loop is greater than the flow rate through the receiver, the thermocline boundary layer in the tank will ascend. In the event that both flow rates are equal, the thermocline zone will remain at a constant level. Heat tracing is used to ensure that the lines will not freeze in a no-flow condition.

The temperature control of the hot molten salt and the cold molten salt is accomplished by flow control. This is done through the use of variable speed drives on the Hot and Cold Molten Salt Pumps. The flow rate of the cold molten salt is adjusted to provide hot molten salt at 560°C (1040°F). This flow rate varies depending on the available solar thermal energy. The flow rate of the hot molten salt is adjusted to provide cold molten salt at 293°C (559°F). The flow rate of this stream is determined by the required power generation.

Operating Mode 4: TS

In Operating Mode 4 a small amount of molten salt will be circulated throughout the system using both the hot and cold molten salt pumps. This is done to maintain the temperature of the system. If the salt temperature approaches the freezing point, flow will be stopped and all lines will be allowed to drain back to the thermocline tank. Heat tracing will then be used to maintain the temperature of the system.

Preheating the Thermocline

Due to the high temperature of the salt system, it is necessary to preheat the thermocline tank before molten salt is added. The preheating system is designed to raise the temperature of the quartzite bed and the tank shell to 288°C (550°F), which is above the melting point of the salt. This is to both prevent damage to the tank from thermal stress, and to ensure that the salt does not freeze on any cold spots during the initial fill. The preheating system defined consists of a rotary blower and a fired heater. The blower moves air through the heater and raises the temperature to 315°C (599°F). The equipment is sized to heat the tank to 288°C (550°F) over a period of 30 days. This temperature was chosen because it is 50°C above the freezing point of the salt, which should provide adequate safety margin. This time period was chosen to ensure that thermal stresses would not damage the tank or the packing, and also to minimize the size of the temporary equipment. The tank also has electric heat tracing.

Charging the System with Salt

To introduce salt into the system, solid salt will be added to the Salt Melting Sump (V-104). The Salt Melting Sump is a small carbon steel tank with an immersion heater (E-102, Salt Melting Heater) and a Salt Charging Pump (P-103). The system has been sized in all design cases to melt

all of the salt for the system in 30 days. After the thermocline system has begun operation, the equipment will be used as necessary to add any makeup salt to the system. In the event of a system upset, provisions have been included to drain salt to the sump as necessary.

Thermocline Tank Efficiencies

As mentioned above, the thermocline tank design was based on the Purdue methodology¹¹. The tank sizing relative to two-tank systems was determined using the efficiency calculations below. The tank efficiency is defined in Equation 3-3, and is roughly a measure of how much of the tank volume is lost in the tank to maintaining the thermocline boundary. The thermocline tank efficiencies for the 95% threshold (threshold temperatures are 395°C and 547°C for trough and central receiver, respectively) are given in Table 3-9. The efficiencies of the two-tank system are assumed to be 100%. The thermal losses¹⁷ constitute energy lost to the atmosphere through the tank insulation. These losses are the same for both the thermocline tank and an equivalent state-of-the-art two-tank system. They include the thermal energy storage system only, and do not include energy lost in electricity generation.

Table 3-9
Molten Salt Thermocline Storage Tank Estimated Thermal Efficiencies

Design Case	MWh _t	Maximum TES Temperature	Technology	Tank Efficiencies	Thermal Losses
1	100	400°C (752°F)	Indirect Trough	89%	0.40%
2		560°C (1040 °F)	Direct Central Receiver	89%	0.38%
3	500	400°C (752°F)	Indirect Trough	89%	0.12%
4		560°C (1040 °F)	Direct Central Receiver	89%	0.10%
5	1000	400°C (752°F)	Indirect Trough	89%	0.08%
6		560°C(1040 °F)	Direct Central Receiver	89%	0.06%
7	3000	400°C (752°F)	Indirect Trough	88%	0.12%
8		560°C (1040 °F)	Indirect Trough	89%	0.07%

Note: Design tank efficiencies estimated by Black & Veatch were determined to be conservative by roughly 15%. This additional design margin effectively accounts for the lack of uniform distribution, wall effects, as-built void coefficient, filler, and as-built height. The result is an extension of the discharge time by over an hour and increased energy extraction from the thermocline tank.

It can be seen from the definition of tank efficiency that η_{95} is a function of both the Reynolds number and the tank height. In the above design cases the tank height is always set at 40 feet (except for Case 8, where the tank height is 47 feet). This means that the Reynolds number is the only variable in the efficiency calculation. The Reynolds number is defined as:

¹⁷ Yang, Z., Garimella, S.V. Molten-salt Thermal Energy Storage in Thermoclines under Different Environmental Boundary Conditions. Applied Energy (2010), doi:10.1016/j.apenergy.2010.04.024.

$$\text{Re} = \frac{u \cdot d}{\nu_c}$$

Equation 3-9

Where u is the salt flow rate, d is the diameter of the filler, and ν_c is the viscosity of the fluid at the cold inlet temperature.

4

CAPITAL COST ESTIMATES

The cost estimates indicate that the proposed thermocline thermal energy storage system has the potential to significantly reduce TES costs compared to a two-tank storage system. There is a wide cost range that depends largely on the scale of the project and the operating temperature. The cost estimates developed in this study include both direct and indirect costs with a contingency of 15% and sales tax. They are presented in January 2010 dollars. A complete breakdown of the equipment and engineering costs is provided in Appendix G. A summary of the total installed capital costs is provided in Table 4-1.

Table 4-1
Molten Salt Thermocline Storage Capital Costs

Storage Capacity (MWh _t)	Technology	Maximum TES Temperature	Thermocline Cost (\$)	Cost Per Unit Capacity (\$/kWh _t)
100	Indirect Trough	400°C (752°F)	24,613,000	246
	Direct Central Receiver	560°C (1040 °F)	13,184,000	132
500	Indirect Trough	400°C (752°F)	53,146,000	106
	Direct Central Receiver	560°C (1040 °F)	30,503,000	61
1000	Indirect Trough	400°C (752°F)	84,219,000	84
	Direct Central Receiver	560°C (1040 °F)	45,955,000	46
1500	Indirect Trough	400°C (752°F)	104,553,000	70
	Direct Central Receiver	560°C (1040 °F)	66,000,000*	44*
3000	Indirect Trough	400°C (752°F)	215,425,000	72
	Direct Central Receiver	560°C (1040 °F)	110,340,000	37
3500	Indirect Trough	400°C (752°F)	254,333,333*	73*
	Direct Central Receiver	560°C (1040 °F)	118,410,000	34

* Values were interpolated or extrapolated from other design case results.

For comparison, estimates were prepared for an equivalent set of two-tank thermal energy storage systems. These estimates are provided in Table 4-2. Both sets of quotes provided in Table 4-1 and Table 4-2 have an uncertainty of approximately +40%/-20% in accordance with AACE Class 4 estimate. The uncertainty with these estimates can be reduced with additional design or demonstration of the concepts identified in this report. Detailed modeling, design or demonstration of the following items will likely be necessary to confirm the conceptual designs presented here:

- Dynamic modeling of thermocline
- Thermal ratcheting
- Tank materials
- Distributor design
- Other tank internals, including structural supports

Table 4-2
Two-Tank Molten Salt Thermal Storage Capital Costs

Storage Capacity (MWh _t)	Technology	Maximum TES Temperature	Two-Tank Cost (\$)	Cost Per Unit Capacity (\$/kWh _t)
100	Indirect Trough	400°C (752°F)	27,532,000	275
	Direct Central Receiver	560°C (1040 °F)	18,059,000	181
500	Indirect Trough	400°C (752°F)	71,348,000	143
	Direct Central Receiver	560°C (1040 °F)	38,783,000	78
1000	Indirect Trough	400°C (752°F)	115,525,000	116
	Direct Central Receiver	560°C (1040 °F)	57,139,000	57
1500	Indirect Trough	400°C (752°F)	166,500,000*	111*
	Direct Central Receiver	560°C (1040 °F)	82,500,000*	55*
3000	Indirect Trough	400°C (752°F)	284,459,000	95
	Direct Central Receiver	560°C (1040 °F)	150,423,000	50
3500	Indirect Trough	400°C (752°F)	312,991,000*	89*
	Direct Central Receiver	560°C (1040 °F)	175,670,000	50

* Values were interpolated or extrapolated from other design case results

It can be seen from this information that the thermocline system offers the lowest installed capital cost at each design capacity. The average savings for the thermocline system is approximately \$25 per kWh_t, which is an average reduction of 24%. For the largest single-tank designs, the savings were 37% for the indirect trough 1500 MWh_t case and 33% for the direct central receiver 3500 MWh_t case. It can also be seen that the direct TES systems are less expensive than indirect thermal energy storage due to the higher charging temperatures, which minimizes the required storage volume. This information is presented graphically in Figure 4-1.

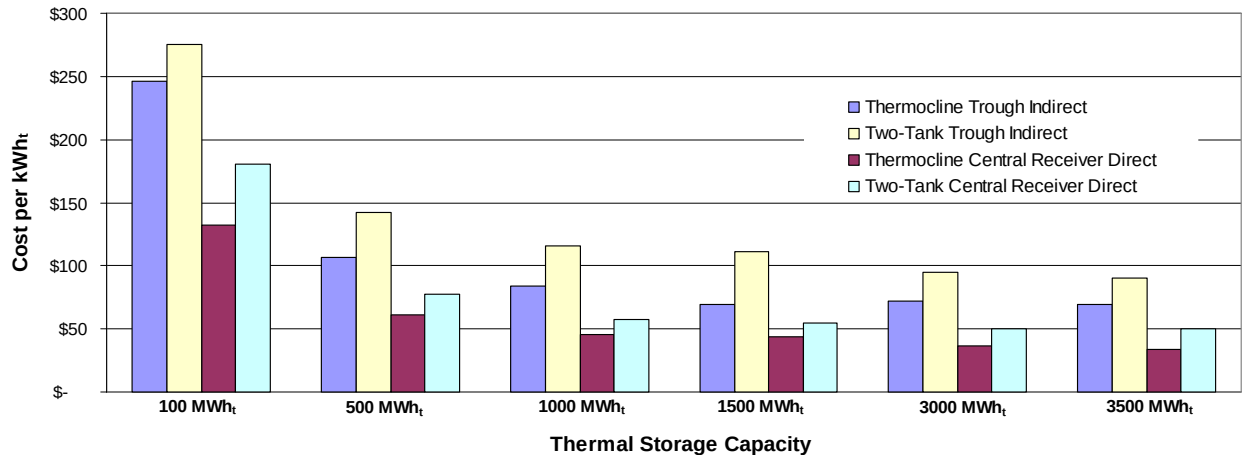


Figure 4-1
Molten Salt Energy Storage Capital Cost Estimates

It can be seen from Figure 4-1 that the direct thermocline central receiver offers the most favorable economics across the range of system sizes, with an installed cost of \$34-\$132/kWh_t. The second most economical option is the direct two-tank central receiver, which has an installed cost of \$50-\$181/kWh_t. The third most economical option is the indirect thermocline parabolic trough design, with a cost of \$70-\$246/kWh_t. The least economical option is the indirect two-tank parabolic trough design, with a price of \$89-\$275/kWh_t.

The main cost advantage for the thermocline system is the substitution of quartzite rock for relatively expensive molten salt. The thermoclines requires roughly half as much salt as the two-tank systems.¹⁸ For Case 7, this amounts to a savings of \$45 million relative to the two-tank design (see Appendix G).

It was also found that direct TES systems are less expensive than indirect systems. This is expected because direct TES does not require oil-to-salt heat exchangers to transfer heat from the working fluid to the storage fluid, and the higher temperature of the direct central receiver system greatly decreases the size of the TES volume required for a given storage capacity. In Case 7 the heat exchangers amount to almost \$48 million, which is 30% of the total direct cost. These heat exchangers are not required in the central receiver design, which can use the molten salt directly without the need for a synthetic oil system. In addition, the cost of a TES system for a given storage capacity depends on the operating temperature of the CSP technology. The size of the storage system is directly proportional to the temperature difference. CSP technologies with a greater differential in the hot storage charging temperature and the cold return temperature will require a smaller volume to store the same amount of energy. It follows that parabolic trough storage operating at 400°C (752°F) requires roughly 2.5 times more storage volume than tower storage operating at a charging temperature of 560°C with a nominal cold return temperature of 290°C for both systems. The cost estimate results show that on average, across

¹⁸ Although the void fraction is 0.25, the salt has about 20% higher volumetric heat capacity than the quartzite filler. Thus more total volume is required to store the same amount of energy in the thermocline than in the hot tank of the two tank system. There is also some volume of the thermocline system that is free of quartzite filler, above the filler in the thermocline and in the surge tanks and interconnecting piping.

the range of sizes, the capital cost for the direct central receiver is 54% of the cost of the indirect parabolic trough.

A significant capital investment for the thermocline designs is the storage tank. The current material selected by EPRI for this tank is 347 stainless steel. However, a detailed material analysis may show that other metals are acceptable for this service. For comparison purposes, cost estimates have been prepared for tanks made of 304 SS, 347 SS, and Inconel 625. This comparison is presented in Appendix I.

Capital cost curves have been produced in Figure 4-2 for the full set of cost data, which show the installed cost per MW_t as a function of storage size. It can be seen from this graph that the installed cost decreases as the capacity of the TES increases. This is expected, as most process equipment benefits from economies of scale. The greatest cost savings occurs between 100 and 500 MWh_t of storage. In this range the capital cost per kWh_t decreases nearly 60%. It is important not to extrapolate the costs beyond the limits of the given curve. Beyond 3500 MWh_t of production it may be necessary to build more thermocline tanks, which would increase the cost per MWh_t . Below 100 MWh_t it may be difficult to purchase equipment, which could drive up costs more significantly than a simple extrapolation may suggest. The curves presented in Figure 4-2 are only a guideline based on a conceptual project, and a detailed cost estimate should be performed for actual project planning purposes.

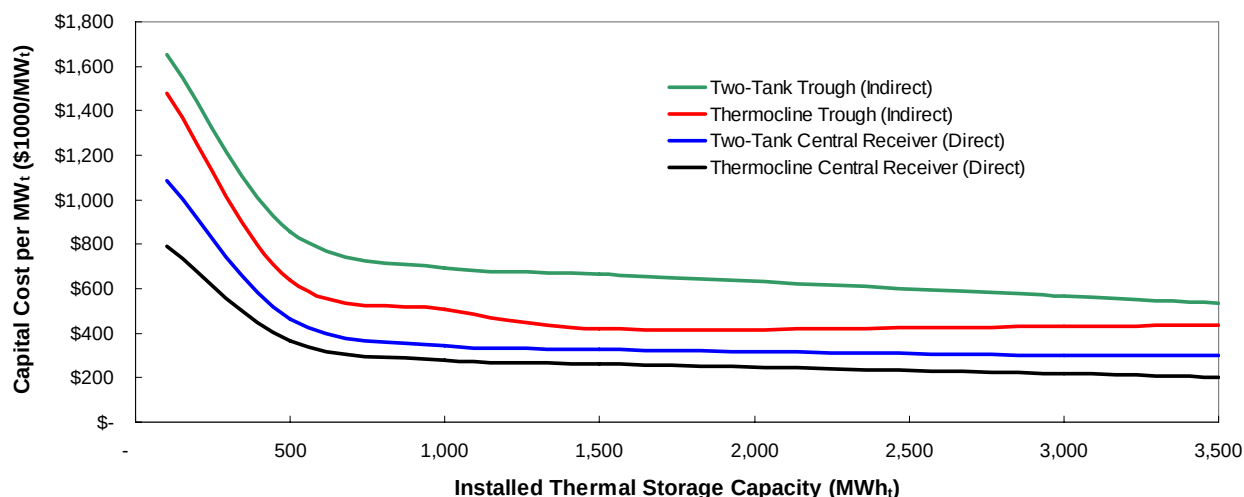


Figure 4-2
Molten Salt Thermal Energy Storage Capital Cost Estimates as a Function of Installed Capacity, $\$/kWh_t$

5

PERFORMANCE ANALYSIS

In conjunction with the structural design study conducted by Black & Veatch, the performance of a thermocline TES system was analyzed by Sandia National Laboratories, the National Renewable Energy Laboratory (NREL), and Purdue University. These analyses investigated the annual performance of a thermocline system compared to a two-tank system, the thermal performance of the thermocline with regards to thermal gradients and natural convection during dwell periods, and the distributor manifold performance.

Annual Performance

The analysis performed by Sandia compared the annual performance of a thermocline system with that of a two-tank TES design for a parabolic trough power plant. The two-tank system represents today's baseline technology as demonstrated at the 50 MWe Andasol plant now operating in Spain. With a solar multiple of ~ 2 , the excess collector field energy at Andasol is stored within a two-tank molten-salt system for later use by the steam turbine. A simplified schematic and some key design parameters¹⁹ for this two-tank system are shown in Figure 5-1 and Table 5-1.

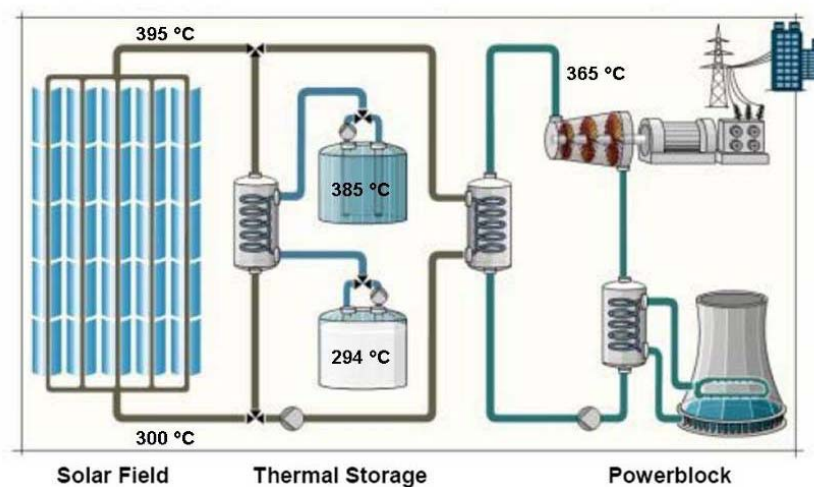


Figure 5-1
Schematic of Andasol-Type Parabolic Trough Plant

¹⁹ Relloso, Sergio and Yolanda Gutierrez, "Real Application of Molten Salt Thermal Storage to Obtain High Capacity Factors in Parabolic Trough Plants," SENER, SolarPACES 2008, Las Vegas, NV

Table 5-1
Key Design Parameters for Andasol-Type Parabolic Trough Plant

Turbine Capacity (gross)	50 MWe
Turbine Efficiency (Full Power)	37%
Storage Thermal Rating	1010 MWh _t , 7.4 hrs
Storage Tank Size*, H x D	2 tanks, 14 m x 37 m each
Storage Fluid	Solar salt (60% NaNO ₃ , 40% KNO ₃)
Storage Heat Exchanger	16,000 m ²
Solar Collector Field Aperture Area	510,000 m ²
Solar Collector Assembly, W x L	5.77 m x 148 m
Solar Collector Row Spacing	16.2 m
Solar Collector Fluid (HTF)	Therminol VP-1

*It was observed that the Black & Veatch tank design volume for 1000 MWh_t is approximately 13% lower than the reported volume of the Andasol system. This may be a result of several factors, such as slightly lower capacity (1000 MWh_t vs. 1010 MWh_t), tank heel level, head space, turbine efficiency used to determine hours of storage, etc.

TRNSYS 16 was used to simulate the annual performance of this two-tank plant. A combination of standard-TRNSYS components, STEC-TRNSYS²⁰ components, and some new system-control components were used to create the model. The logic model is depicted in Figure 5-2 and Figure 5-3 and the important model parameters are listed in Table 5-1 and Table 5-2.

²⁰ STEC components can be downloaded from <http://sel.me.wisc.edu/trnsys/trnlib/stec/stec.htm>

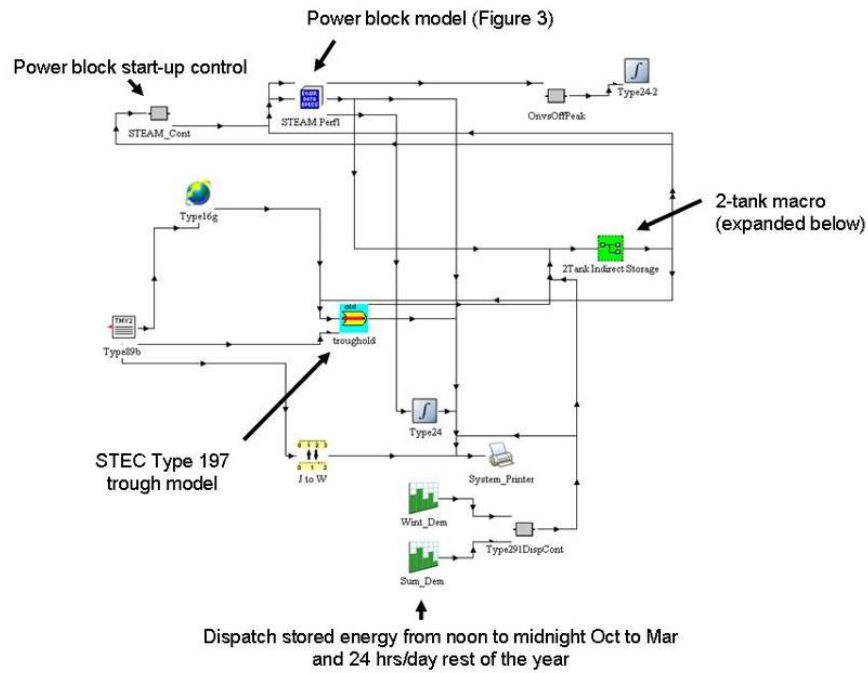


Figure 5-2
TRNSYS Model of Andasol-Type Power Plant

2-Tank storage control algorithm determines HTF flow split between storage and Rankine cycle

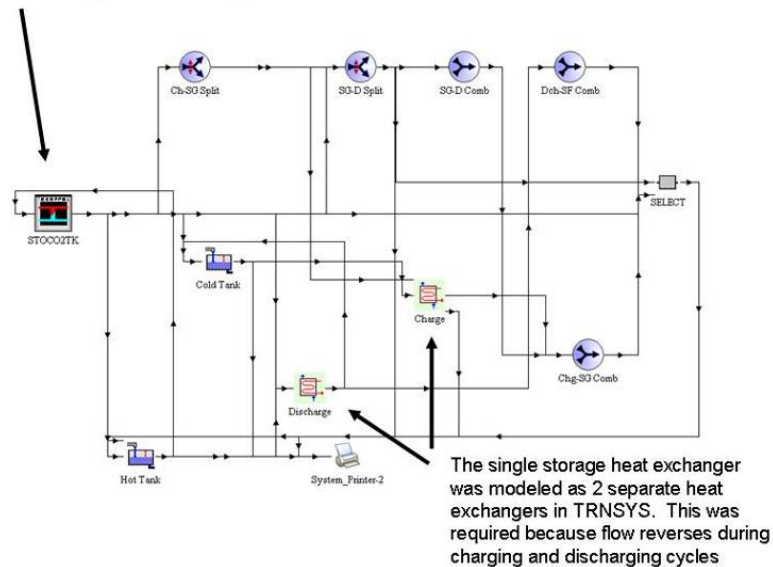


Figure 5-3
Expanded TRNSYS Two-Tank Macro

Table 5-2
Additional Model Parameters for TRNSYS

Solar collector optical efficiency	80%
Heat collection element thermal losses	Based on original LUZ cermet coating
Solar field cleanliness	95%
Storage heat exchanger transfer coefficient	1200 W/m ² -°C
Turbine startup time	15 to 30 minutes
Minimum HTF temperature to Steam Generator	300°C (570°F)

Rather than developing a detailed model of all components in the steam-Rankine power block, the system was represented by two transfer functions. The inputs to the functions were HTF flow rate and exit temperature, and the outputs were HTF return temperature and turbine-generator power output. The transfer functions depicted in Figure 5-4 and Figure 5-5 were developed by scaling up from 30 to 50 MW_e a similar empirical model developed for the SEGS VI power block²¹.

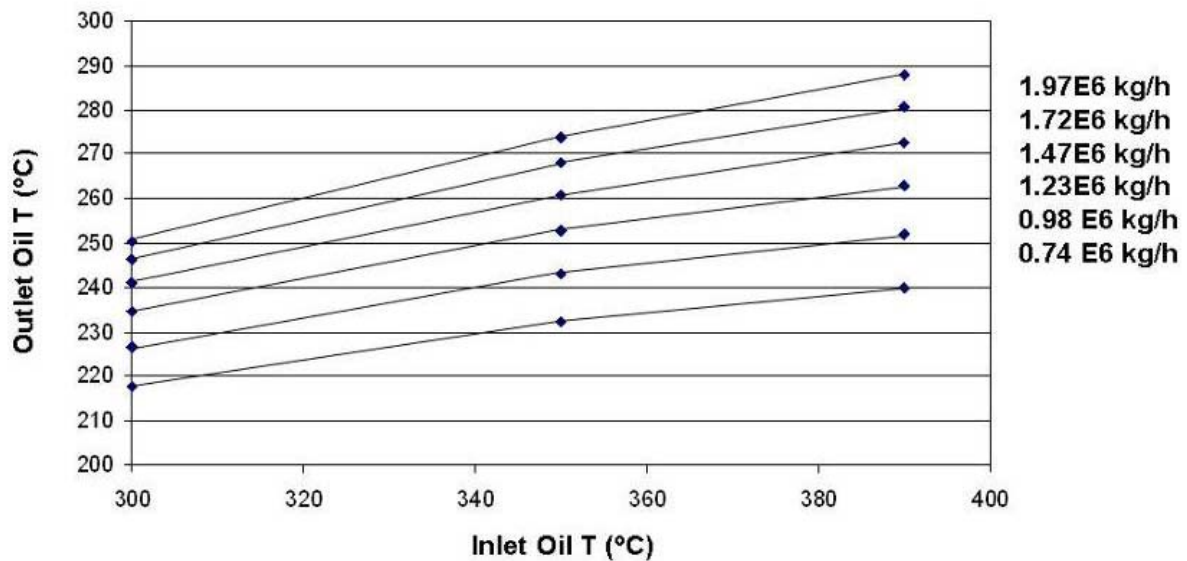


Figure 5-4
Empirical Model of a 50 MWe Steam-Rankine Power Block – HTF Return Temperature

²¹ Angela Patnode, Simulation and Performance Evaluation of Parabolic Trough Solar Power Plants, Masters Thesis, University of Wisconsin, 2006.

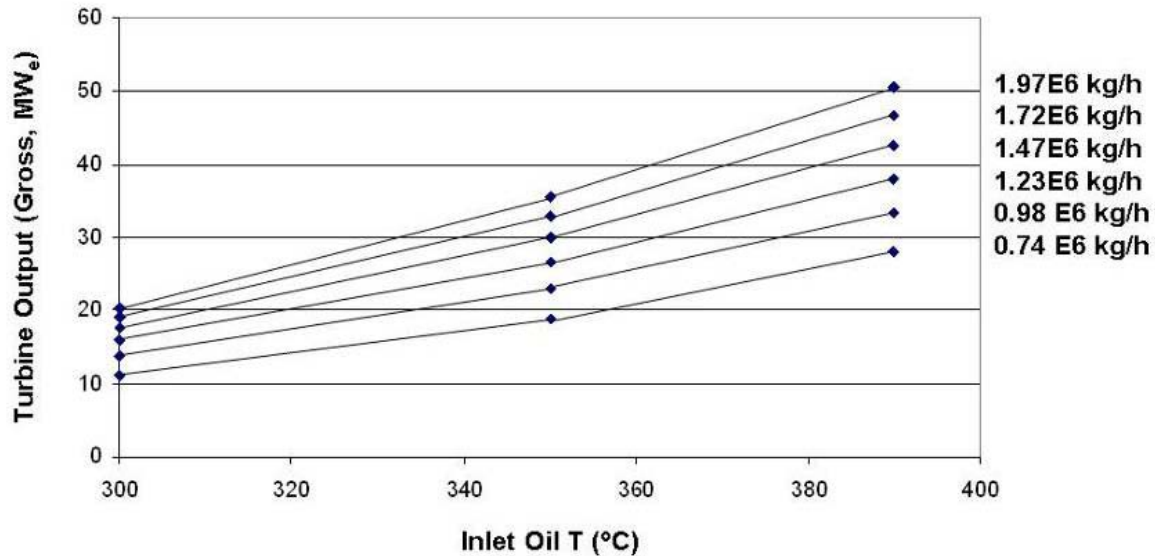


Figure 5-5
Empirical Model of a 50 MWe Steam-Rankine Power Block – Turbine Generator Output

The minimum oil-inlet temperature to the steam generator is shown in Figure 5-5 to be 300°C (570°F). This is based on actual experience at SEGS VI; near noontime in the winter, the field outlet temperature is dropped from the design value of 390°C (734°F) to ~300°C (570°F) and the Rankine cycle temperature and pressure is dropped to maintain 50°C of superheat (called sliding pressure operation). The Kramer Junction plant staff and Sandia’s evaluation have found that this operating approach maximizes the overall solar-to-electric efficiency.²² Establishing the minimum oil operating temperature is very important to evaluation of thermocline storage, as described later.

The two-tank TRNSYS model was run using a three-minute time step and the hourly TMY file for Tucumcari, New Mexico. This weather file was chosen because the annual DNI is similar to southern Spain, i.e., 2.3 MWh/m²-yr. The model predicted an annual gross electricity output of 152 GWh. This value is similar to values predicted for Andasol prior to plant startup²³, indicating that the model’s results are reasonable.

The next step in the analysis was to replace the two-tank model with a thermocline model. The thermocline chosen for this study is Case 5 defined by Black and Veatch, in Chapter 3: a 1000 MWh_t thermocline with a height of 12.2 m (40 ft) and a diameter of 46.3 m (152 ft). A simplified schematic is shown in Figure 5-6.

²² Lippke, Frank, *Simulation of the Part-Load Behavior of a 30 MWe SEGS Plant*, SAND95-1293, June 1995. This report also describes the basis for the STEC Type 197 trough model.

²³ Geyer, et. al., “Dispatchable Solar Electricity for Summerly Peak Loads from the Solar Thermal Projects Andasol 1 Y Andasol 2 (sic),” Solar Millennium, SolarPACES 2006, Seville, Spain. The paper quotes 3589 full load hours with 12% from fossil. Thus, solar-only performance = 0.88*49.9 MWe*3589 = 157.6 GWh.

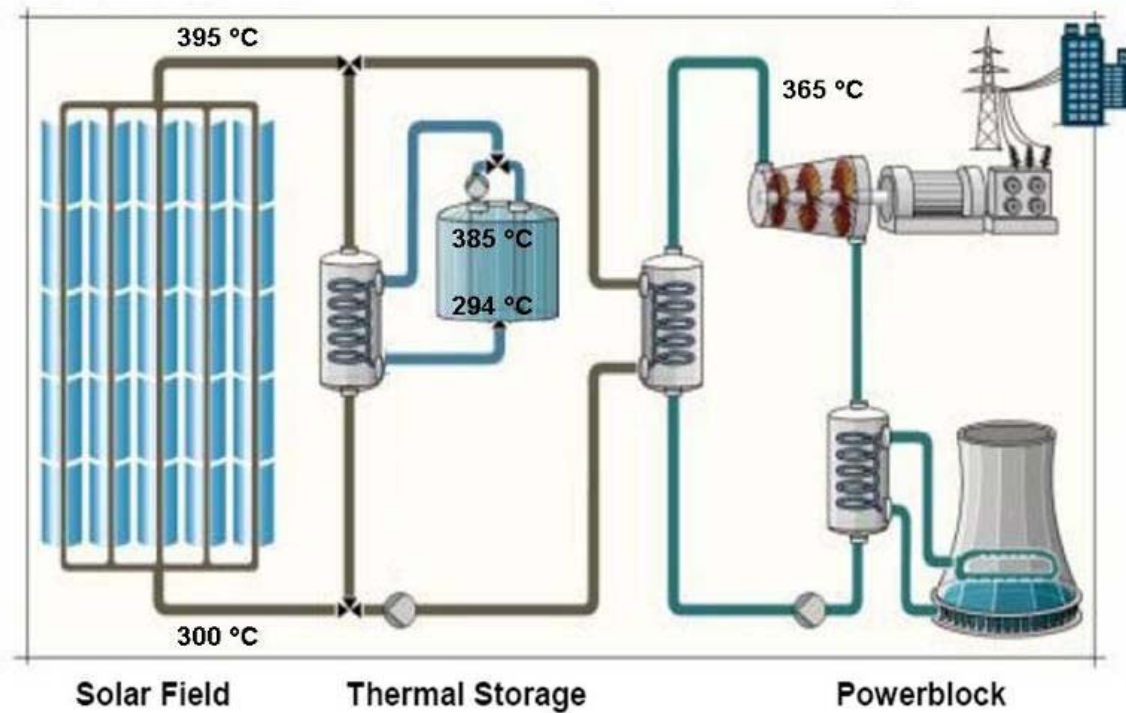


Figure 5-6
Andasol-Type Plant with 1000 MWh Thermocline Storage System

The TRNSYS logic model for this plant is depicted in Figure 5-7. Only the storage portion of the model is shown since the remainder is the same as the two-tank model. The void fraction of the tank is assumed to be 24% based on experience at the Solar One thermocline tank. This is comparable to the 25% void fraction assumed in the Black & Veatch design. The density of the rock is 1940 kg/m^3 and the specific heat is 880 J/kg-C . The basis of the thermocline tank model was the STEC Type 502, which was developed for SolarPACES a few years ago and is a variation of the standard TRNSYS Type 10 component. The tank is divided into several equally-sized control volumes (23 stacked cylinders used here) and a first-order differential equation describes the energy balance of each cylinder.

Thermocline storage control algorithm determines HTF flow split between storage and Rankine cycle

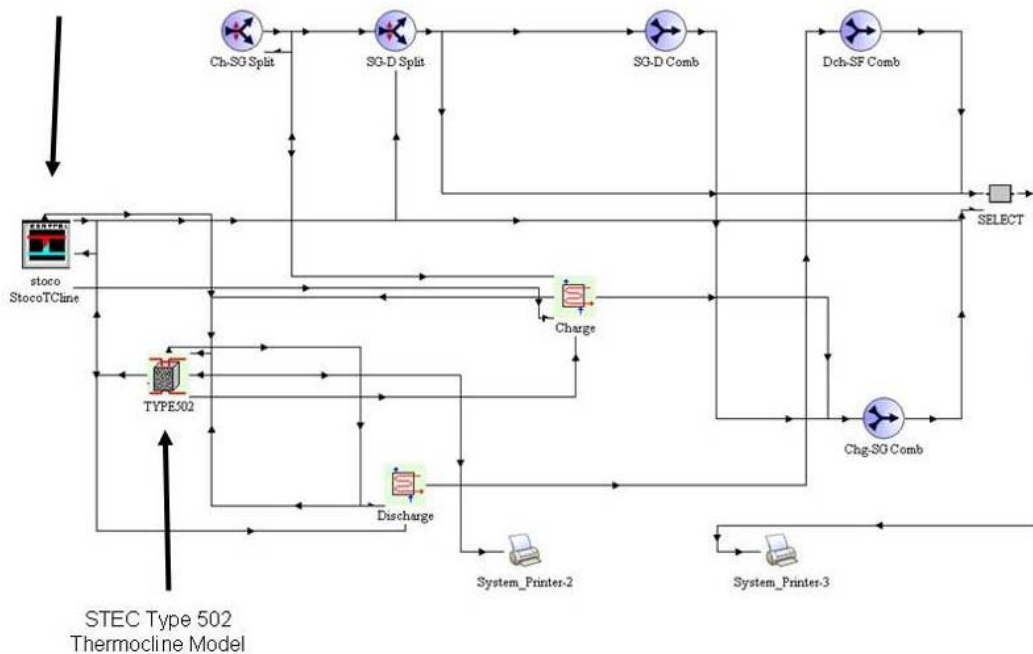


Figure 5-7
TRNSYS Model of Thermocline Storage System

The thermocline TRNSYS model was run using the same time step and the hourly TMY file as for the two-tank system. The model predicted an annual gross electricity output of 153 GWh. This value is virtually identical to the two-tank plant, which suggests that it should be possible to develop a thermocline system that achieves the same overall power plant performance as a two-tank system.

Previous investigators of thermoclines have assumed that thermocline exit temperatures must remain high to produce useful electricity. As discussed previously, the SEGS VI power block is designed to operate up to 90°C below the design value, i.e., 300°C (570°F) vs. 390°C (734°F). Thermocline plants would not perform as well as a two-tank plant if the degradation in temperature was not allowed. To estimate the effect, the TRNSYS models were rerun assuming a minimum temperature of 340°C (644°F) and 360°C (680°F). The results of this investigation are shown in Table 5-3.

Table 5-3
Comparison of Minimum Temperature Limitation Results

Min Oil T to Power Block	2-Tank Plant Electricity	Thermocline Plant Electricity
300°C	152 GWh	149 GWh
340°C	149 GWh	135 GWh
360°C	146 GWh	125 GWh

Thermal Performance

The thermal performance of the thermocline systems was evaluated by both NREL and Purdue University. While the solid fill used within a thermocline tank helps to lower the overall cost of the TES by reducing the amount of relatively expensive molten salt required, the presence of solids in the tank can cause the temperature gradients at the hot/cold salt interface to spread out more than in the absence of solids. This spreading reduces the effective capacity of the tank. To assess its behavior and the overall thermal cycling efficiency of the TES, it is necessary to model the thermal/hydraulic mechanisms that occur in the thermocline tank.

NREL Analysis

The thermocline system analyzed by NREL was modeled in two separate steps by first modeling the core of the tank and then analyzing its wall regions. The core region has adiabatic boundary conditions, whereas the wall region's external wall loses heat to the environment.

The core region model should be applicable to tanks of all sizes (diameters). As per Black & Veatch, the largest tank feasible for TES is estimated to be about 15.2 m (50 ft) in height and about 48.8 m (160 ft) in diameter. This size represents a volumetric capacity of about 28,400 m³ (over 7 million gallons). Based on their assumed temperature difference of 107°C, such a tank represents a storage capacity of about 1500 MWh_t or a power generation capacity of about 85 MW_e over a 6-hour period.

The NREL model used solid fillers shaped in a structurally regular manner to eliminate lateral structural loads to the side walls of the tank. The selected structural solid fills were in the form of hexagonal rods or a honeycomb-like structure that spans the entire height of the tank (Note: this design was not consistent with the Black & Veatch design analysis; capital costs were based on a quartzite rock/sand mixture). The void fraction, denoted by α , is filled by the molten salt.

Table 5-4 summarizes the key assumptions made for the numerical model.

Table 5-4
Modeling Assumptions

Quantity	Value	Units	Remarks
Hot salt temperature	385	(°C)	
Cold salt temperature	300	(°C)	
Tank void fraction	α	(---)	Variable (between 0 and 1)
Tank height	14	(m)	
Tank discharge time	6*3600	(s)	
Molten salt velocity	$0.648/\alpha$	(mm/s)	Varies with tank void fraction

For a nominal 600 MW_t tank, Figure 5-8 shows the variation of the required tank volume, equivalent density of the fluid/filler mixture and the fluid residence time, as functions of the tank void fraction, α . When $\alpha=1$, the tank contains fluid only, and when $\alpha=0$, it is all solid.

Due to the higher density of the solid fill, the bulk density of the mixture increases with decreasing α . The solid also has a lower specific heat capacity than the fluid. Therefore, increasing the amount of solid increases the required tank volume.

Also plotted in this figure is the fluid residence or breakthrough time, which represents the time to replace the tank fluid once. At a residence time of 6 hours the tank is completely filled with only fluid ($\alpha=1$). As more solid is introduced, the void fraction decreases, and the fluid is forced to move through the tank more rapidly, resulting in decreased residence times. For example, the residence time for the fluid is only 1.5 hours when the tank has a void fraction of 0.25. These results hold true for all size tanks. Increased thermal capacity for the tank will increase its volume proportionately.

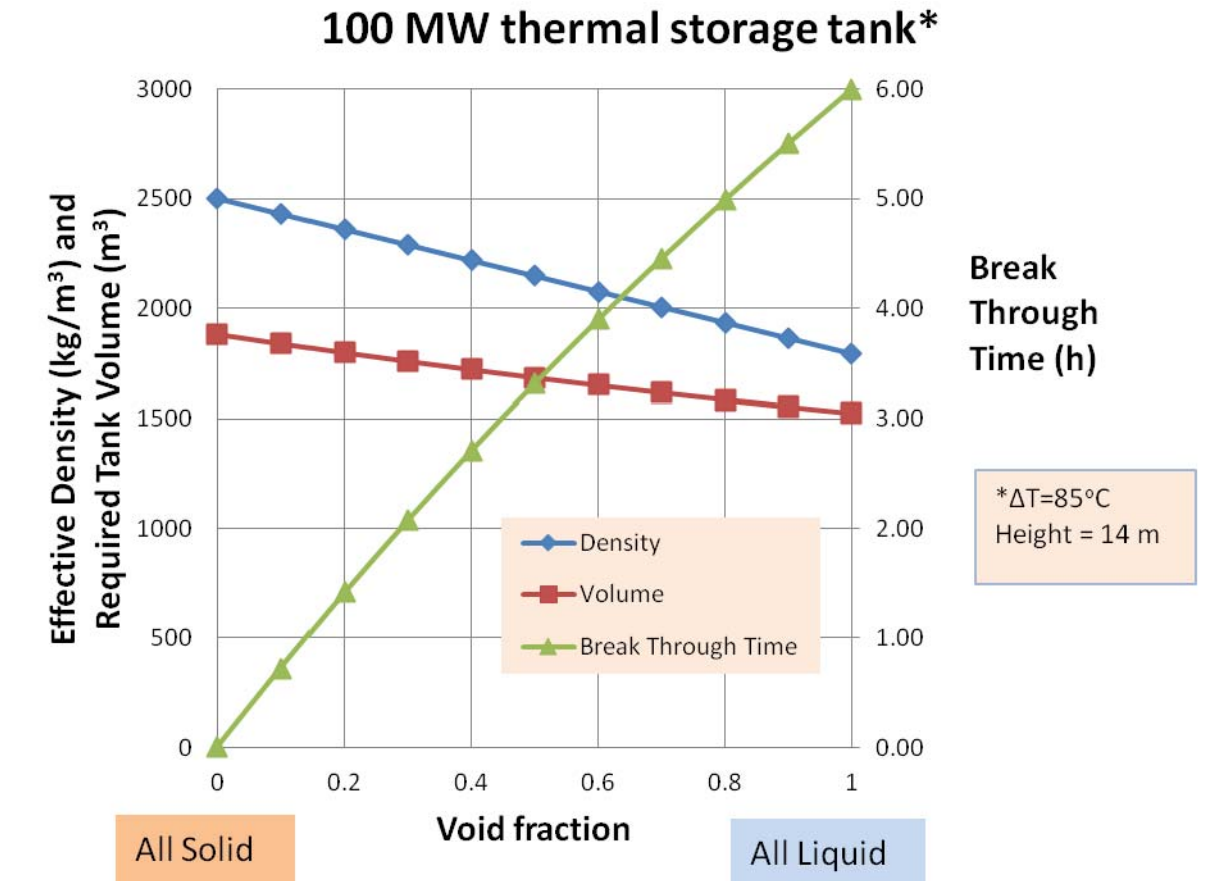


Figure 5-8
Variation in Key Tank Parameters with Tank Void Fraction

Core Model

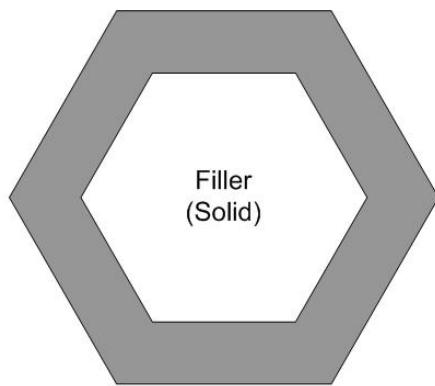
The thermocline behavior was first modeled in the core of the tank. The filler was assumed to be made of vertical rods of hexagonal cross-section. This arrangement was selected because it was a relatively easy geometry to model computationally. The side-to-side distance for the hexagon was selected to be 1 inch (25.4 mm). The thickness of the fluid region can be changed to obtain various void fractions for the simulations (see Figure 5-9).

The cross section of the filler is projected along a z-axis to form the simulated solid volume. The overall length of the model was set at 14 m (46 ft), matching the maximum liquid height originally determined by Black & Veatch.

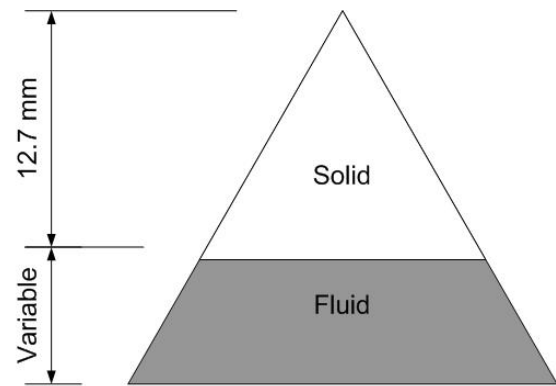
Molten salt and solid properties as used in the simulation are listed in Table 5-5. The properties of the solid are representative of likely solid fill materials such as reinforced concrete or silica fire bricks. The fluid volume was varied to obtain a range of void fraction values in the simulations.

Table 5-5
Summary of Molten Salt and Filler Material Properties Used in the Simulation

Property	Fluid	Solid	Units
Density	1794	2500	kg/m ³
Specific heat	1549	900	J/kg-K
Thermal conductivity	0.5365	2	W/m-K
Viscosity	0.0021	---	Pa-s
Thermal diffusivity (*10 ⁶)	19.31	88.9	m ² /s



Filler surrounded by Molten Salt
(Fluid)



A sixty-degree segment
showing fluid thickness as
a variable.

Filler (Structured) and Fluid Cross section
(as modeled in CFD)

Figure 5-9
Cross Sectional View of Filler and Fluid Volumes

Based on the geometry and the low velocities for the salt, the entire flow field is laminar. The flow is assumed to be uniform across the entire cross-section of the TES tank.

The results of the simulation are shown in Figure 5-10. This figure plots the tank outlet temperature as a function of time as it is emptied of hot fluid. The cold fluid is introduced at the bottom of the tank at the inlet. The inlet velocity for the illustrated results (with a void fraction value of 0.25) is 2.593 mm/s. This velocity is four times greater than what would occur without the filler.

Two temperatures are plotted as a function of time over 8 hours. The outlet temperature of the fluid exiting the thermocline tank is indicated using red squares for the simulation. An average solid filler temperature is indicated using blue diamonds. The elapsed time is indicated on the x-axis in minutes. All temperatures are normalized using the temperature difference between the maximum and minimum temperatures, as indicated below:

The temperature of the outlet fluid, τ_{fo} , is normalized as:

$$\tau_{fo} = \frac{T_{fo} - T_{min}}{T_{max} - T_{min}} \quad \text{Equation 5-1}$$

and the solid average temperature is normalized as:

$$\tau_{sa} = \frac{T_{sa} - T_{min}}{T_{max} - T_{min}} \quad \text{Equation 5-2}$$

where, T_{sa} is the average solid temperature. T_{max} and T_{min} are 385°C (725°F) and 300°C (570°F), respectively, as listed in Table 5-4.

From this figure, it can be seen that the average solid temperature decreases mostly linearly, except at the tail end when the entire solid has cooled to the incoming fluid temperature. The fluid outlet temperature remains constant for up to 4.5 hours and then begins to drop off gradually.

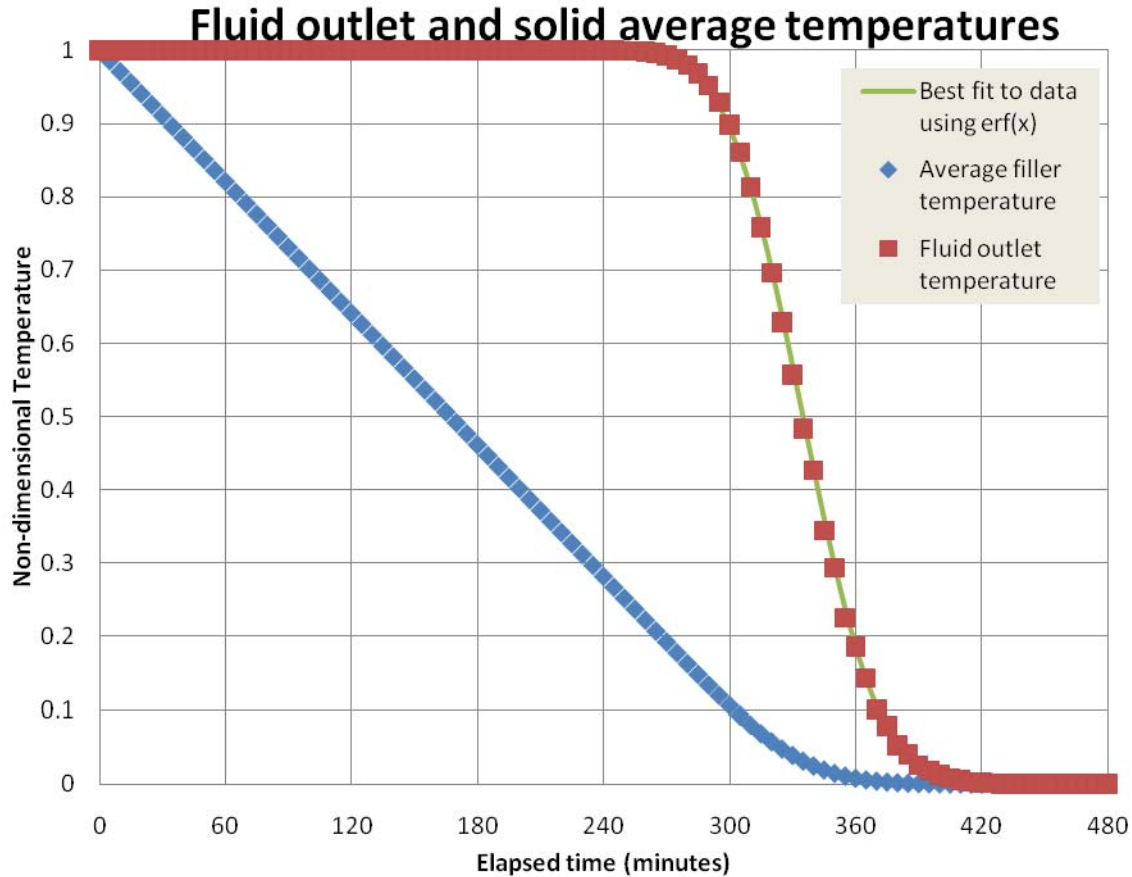


Figure 5-10
Variation of Fluid Outlet and Solid Average (Non-dimensional) Temperatures as Functions of Elapsed Time (Minutes)

The drop-off in the outlet temperature is the result of the spread of the thermocline interface between hot and cold fluids. In simple one-dimensional conduction, partial differential equations for thermal diffusion lead to a temperature distribution that is fully represented by an error function²⁴.

The approximation for the error function follows the form:

$$\text{erf}(x) = \text{sqrt}\left\{1 - \exp\left(-\frac{4x^2}{\pi}\right)\right\}$$

Equation 5-3

²⁴ The error function, normally denoted as $\text{erf}(x)$ is defined as $\text{erf}(x) = \int_0^x e^{-x^2} dx$. In this case, for ease of evaluation, this function was approximated as: $\text{erf}(x) = \text{sqrt}\left\{1 - \exp\left(-\frac{4x^2}{\pi}\right)\right\}$. See Oona's approximation provided by S. Kalkaja. This approximation differs from its actual value by less than 0.8%.

The fluid outlet temperature is expressed as:

$$\tau_{fo} = \{1 - \text{erf}(\delta)\} \quad \text{Equation 5-4}$$

where δ is a non-dimensional time, based on the time for breakthrough, defined as:

$$\delta = \frac{t - t_r - t_{dy}}{t_s} \quad \delta = (t - t_r - t_{dy}) / t_s \quad \text{Equation 5-5}$$

Here, three time parameters are indicated, and all times are represented in seconds.

t_r is the residence time for the fluid, or the tank height divided by the actual fluid vertical absolute velocity.

t_{dy} is a delay caused by the heat transfer that occurs between the solid and the liquid.

t_s represents the inverse of the slope of the decline in temperature versus time, which indicates the spread of the thermocline region.

Using these parameters, a fit for the variation of fluid outlet temperature versus time was found and is shown in the figure by the green line. For this fit, the “time” parameters were found to be:

$$t_r = 5400 \text{ s}; t_{dy} = 14700 \text{ s}; \text{ and } t_s = 2400 \text{ s}.$$

This fit was developed for this particular case, where the liquid height is 14 m (46 ft) with solid fill resulting in a void fraction of 0.25. The solid fillers assumed a characteristic dimension of 2.54 cm (1 inch). This dimension was chosen such that the fillers would have some structural stability and rigidity while maintaining the size small enough that the heat transfer occurs relatively quickly. Smaller diameter fillers would react more quickly, resulting in smaller spreading of the thermocline. The tank is filled with hot fluid initially and the cold fluid is introduced from the bottom.

Although this curve fit determines the overall tank effectiveness of large thermocline tanks for indirect trough applications, the normalized temperature means that it also should hold for other system applications, such as direct molten salt troughs and for higher temperature central receiver systems.

Wall Model

The behavior of the thermocline along the outer wall of the tank was modeled next. In contrast to the core of the tank, the flow and heat transfer that occur next to the outer wall of the tank is dominated by natural convection in the fluid. The wall flow influence extends about 0.5 m (1.5 ft) into the tank from the wall. This distance is small compared to the overall diameter of the tank, which ranges from 14 m to 50 m (46 ft to 164 ft). Therefore, the flow and heat transfer in this region can be handled as a two-dimensional (2-D) flow.

The flow field was modeled using a 2-D flow in a region that is 0.5-m (1.5-ft) wide and 14-m (46-ft) tall. For this study, the region is considered void of any solids. The fluid domain, shown in Figure 5-11, contains a right-side wall on which a heat loss of 100 W/m^2 is imposed. This loss is estimated to range from 50 to 150 W/m^2 , depending upon the type of insulation used and the potential presence of flow diverters inside the tank. The left boundary condition of the model is set at a uniform temperature of 300°C (570°F) for this simulation.

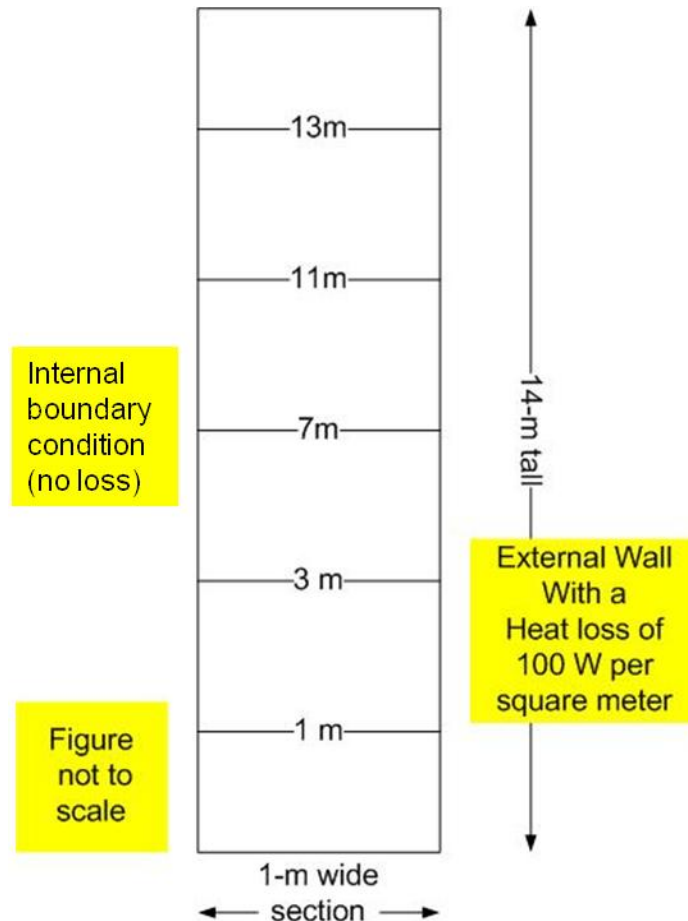


Figure 5-11
Two-Dimensional Representation of Wall Flow and Heat Transfer in the Simulated Domain

Five horizontal lines at vertical heights of 0, 1, 7, 13, and 14 m are also indicated in this figure. The lines at 0 and 14 m represent the bottom and top of the domain (tank), respectively. In the following figures these lines represent the vertical profiles of temperatures and velocities at various heights in the tank.

Figure 5-12 illustrates the temperature profiles in the tank. The temperature falls adjacent to the external (right) wall and increases adjacent to the left boundary condition. The drop in temperature next to the external wall is about 3°C to 4°C at the imposed heat flux of 100 W/m^2 . This decline is confined to a fairly small thermal boundary layer region of about 5 cm.

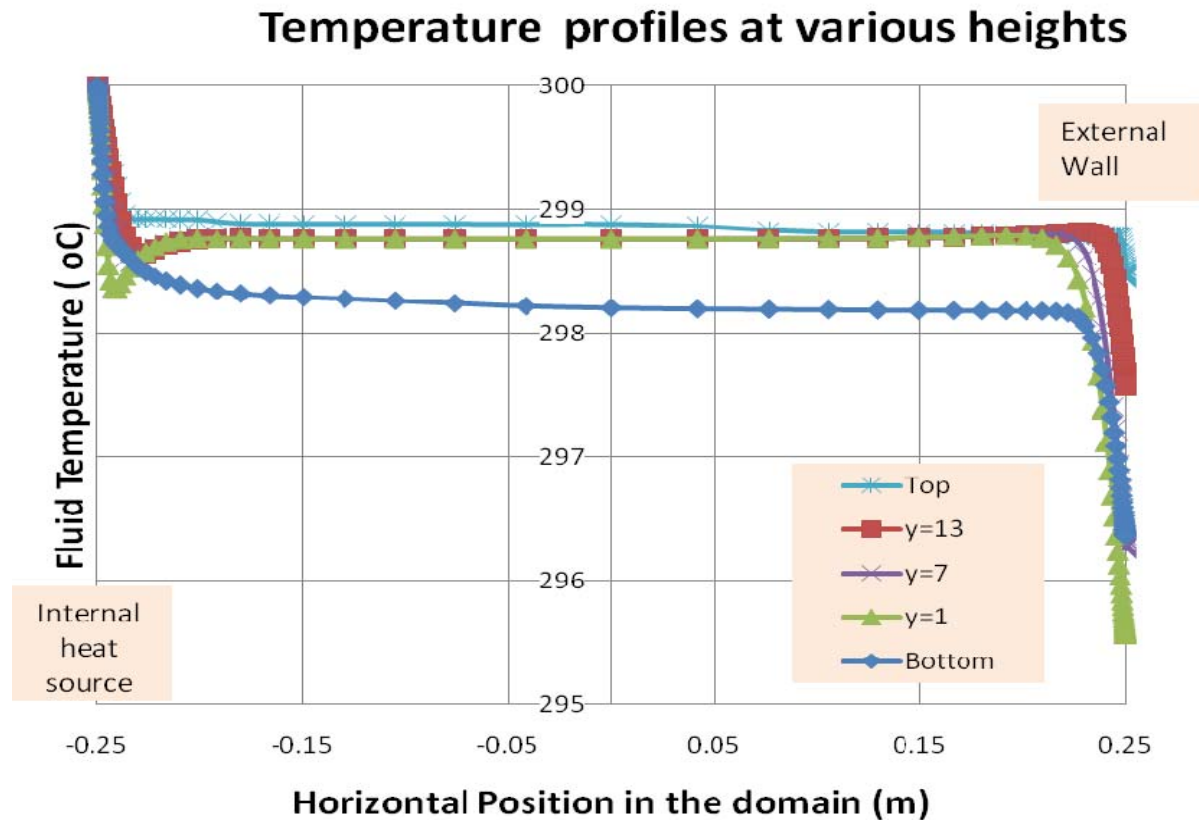


Figure 5-12
Temperature Profiles at Varied Heights with 100 W/m^2 Heat Flux on External Wall and 300°C Constant Temperature on Internal Wall

Figure 5-13 shows the vertical velocity profiles at various heights. These profiles look similar at the three heights indicated. The velocity goes to zero at the no-slip internal boundary condition and external wall. Adjacent to the outer wall at right, the colder fluid is sinking with gravity at a nominal maximum velocity of about -25 mm/s . This profile extends horizontally for about 30 cm (1 ft) into the tank. Adjacent to the internal boundary condition, the hot fluid rises, reaching a maximum velocity of about 70 mm/s . Note that only the wall profiles adjacent to the outer wall are of interest. However, to simulate steady state, a heat source on the left boundary condition is necessary to provide heat to the modeled volume.

These profiles were generated with no fillers adjacent to the wall. Any type of filler material or other flow diverters will reduce the circulation velocity induced in the fluid.

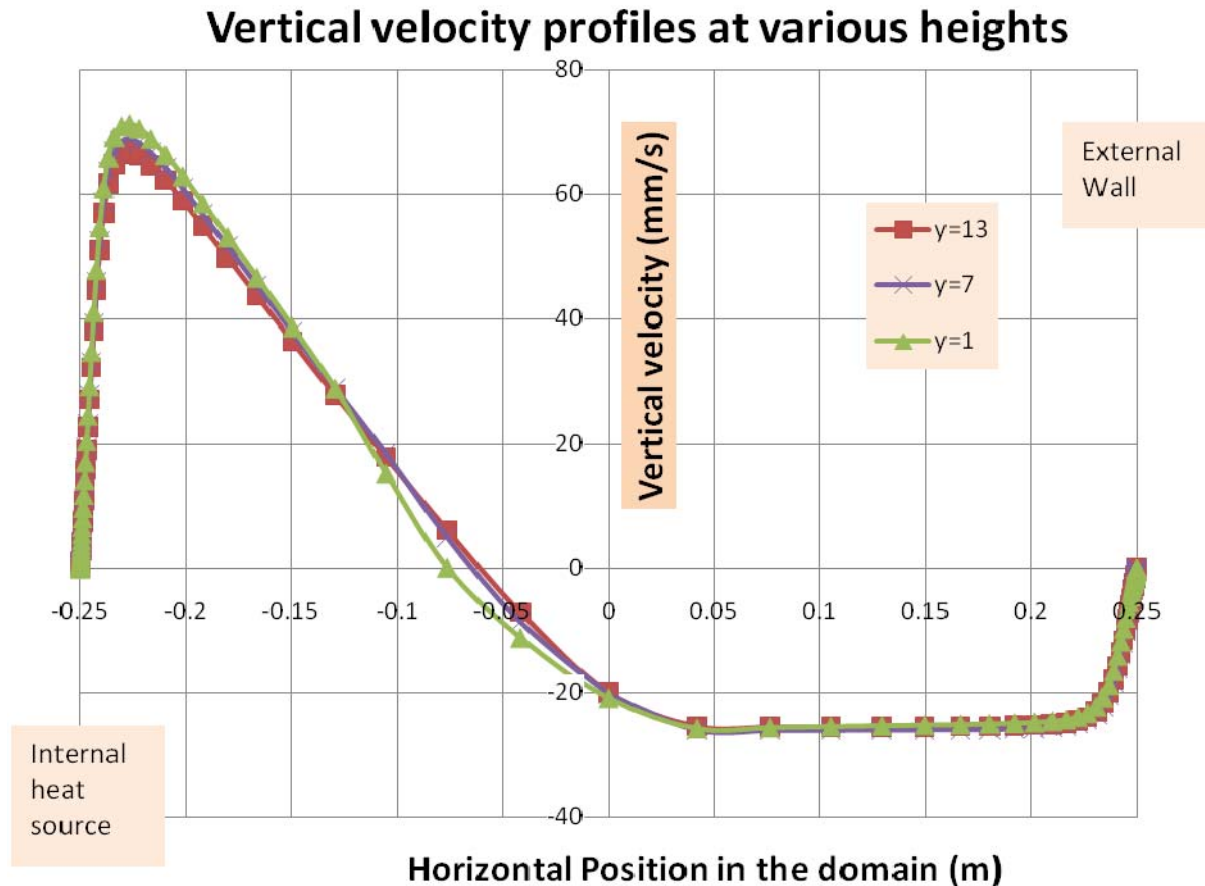


Figure 5-13
Variations of Vertical Velocity with Distance from Walls at Various Heights in the Tank

NREL Conclusions

Over the limited number of simulations conducted, the thermocline system inside the tank behaves reasonably well and as expected. The filler material used tends to spread the temperature gradients more than for the case with no fillers. Thus, the filler effectively decreases the usable tank height if the outlet temperature is limited to a value slightly less than the hottest available temperature. However, as indicated in Sandia's annual performance analysis, it may be feasible for a power block to operate with outlet temperatures as low as 90°C below design temperature.

For use of multiple tanks, operation can occur with tanks operating in parallel or in series. Parallel operation is captured well with these simulations. In parallel tanks, all tanks are emptied simultaneously over the discharge time and behave in a similar manner. In the case of series operation, one advantage that results is that the thermocline would be present in only one tank. The tanks also empty faster. Such operation might be advantageous to tailor power production to the times of high demand as well. How the thermocline propagates from one tank to another depends on the piping and the overall hydraulic system design. The thermocline is likely to spread in height as it passes through transitions from one tank to the next. Further investigations are needed to assess the advantages and disadvantages of parallel and series operations.

The NREL results are limited in scope to a single tank undergoing the first cycle of discharge over a period of 6 hours. The results show that the tank outlet temperature can begin to show a decline as early as 1.5 hours before the expected time of 6 hours (without diffusion at the thermocline interface). The presence of fillers promotes the diffusion and spread of the thermocline farther than for the case without fillers, reducing the effective thermal storage capacity of the tank. The filler may reduce the quantity of salt needed, but may also increase the cost of the tank.

Depending on how the plant is operated, the hot fluid may remain stagnant in the TES system over many hours before being discharged. In such cases, the thermocline can be expected to increase in width in proportion to the square root of the elapsed time. Longer discharge times will also increase the spread of the thermocline. Many cycles of charge/discharge would have to be simulated to arrive at the effective long-term behavior of the tank and its effective capacity.

Purdue Analysis

As a follow-on to the analysis performed by NREL, Purdue University performed additional analysis using computational fluid dynamics (CFD) to investigate the performance of the discharge flow as well as the susceptibility to natural convection during dwell conditions and over multiple cycles.

Tank Discharge Performance

For Purdue's analysis of the thermocline tank discharge, the Case 5 design from Black & Veatch's analysis was rendered and meshed with GAMBIT (Figure 5-14) and exported to the FLUENT CFD solver. To manage computational requirements, an axisymmetric boundary condition was assumed along the tank centerline, which is a valid assumption as the transport equations contain no circumferential dependence. For the discharge half-cycle, a uniform velocity inflow was assumed at the tank bottom. Inside the filler bed, flow momentum was computed according to Darcy's Law with the Brinkman-Forchheimer extension. To track the thermal response of both the fluid and solid, co-located temperature storage was enacted throughout the domain mesh.

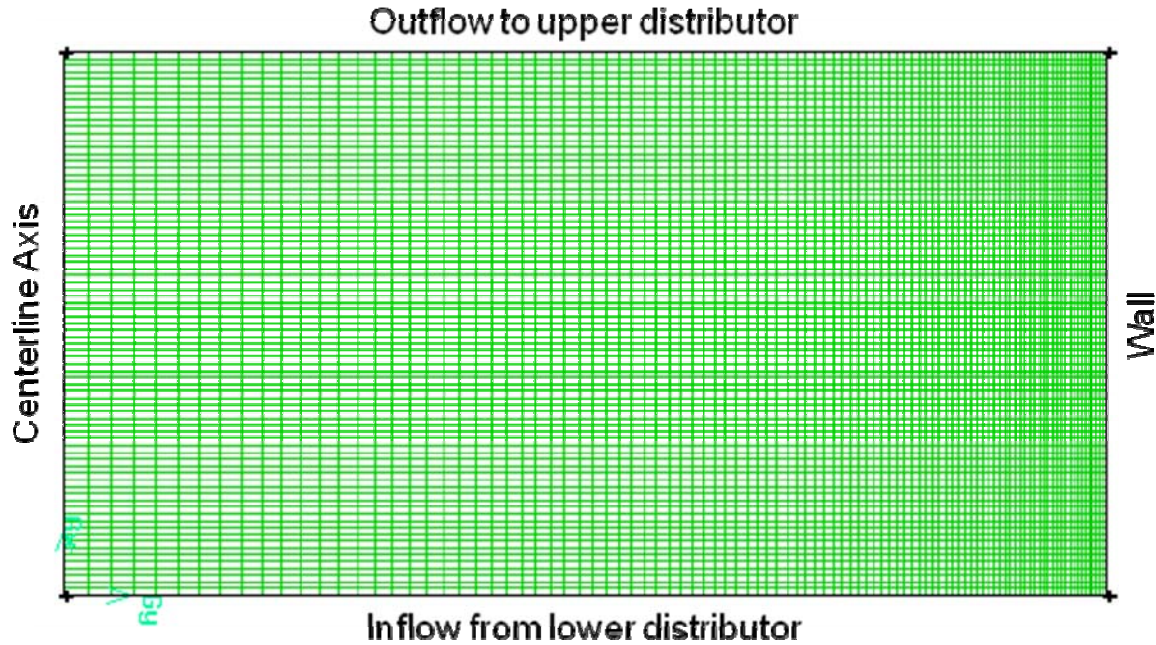


Figure 5-14
GAMBIT Mesh for Case 5 Tank Dimensions: Discharge Half Cycle

To meet the desired discharge energy of 1000 MWh_t in six hours for Case 5, the inflow velocity (equal to the superficial velocity in the fillerbed) is determined according to:

$$u_{in} = \frac{A_d}{A_t} \frac{E}{\rho_{1c} C_{p,1} (T_h - T_c) \Delta t} \quad \text{Equation 5-6}$$

The distributor area (A_d) is defined as the sum of 4187 distribution points in the manifold design for Case 5. For Hitec molten salt²⁵ operating between 287-400°C (549-752°F), the required inflow velocity is 0.299 mm/s. This temperature difference closely matches the design conditions evaluated by Black & Veatch, and is slightly different than the NREL assumption of 300-385°C (572-725°F). In each case, the fillerbed was considered to be composed of particles with a mean diameter of 1 inch (0.0254 m) and a void fraction of 0.22. The solid density and specific heat are defined as 2201 kg/m³ and 964 J/kg-K, respectively. For reference, Hitec molten salt is composed of 53% potassium nitrate, 40% sodium nitrite, and 7% sodium nitrate. Compared to the binary mixture at 300°C assumed by Black & Veatch, Hitec has 4% greater specific heat, 14% lower density, and 3% lower viscosity.²⁶ Despite these small property differences, the overall trends for tank discharge behavior should be similar between the Hitec and binary mixture.

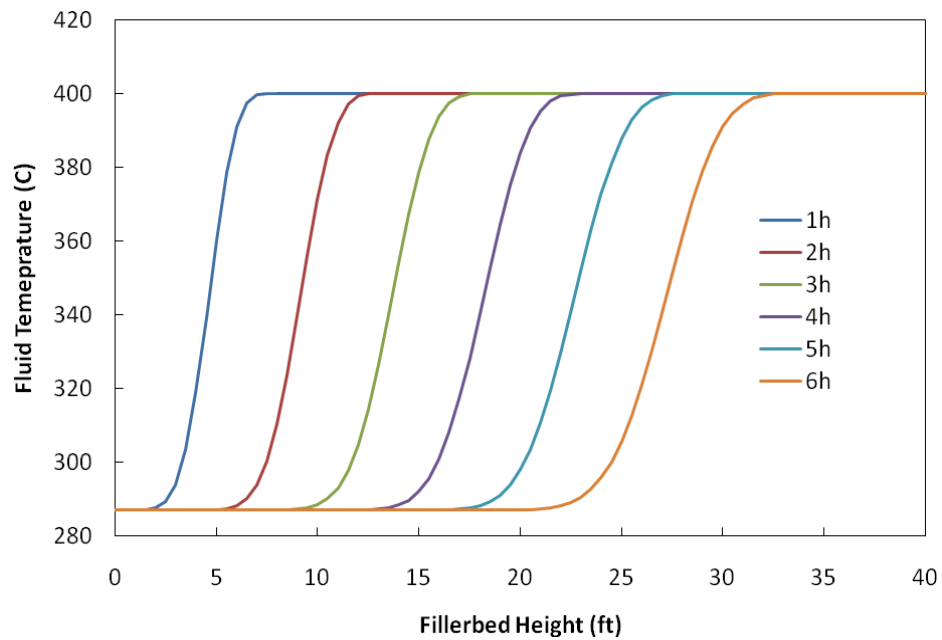
To investigate the effects of external losses, two separate boundary conditions were applied at the tank wall – adiabatic, and a fixed heat loss of 100 W/m², as stipulated in NREL’s analysis.

²⁵ HITEC Heat Transfer Salt, Costal Chemical Co., L.L.C., Brenntag Company, www.coastalchem.com

²⁶ Kearney *et al.*, JSSE, vol. 135, 2003.

Prior to discharge, the tank domain is initialized to 400°C (752°F). Cold fluid enters the lower boundary at the defined inflow velocity to simulate the discharge process. To track the progression of the heat exchange zone through the filler bed, the fluid temperature profiles at the wall were plotted in Figure 5-15 for both wall boundary conditions considered. To improve understanding of the temperature loss, the axial centerline temperature profiles were also plotted for the non-adiabatic case Figure 5-15(b). Also provided in Figure 5-16 is a contour plot of the fluid temperature at the end of the discharge half cycle for the 100 W/m² wall condition.

As expected, the adiabatic tank maintained a fluid outflow at 400°C throughout the discharge half cycle. In contrast, the tank with a 100 W/m² heat loss at the wall experienced some temperature decay in both the hot and cold regions outside of the heat exchange zone. It should be noted, however, that this decay only exists very close to the tank wall. Beyond a distance of two feet from the wall, the fillerbed behaves similar to the adiabatic condition. Also, the decay is limited to a few degrees of temperature drop within the six-hour discharge period. As the minimum required fluid temperature for steam generation was defined as 364°C (687°F) in the Design Requirements Table (see Appendix A), the non-adiabatic tank appears viable for the desired operation.



(a) Adiabatic

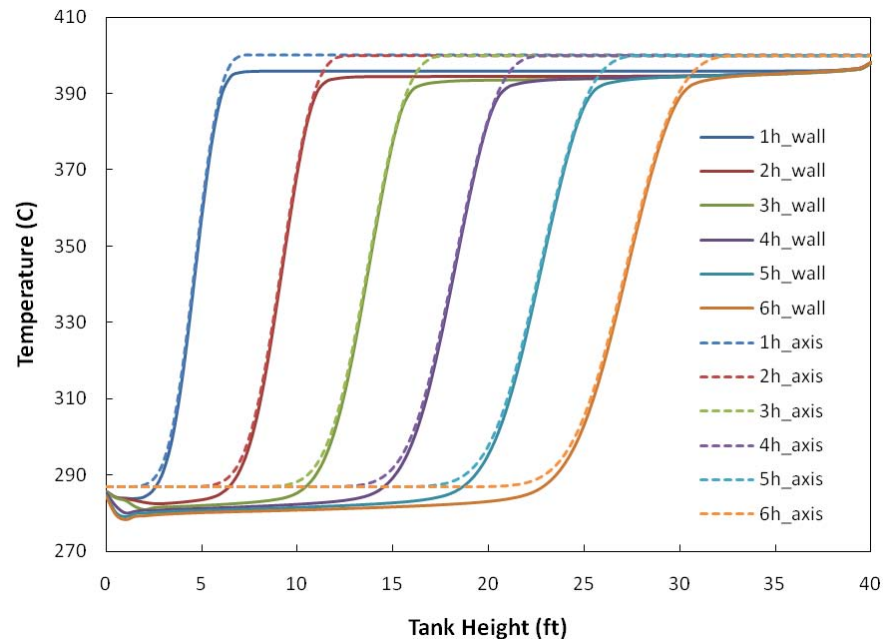
(b) 100 W/m² Heat Loss

Figure 5-15
Fluid Temperature Profiles During a Discharge Half-Cycle for (a) an Adiabatic Wall Boundary, and (b) a Heat Loss Boundary Condition.

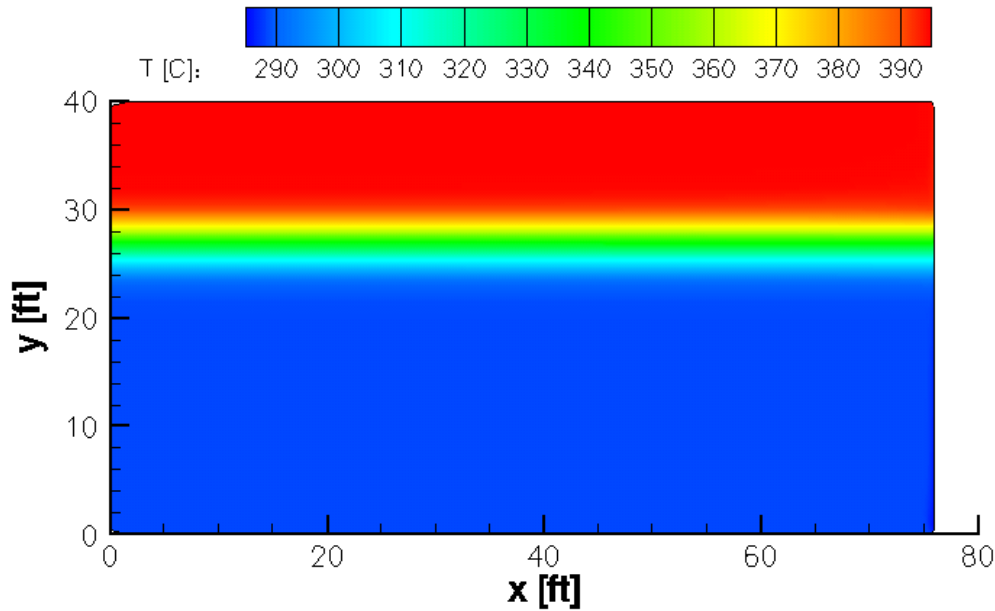


Figure 5-16
Fluid Temperature Distribution after Six Hours of Discharge for 100 W/m² Heat Loss at the Tank Wall

Some simplifying assumptions, such as the inclusion of monodisperse solid and uniform packing structure, were employed in the modeling results above. In addition, the discharge process has thus far incorporated the presence of the tank wall as two different boundary conditions (adiabatic and fixed heat flux) along the fillerbed exterior. In reality, this surface would be coupled to the tank wall behavior, which is itself driven by external convection to the surroundings. In the following discussion, more realistic tank operation is simulated by relaxing these assumptions.

The preferred composition of a thermocline fillerbed is a mixture of rock and sand-scale quartzite, which improves the overall packing structure as well as the thermal exchange behavior at the solid interface. However, the use of sand creates potential for distributor clogging or entrainment during periods of tank operation (charge and discharge processes). These unwanted phenomena are avoided through the inclusion of rock stratification within the fillerbed. Near the two distributor manifolds on either end, the fillerbed is composed only of the rock-scale quartzite. Sandwiched between these two thin layers is the mixture of rock and sand. With this modified structure, the rock layers isolate the sand and minimize contact with the distributor manifolds.

To enforce this stratified structure within the CFD simulation, the tank domain is modified to include three different porous media regions for the rock and mixture, respectively, with distinct properties in each region. Given the dimensions for the Case 5 tank (152 ft diameter, 40 ft height), the pure rock zones are defined at the lower and upper three feet of fillerbed while the mixture is defined at the remaining interior. The equivalent diameters of the rock and sand are

defined as 1.905 cm and 0.2 cm, with a 2:1 ratio of rock to sand in the mixture region²⁷. A Sauter mean diameter of 0.5 cm is calculated for the mixture to serve as an equivalent diameter for the region. The void fractions of the rock and mixture regions are assumed to be 0.4 and 0.22, respectively.

With known geometry and void fraction, the permeability of both regions is determined as²⁸:

$$K = \frac{d^2 \varepsilon^3}{175(1-d)^3} \quad \text{Equation 5-7}$$

Due to its larger particle size and void fraction, the permeability of the rock is greater by a factor of 147.5, a two orders of magnitude difference. The increased permeability of the rock layer generates the potential for natural convection currents to develop along the tank wall due to losses to the surroundings. These currents disrupt flow uniformity as well as the thermocline region itself by developing radial temperature gradients. To inhibit this behavior, the lower-permeability mixture region is extended along the wall surface to provide a buffer from the rock layers. These extensions encompass a two-foot radial distance along the entire tank wall (Figure 5-17).

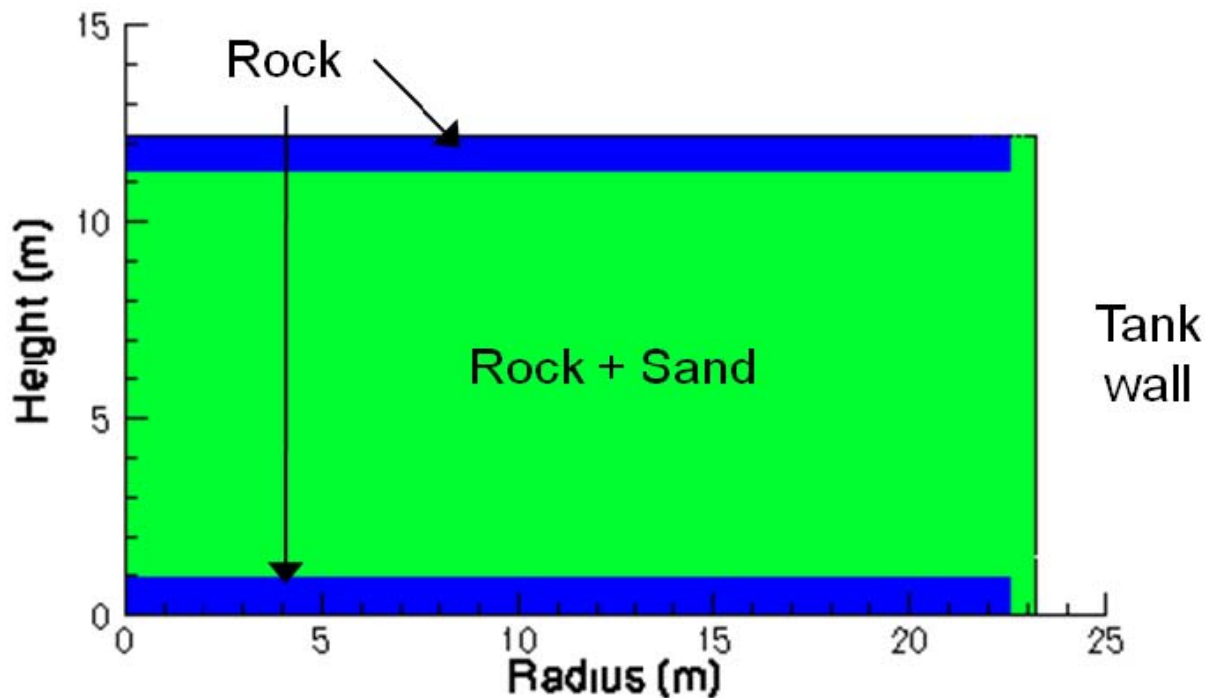


Figure 5-17
CFD Domain for Simulation of Thermocline Tank Operation

²⁷ Pacheco, J.E., Showalter, S.K., Kolb, W.J., 2002. Thermocline thermal storage system for parabolic trough plants. *J. Sol. Energy Eng.* 124, 153-159.

²⁸ Beckermann, C., and Viskanta, R., 1988, "Natural convection solid/liquid phase change in porous media," *Int. J. Heat Mass Transfer*, 31, pp. 35-46.

The simplified modeling work already reported investigated the tank discharge process for both adiabatic and fixed heat flux boundary conditions along the fillerbed exterior surface. A tank wall region was not included in either case. In reality, the thermal response of the fillerbed to external losses is coupled to the response of the tank wall itself. To include this coupling, a two inch thick tank wall was added to the model domain. The thermal transport properties of this wall are taken from known data for AISI 347 stainless steel, as listed in Table 5-6.

Table 5-6
Thermal Transport Properties of AISI 347 Stainless Steel²⁹

Property	Units	Value
Thermal Conductivity	W/m-K	20
Density	kg/m ³	8000
Specific Heat	J/kg-K	500

To simulate the external losses, a convection boundary condition is defined along the external tank surface with a fixed convection coefficient of 10 W/m²-K and an ambient temperature of 27°C (81°F).

For simulation of the discharge cycle, the entire domain was initialized to the temperature of the hot operation limit of the molten salt (400°C). Corresponding to the specified discharge power, an inlet velocity of 0.299 mm/s at the cold operation limit (287°C) was imposed at the tank bottom to simulate the cold fluid entering the tank. To resolve temperature and flow phenomena in the mixture extensions at the top and bottom edges, the domain mesh was regenerated with successive refinements along the tank wall. As with the previous analysis, the implicit time step interval remained fixed at 1.44 seconds.

It should be noted that the temperature initialization described included the tank wall. As the thermal response of the wall (as well as the adjacent filler material) was driven both by tank operation and external convection, thermal equilibrium with the surroundings could not be assumed during the first discharge cycle. Multiple discharge and charge cycles were therefore modeled in succession until the domain developed a periodic response with time. For the charge process, the same inlet velocity was applied at the top of the fillerbed for the hot molten salt temperature. As no dwell time was enforced between the half cycles, the start of the discharge process corresponds to the end of the previous charge process. As a consequence, the temperature data associated with time zero of a discharge process were actually prior to the flow reversal associated with cycle transition in the thermocline tank.

Periodic conditions were achieved after three full cycles of discharging and charging. The molten salt temperature data during the fourth subsequent discharge cycle at both the inside wall and the centerline are plotted in Figure 5-18.

²⁹ Online materials information resource, <http://www.matweb.com>

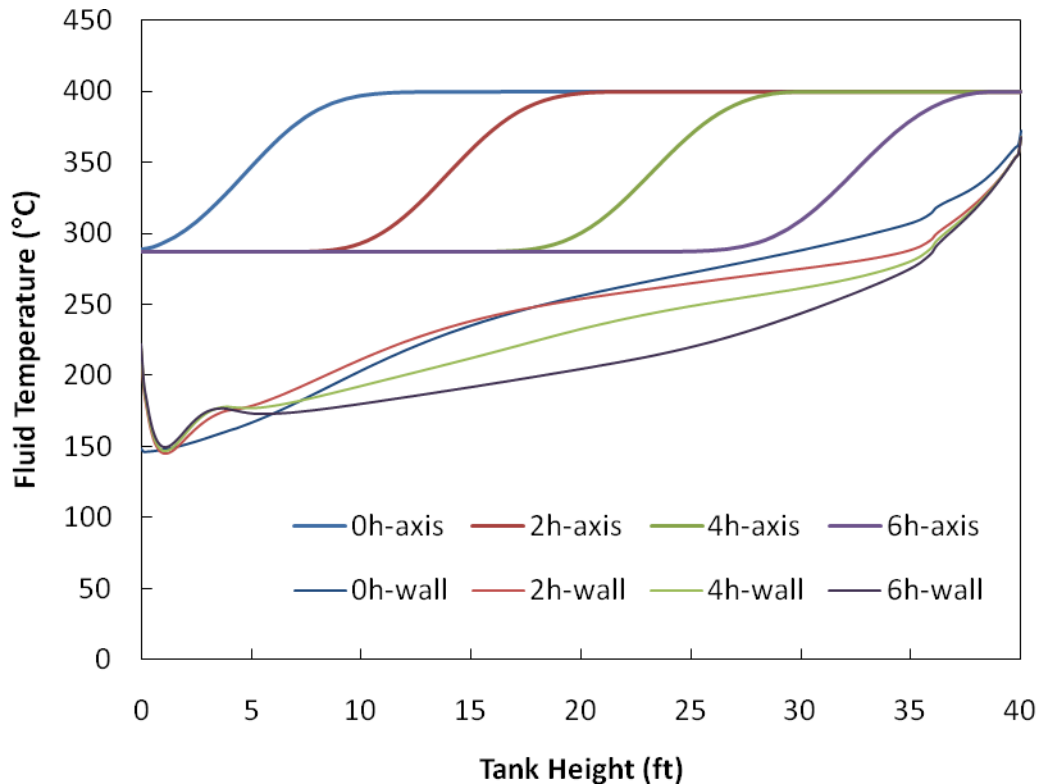


Figure 5-18
Molten Salt Temperature Profiles during the Discharge Process

Results are shown for four different times during the discharge cycle (zero, two, four, and six hours). Due to the large diameter of the thermocline tank, the influence of the external convection losses does not span the entire domain. Thus the centerline profiles remained within the hot and cold operation limits of the molten salt, as was the case with the previous adiabatic model. Due to the periodic nature of the charge-discharge cycles, it can be seen that the bottom of the fillerbed remained below the hot temperature limit at time zero of the discharge process.

Unlike the behavior along the centerline, the convection losses lead to significant cooling of the molten salt near the tank wall. As seen in Figure 5-18 above, the overall trend still shows an increasing temperature with height from 150 to 350°C (302 to 662°F), consistent with the imposed temperature gradient across the fillerbed throughout the six-hour discharge. Apart from this trend, some spatial oscillations were present along the wall at the tank bottom. The negative temperature gradient present at zero height (at the left end of the x-axis) is attributed to the cooling of the tank wall below the cold operation fluid temperature. Thus the entering cold fluid was warmer than the adjacent wall and led to additional cooling with height. The zero-time profile does not show such a negative temperature gradient, as it precedes flow reversal for discharge. The negative gradient is only seen at the later times.

The temperature fluctuation in space at the lower heights is attributed to flow disruptions in the sand-rock mixture extensions along the wall. While these were included with the intention of diminishing natural convection currents, the extensions also created vertical interfaces between the rock and mixture layers. Due to the large disparity in permeability between the rock and mixture (two orders of magnitude as previously reported), the fluid prefers to travel through the more conductive rock layer. At the vertical intersection of the different media, radial velocities developed to allow transport of the fluid from the mixture region to the rock layer. As a result, a swirl flow developed on the rock side of the intersection while fluid flow in the mixture-extension region decayed to stagnant conditions. The temperature and velocity field in the lower mixture-extension region after six hours of discharge are provided in Figure 5-19.

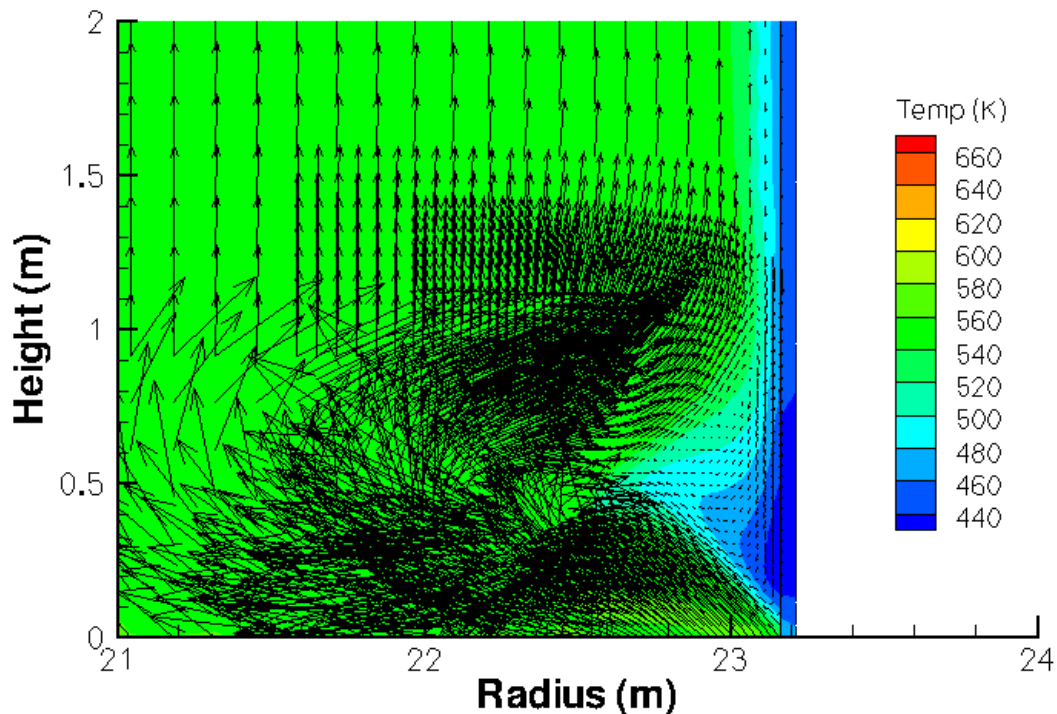


Figure 5-19
Temperature and Velocity Vectors along the Tank Wall in the Lower Mixture-Extension Region

These radial velocities generate large vortices as well as flow stagnation within the lower extension. This stagnant region prevents heating of the fluid by advective transport; this region thus experiences further cooling due to external wall heat loss. It is noted that the vortices exhibit turbulent characteristics and may not have been fully resolved despite the localized mesh refinement. As the fluid velocity and temperature are coupled, this may also lead to some discrepancies in the temperature results; however, the effects of this very limited and small region are expected to be negligible, and the mesh size was chosen to limit computation time.

To illustrate the influence of the external losses and mixture and rock layers on the tank operation, the discharge temperature and axial velocity profiles along the top of the fillerbed

after six hours of discharge are plotted in Figure 5-20. Near the tank centerline, the discharge temperature and velocity match the design conditions. As expected, outflow through the mixture-extension region was minimal due to its lower permeability. Outflow through the rock layer correspondingly increases to satisfy mass balance; this is apparent from the annulus of larger velocities present between a radius of 40 and 65 ft (14 and 20 m).

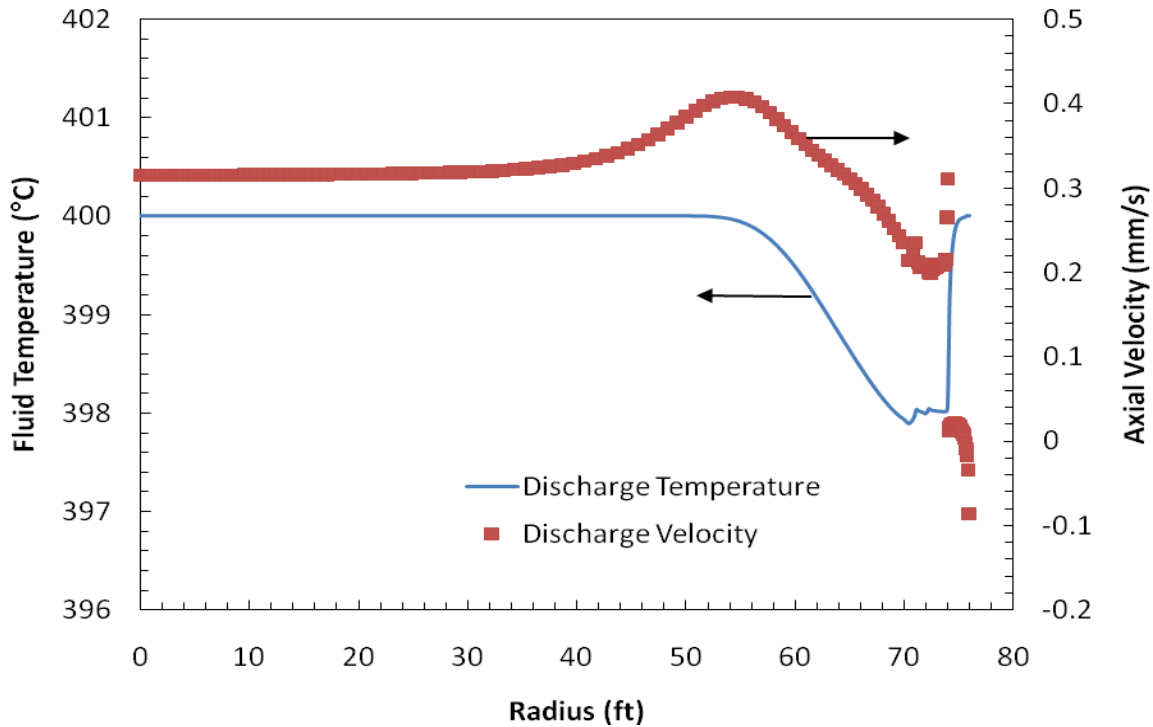


Figure 5-20
Exit Fluid Temperature and Velocity Profiles Six Hours into the Discharge Process

While some cooling of the fluid outflow occurs beyond a radius of 50 ft (15 m), the salt remains within two degrees of the hot operation limit. Some reverse flow is also present due to residual natural convection in the mixture-extension region. The small temperature increase near the tank wall is a consequence of the flow entering the domain at the hot operation limit (no distributor region is included above the fillerbed in the model). This fluid supply creates an additional but negligible energy transport into the CFD domain. Using the reported temperature and velocity profiles in Figure 5-21, the total outflow power equals 164.3 MW after six hours of tank discharge, which is only slightly lower than the 166.6 MW design value (for 1000 MWh_t of energy transport).

To quantify the amount of loss through the tank wall, temperature profiles along the tank wall exterior were recorded throughout the discharge process and converted to an overall convection transfer rate at the tank surface. Integrating these convection rates with time yielded a total loss of 21.9 MWh_t of heat loss during the discharge process.

Tank Behavior during Dwell Conditions

To simulate the thermal performance of the molten salt thermocline tank during dwell conditions between cycles, a constant heat flux of 100 W/m^2 was applied at the tank wall to simulate external losses. Near the wall, the molten salt is cooled, which generates a density gradient. Buoyancy forces cause the denser fluid to flow down the wall and natural convection currents develop in the tank. To accurately model the thermal performance of the thermocline tank with this loss, the existing mesh was significantly refined at the tank wall to capture the velocity profiles associated with natural convection. The control volumes in this region were discretized to a length of 1.5 inches along the wall. Grid-independence was verified by using control volumes that are half this size (0.75 inch). A time step of 5 seconds was used for the duration of the dwell time.

To better approximate the no-flow conditions associated with the stagnant tank, the bottom of the fillerbed domain was changed to a wall boundary condition to prevent fluid from leaving the domain. Moreover, due to the density change of the molten salt with temperature, the bulk volume of the molten salt in the tank decreased with dwell time. In an actual thermocline tank, this reduction would generate some void space as the fluid level would drop to a lower tank height. To maintain continuity in the CFD model, the upper boundary condition is rendered permeable for additional molten salt to enter the domain and make up for the lost volume associated with the thermal contraction. While continuity is ensured in this manner, this permeability distorts the energy balance of the stagnant tank as the molten salt enters at the hot temperature and thus transports some additional energy into the domain. However, this artificial energy transfer is approximately two orders of magnitude less than the fixed heat loss through the tank wall, and may be readily neglected.

A total dwell time of eight hours was simulated. The thermal response at the wall is shown in Figure 5-21 at three different heights in the tank of 0, 20, and 40 feet (the bottom, middle, and top of the tank, respectively).

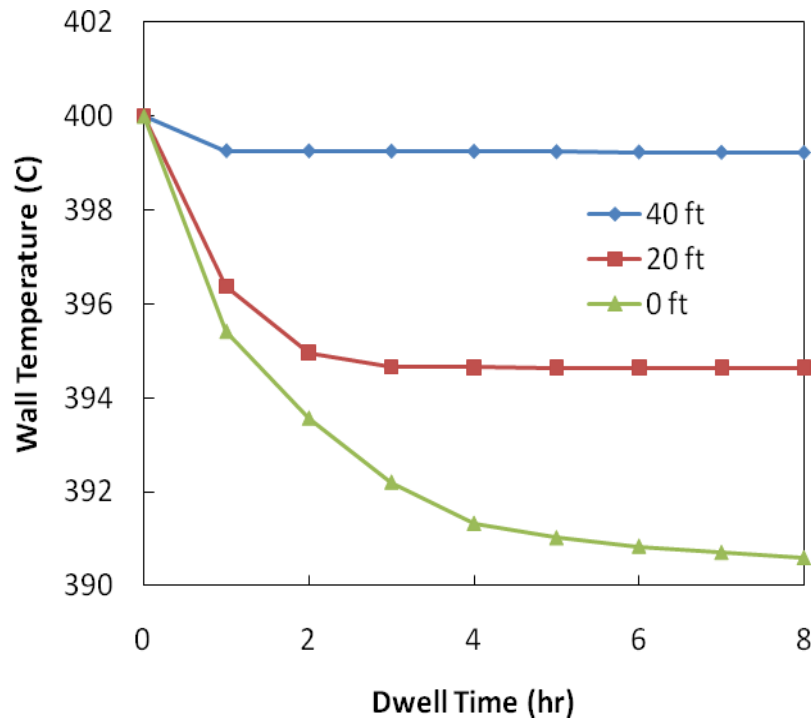


Figure 5-21
Wall Temperature of the Thermocline Tank as a Function of Dwell Time

As seen in the plot, the top of the tank appears largely insensitive to the external tank losses. It should be remembered that the top of the tank was replenished with hot molten salt to make up for the density change of the cooled fluid and may have distorted the local temperature values to some extent. At the midpoint of the tank height, the wall temperature shows an initial decrease but decays to a constant temperature of approximately 395°C (743°F) after 3.5 hours of dwell time. This temperature response can be attributed to the development of the natural convection velocity profile along the wall (shown in Figure 5-22). As the molten salt velocity along the wall develops with time, the temperatures in the fully developed regions approach an approximately steady-state condition.

In contrast, the bottom of the tank continues to cool during the entire dwell time. Due to the impermeable wall boundary condition stipulated for the bottom of the stagnant, the natural convection currents do not develop in this corner and the fluid remains relatively stagnant with time. As such, this area of the tank is not replenished with warmer fluid from above and continues to cool due to heat losses.

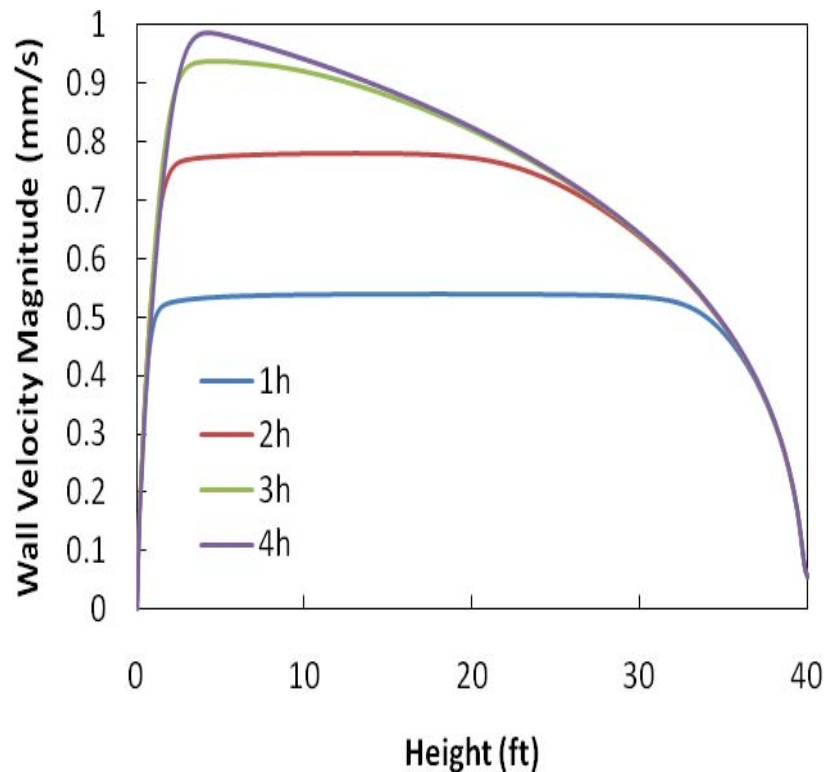


Figure 5-22
Molten Salt Velocity Along the Tank Wall During the Initial Four Hours of Dwell Time

As with the tank discharge performance, further analysis was conducted by Purdue using more realistic tank assumptions to further understand the dwell-time performance of the thermocline. Using the CFD model with rock stratification and heat losses at the tank wall, forced fluid flow was stopped after a discharge half-cycle during periodic operation so that the tank thermal response during dwell conditions could be investigated. To prevent any outflow during this period, the bottom of the fillerbed was converted to a wall boundary condition. These conditions were applied for four hours of dwell time. As with the discharge process, the total heat loss from external convection was determined from the exterior temperature profiles and found to be 12.6 MWh_t during the total period of the dwell time. The fluid temperature throughout the fillerbed at this time is provided in Figure 5-23.

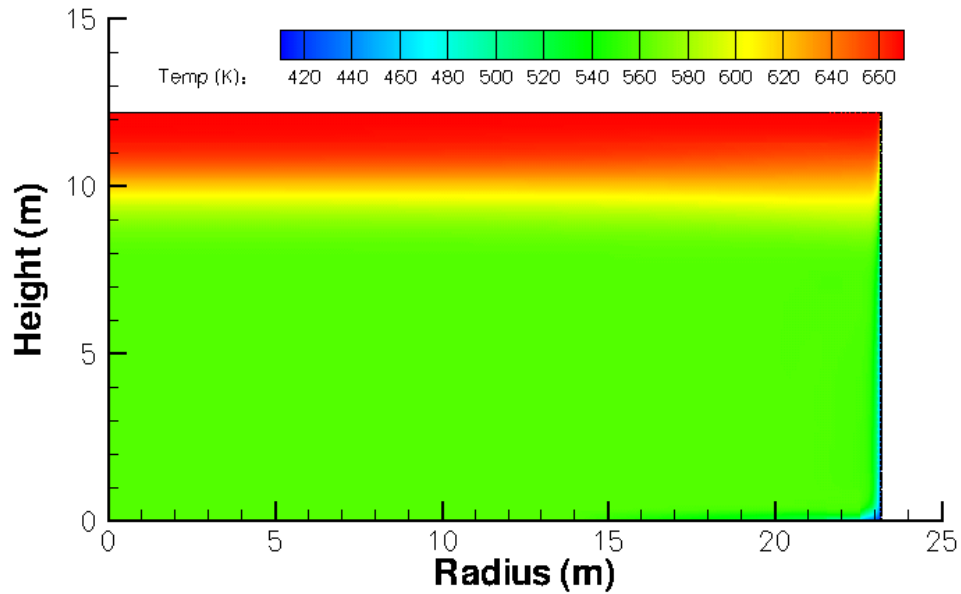


Figure 5-23
Fluid Temperature Distribution after Four Hours of Dwell Conditions

The thermocline stratification is retained throughout most of the tank region with convection losses still being limited to the region very close to the tank wall. To resolve the external convection effects, the temperature response at five height locations along the interior wall surface is plotted in Figure 5-24 against dwell time.

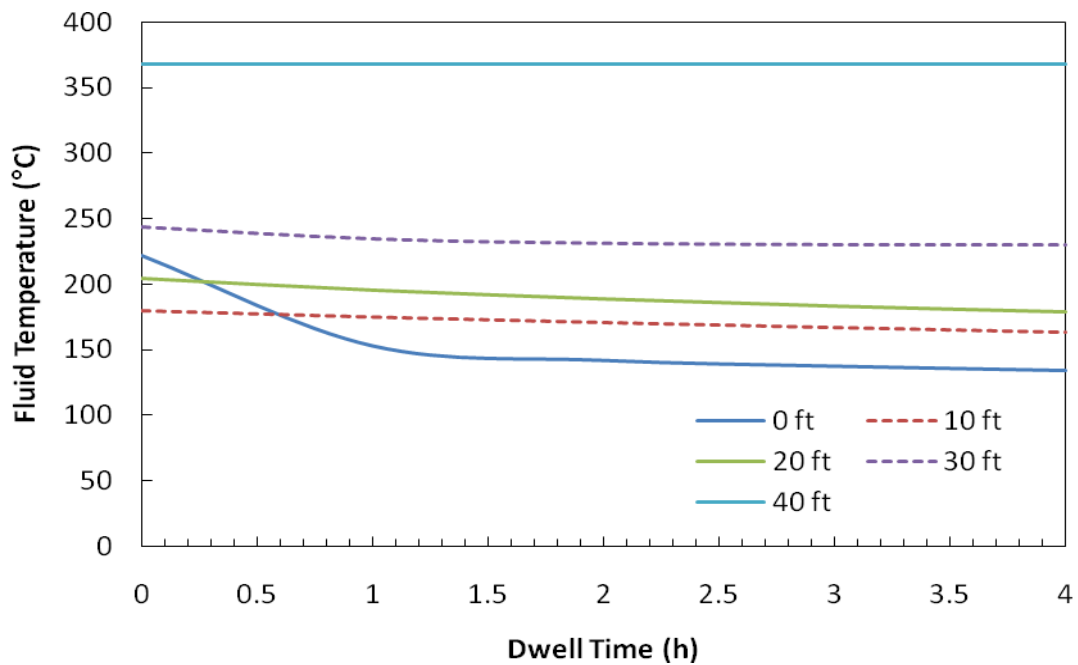


Figure 5-24
Interior Wall Temperatures in the Thermocline Tank during the Dwell Time

The largest temperature change occurs at the bottom of the fillerbed during the first hour of dwell conditions. As mentioned in the discussion of the discharge cycle, the bottom of the fillerbed was influenced by molten salt entering at a warmer temperature than the adjacent wall. As this inflow was not present during the dwell state, the local fillerbed could no longer support a greater temperature than the surroundings which led to the cooling shown in the figure. After this initial effect, the bottom (as well as the other wall locations) maintained a minimal temperature decrease with time.

Discussion and Comparison of NREL and Purdue Thermal Performance Results

Comparing the results from the models developed at Purdue University with the work performed by NREL, discrepancies are evident with respect to the overall performance predictions of the thermocline system. The reasons for the differences between the two sets of predictions are provided below, for both the discharge model and the natural convection analysis.

Discharge Cycle Model

In the NREL thermocline discharge model, the superficial velocity (the product of real fluid velocity and fillerbed void fraction) was constrained to a fluid flow residence time equal to the tank discharge time. For the specified liquid height and discharge time of 14 meters and 6 hours, respectively, the resultant superficial velocity was 0.648 mm/s. The real velocity was then varied according to the user-defined bed void fraction. In reality, the fluid flow does not have a “superficial” residence time in the tank. The residence time of the fluid is the ratio of the tank height to the real fluid velocity. The merit of superficial velocity in a porous medium is to correct for the deviation in mass flow due to the presence of solids in the cross-sectional tank area. Thus by constraining the superficial velocity, the residence time of the fluid flow in the tank still varied with the user-defined void fraction.

In the Purdue analysis, the fluid velocity was determined from the required discharge power and the diameter of the tank and not by constraining it to a residence time. As thermocline tanks are sized for a required storage energy and cycle time, the discharge power is subsequently equal to the ratio of this energy to the cycle time. In addition, the power associated with the discharge flow can also be related to the molten salt mass flow rate as:

$$P = \dot{m}C_p(T_h - T_c) \quad \text{Equation 5-8}$$

For known material properties and tank area, this mass flow rate can be converted to a discharge outflow velocity using:

$$\dot{m} = \rho_h UA \quad \text{Equation 5-9}$$

This would be the velocity at the outflow of the fillerbed, and therefore equal to the superficial velocity of the molten salt inside the fillerbed. Using the model parameters listed by NREL for a 50 ft diameter tank, the superficial velocity would be 0.387 mm/s to satisfy the outflow power associated with the tank energy and discharge cycle time. This result is entirely independent of

the tank height. Applying the artificial residence time constraint instead generated a superficial velocity that is 67% greater than necessary for nominal tank operation. As a consequence, the outflow supplied excess thermal power which drained the stored tank energy too quickly, as noted by the sharp temperature decrease at 4.5 hours in Figure 5-10.

Dwell-Time Model

In both the NREL and Purdue models for a stagnant tank filled with hot molten salt, natural convection currents are generated by applying a constant heat flux of 100 W/m^2 at the tank wall to simulate external losses to the surroundings. In the NREL simulation of the wall heat transfer, the left side of the domain has a fixed temperature imposed, which causes a big slope in the liquid temperature profiles which does not truly represent the real situation. Since the heat transfer in the fluid region is dominated by convection, it is not likely for the fluid to have a sudden change in temperature in the regions far away from the wall.

Nomenclature

A	Area
C_p	Specific heat
d	Filler diameter
D	Diameter
E	Energy
K	Permeability
p	Pressure
Q	Flow rate
T	Temperature
u	Velocity

Greek

ε	Void fraction
λ	Friction factor
μ	Viscosity
ρ	Density

Subscript

l	fluid
c	Cold
d	Distributor
h	Hot
m	Mean
t	Tank

6

CONCLUSIONS

Thermal energy storage holds the promise of providing an efficient and cost-effective way to transform solar thermal energy from a variable resource into a firm, more dispatchable source of electricity. Studies have shown that a thermal energy storage system improves the performance of a solar thermal plant by smoothing power production and/or allowing it to continue during brief periods of lost solar resource, such as intermittent cloud cover, or into the evening hours when the solar resource is no longer available. Used in this way, thermal storage can increase the utilization of the power block and the annual capacity factor of a CSP facility, decrease the levelized cost of electricity produced by the plant, and facilitate the incorporation of solar thermal power plants into the utility grid.

This study showed the potential of thermocline storage systems to significantly reduce the cost of TES. The potential cost advantage for the thermocline is the use of a single tank instead of two tanks and a much lower volume of solar salt. For a 1000 MWh_t system, this means a tank volume of 20,600 m³ instead of two 16,300 m³ tanks, and a salt volume of roughly 5,000 m³ instead of about 18,000 m³ for the two tanks. These systems require relatively simple designs, involving chemically inert media that remain in a single phase throughout the storage charging and discharging process. The direct central receiver thermocline systems appear to be the lowest cost option on an absolute basis. The study confirmed the feasibility of achieving total (direct & indirect) capital costs below \$35/kWh_t and \$70/kWh_t for direct and indirect systems, respectively, compared to \$50/kWh_t and \$90/kWh_t for the equivalent two-tank cases. For the 500-1500 MWh_t size range the indirect parabolic trough thermocline systems have the greatest cost benefit on a percentage basis compared to two-tank; the 1500 MWh_t indirect trough case has a 37% cost reduction compared to 20% for the direct central receiver. For the larger tank sizes, 3000-3500 MWh_t the direct systems show greater benefit than the indirect systems, a 33% vs. 23% advantage over the equivalent two-tank designs for the 3500 MWh_t cases. For all systems 500 MWh_t and larger, the cost benefit was 20% or more compared to the corresponding two-tank designs. In the future a direct storage system for parabolic trough also should be considered to further reduce cost and efficiency losses. Developing cost effective trough storage is a significant R&D challenge that will likely require development of a new lower cost HTF.

On the spectrum of commercial maturity, the two-tank indirect molten salt system represents the current state-of-the-art for thermal energy storage, with two projects recently completed and several more scheduled to begin construction. Two-tank direct systems are close behind, both for central receiver CSP plants and parabolic trough plants if molten salt proves viable in a parabolic trough collector field. The logical next step for the thermocline system is a pilot plant or a small commercial unit that can validate the operation of the storage system outside of the laboratory. Given the expanding number of CSP plant deployments, there is ample room for further development in the field of thermal energy storage. In addition to further testing and validation in the field, standardized design and modeling procedures would be a valuable addition to TES

system design to improve the design process and speed up production and deployment of thermal energy storage systems. The development of the thermocline TES would benefit greatly from improved modeling and tank design techniques, which would offer a more complete picture of design and performance. Testing and operation of a larger scale pilot thermocline will also demonstrate the capabilities of the system.

Next Steps

Two potential courses of action are under consideration for further development of the thermocline technology under EPRI's Generation Technology Industry Demonstration Program.

- The first is a front-end engineering design of a thermocline storage system to reduce the spread in the budget estimation and develop a complete design for the first commercial-scale demonstration plant. This follow-on study would include the following:
 - Detailed design of storage tank shell, reinforcements, internal structures, distributors systems, civil foundations, etc.
 - Detailed process diagrams and P&IDs
 - Main equipment specifications for salt pumps, heat exchangers, and auxiliary equipment
 - Interconnection with the solar plant
 - Operating modes and transient analysis
 - Project schedule
- In the second approach EPRI would develop a small capacity pilot project that could resolve outstanding R&D questions. This project would likely be done in collaboration with the national laboratories and would include testing and modeling activities. The equipment could potentially be sited at the Solar Technology Acceleration Center (SolarTAC) in Colorado, at Sandia in Albuquerque, or at a thermal plant (solar or fossil). Sandia plans to study thermal ratcheting in 2010. Detailed finite element analysis (FEA) models of various thermocline tank geometries, such as vertical and sloped tank walls, will be analyzed with ANSYS software. The temperature difference between the top and bottom of the tank will be varied from 100°C (today's trough) to 350°C (next-generation super-critical tower). Low-cost methods of eliminating the thermal ratcheting issue will be proposed, e.g., sloped walls, inserts, etc. If the analytical work indicates that ratcheting is a manageable issue, Sandia plans to build a lab-scale prototype to validate the FEA models of the most promising solution(s) in 2011. Further work is ongoing at NREL to develop a more universal fit for evaluation of thermocline-based thermal energy storage systems. NREL plans to develop a set of approximations for the thermocline tank behavior for a variety of tank and filler geometries, heat-transfer parameters, and operating conditions. The objectives of the pilot project would include:
 - Validate model performance across different operating strategies
 - Determine potential impact, if any, due to thermal ratcheting
 - Review performance of active components (pumps, valves, seals, etc.) operating in molten salt environment
 - Examine transients and fatigue effects on materials due to thermal cycling

A

DESIGN REQUIREMENTS

Table A-1
Design Requirements

ITEM	REQUIREMENT
Approach	Design single thermocline tank or multiple modular thermocline tanks for each design case as appropriate.
Storage Fluid	Molten binary nitrate salts: 60%Na/40%K
Density at 300°C (572°F)	1830 kg/m ³
Viscosity at 300°C (572°F)	3.26 cP
Heat Capacity at 300°C (572°F)	1,495 J/kg-K
Filler	75% of the volume filled with rocks and sand
Material	Quartzite (SiO ₂) 1- 2" sphere equivalent
Rock density	165 lbm/ft ³
Design Storage Temperatures (Direct)	HOT-Top of the tank 585°C (1085°F)
	COLD-Bottom of the tank 287°C (550°F)
Design Storage Temperatures (Indirect)	HOT-Top of the tank 400°C (752°F)
	COLD-Bottom of the tank 287°C (550°F)
Working Capacity: Full Power TES Hours	SIX (6) HOURS
Minimum Heel	0.9 m (3 feet)
Cover Gas	Vented to atmosphere; NO _x scrubbing system for initial salt fill
Site Conditions (See Environmental Conditions Document below for additional details)	
Seismic	API 650 Zone 3
Wind	40.2 m/sec (90 mph)
Soil	480 kPa (10,000 psi) allowable bearing
Instrumentation	Level, pressure and temperatures indication

Table A-1
Design Requirements (Continued)

ITEM	REQUIREMENT
Tank Valving	Vented to atmosphere
Auxiliaries	Determined that no active convection cooling is required
	Tank warm-up/start-up system – hot air to heat tank internals, filler, etc.
	Tank thermal insulation and supporting structures as required
	Earthen impoundment dikes to contain a salt spill of the complete content of the tank above ground
	Upper and lower manifolds and internal plumbing to minimize entrainment
	Inlet/outlet ports design details (orientation and geometry including baffles)
	Salt drain tank for gravity drain of auxiliary piping and heat exchangers, if required
	Auxiliary piping, hot/cold pumps and valves as required by system design.
	Auxiliary salt melter
	BOP-salt-to-steam heat exchanger (solar steam generator); LMTD 7°C-salt side
	System heat exchanger with solar field HTF- (thermal oil) for indirect TES system; LMTD 7°C-salt side
	Any other auxiliary system required for the start-up/operation of the TES thermocline system (e.g., salt maintenance system)
Design Life	30 years
Design Standards	
Basic Design	API 650
Seismic Design	API 650 APP E
Foundations	ACI 318
Piping	B 31.1
High Temperature Allowable Stresses	ASME Section VIII

Table A-1
Design Requirements (Continued)

ITEM	REQUIREMENT
TES Storage Tank Design	Vertical, cylindrical with dome roof
	Tank ullage space provided over the upper manifold
	Dome roofs self supporting
	Tank height added as freeboard to allow for salt sloshing during safe shutdown earthquake (SSE)
	Tank bottom design developed by B&V to minimize thermal stresses (e.g., sloped-wall conical bottom concept) and cost
Tank Height/Diameter	Optimized for capacity vs. cost
	Taller thermocline tanks with smaller diameters are favored over shorter tanks with larger diameters. Soil bearing capacity limits tank height.
Piping/Valves/Instrumentation Heat Tracing	All TES salt piping, valves, instrumentation to be heat traced to prevent salt freeze-up during operation and allow pre-heating during start-up.
Expansion Joints	Expansion joints included, if required
Solar Steam Generator Subsystem	Only the molten salt side
Steam Generator Configuration	Forced recirculation
Heat Exchanger	Welded plate and frame
Thermal Rating (MW_t)	16.7 to 583.3 MW_t
LMTD	5°C
Materials	
Tanks operating up to 425°C (800°F)	Carbon steel SA-285C
Tanks operating above 425°C up to 560°C (1040°F)	Stainless steel SA-240-347
Side Wall Thermal Insulation	Mineral wool
Base Insulation	Combination of foam glass and insulating fire brick
Salt Cold Piping	ASTM A 106 Grade B seamless carbon steel
Salt Hot Piping	ASTM A312, TP 304 seamless stainless steel

Table A-1
Design Requirements (Continued)

ITEM	REQUIREMENT
Operation	
TES Operating Modes	Initial charging
Process Diagrams Including All Operating Modes For Direct And Indirect Schemes.	Full discharge, full charge, partial charge, partial discharge, stand-by
Steam Conditions for CSP Central Receiver Direct TES	12.5 MPa/550°C (1,815 psia/1022°F)
Allowable Max Temperature Decay (or Rise) Ratio (Direct TES)	0.15*
*Minimum Steam Generator Supply Temperature	560-45=515°C (959.4°F)
*Maximum Solar Field Temperature	287+45=336°C (637°F)
Steam Conditions for Parabolic Trough Indirect TES	10 MPa/370°C (1,450 psia/700°F)
Allowable Max Temperature Decay (or Rise) Ratio (Indirect TES)	0.15*
*Minimum Steam Generator Supply Temperature	375-14=364°C (682°F)
*Maximum Solar Field Temperature	287+14=301°C (574°F)
Economic Parameters For Insulation Optimization	
MWhe/kWhe	\$350 MWhe/\$0.35 kWhe
MWh _i /kWh _i	\$135 MWh _i /\$0.135 kWh _i
Parametric Range For Sensitivity Analysis	Range +/-25%
WACC	8.45%
Physical Plant Depreciation Period	5 years
Design Battery Limits Definition	Process diagrams for direct/indirect TES designs

ATTACHMENT B

ENVIRONMENTAL CONDITIONS

BARSTOW, CALIFORNIA

011888

A-43

Environmental Conditions -
Barstow, California

1. GENERAL

1.1 Scope

This document lists the environmental conditions for the 10 MWe Solar Central Receiver Pilot Plant to be located at the Southern California Edison (SCE) Cool Water Site near Barstow, California.

2. DOCUMENTS

The following form a part of this document to the extent stated herein.

MIL-STD-810B Environmental Test Methods

Uniform Building Code - 1976 Edition, Volume I by
International Conference of Building Officials

3. ENVIRONMENTS

Environmental conditions include winds and gusts, temperature extremes, rain, sleet, hail, snow, earthquake and soil conditions as follows:

3.1 Wind

The wind speed specifications at a reference height of 10m (30 ft) shall be:

3.1.1 Speed Frequency

Speed, m/s (mph)		Frequency, Percent
0-2	(0-4.5)	29
2-4	(4.5-9.0)	21
4-6	(9.0-13.5)	19
6-8	(13.5-18.0)	14
6-10	(18.0-22.5)	8
10-12	(22.5-27.0)	5
12-14	(27.0-31.5)	3
14-	(31.5-)	Less than 1

Environmental Conditions -
Barstow, California

1. GENERAL

1.1 Scope

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Speed, m/s (mph)		Frequency, Percent
0-2	(0-4.5)	29
2-4	(4.5-9.0)	21
4-6	(9.0-13.5)	19
6-8	(13.5-18.0)	14
6-10	(18.0-22.5)	8
10-12	(22.5-27.0)	5
12-14	(27.0-31.5)	3
14-	(31.5-)	Less than 1

- 3.3.4 Snow - Maximum 24-hour rate: 0.3m (1 ft.); maximum loading: 250 Pa (5 lbs/ft²).

3.4 Insolation

- 3.4.1 Maximum Flux. Direct normal nominal insolation of 1100 watts/square metre maximum at the plant site.

- 3.4.2 Rate of Change. The maximum rate of change of incident flux shall be assumed as that which would result from the passage of an opaque cloud across an otherwise clear sky where the sharp leading or trailing edges of the shadow move across the pilot plant site at a velocity of 20 m/s (45 mph).

3.5 Earthquake

Seismic zone 3 (Uniform Bldg. Code).

3.6 Soil Properties

The surface deposits of silty sand, which extends to depths of from 0.3m to 1.5m (1 ft to 5 ft), are only moderately firm and become weaker when wet. The sand below a depth of about 1.5m (5 ft) is firm but contains thin layers of relatively soft silt. In general, the sand is firmer below 3m (10 ft) and layers of soft silt were not encountered.

3.6.1 G. Shear Modulus. and E. Secant Modulus

$$\text{Shear Modulus} = G = \frac{\beta(h+z)}{2(1+\mu)} = \frac{\beta v}{2.6}$$

$$\text{Secant Modulus} = E = \beta(h+z) = \beta y$$

Where β = function of soil depth (see chart)

h = depth of burden

z = depth from burden

μ = constant = 0.3

y = depth from surface

Soil Depth (v)		Density		γ		Secant Modulus at the free surface	
(m)	(ft)	(kg/m ³)	(Lb/ft ³)	(MPa/m)	(psi/ft)	(MPa)	(psi)
0-1.5	(0-5)	1600	(100)	1.7	(75)	0.7	(100)
1.5-3.0	(5-10)	1840	(115)	2.5	(110)	1.4	(200)
3.0-	(10-)	1920	(120)	3.4	(150)	2.1	(300)

At depth of 1.5m (5 ft) $G = 10$ MPa, (1500 psi)
 $E = 28$ MPa (4000 psi)

At depth of 3.0m (10 ft) $G = 32$ MPa, (4600 psi)
 $E = 83$ MPa (12,000 psi)

Geophysical data (from seismic refraction investigation) indicates a shear modulus at a strain rate of 10^{-4} m/m (in/in) of between 100 to 140 MPa (15,000 to 20,000 psi) at a depth of 3m (10 ft).

3.6.2. Bearing Capacity (allowable for standard spread or mat-type foundations).

Depth, m(ft)		Load kPa(psf)	
0.6	(2)	70	(1,500)
1.5	(5)	240	(5,000)
3.0	(10)	480	(10,000)

3.6.3 Penetration Data. The number of blows required to drive a materials sampler 300mm (12 in) was recorded for a number of borings. To a depth of 7.5m (25 ft) a weight of 725 kg (1600 lb) (Kelly weight) falling a distance of 0.3m (1 ft) was used to drive the 76mm (3 in) diameter sampler.

Number of Blows to Drive an LC&A Sampler 300m (12 in)

Depth (m)	(ft)	Average	Range	No. of Samples
0 - 1.5	(0-5)	2.6	0 - 9	74
1.5 - 3.0	(5-10)	4.2	1 - 10	42
3.0 - 4.5	(10-15)	5.9	2 - 11	31
4.5 - 6.0	(15-20)	8.0	4 - 15	19
6.0 - 7.5	(20-25)	9.0	5 - 20	17

Note: There does not appear to be a direct conversion between these data and a standard penetration test, so the results are qualitative in nature

3.6.4 Water Table. Below 30 m (100 ft)

B

THERMOCLINE PROCESS FLOW DIAGRAMS

Thermocline Process Flow Diagrams

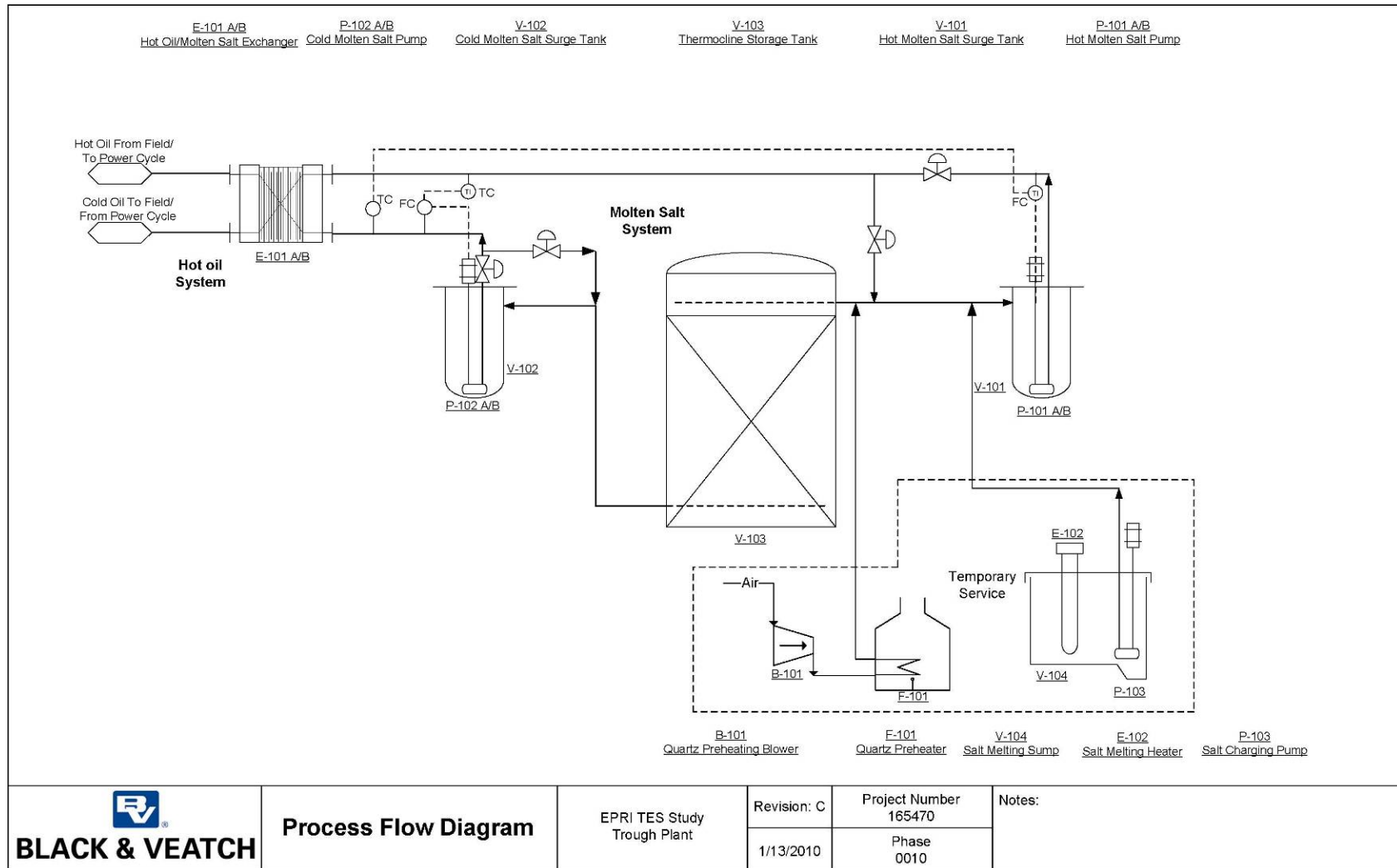


Figure B-1
Indirect Thermocline Process Flow Diagram

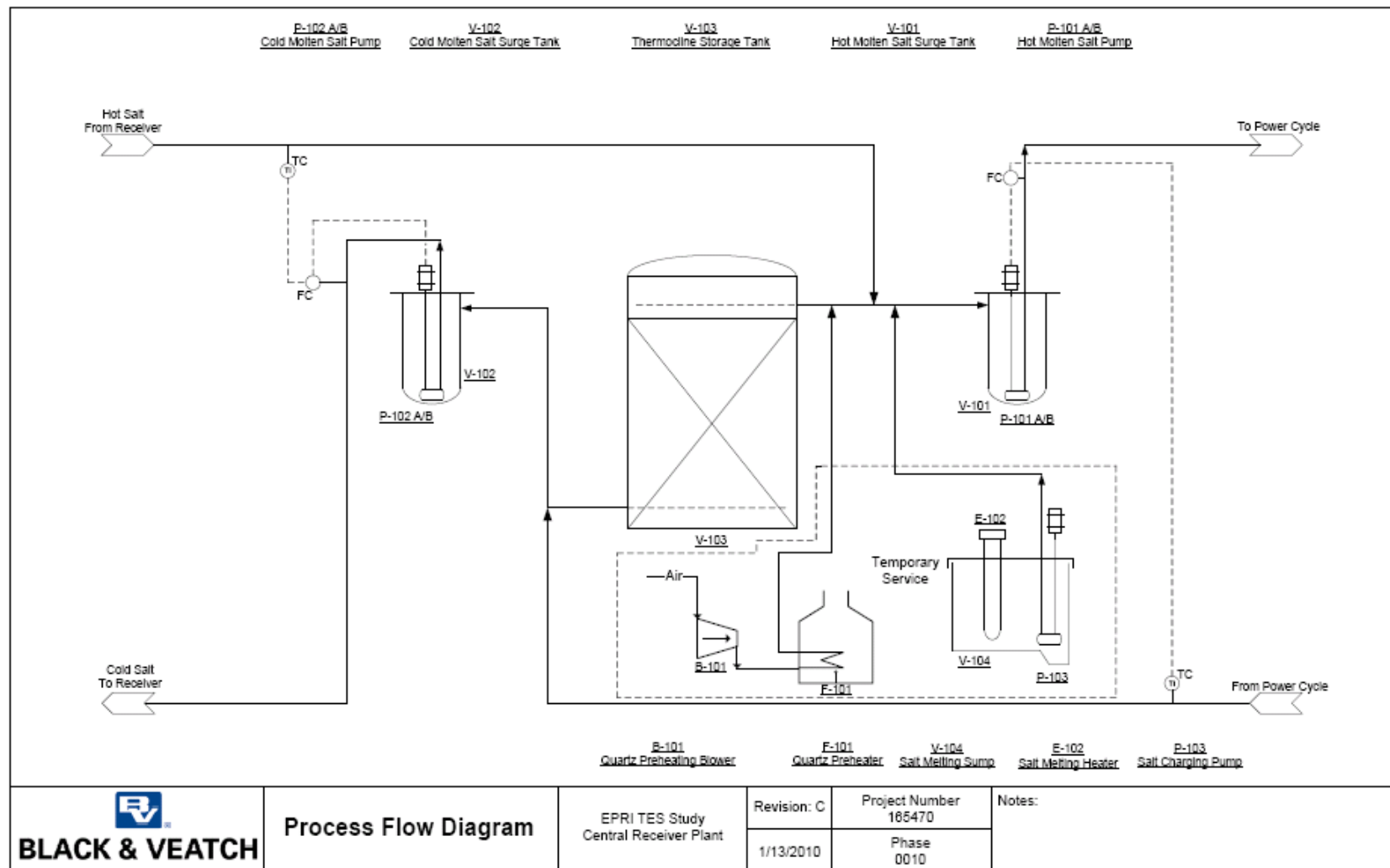


Figure B-2
Direct Thermocline Process Flow Diagram

C

HEAT AND MATERIAL BALANCE

 BLACK & VEATCH						Heat and Material Balance Summary	
						PROJECT: EPRI Thermocline Storage System	
						LOCATION: Barstow, California	
						DWG. NO.: D01-0000	
						REV.: B	

Design Case	Working Storage (MWh _{thermal})	Charging Salt Flowrate (lb/hr)	Discharging Salt Flowrate (lb/hr)	T _{in, (°F)}	Operating Pressure (psia)	Process Type
1	100	1,488,444	826,913	752	80	Trough
2	100	596,493	331,595	1040	280	Central Receiver
3	500	7,442,220	4,134,567	752	80	Trough
4	500	2,982,453	1,656,924	1040	280	Central Receiver
5	1000	14,884,441	8,269,134	752	80	Trough
6	1000	5,964,926	3,313,848	1040	280	Central Receiver
7	3000	44,653,323	24,807,402	752	80	Trough
8	3000	17,894,777	9,941,543	1040	280	Central Receiver

There are 3 operating modes for each design case.

Operating Mode 1:
In this mode, molten salt is heated by the solar energy collection field and returned to the tank directly. There is no power generation taking place, and thermal energy is being stored in the tank.

Operating Mode 2:
In this mode, hot molten salt is being used to generate power. No molten salt is being heated in this scenario. As such, thermal energy is being removed from the thermocline tank.

Operating Mode 3:
In this operating mode, molten salt is being heated by the solar energy collection field and used to generate electricity. This mode is not available in the trough configurations (Case 1, 3, 5, 7). In this mode, thermal energy is either removed or added to the thermocline tank depending on the solar input. The values provided in the heat and material balance assume the power plant and the solar collection field are operating at 100% capacity. The capacity of the solar field is 1.8 times the capacity of the power plant.

Operating Mode 4:
This operating mode occurs when solar input drops below a certain level. In this mode a small amount of molten salt is circulated throughout the system to maintain the temperature above the freezing point of the salt. This reduces the energy stored in the tank. If the temperature is in danger of dropping below the freezing point, all lines will be drained and the heat tracing will be activated. The lines between the thermocline tank and the surge tanks cannot be drained.

Design Case	Solar Charging System (inches)	Power Generation System (inches)
1	10	8
2	8	6
3	20	16
4	14	12
5	24	20
6	18	16
7	36	30
8	26	20

Figure C-1
Heat and Material Balance Spreadsheet, Page 1

Heat and Material Balance

 BLACK & VEATCH		MATERIAL BALANCE									
		NO.	DATE	REVISION		DWG.	CHK.	APP.	PROJECT: EPRI Thermocline Storage System		
		A	10-Nov-09	Preliminary Issue		RJH	MAL		CASE: Design Case 1: 100MWh Storage at 752°F		
		B	01-Dec-09	Updated With Comments		RJH	MAL		LOCATION: Barstow, California		
									DWG. NO. D01-0001		
Stream Name		Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger	Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger	REV. 8	
Stream Number	Units	1	2	3	4	1	2	3	4		
Operating Mode	Mode	1 - Heating Solar Salt Without Power Generation (Charging)				2 - Generating Power Without Heating Solar Salt (Discharging)					
Overall Properties	Temperature	°F	556	556	752	559	559	752	752		
	Pressure	psia	85	80	85	85	80	85	80		
	Mass Flow	lb/hr	-	1,488,444	1,488,444	826,913	-	-	826,913		
	Volume Flow (actual)	ft ³ /hr	-	12,709	13,320	7,685	-	-	7,400		
	Volume Flow (actual)	USGPM	-	1,590	1,661	983	-	-	923		
	Density	lb/ft ³	117	117	112	117	117	112	112		
	Viscosity	cP	3.28	3.78	1.75	3.28	3.28	1.75	1.75		
	Thermal Conductivity	Btu/hr-ft-F	0.31	0.31	0.31	0.31	0.31	0.31	0.31		
	Mass Heat Capacity	Btu/lb-F	0.357	0.357	0.357	0.357	0.357	0.357	0.357		
Piping Components	Line Size	in	-	10.0	10.0	8.0	-	-	8.0		
	60% NaNO ₂ /40% KNO ₃	lb/hr	-	1,488,444	1,488,444	826,913	-	-	826,913		
Total Mass Flow	lb/hr	-	-	1,488,444	1,488,444	826,913	-	-	826,913		

NOTES:

- Standard conditions above defined as 60°F and 14.696 psia.
- Current pressures are estimated.

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Figure C-2
Heat and Material Balance Spreadsheet, Page 2

 BLACK & VEATCH		MATERIAL BALANCE											
		NO.	DATE	REVISION	DWG.	CHK.	APP.	PROJECT: EPRI Thermocline Storage System					
		A	10-Nov-09	Preliminary Issue	RJH	MAL		CASE: Design Case 2: 100MWh Storage at 1040 °F					
		B	01-Dec-09	Updated With Comments	RJH	MAL		LOCATION: Barstow, California					
								DWG. NO. D01-0002					
								REV. B					
Stream Name		Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation	Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation	Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation
Stream Number	Units	1	2	3	4	1	2	3	4	1	2	3	4
Operating Mode	Mode	1 - Heating Solar Salt Without Power Generation (Charging)				2 - Generating Power Without Heating Solar Salt (Discharging)				3 - Generating Power While Heating the Solar Salt			
Overall Properties	Temperature	F	559	559	1,040	559	559	1,040	1,040	559	559	1,040	1,040
	Pressure	psia	265	280	80	265	280	80	65	265	280	80	65
	Mass Flow	lb/hr	-	596,493	596,493	-	331,385	-	331,385	331,385	596,493	596,493	331,385
	Volume Flow (actual)	ft ³ /hr	-	5,110	5,667	-	2,839	-	3,160	2,839	5,110	5,667	3,160
	Volume Flow (actual)	USGPM	-	637	709	-	354	-	394	354	637	709	394
	Density	lb/ft ³	117	117	105	117	117	105	105	117	117	105	105
	Viscosity	cP	3.26	3.26	1.05	3.26	3.26	1.05	1.05	3.26	3.26	1.05	1
	Thermal Conductivity	Btu/hr-ft-F	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0
	Mass Heat Capacity	Btu/lb-F	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357
Piping Components	Line Size	in	8.0	8.0	8.0	6.0	6.0	8.0	8.0	6.0	6.0	8.0	8.0
	60% NaNO ₂ /40% KNO ₃	lb/hr	-	596,493	596,493	-	331,385	-	331,385	331,385	596,493	596,493	331,385
Total Mass Flow	lb/hr	-	596,493	596,493	-	331,385	-	-	331,385	331,385	596,493	596,493	331,385

NOTES:

- Standard conditions above defined as 60°F and 14.696 psia.
- Current pressures are estimated.

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Figure C-3
Heat and Material Balance Spreadsheet, Page 3

Heat and Material Balance

 BLACK & VEATCH		MATERIAL BALANCE											
		NO.	DATE	REVISION	DWG.	CHK.	APP.	PROJECT: EPRI Thermocline Storage System					
		A	10-Nov-09	Preliminary Issue	RJH	MAL		CASE: Design Case 3: 500MWh Storage at 752°F					
		B	01-Dec-09	Updated With Comments	RJH	MAL		LOCATION: Barstow, California					
								DWG. NO.: 001-0003					
Stream Name		Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger	Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger				
Stream Number	Units	1	2	3	4	1	2	3	4				
Operating Mode	Mode	1 - Heating Solar Salt Without Power Generation (Charging)				2 - Generating Power Without Heating Solar Salt (Discharging)							
Overall Properties	Temperature	559	559	752	752	559	559	752	752				
	Pressure	85	85	85	89	85	100	70	130				
	Mass Flow	-	7,442,220	7,442,220	-	4,134,567	-	-	4,134,567				
	Volume Flow (actual)	-	83,750	88,099	-	35,417	-	-	37,000				
	Volume Flow (actual)	-	7,968	8,203	-	4,416	-	-	4,613				
	Density	117	117	112	112	117	117	112	112				
	Viscosity	3.28	3.28	1.75	1.75	3.28	3.28	1.75	1.75				
	Thermal Conductivity	0.51	0.51	0.51	0.51	0.51	0.51	0.51	0.51				
Piping Components	Line Size	in	20.0	20.0	-	16.0	-	-	16.0				
	60% NaNO ₂ /40% KNO ₃	lb/hr	-	7,442,220	7,442,220	-	4,134,567	-	4,134,567				
Total Mass Flow		lb/hr	-	7,442,220	7,442,220	-	4,134,567	-	4,134,567				

NOTES:

1. Standard conditions above defined as 60°F and 14.696 psia.
2. Current pressures are estimated.

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Figure C-4
Heat and Material Balance Spreadsheet, Page 4

 BLACK & VEATCH		MATERIAL BALANCE											
		NO.	DATE	REVISION		DWG.	CHK.	APP.	PROJECT: EPRI Thermocline Storage System				
		A	10-Nov-09	Preliminary Issue		RJH	MAL		CASE: Design Case 4: 500MWh Storage at 1040°F				
		B	01-Dec-09	Updated With Comments		RJH	MAL		LOCATION: Barstow, California				
									REV. B				
									DWG. NO. D01-0004				
Stream Name		Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation	Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation	Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation
Stream Number		1	2	3	4	1	2	3	4	1	2	3	4
Operating Mode		1 - Heating Solar Salt Without Power Generation (Charging)				2 - Generating Power Without Heating Solar Salt (Discharging)				3 - Generating Power While Heating the Solar Salt			
Overall Properties	Temperature	F	559	559	1,040	559	559	1,040	1,040	559	559	1,040	1,040
	Pressure	psia	265	280	80	265	280	80	65	265	280	80	65
	Mass Flow	lb/hr	-	2,982,463	2,982,463	-	1,656,924	-	1,656,924	1,656,924	2,982,463	2,982,463	1,656,924
	Volume Flow (actual)	ft ³ /hr	-	25,548	26,437	-	14,193	-	15,798	14,193	25,548	26,437	15,798
	Volume Flow (actual)	USGPM	-	3,185	3,545	-	1,770	-	1,970	1,770	3,185	3,545	1,970
	Density	lb/ft ³	117	117	105	117	117	105	105	117	117	105	105
	Viscosity	cP	3.26	3.26	1.05	3.26	3.26	1.05	1.05	3.26	3.26	1.05	1
Piping Components	Thermal Conductivity	Btu/hr-ft-F	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0
	Mass Heat Capacity	Btu/lb-F	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357
	Line Size	in	14.0	14.0	14.0	12.0	12.0	12.0	12.0	14.0	14.0	14.0	12.0
Total Mass Flow		lb/hr	-	2,982,463	2,982,463	-	1,656,924	-	1,656,924	1,656,924	2,982,463	2,982,463	1,656,924

NOTES:

- Standard conditions above defined as 60°F and 14.696 psia.
- Current pressures are estimated

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Figure C-5
Heat and Material Balance Spreadsheet, Page 5

Heat and Material Balance

 BLACK & VEATCH		MATERIAL BALANCE									
		NO.	DATE	REVISION	DWG.	CHK.	APP.	PROJECT: EPR1 Thermocline Storage System			
		A	10-Nov-09	Preliminary Issue	RJH	MAL		CASE: Design Case 5: 1000MWh Storage at 752°F			
		B	01-Dec-09	Updated With Comments	RJH	MAL		LOCATION: Bartow, California			
								DWG. NO. D05-0001			
								REV. B			
Stream Name		Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger			Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger
Stream Number	Units	1	2	3	4			1	2	3	4
Operating Mode	Mode	1 - Heating Solar Salt Without Power Generation (Charging)						2 - Generating Power Without Heating Solar Salt (Discharging)			
Overall Properties	Temperature	569	569	752	752			569	569	752	752
	Pressure	85	80	85	80			85	100	70	100
	Mass Flow	-	14,884,441	14,884,441	-			8,269,134	-	-	8,269,134
	Volume Flow (actual)	-	127,501	133,196	-			70,834	-	-	73,999
	Volume Flow (ideal)	-	15,896	16,607	-			6,831	-	-	9,226
	Density	117	117	112	112			117	117	112	112
	Viscosity	3.26	3.26	1.75	1.75			3.26	3.26	1.75	1.75
	Thermal Conductivity	0.31	0.31	0.31	0.31			0.31	0.31	0.31	0.31
	Mass Heat Capacity	0.357	0.357	0.357	0.357			0.357	0.357	0.357	0.357
Piping	Line Size	24.0	24.0	24.0	24.0			20.0	20.0	20.0	20.0
Components	60% NaNO ₃ /40% KNO ₃	-	14,884,441	14,884,441	-			8,269,134	-	-	8,269,134
Cold Mass Flow	lb/hr	-	14,884,441	14,884,441	-			8,269,134	-	-	8,269,134

NOTES:

- Standard conditions above defined as 60°F and 14.696 psia.
- Current pressures are estimated.

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Figure C-6
Heat and Material Balance Spreadsheet, Page 6

 BLACK & VEATCH		MATERIAL BALANCE											
		NO.	DATE	REVISION	DWG.	CHK.	APP.	PROJECT: EPRI Thermocline Storage System					
		A	10-Nov-09	Preliminary Issue	RJH	MAL		CASE: Design Case 6: 1000MWh Storage at 1040°F					
		B	01-Dec-09	Updated With Comments	RJH	MAL		LOCATION: Barstow, California					
								DWG. NO. D01-0006					
								REV. B					
Stream Name		Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation			Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation		
Stream Number	Units	1	2	3	4			1	2	3	4		
Operating Mode	Mode	1 - Heating Solar Salt Without Power Generation (Charging)						2 - Generating Power Without Heating Solar Salt (Discharging)					
Overall Properties	Temperature	F	559	559	1,040	1,040		559	559	1,040	1,040		
	Pressure	psia	265	280	80	65		265	280	80	65		
	Mass Flow	lb/hr	-	5,964,926	5,964,926	-		3,313,848	-	-	3,313,848		
	Volume Flow (actual)	ft ³ /hr	-	51,096	56,874	-		28,367	-	-	31,597		
	Volume Flow (actual)	USGPM	-	6,370	7,091	-		3,539	-	-	3,659		
	Density	lb/ft ³	117	117	105	105		117	117	105	105		
	Viscosity	cP	3.26	3.26	1.05	1.05		3.26	3.26	1.05	1.05		
	Thermal Conductivity	Btu/hr-ft-F	0.31	0.31	0.31	0.31		0.31	0.31	0.31	0.31		
Piping	Line Size	in	18.0	18.0	18.0	18.0		18.0	18.0	18.0	18.0		
Components	90% NaNO ₂ /40% KNO ₃	lb/hr	-	5,964,926	5,964,926	-		3,313,848	-	-	3,313,848		
Total Mass Flow	lb/hr	-	5,964,926	5,964,926	-		3,313,848	-	-	3,313,848	-		

NOTES:

- Standard conditions above defined as 60°F and 14.696 psia.
- Current pressures are estimated

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Figure C-7
Heat and Material Balance Spreadsheet, Page 7

Heat and Material Balance

 BLACK & VEATCH		MATERIAL BALANCE									
		NO.	DATE	REVISION		DWG.	CHK.	APP.	PROJECT: EPRI Thermocline Storage System		
		A	10-Nov-09	Preliminary Issue		RJH	MAL		CASE: Design Case 7: 3000MWh Storage at 752°F		
		B	01-Dec-09	Updated With Comments		RJH	MAL		LOCATION: Barstow, California		
									DWG. NO. D01-0007		
									REV. B		
Stream Name		Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger	Cold Salt From Heat Exchanger to Thermocline	Cold Salt From Thermocline to Heat Exchanger	Hot Salt From Heat Exchanger to Thermocline	Hot Salt From Thermocline to Heat Exchanger		
Stream Number		1	2	3	4	1	2	3	4		
Operating Mode		1 - Heating Solar Salt Without Power Generation (Charging)				2 - Generating Power Without Heating Solar Salt (Discharging)					
Overall Properties		559	559	752	752	559	559	752	752		
Temperature		F	F	F	F	F	F	F	F		
Pressure		psia	85	80	85	85	100	70	100		
Mass Flow		lb/hr	-	44,653,323	44,653,323	-	-	-	24,807,402		
Volume Flow (actual)		ft ³ /hr	-	382,602	382,602	-	-	-	221,898		
Volume Flow (actual)		USGPM	-	47,989	47,989	-	-	-	27,678		
Density		lb/ft ³	117	117	112	117	117	112	112		
Viscosity		cP	3.26	3.26	1.75	3.26	3.26	1.75	1.75		
Thermal Conductivity		Btu/hr-ft-F	0.31	0.31	0.31	0.31	0.31	0.31	0.31		
Mass Heat Capacity		Btu/lb-F	0.357	0.357	0.357	0.357	0.357	0.357	0.357		
Piping Components		in	36.0	36.0	36.0	30.0	30.0	30.0	30.0		
Line Size		lb/hr	-	44,653,323	44,653,323	-	-	-	24,807,402		
Components		90% NaNO ₂ /40% KNO ₃	-	44,653,323	44,653,323	-	-	-	24,807,402		
Total Mass Flow		lb/hr	-	44,653,323	44,653,323	-	-	-	24,807,402		

NOTES:

- Standard conditions above defined as 60°F and 14.696 psia.
- Current pressures are estimated

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Figure C-8
Heat and Material Balance Spreadsheet, Page 8

 BLACK & VEATCH		MATERIAL BALANCE											
		NO.	DATE	REVISION		DWG.	CHK.	APP.	PROJECT: EPRI Thermocline Storage System				
		A	10-Nov-09	Preliminary Issue		RJH	MAL		CASE: Design Case 8: 3000MWh Storage at 1040°F				
		B	01-Dec-09	Updated With Comments		RJH	MAL		LOCATION: Barstow, California				
									REV. B				
									DWG. NO. D01-0008				
Stream Name		Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation	Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation	Cold Salt From Power Generation to Thermocline	Cold Salt From Thermocline to Solar Field	Hot Salt From Solar Field to Thermocline	Hot Salt From Thermocline to Power Generation
Stream Number	Units	1	2	3	4	1	2	3	4	1	2	3	4
Operating Mode	Mode	1 - Heating Solar Salt Without Power Generation (Charging)				2 - Generating Power Without Heating Solar Salt (Discharging)				3 - Generating Power While Heating the Solar Salt			
Overall Properties	Temperature	F	559	559	1,040	559	559	1,040	1,040	559	559	1,040	1,040
	Pressure	psia	265	280	80	265	280	80	65	265	280	80	65
	Mass Flow	lb/hr	-	17,894,777	17,894,777	-	-	-	9,941,543	9,941,543	17,894,777	17,894,777	9,941,543
	Volume Flow (actual)	ft ³ /hr	-	153,287	170,623	-	-	-	94,791	85,180	153,287	170,623	94,791
	Volume Flow (design)	USGPM	-	19,111	21,272	-	-	-	11,818	10,617	19,111	21,272	11,818
	Density	lb/ft ³	117	117	105	117	117	105	105	117	117	105	105
	Viscosity	cP	3.26	3.26	1.05	3.26	3.26	1.05	1.05	3.26	3.26	1.05	1
	Thermal Conductivity	Btu/hr-ft-F	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0.31	0
Piping Components	Mass Heat Capacity	Btu/lb-F	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357	0.357
	Line Size	in	20.0	26.0	26.0	20.0	20.0	20.0	20.0	20.0	26.0	26.0	20.0
Total Mass Flow		lb/hr	-	17,894,777	17,894,777	-	-	-	9,941,543	9,941,543	17,894,777	17,894,777	9,941,543

NOTES:

- Standard conditions above defined as 60°F and 14.696 psia.
- Current pressure drops are estimated.

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Figure C-9
Heat and Material Balance Spreadsheet, Page 9

D

MAXIMUM TANK HEIGHT CALCULATIONS

This analysis is a reference to a civil engineering study prepared by B&V. The goal of this analysis was to determine the maximum feasible height of a thermocline tank.

In designing a thermocline system, it is desirable to maximize the height of the storage tank. Increasing the height of the tank improves the efficiency such that the volume of the tank can be reduced. The smaller tank volume means that less salt is required to store a given amount of energy, which reduces the overall cost of the project. The goal of this study is to determine the maximum feasible tank height for this project.

The main consideration in this study is the amount of pressure that the soil can support. Based on the environmental conditions of Barstow California, it was found that a spread foundation can support approximately 10,000 lb/ft².

The bearing stress was calculated for tank diameters of 70, 75, 80, and 85 feet. The stress was calculated for all heights between 8 and 40 feet tall. It should be noted that this is a liquid height, and this does not include any empty space at the top of the tank. This space does not contribute significantly to the bearing pressure. The reference parameters for the calculations include:

- Dia (ft): The diameter of the thermocline tank, in feet
- Height (ft): The liquid height in the thermocline tank
- Area (ft²): The area of the tank foundation
- Weight (K): The total mass of the thermocline tank contents in Kips (1,000 lb increments)
- V (K): UBC 97 Limiting base shear
- OTM (ft-K): Overturning moment
- P/A (K/ft²): The weight of the tank divided by the area
- S (ft³): Section modulus (inertia / distance to the edge of the tank)
- M/S (K/ft²): The overturning moment divided by the section modulus
- Bearing (ksf): The estimated bearing pressure on the soil
- $e/D < 0.123$: The overturning moment divided by the weight of the structure * diameter

Maximum Tank Height Calculations

The last term is important for calculating the solar bearing pressure. When the foundation remains in compression and e/D is less than 0.123, the soil bearing pressure can be computed by:

$$\text{Bearing Pressure} = \text{Vertical Load} / \text{Area} + \text{Moment} / \text{Section Modulus}$$

It can be seen from Table D-1 that the liquid level varies with the tank diameter. It was determined that 39 ft could be reliably assumed for most design cases where a typical foundation is assumed. For the purposes of this study, 39 ft was selected as the maximum liquid height for Cases 1-7. For Case 8, a liquid height of 46 ft was selected.

Table D-1
Bearing Stress Calculations

Density (pcf) = 146.309
V = 0.388

Allowable Soil Pressure (ksf)			
at 2' =	1.500	at 5' =	5.000
		at 10' =	10.000

Dia (ft) =	Height (ft)	Area (sqft)	Weight (K)	V (K)	OTM (ft-K)	P/A (ksf)	S (cf)	M/S (ksf)	Bearing (ksf)	e/D < 0.123
160	20	20,106	58,834	22,824	228,237	2.926	402,124	0.568	3.494	0.0242
160	21	20,106	61,776	23,965	251,631	3.072	402,124	0.626	3.698	0.0255
160	22	20,106	64,718	25,106	276,166	3.219	402,124	0.687	3.906	0.0267
160	23	20,106	67,659	26,247	301,843	3.365	402,124	0.751	4.116	0.0279
160	24	20,106	70,601	27,388	328,661	3.511	402,124	0.817	4.329	0.0291
160	25	20,106	73,543	28,530	356,620	3.658	402,124	0.887	4.545	0.0303
160	26	20,106	76,485	29,671	385,720	3.804	402,124	0.959	4.763	0.0315
160	27	20,106	79,426	30,812	415,961	3.950	402,124	1.034	4.985	0.0327
160	28	20,106	82,368	31,953	447,344	4.097	402,124	1.112	5.209	0.0339
160	29	20,106	85,310	33,094	479,868	4.243	402,124	1.193	5.436	0.0352
160	30	20,106	88,252	34,235	513,532	4.389	402,124	1.277	5.666	0.0364
160	31	20,106	91,193	35,377	548,339	4.536	402,124	1.364	5.899	0.0376
160	32	20,106	94,135	36,518	584,286	4.682	402,124	1.453	6.135	0.0388
160	33	20,106	97,077	37,659	621,374	4.828	402,124	1.545	6.373	0.0400
160	34	20,106	100,018	38,800	659,604	4.975	402,124	1.640	6.615	0.0412
160	35	20,106	102,960	39,941	698,975	5.121	402,124	1.738	6.859	0.0424
160	36	20,106	105,902	41,083	739,487	5.267	402,124	1.839	7.106	0.0436
160	37	20,106	108,844	42,224	781,140	5.413	402,124	1.943	7.356	0.0449
160	38	20,106	111,785	43,365	823,934	5.560	402,124	2.049	7.609	0.0461
160	39	20,106	114,727	44,506	867,870	5.706	402,124	2.158	7.864	0.0473
160	40	20,106	117,669	45,647	912,947	5.852	402,124	2.270	8.123	0.0485
160	41	20,106	120,610	46,789	959,165	5.999	402,124	2.385	8.384	0.0497
160	42	20,106	123,552	47,930	1,006,524	6.145	402,124	2.503	8.648	0.0509
160	43	20,106	126,494	49,071	1,055,024	6.291	402,124	2.624	8.915	0.0521
160	44	20,106	129,436	50,212	1,104,665	6.438	402,124	2.747	9.185	0.0533
160	45	20,106	132,377	51,353	1,155,448	6.584	402,124	2.873	9.457	0.0546
160	46	20,106	135,319	52,494	1,207,372	6.730	402,124	3.002	9.733	0.0558
160	47	20,106	138,261	53,636	1,260,437	6.877	402,124	3.134	10.011	0.0570
160	48	20,106	141,202	54,777	1,314,643	7.023	402,124	3.269	10.292	0.0582
160	49	20,106	144,144	55,918	1,369,991	7.169	402,124	3.407	10.576	0.0594
160	50	20,106	147,086	57,059	1,426,479	7.315	402,124	3.547	10.863	0.0606
160	51	20,106	150,028	58,200	1,484,109	7.462	402,124	3.691	11.152	0.0618
160	52	20,106	152,969	59,342	1,542,880	7.608	402,124	3.837	11.445	0.0630

Table D-1
Bearing Stress Calculations (Continued)

Dia (ft) =	Height (ft)	Area (sqft)	Weight (K)	V (K)	OTM (ft-K)	P/A (ksf)	S (cf)	M/S (ksf)	Bearing (ksf)	e/D < 0.123
150	20	17,671	51,710	20,060	200,599	2.926	331,340	0.605	3.532	0.0259
150	21	17,671	54,295	21,063	221,160	3.072	331,340	0.667	3.740	0.0272
150	22	17,671	56,881	22,066	242,724	3.219	331,340	0.733	3.951	0.0284
150	23	17,671	59,466	23,069	265,292	3.365	331,340	0.801	4.166	0.0297
150	24	17,671	62,052	24,072	288,862	3.511	331,340	0.872	4.383	0.0310
150	25	17,671	64,637	25,075	313,435	3.658	331,340	0.946	4.604	0.0323
150	26	17,671	67,223	26,078	339,012	3.804	331,340	1.023	4.827	0.0336
150	27	17,671	69,808	27,081	365,591	3.950	331,340	1.103	5.054	0.0349
150	28	17,671	72,394	28,084	393,173	4.097	331,340	1.187	5.283	0.0362
150	29	17,671	74,979	29,087	421,759	4.243	331,340	1.273	5.516	0.0375
150	30	17,671	77,565	30,090	451,347	4.389	331,340	1.362	5.751	0.0388
150	31	17,671	80,150	31,093	481,938	4.536	331,340	1.455	5.990	0.0401
150	32	17,671	82,736	32,096	513,532	4.682	331,340	1.550	6.232	0.0414
150	33	17,671	85,321	33,099	546,130	4.828	331,340	1.648	6.476	0.0427
150	34	17,671	87,907	34,102	579,730	4.975	331,340	1.750	6.724	0.0440
150	35	17,671	90,492	35,105	614,333	5.121	331,340	1.854	6.975	0.0453
150	36	17,671	93,078	36,108	649,940	5.267	331,340	1.962	7.229	0.0466
150	37	17,671	95,663	37,111	686,549	5.413	331,340	2.072	7.485	0.0478
150	38	17,671	98,249	38,114	724,161	5.560	331,340	2.186	7.745	0.0491
150	39	17,671	100,834	39,117	762,776	5.706	331,340	2.302	8.008	0.0504
150	40	17,671	103,420	40,120	802,395	5.852	331,340	2.422	8.274	0.0517
150	41	17,671	106,005	41,123	843,016	5.999	331,340	2.544	8.543	0.0530
150	42	17,671	108,591	42,126	884,640	6.145	331,340	2.670	8.815	0.0543
150	43	17,671	111,176	43,129	927,267	6.291	331,340	2.799	9.090	0.0556
150	44	17,671	113,762	44,132	970,897	6.438	331,340	2.930	9.368	0.0569
150	45	17,671	116,347	45,135	1,015,531	6.584	331,340	3.065	9.649	0.0582
150	46	17,671	118,933	46,138	1,061,167	6.730	331,340	3.203	9.933	0.0595
150	47	17,671	121,518	47,141	1,107,806	6.877	331,340	3.343	10.220	0.0608
150	48	17,671	124,104	48,144	1,155,448	7.023	331,340	3.487	10.510	0.0621
150	49	17,671	126,689	49,147	1,204,093	7.169	331,340	3.634	10.803	0.0634
150	50	17,671	129,275	50,150	1,253,741	7.315	331,340	3.784	11.099	0.0647
150	51	17,671	131,860	51,153	1,304,393	7.462	331,340	3.937	11.398	0.0659
150	52	17,671	134,446	52,156	1,356,047	7.608	331,340	4.093	11.701	0.0672

Table D-1
Bearing Stress Calculations (Continued)

Dia (ft) =	Height (ft)	Area (sqft)	Weight (K)	V (K)	OTM (ft-K)	P/A (ksf)	S (cf)	M/S (ksf)	Bearing (ksf)	e/D < 0.123
140	20	15,394	45,045	17,474	174,744	2.926	269,392	0.649	3.575	0.0277
140	21	15,394	47,297	18,348	192,655	3.072	269,392	0.715	3.788	0.0291
140	22	15,394	49,550	19,222	211,440	3.219	269,392	0.785	4.004	0.0305
140	23	15,394	51,802	20,096	231,099	3.365	269,392	0.858	4.223	0.0319
140	24	15,394	54,054	20,969	251,631	3.511	269,392	0.934	4.445	0.0333
140	25	15,394	56,306	21,843	273,037	3.658	269,392	1.014	4.671	0.0346
140	26	15,394	58,559	22,717	295,317	3.804	269,392	1.096	4.900	0.0360
140	27	15,394	60,811	23,590	318,470	3.950	269,392	1.182	5.133	0.0374
140	28	15,394	63,063	24,464	342,498	4.097	269,392	1.271	5.368	0.0388
140	29	15,394	65,315	25,338	367,399	4.243	269,392	1.364	5.607	0.0402
140	30	15,394	67,568	26,212	393,173	4.389	269,392	1.459	5.849	0.0416
140	31	15,394	69,820	27,085	419,822	4.536	269,392	1.558	6.094	0.0429
140	32	15,394	72,072	27,959	447,344	4.682	269,392	1.661	6.342	0.0443
140	33	15,394	74,324	28,833	475,740	4.828	269,392	1.766	6.594	0.0457
140	34	15,394	76,577	29,706	505,009	4.975	269,392	1.875	6.849	0.0471
140	35	15,394	78,829	30,580	535,153	5.121	269,392	1.987	7.107	0.0485
140	36	15,394	81,081	31,454	566,170	5.267	269,392	2.102	7.369	0.0499
140	37	15,394	83,333	32,328	598,060	5.413	269,392	2.220	7.633	0.0513
140	38	15,394	85,586	33,201	630,825	5.560	269,392	2.342	7.901	0.0526
140	39	15,394	87,838	34,075	664,463	5.706	269,392	2.467	8.173	0.0540
140	40	15,394	90,090	34,949	698,975	5.852	269,392	2.595	8.447	0.0554
140	41	15,394	92,342	35,822	734,360	5.999	269,392	2.726	8.725	0.0568
140	42	15,394	94,595	36,696	770,620	6.145	269,392	2.861	9.006	0.0582
140	43	15,394	96,847	37,570	807,753	6.291	269,392	2.998	9.290	0.0596
140	44	15,394	99,099	38,444	845,759	6.438	269,392	3.140	9.577	0.0610
140	45	15,394	101,351	39,317	884,640	6.584	269,392	3.284	9.868	0.0623
140	46	15,394	103,604	40,191	924,394	6.730	269,392	3.431	10.162	0.0637
140	47	15,394	105,856	41,065	965,022	6.877	269,392	3.582	10.459	0.0651
140	48	15,394	108,108	41,938	1,006,524	7.023	269,392	3.736	10.759	0.0665
140	49	15,394	110,360	42,812	1,048,899	7.169	269,392	3.894	11.063	0.0679
140	50	15,394	112,613	43,686	1,092,148	7.315	269,392	4.054	11.370	0.0693
140	51	15,394	114,865	44,560	1,136,271	7.462	269,392	4.218	11.680	0.0707
140	52	15,394	117,117	45,433	1,181,267	7.608	269,392	4.385	11.993	0.0720

Table D-1
Bearing Stress Calculations (Continued)

Dia (ft) =	Height (ft)	Area (sqft)	Weight (K)	V (K)	OTM (ft-K)	P/A (ksf)	S (cf)	M/S (ksf)	Bearing (ksf)	e/D < 0.123
130	15	13,273	29,130	11,300	84,753	2.195	215,690	0.393	2.588	0.0224
130	16	13,273	31,072	12,054	96,430	2.341	215,690	0.447	2.788	0.0239
130	17	13,273	33,014	12,807	108,860	2.487	215,690	0.505	2.992	0.0254
130	18	13,273	34,956	13,560	122,044	2.634	215,690	0.566	3.199	0.0269
130	19	13,273	36,898	14,314	135,981	2.780	215,690	0.630	3.410	0.0283
130	20	13,273	38,840	15,067	150,672	2.926	215,690	0.699	3.625	0.0298
130	21	13,273	40,782	15,821	166,116	3.072	215,690	0.770	3.843	0.0313
130	22	13,273	42,724	16,574	182,313	3.219	215,690	0.845	4.064	0.0328
130	23	13,273	44,666	17,327	199,264	3.365	215,690	0.924	4.289	0.0343
130	24	13,273	46,608	18,081	216,967	3.511	215,690	1.006	4.517	0.0358
130	25	13,273	48,550	18,834	235,425	3.658	215,690	1.091	4.749	0.0373
130	26	13,273	50,492	19,587	254,635	3.804	215,690	1.181	4.985	0.0388
130	27	13,273	52,434	20,341	274,599	3.950	215,690	1.273	5.223	0.0403
130	28	13,273	54,376	21,094	295,317	4.097	215,690	1.369	5.466	0.0418
130	29	13,273	56,318	21,847	316,788	4.243	215,690	1.469	5.712	0.0433
130	30	13,273	58,260	22,601	339,012	4.389	215,690	1.572	5.961	0.0448
130	31	13,273	60,202	23,354	361,989	4.536	215,690	1.678	6.214	0.0463
130	32	13,273	62,144	24,107	385,720	4.682	215,690	1.788	6.470	0.0477
130	33	13,273	64,086	24,861	410,204	4.828	215,690	1.902	6.730	0.0492
130	34	13,273	66,028	25,614	435,442	4.975	215,690	2.019	6.993	0.0507
130	35	13,273	67,970	26,368	461,433	5.121	215,690	2.139	7.260	0.0522
130	36	13,273	69,912	27,121	488,177	5.267	215,690	2.263	7.530	0.0537
130	37	13,273	71,854	27,874	515,674	5.413	215,690	2.391	7.804	0.0552
130	38	13,273	73,796	28,628	543,925	5.560	215,690	2.522	8.082	0.0567
130	39	13,273	75,738	29,381	572,930	5.706	215,690	2.656	8.362	0.0582
130	40	13,273	77,680	30,134	602,687	5.852	215,690	2.794	8.647	0.0597
130	41	13,273	79,622	30,888	633,198	5.999	215,690	2.936	8.934	0.0612
130	42	13,273	81,564	31,641	664,463	6.145	215,690	3.081	9.226	0.0627
130	43	13,273	83,506	32,394	696,481	6.291	215,690	3.229	9.520	0.0642
130	44	13,273	85,448	33,148	729,252	6.438	215,690	3.381	9.819	0.0656
130	45	13,273	87,390	33,901	762,776	6.584	215,690	3.536	10.120	0.0671
130	46	13,273	89,332	34,655	797,054	6.730	215,690	3.695	10.426	0.0686
130	47	13,273	91,274	35,408	832,085	6.877	215,690	3.858	10.734	0.0701

Table D-1
Bearing Stress Calculations (Continued)

Dia (ft) =	Height (ft)	Area (sqft)	Weight (K)	V (K)	OTM (ft-K)	P/A (ksf)	S (cf)	M/S (ksf)	Bearing (ksf)	e/D < 0.123
120	15	11,310	24,821	9,629	72,216	2.195	169,646	0.426	2.620	0.0242
120	16	11,310	26,475	10,271	82,165	2.341	169,646	0.484	2.825	0.0259
120	17	11,310	28,130	10,913	92,757	2.487	169,646	0.547	3.034	0.0275
120	18	11,310	29,785	11,554	103,990	2.634	169,646	0.613	3.247	0.0291
120	19	11,310	31,440	12,196	115,866	2.780	169,646	0.683	3.463	0.0307
120	20	11,310	33,094	12,838	128,383	2.926	169,646	0.757	3.683	0.0323
120	21	11,310	34,749	13,480	141,542	3.072	169,646	0.834	3.907	0.0339
120	22	11,310	36,404	14,122	155,344	3.219	169,646	0.916	4.134	0.0356
120	23	11,310	38,058	14,764	169,787	3.365	169,646	1.001	4.366	0.0372
120	24	11,310	39,713	15,406	184,872	3.511	169,646	1.090	4.601	0.0388
120	25	11,310	41,368	16,048	200,599	3.658	169,646	1.182	4.840	0.0404
120	26	11,310	43,023	16,690	216,967	3.804	169,646	1.279	5.083	0.0420
120	27	11,310	44,677	17,332	233,978	3.950	169,646	1.379	5.330	0.0436
120	28	11,310	46,332	17,974	251,631	4.097	169,646	1.483	5.580	0.0453
120	29	11,310	47,987	18,616	269,926	4.243	169,646	1.591	5.834	0.0469
120	30	11,310	49,641	19,257	288,862	4.389	169,646	1.703	6.092	0.0485
120	31	11,310	51,296	19,899	308,440	4.536	169,646	1.818	6.354	0.0501
120	32	11,310	52,951	20,541	328,661	4.682	169,646	1.937	6.619	0.0517
120	33	11,310	54,606	21,183	349,523	4.828	169,646	2.060	6.889	0.0533
120	34	11,310	56,260	21,825	371,027	4.975	169,646	2.187	7.162	0.0550
120	35	11,310	57,915	22,467	393,173	5.121	169,646	2.318	7.438	0.0566
120	36	11,310	59,570	23,109	415,961	5.267	169,646	2.452	7.719	0.0582
120	37	11,310	61,224	23,751	439,391	5.413	169,646	2.590	8.003	0.0598
120	38	11,310	62,879	24,393	463,463	5.560	169,646	2.732	8.292	0.0614
120	39	11,310	64,534	25,035	488,177	5.706	169,646	2.878	8.584	0.0630
120	40	11,310	66,189	25,677	513,532	5.852	169,646	3.027	8.879	0.0647
120	41	11,310	67,843	26,319	539,530	5.999	169,646	3.180	9.179	0.0663
120	42	11,310	69,498	26,960	566,170	6.145	169,646	3.337	9.482	0.0679
120	43	11,310	71,153	27,602	593,451	6.291	169,646	3.498	9.789	0.0695
120	44	11,310	72,807	28,244	621,374	6.438	169,646	3.663	10.100	0.0711
120	45	11,310	74,462	28,886	649,940	6.584	169,646	3.831	10.415	0.0727
120	46	11,310	76,117	29,528	679,147	6.730	169,646	4.003	10.734	0.0744
120	47	11,310	77,772	30,170	708,996	6.877	169,646	4.179	11.056	0.0760

Table D-1
Bearing Stress Calculations (Continued)

Dia (ft) =	Height (ft)	Area (sqft)	Weight (K)	V (K)	OTM (ft-K)	P/A (ksf)	S (cf)	M/S (ksf)	Bearing (ksf)	e/D < 0.123
110	15	9,503	20,856	8,091	60,681	2.195	130,671	0.464	2.659	0.0264
110	16	9,503	22,247	8,630	69,042	2.341	130,671	0.528	2.869	0.0282
110	17	9,503	23,637	9,170	77,941	2.487	130,671	0.596	3.084	0.0300
110	18	9,503	25,028	9,709	87,381	2.634	130,671	0.669	3.302	0.0317
110	19	9,503	26,418	10,248	97,359	2.780	130,671	0.745	3.525	0.0335
110	20	9,503	27,808	10,788	107,877	2.926	130,671	0.826	3.752	0.0353
110	21	9,503	29,199	11,327	118,935	3.072	130,671	0.910	3.983	0.0370
110	22	9,503	30,589	11,867	130,532	3.219	130,671	0.999	4.218	0.0388
110	23	9,503	31,980	12,406	142,668	3.365	130,671	1.092	4.457	0.0406
110	24	9,503	33,370	12,945	155,344	3.511	130,671	1.189	4.700	0.0423
110	25	9,503	34,761	13,485	168,559	3.658	130,671	1.290	4.948	0.0441
110	26	9,503	36,151	14,024	182,313	3.804	130,671	1.395	5.199	0.0458
110	27	9,503	37,541	14,563	196,607	3.950	130,671	1.505	5.455	0.0476
110	28	9,503	38,932	15,103	211,440	4.097	130,671	1.618	5.715	0.0494
110	29	9,503	40,322	15,642	226,812	4.243	130,671	1.736	5.979	0.0511
110	30	9,503	41,713	16,182	242,724	4.389	130,671	1.858	6.247	0.0529
110	31	9,503	43,103	16,721	259,176	4.536	130,671	1.983	6.519	0.0547
110	32	9,503	44,493	17,260	276,166	4.682	130,671	2.113	6.795	0.0564
110	33	9,503	45,884	17,800	293,696	4.828	130,671	2.248	7.076	0.0582
110	34	9,503	47,274	18,339	311,766	4.975	130,671	2.386	7.360	0.0600
110	35	9,503	48,665	18,879	330,375	5.121	130,671	2.528	7.649	0.0617
110	36	9,503	50,055	19,418	349,523	5.267	130,671	2.675	7.942	0.0635
110	37	9,503	51,446	19,957	369,211	5.413	130,671	2.826	8.239	0.0652
110	38	9,503	52,836	20,497	389,438	5.560	130,671	2.980	8.540	0.0670
110	39	9,503	54,226	21,036	410,204	5.706	130,671	3.139	8.845	0.0688
110	40	9,503	55,617	21,575	431,510	5.852	130,671	3.302	9.155	0.0705
110	41	9,503	57,007	22,115	453,355	5.999	130,671	3.469	9.468	0.0723
110	42	9,503	58,398	22,654	475,740	6.145	130,671	3.641	9.786	0.0741
110	43	9,503	59,788	23,194	498,664	6.291	130,671	3.816	10.107	0.0758
110	44	9,503	61,179	23,733	522,127	6.438	130,671	3.996	10.433	0.0776
110	45	9,503	62,569	24,272	546,130	6.584	130,671	4.179	10.763	0.0793
110	46	9,503	63,959	24,812	570,672	6.730	130,671	4.367	11.097	0.0811
110	47	9,503	65,350	25,351	595,753	6.877	130,671	4.559	11.436	0.0829

Table D-1
Bearing Stress Calculations (Continued)

100	15	7,854	17,237	6,687	50,150	2.195	98,175	0.511	2.705	0.0291
100	16	7,854	18,386	7,132	57,059	2.341	98,175	0.581	2.922	0.0310
100	17	7,854	19,535	7,578	64,414	2.487	98,175	0.656	3.143	0.0330
100	18	7,854	20,684	8,024	72,216	2.634	98,175	0.736	3.369	0.0349
100	19	7,854	21,833	8,470	80,462	2.780	98,175	0.820	3.599	0.0369
100	20	7,854	22,982	8,915	89,155	2.926	98,175	0.908	3.834	0.0388
100	21	7,854	24,131	9,361	98,293	3.072	98,175	1.001	4.074	0.0407
100	22	7,854	25,280	9,807	107,877	3.219	98,175	1.099	4.318	0.0427
100	23	7,854	26,429	10,253	117,907	3.365	98,175	1.201	4.566	0.0446
100	24	7,854	27,579	10,699	128,383	3.511	98,175	1.308	4.819	0.0466
100	25	7,854	28,728	11,144	139,305	3.658	98,175	1.419	5.077	0.0485
100	26	7,854	29,877	11,590	150,672	3.804	98,175	1.535	5.339	0.0504
100	27	7,854	31,026	12,036	162,485	3.950	98,175	1.655	5.605	0.0524
100	28	7,854	32,175	12,482	174,744	4.097	98,175	1.780	5.877	0.0543
100	29	7,854	33,324	12,927	187,448	4.243	98,175	1.909	6.152	0.0563
100	30	7,854	34,473	13,373	200,599	4.389	98,175	2.043	6.433	0.0582
100	31	7,854	35,622	13,819	214,195	4.536	98,175	2.182	6.717	0.0601
100	32	7,854	36,771	14,265	228,237	4.682	98,175	2.325	7.007	0.0621
100	33	7,854	37,921	14,711	242,724	4.828	98,175	2.472	7.301	0.0640
100	34	7,854	39,070	15,156	257,658	4.975	98,175	2.624	7.599	0.0659
100	35	7,854	40,219	15,602	273,037	5.121	98,175	2.781	7.902	0.0679
100	36	7,854	41,368	16,048	288,862	5.267	98,175	2.942	8.209	0.0698
100	37	7,854	42,517	16,494	305,133	5.413	98,175	3.108	8.521	0.0718
100	38	7,854	43,666	16,939	321,849	5.560	98,175	3.278	8.838	0.0737
100	39	7,854	44,815	17,385	339,012	5.706	98,175	3.453	9.159	0.0756
100	40	7,854	45,964	17,831	356,620	5.852	98,175	3.632	9.485	0.0776
100	41	7,854	47,113	18,277	374,674	5.999	98,175	3.816	9.815	0.0795
100	42	7,854	48,263	18,723	393,173	6.145	98,175	4.005	10.150	0.0815
100	43	7,854	49,412	19,168	412,119	6.291	98,175	4.198	10.489	0.0834
100	44	7,854	50,561	19,614	431,510	6.438	98,175	4.395	10.833	0.0853
100	45	7,854	51,710	20,060	451,347	6.584	98,175	4.597	11.181	0.0873
100	46	7,854	52,859	20,506	471,630	6.730	98,175	4.804	11.534	0.0892
100	47	7,854	54,008	20,951	492,358	6.877	98,175	5.015	11.892	0.0912

Table D-1
Bearing Stress Calculations (Continued)

90	15	6,362	13,962	5,416	40,621	2.195	71,569	0.568	2,762	0.0323
90	16	6,362	14,892	5,777	46,218	2.341	71,569	0.646	2,987	0.0345
90	17	6,362	15,823	6,138	52,176	2.487	71,569	0.729	3,216	0.0366
90	18	6,362	16,754	6,499	58,495	2.634	71,569	0.817	3,451	0.0388
90	19	6,362	17,685	6,860	65,174	2.780	71,569	0.911	3,691	0.0409
90	20	6,362	18,616	7,222	72,216	2.926	71,569	1.009	3,935	0.0431
90	21	6,362	19,546	7,583	79,618	3.072	71,569	1.112	4,185	0.0453
90	22	6,362	20,477	7,944	87,381	3.219	71,569	1.221	4,440	0.0474
90	23	6,362	21,408	8,305	95,505	3.365	71,569	1.334	4,700	0.0496
90	24	6,362	22,339	8,666	103,990	3.511	71,569	1.453	4,964	0.0517
90	25	6,362	23,269	9,027	112,837	3.658	71,569	1.577	5,234	0.0539
90	26	6,362	24,200	9,388	122,044	3.804	71,569	1.705	5,509	0.0560
90	27	6,362	25,131	9,749	131,613	3.950	71,569	1.839	5,789	0.0582
90	28	6,362	26,062	10,110	141,542	4.097	71,569	1.978	6,074	0.0603
90	29	6,362	26,993	10,471	151,833	4.243	71,569	2.121	6,364	0.0625
90	30	6,362	27,923	10,832	162,485	4.389	71,569	2.270	6,660	0.0647
90	31	6,362	28,854	11,193	173,498	4.536	71,569	2.424	6,960	0.0668
90	32	6,362	29,785	11,554	184,872	4.682	71,569	2.583	7,265	0.0690
90	33	6,362	30,716	11,916	196,607	4.828	71,569	2.747	7,575	0.0711
90	34	6,362	31,646	12,277	208,703	4.975	71,569	2.916	7,891	0.0733
90	35	6,362	32,577	12,638	221,160	5.121	71,569	3.090	8,211	0.0754
90	36	6,362	33,508	12,999	233,978	5.267	71,569	3.269	8,536	0.0776
90	37	6,362	34,439	13,360	247,158	5.413	71,569	3.453	8,867	0.0797
90	38	6,362	35,370	13,721	260,698	5.560	71,569	3.643	9,202	0.0819
90	39	6,362	36,300	14,082	274,599	5.706	71,569	3.837	9,543	0.0841
90	40	6,362	37,231	14,443	288,862	5.852	71,569	4.036	9,888	0.0862
90	41	6,362	38,162	14,804	303,486	5.999	71,569	4.240	10,239	0.0884
90	42	6,362	39,093	15,165	318,470	6.145	71,569	4.450	10,595	0.0905
90	43	6,362	40,023	15,526	333,816	6.291	71,569	4.664	10,956	0.0927
90	44	6,362	40,954	15,887	349,523	6.438	71,569	4.884	11,321	0.0948
90	45	6,362	41,885	16,248	365,591	6.584	71,569	5.108	11,692	0.0970
90	46	6,362	42,816	16,610	382,020	6.730	71,569	5.338	12,068	0.0991
90	47	6,362	43,747	16,971	398,810	6.877	71,569	5.572	12,449	0.1013

Table D-1
Bearing Stress Calculations (Continued)

80	10	5,027	7,354	2,853	14,265	1.463	50,265	0.284	1.747	0.0242
80	11	5,027	8,090	3,138	17,260	1.609	50,265	0.343	1.953	0.0267
80	12	5,027	8,825	3,424	20,541	1.756	50,265	0.409	2.164	0.0291
80	13	5,027	9,561	3,709	24,107	1.902	50,265	0.480	2.382	0.0315
80	14	5,027	10,296	3,994	27,959	2.048	50,265	0.556	2.605	0.0339
80	15	5,027	11,031	4,279	32,096	2.195	50,265	0.639	2.833	0.0364
80	16	5,027	11,767	4,565	36,518	2.341	50,265	0.726	3.067	0.0388
80	17	5,027	12,502	4,850	41,225	2.487	50,265	0.820	3.307	0.0412
80	18	5,027	13,238	5,135	46,218	2.634	50,265	0.919	3.553	0.0436
80	19	5,027	13,973	5,421	51,496	2.780	50,265	1.024	3.804	0.0461
80	20	5,027	14,709	5,706	57,059	2.926	50,265	1.135	4.061	0.0485
80	21	5,027	15,444	5,991	62,908	3.072	50,265	1.252	4.324	0.0509
80	22	5,027	16,179	6,277	69,042	3.219	50,265	1.374	4.592	0.0533
80	23	5,027	16,915	6,562	75,461	3.365	50,265	1.501	4.866	0.0558
80	24	5,027	17,650	6,847	82,165	3.511	50,265	1.635	5.146	0.0582
80	25	5,027	18,386	7,132	89,155	3.658	50,265	1.774	5.431	0.0606
80	26	5,027	19,121	7,418	96,430	3.804	50,265	1.918	5.722	0.0630
80	27	5,027	19,857	7,703	103,990	3.950	50,265	2.069	6.019	0.0655
80	28	5,027	20,592	7,988	111,836	4.097	50,265	2.225	6.322	0.0679
80	29	5,027	21,327	8,274	119,967	4.243	50,265	2.387	6.630	0.0703
80	30	5,027	22,063	8,559	128,383	4.389	50,265	2.554	6.943	0.0727
80	31	5,027	22,798	8,844	137,085	4.536	50,265	2.727	7.263	0.0752
80	32	5,027	23,534	9,129	146,071	4.682	50,265	2.906	7.588	0.0776
80	33	5,027	24,269	9,415	155,344	4.828	50,265	3.090	7.919	0.0800
80	34	5,027	25,005	9,700	164,901	4.975	50,265	3.281	8.255	0.0824
80	35	5,027	25,740	9,985	174,744	5.121	50,265	3.476	8.597	0.0849
80	36	5,027	26,475	10,271	184,872	5.267	50,265	3.678	8.945	0.0873
80	37	5,027	27,211	10,556	195,285	5.413	50,265	3.885	9.299	0.0897
80	38	5,027	27,946	10,841	205,984	5.560	50,265	4.098	9.658	0.0921
80	39	5,027	28,682	11,127	216,967	5.706	50,265	4.316	10.022	0.0946
80	40	5,027	29,417	11,412	228,237	5.852	50,265	4.541	10.393	0.0970
80	41	5,027	30,153	11,697	239,791	5.999	50,265	4.770	10.769	0.0994
80	42	5,027	30,888	11,982	251,631	6.145	50,265	5.006	11.151	0.1018

Table D-1
Bearing Stress Calculations (Continued)

70	10	3,848	5,631	2,184	10,921	1.463	33,674	0.324	1.787	0.0277
70	11	3,848	6,194	2,403	13,215	1.609	33,674	0.392	2.002	0.0305
70	12	3,848	6,757	2,621	15,727	1.756	33,674	0.467	2.223	0.0333
70	13	3,848	7,320	2,840	18,457	1.902	33,674	0.548	2.450	0.0360
70	14	3,848	7,883	3,058	21,406	2.048	33,674	0.636	2.684	0.0388
70	15	3,848	8,446	3,276	24,573	2.195	33,674	0.730	2.924	0.0416
70	16	3,848	9,009	3,495	27,959	2.341	33,674	0.830	3.171	0.0443
70	17	3,848	9,572	3,713	31,563	2.487	33,674	0.937	3.425	0.0471
70	18	3,848	10,135	3,932	35,386	2.634	33,674	1.051	3.684	0.0499
70	19	3,848	10,698	4,150	39,427	2.780	33,674	1.171	3.951	0.0526
70	20	3,848	11,261	4,369	43,686	2.926	33,674	1.297	4.224	0.0554
70	21	3,848	11,824	4,587	48,164	3.072	33,674	1.430	4.503	0.0582
70	22	3,848	12,387	4,805	52,860	3.219	33,674	1.570	4.789	0.0610
70	23	3,848	12,950	5,024	57,775	3.365	33,674	1.716	5.081	0.0637
70	24	3,848	13,514	5,242	62,908	3.511	33,674	1.868	5.380	0.0665
70	25	3,848	14,077	5,461	68,259	3.658	33,674	2.027	5.685	0.0693
70	26	3,848	14,640	5,679	73,829	3.804	33,674	2.192	5.997	0.0720
70	27	3,848	15,203	5,898	79,618	3.950	33,674	2.364	6.315	0.0748
70	28	3,848	15,766	6,116	85,624	4.097	33,674	2.543	6.639	0.0776
70	29	3,848	16,329	6,334	91,850	4.243	33,674	2.728	6.971	0.0804
70	30	3,848	16,892	6,553	98,293	4.389	33,674	2.919	7.308	0.0831
70	31	3,848	17,455	6,771	104,955	4.536	33,674	3.117	7.652	0.0859
70	32	3,848	18,018	6,990	111,836	4.682	33,674	3.321	8.003	0.0887
70	33	3,848	18,581	7,208	118,935	4.828	33,674	3.532	8.360	0.0914
70	34	3,848	19,144	7,427	126,252	4.975	33,674	3.749	8.724	0.0942
70	35	3,848	19,707	7,645	133,788	5.121	33,674	3.973	9.094	0.0970
70	36	3,848	20,270	7,863	141,542	5.267	33,674	4.203	9.470	0.0998
70	37	3,848	20,833	8,082	149,515	5.413	33,674	4.440	9.854	0.1025
70	38	3,848	21,396	8,300	157,706	5.560	33,674	4.683	10.243	0.1053
70	39	3,848	21,959	8,519	166,116	5.706	33,674	4.933	10.639	0.1081
70	40	3,848	22,523	8,737	174,744	5.852	33,674	5.189	11.042	0.1108
70	41	3,848	23,086	8,956	183,590	5.999	33,674	5.452	11.451	0.1136
70	42	3,848	23,649	9,174	192,655	6.145	33,674	5.721	11.866	0.1164

E

STEEL SENSITIZATION

If stainless steel is heated above 500°C (932°F) there is a risk that chrome may react with carbon to form chrome carbides. These chrome carbides deplete the surrounding areas of chrome, which decreases the corrosion resistance of the steel. This phenomenon is called sensitization. Over time, this leads to preferential corrosion of the affected area, which can lead to cracking and possible failure of the steel.

This phenomenon was investigated in an article that was published in a Nippon Steel Technical Report in 2004.³⁰ The most relevant piece of information provided in this article is a graph indicating the temperatures and conditions at which sensitization occurs.

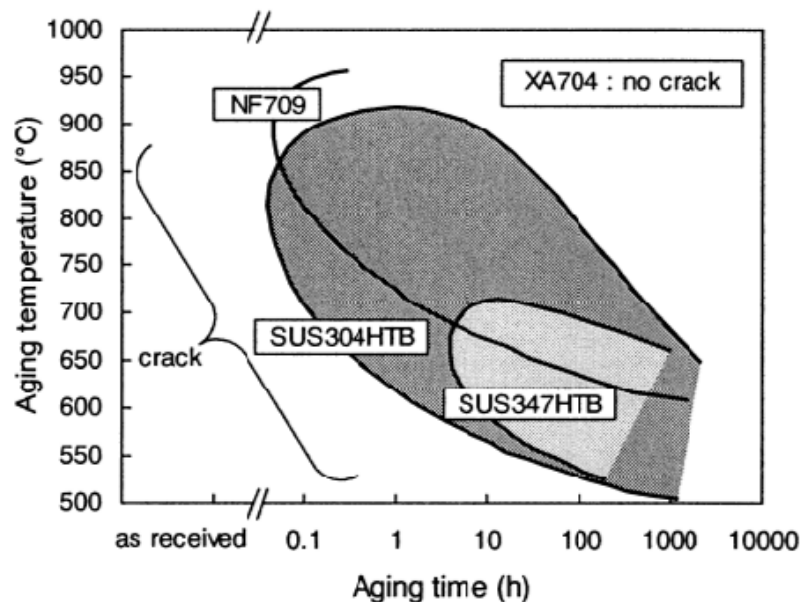


Figure E-1
Steel Sensitization

The thermocline process operates at 560°C (1040 °F). At these conditions, Stainless Steel 347 (SUS347HTB in Figure E-1) became sensitized after approximately 20 hours of exposure. For this reason, Black & Veatch selected Inconel 625 for the high temperature thermocline process. The addition of significant amounts of nickel, chrome, niobium, and titanium decrease the sensitization potential of the metal.

³⁰ “Development of New Austenitic Stainless Steel Boiler Tube with High Strength at Elevated Temperatures and Intergranular Corrosion Resistance.” Nippon Steel Technical Report No 90,; July 2004.

F

EQUIPMENT LIST

Table F-1
Equipment List Summary

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
Equipment List Summary

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Design Case	Working Storage (MWh)	# Tanks	Tank ID (ft)	Tank Height (ft)	Surge Tank ID (ft)	Surge Tank Height (ft)	Hot Salt Flow Rate (lb/hr)	Number of Operating Hot Pumps	Hot Pump Motor Size (hp)	Hot Pump ΔP (psia)	Cold Salt Flow Rate (lb/hr)	Number of Operating Cold Pumps	Cold Pump Motor Size (hp)	Cold Pump ΔP (psia)	Cold Salt Storage Temperature (°F)	Hot Salt Storage Temperature (°F)	Preheating Blower Power (hp)	Preheating Fired Heater Duty (MMBtu/h)
1	100	1	48	40	4.0	10.0	1,488,444	1	75	60	826,913	1	60	60	559	752	290	8.3
2	100	1	31	40	5.0	8.0	596,493	1	30	60	331,385	1	200	260	559	1040	125	3.4
3	500	1	107	40	7.5	10.0	7,442,220	3	125	60	4,134,567	3	125	60	559	752	1400	39.9
4	500	1	69	40	8.0	10.0	2,982,463	1	150	60	1,656,924	1	900	260	559	1040	550	15.6
5	1000	1	152	40	11.0	10.0	14,884,441	5	150	60	8,269,134	5	150	60	559	752	2800	79.8
6	1000	1	98	40	11.0	10.0	5,964,926	2	150	60	3,313,848	2	800	260	559	1040	1100	31.3
7	3000	2	160	50	20.0	16.0	44,653,323	13	200	60	24,807,402	12	200	60	559	752	2800	79.8
8	3000	1	155	47	14.0	16.0	17,894,777	6	150	60	9,941,543	5	900	260	559	1040	1700	47.6

Table F-2
Equipment List Summary – 100 MWh Indirect Trough

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
PRELIMINARY Equipment List
Design Case 1: 100MWh Storage at 752°F

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	1	Vertical	TL-TL = 10'	ID = 4'	Carbon Steel			Atm	800 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 17" of mineral wool.
V-102	Cold Molten Salt Surge Tank	1	Vertical	TL-TL = 10'	ID = 4'	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	1	API-650	TL-TL = 40'	ID = 48'	Carbon Steel			Atm	800 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 17" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 6'	ID = 9'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
E-101A/B	Hot Oil/Molten Salt Exchanger	2	Plate and Frame	Duty		Carbon Steel			100.0	800°F	Fouling duty to be included in area calculation by vendor.
				56.9 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				200 kW							
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				8.3 MMBtu/h							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	2	Vertical Submerged	1661 gpm	dP = 60 psi	By vendor	75.0		65.0	800 °F	1 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	2	Vertical Submerged	1590 gpm	dP = 60 psi	By vendor	60.0		65.0	600 °F	1 operating + 1 installed spare
P-103	Salt Charging Pump	2	Vertical Submerged	225 gpm	dP = 60 psi	By vendor	15.0		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	22.0 MMSCFD	dp = 10 inH ₂ O	Carbon Steel	290.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing	4015 ton		Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt	1230 ton		60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.
M-103	Heat Transfer Oil								150.0	800°F	Exact oil specification to be selected at a later date.

Table F-3
Equipment List Summary – 100 MWh Direct Central Receiver

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
PRELIMINARY Equipment List
Design Case 2: 100MWh Storage at 1040°F

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	1	Vertical	TL-TL = 8'	ID = 5'	Inconel 625			Atm	1050 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 23" of mineral wool.
V-102	Cold Molten Salt Surge Tank	1	Vertical	TL-TL = 8'	ID = 5'	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	1	API-650	TL-TL = 40'	ID = 31'	Inconel 625			Atm	1050 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 23" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 6'	ID = 6'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				4.6 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				75 kW							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	2	Vertical Submerged	709 gpm	dP = 60 psi	By vendor	30.0		65.0	1050 °F	1 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	2	Vertical Submerged	637 gpm	dP = 260 psi	By vendor	200.0		265.0	600 °F	1 operating + 1 installed spare
P-103	Salt Charging Pump	1	Vertical Submerged	94 gpm	dP = 60 psi	By vendor	7.5		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	3.4 MMSCFD	dp = 10 inH ₂ O	Carbon Steel	125.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing	1700 ton		Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt	480 ton		60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.

Table F-4
Equipment List Summary – 500 MWh Indirect Trough

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
PRELIMINARY Equipment List
Design Case 3: 500MWh Storage at 752°F

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	1	Vertical	TL-TL = 10'	ID = 7' 6"	Carbon Steel			Atm	800 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 17" of mineral wool.
V-102	Cold Molten Salt Surge Tank	1	Vertical	TL-TL = 10'	ID = 7' 6"	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	1	API-650	TL-TL = 40'	ID = 107'	Carbon Steel			Atm	800 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 17" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 10'	ID = 15'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
E-101A/B	Hot Oil/Molten Salt Exchanger	2	Plate and Frame	Duty		Carbon Steel			100.0	800°F	Fouling duty to be included in area calculation by vendor.
				284.3 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				900 kW							
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				39.9 MMBtu/h							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	4	Vertical Submerged	2767 gpm	dP = 60 psi	By vendor	125.0		65.0	800 °F	3 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	4	Vertical Submerged	2649 gpm	dP = 60 psi	By vendor	125.0		65.0	600 °F	3 operating + 1 installed spare
P-103	Salt Charging Pump	1	Vertical Submerged	1124 gpm	dP = 60 psi	By vendor	50.0		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	106 MMSCFD	dp = 10 in H ₂ O	Carbon Steel	290.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing	20000 ton		Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt	6100 ton		60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.
M-103	Heat Transfer Oil								150.0	800°F	Exact oil specification to be selected at a later date.

Table F-5
Equipment List Summary – 500 MWh, Direct Central Receiver

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
PRELIMINARY Equipment List
Design Case 4: 500MWh Storage at 1040°F

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	1	Vertical	TL-TL = 8'	ID = 10'	Inconel 625			Atm	1050 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 23" of mineral wool.
V-102	Cold Molten Salt Surge Tank	1	Vertical	TL-TL = 8'	ID = 10'	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	1	API-650	TL-TL = 40'	ID = 69'	Inconel 625			Atm	1050 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 23" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 10'	ID = 10'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				15.6 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				350 kW							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	2	Vertical Submerged	3545 gpm	dP = 60 psi	By vendor	150.0		65.0	1050 °F	1 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	2	Vertical Submerged	3185 gpm	dP = 260 psi	By vendor	900.0		265.0	600 °F	1 operating + 1 installed spare
P-103	Salt Charging Pump	1	Vertical Submerged	472 gpm	dP = 60 psi	By vendor	25.0		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	41.6 MMSCFD	dp = 10 inH ₂ O	Carbon Steel	550.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing		8435 ton	Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt		2400 ton	60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.

Table F-6
Equipment List Summary – 1000 MWh Indirect Trough

Barstow, California
Project Number 165470.0010

PRELIMINARY Equipment List
Design Case 5: 1000MWh Storage at 752°F

Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	1	Vertical	TL-TL = 10'	ID = 11'	Carbon Steel			Atm	800 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 17" of mineral wool.
V-102	Cold Molten Salt Surge Tank	1	Vertical	TL-TL = 10'	ID = 11'	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	1	API-650	TL-TL = 40'	ID = 152'	Carbon Steel			Atm	800 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 17" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 10'	ID = 21'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
E-101A/B	Hot Oil/Molten Salt Exchanger	2	Plate and Frame	Duty		Carbon Steel			100.0	800°F	Fouling duty to be included in area calculation by vendor.
				568.7 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				1800 kW							
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				79.8 MMBtu/h							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	6	Vertical Submerged	3321 gpm	dP = 60 psi	By vendor	150.0		65.0	800 °F	5 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	6	Vertical Submerged	3179 gpm	dP = 60 psi	By vendor	150.0		65.0	600 °F	5 operating + 1 installed spare
P-103	Salt Charging Pump	1	Vertical Submerged	2248 gpm	dP = 60 psi	By vendor	100.0		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	211.9 MMSCFD	dp = 10 in H ₂ O	Carbon Steel	290.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing		40200 ton	Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt		12120 ton	60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.
M-103	Heat Transfer Oil								150.0	800°F	Exact oil specification to be selected at a later date.

Table F-7
Equipment List Summary – 1000 MWh, Direct Central Receiver

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
PRELIMINARY Equipment List
Design Case 6: 1000MWh Storage at 1040°F

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	1	Vertical	TL-TL = 8'	ID = 11'	Inconel 625			Atm	1050 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 23" of mineral wool.
V-102	Cold Molten Salt Surge Tank	1	Vertical	TL-TL = 8'	ID = 11'	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	1	API-650	TL-TL = 40'	ID = 98'	Inconel 625			Atm	1050 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 23" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 10'	ID = 14'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				31.3 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				700 kW							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	3	Vertical Submerged	3185 gpm	dP = 60 psi	By vendor	150.0		65.0	1050 °F	2 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	3	Vertical Submerged	3545 gpm	dP = 260 psi	By vendor	800.0		265.0	600 °F	2 operating + 1 installed spare
P-103	Salt Charging Pump	1	Vertical Submerged	944 gpm	dP = 60 psi	By vendor	50.0		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	83.1 MMSCFD	dp = 10 inH ₂ O	Carbon Steel	1100.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing		16900 ton	Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt		4740 ton	60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.

Table F-8
Equipment List Summary – 3000 MWh Indirect Trough

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
PRELIMINARY Equipment List
Design Case 7: 3000MWh Storage at 752°F

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	3	Vertical	TL-TL = 16'	ID = 20'	Carbon Steel			Atm	800 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 17" of mineral wool.
V-102	Cold Molten Salt Surge Tank	3	Vertical	TL-TL = 16'	ID = 20'	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	3	API-650	TL-TL = 50'	ID = 160'	Carbon Steel			Atm	800 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 17" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 10'	ID = 21'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
E-101A/B	Hot Oil/Molten Salt Exchanger	2	Plate and Frame	Duty		Carbon Steel			100.0	800°F	Fouling duty to be included in area calculation by vendor.
				1,706 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				1800 kW							
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				220.9 MMBtu/h							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	14	Vertical Submerged	3832 gpm	dP = 60 psi	By vendor	200.0		65.0	800 °F	13 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	13	Vertical Submerged	3974 gpm	dP = 60 psi	By vendor	200.0		65.0	600 °F	12 operating + 1 installed spare
P-103	Salt Charging Pump	1	Vertical Submerged	2248 gpm	dP = 60 psi	By vendor	100.0		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	211.9 MMSCFD	dp = 10 in H ₂ O	Carbon Steel	290.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing		120465 ton	Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt		36261 ton	60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.
M-103	Heat Transfer Oil								150.0	800°F	Exact oil specification to be selected at a later date.

Table F-9
Equipment List Summary – 3000 MWh, Direct Central Receiver

EPRI Thermocline Study
Barstow, California
Project Number 165470.0010

BLACK & VEATCH
PRELIMINARY Equipment List
Design Case 8: 3000MWh Storage at 1040°F

Date: 01/13/10
Rev: C
By: RJH
Checked: MAL

Tag	Service	Qty	Type	Size			Electrical		Mech. Design		Notes
				Parameter 1	Parameter 2	MOC	Oper. (HP)	Conn. (HP)	Pres (psig)	Temp (F)	
VESSELS & TANKS											
V-101	Hot Molten Salt Surge Tank	2	Vertical	TL-TL = 16'	ID = 11'	Inconel 625			Atm	1050 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 23" of mineral wool.
V-102	Cold Molten Salt Surge Tank	2	Vertical	TL-TL = 16'	ID = 11'	Carbon Steel			Atm	600 °F	Install internal baffles to minimize fluid turbulence in tank. Insulated with 13" of mineral wool.
V-103	Thermocline Storage Tank	1	API-650	TL-TL = 47'	ID = 155'	Inconel 625			Atm	1050 °F	Contains 36 ft of packed quartz and quartz sand, void fraction = 0.25. Insulated with 23" of mineral wool.
V-104	Salt Melting Sump	1	Vertical	TL-TL = 10'	ID = 17'	Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
HEAT EXCHANGERS & FIRED EQUIPMENT											
F-101	Quartz Preheater	1	Gas Fired Heater	Duty		Carbon Steel			15.0	600 °F	Temporary Service: Fired heater used to heat air. Used to preheat the thermocline quartz before adding salt.
				47.6 MMBtu/h							
E-102	Salt Melting Heater	1	Electric	Duty		Carbon Steel			Atm	500 °F	Temporary Service. Used to melt salt for initial charging and any required upkeep.
				1100 kW							
ROTATING EQUIPMENT											
P-101 A/B	Hot Molten Salt Pump	7	Vertical Submerged	3545 gpm	dP = 60 psi	By vendor	200.0		65.0	1050 °F	6 operating + 1 installed spare
P-102 A/B	Cold Molten Salt Pump	6	Vertical Submerged	3822 gpm	dP = 260 psi	By vendor	900.0		265.0	600 °F	5 operating + 1 installed spare
P-103	Salt Charging Pump	1	Vertical Submerged	1416 gpm	dP = 60 psi	By vendor	60.0		65.0	500 °F	Temporary Service: 1 operating and 1 warehouse spare. Used during initial charging.
B-101	Quartz Preheating Blower	1	Rotary	118.2 MMSCFD	dp = 10 inH ₂ O	Carbon Steel	1700.0		15.0	150 °F	Temporary Service: Used to preheat the thermocline quartz before adding salt.
Tank Packing and Process Chemicals											
M-101	Quartzite Packing		41810 ton	Void Fraction: 0.25	Density: 165 lb/ft3	By vendor			1050.0		Packing to consist of a 50:50 mixture of 1-2" quartzite spheres mixed with a fine quartzite sand.
M-102	Molten Salt		11739 ton	60:40 wt NaNO3:KNO3	Standard Grade	By vendor			1050.0		Sufficient molten salt to be provided to fill V-103 to a normal liquid level of 39 feet.

G

COMPLETE DESIGN ESTIMATE

The cost estimates indicate that the proposed thermocline thermal energy storage system has the potential to significantly reduce TES costs compared to a two-tank storage system. There is a wide cost range that depends largely on the scale of the project and the operating temperature. The cost estimates developed in this study include both direct and indirect costs with a contingency of 15% and sales tax. They are presented in January 2010 dollars.

The results in Figure G-1 and Figure G-2 represent the total installed capital costs (direct and indirect) for the thermocline and two-tank systems, respectively, including 15% contingency and sales tax. Table G-1, and Table G-2 show the detailed breakdown of these costs. Values for the systems not included in the tables were calculated through linear interpolation or linear extrapolation.

Estimates provided in Table G-1 and Table G-2 have an uncertainty of approximately +40%/-20% in accordance with AACE Class 4 estimate. The uncertainty with these estimates can be reduced with additional design or demonstration of the concepts identified in this report. Detailed modeling, design or demonstration of the following items will likely be necessary to confirm the conceptual designs presented here:

- Dynamic modeling of thermocline
- Thermal ratcheting
- Tank materials
- Distributor design
- Other tank internals, including structural supports

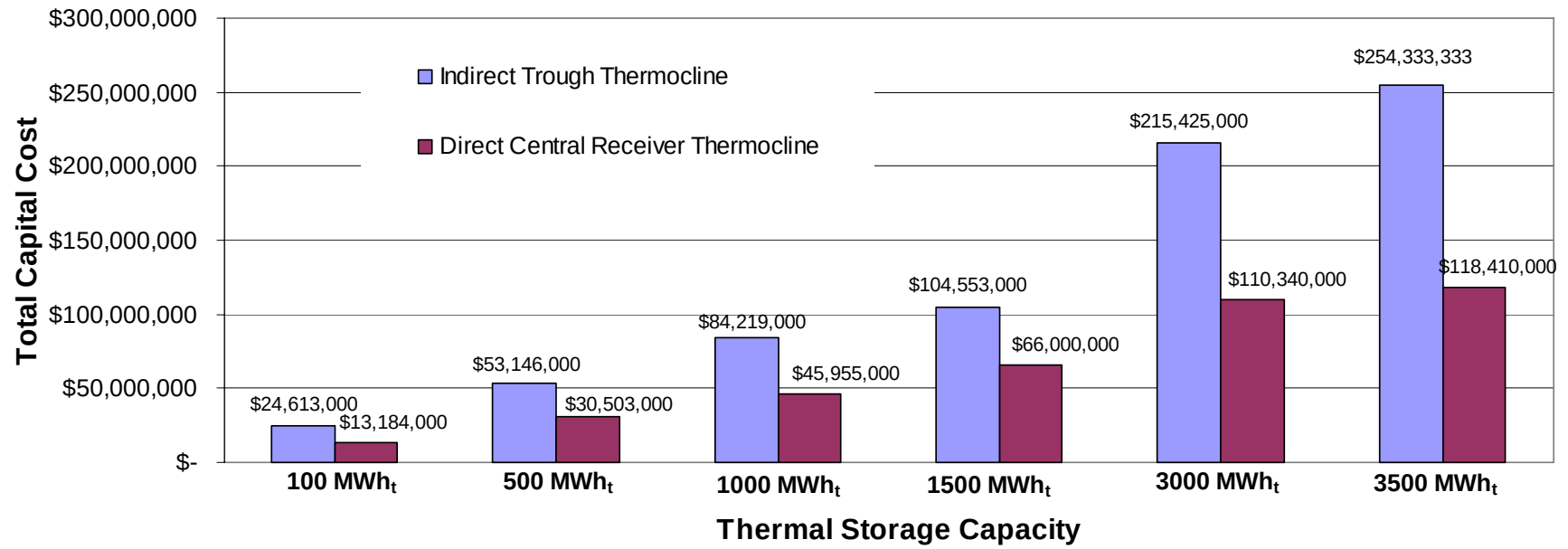


Figure G-1
Total Capital Cost (Direct & Indirect) for Thermocline Design Cases

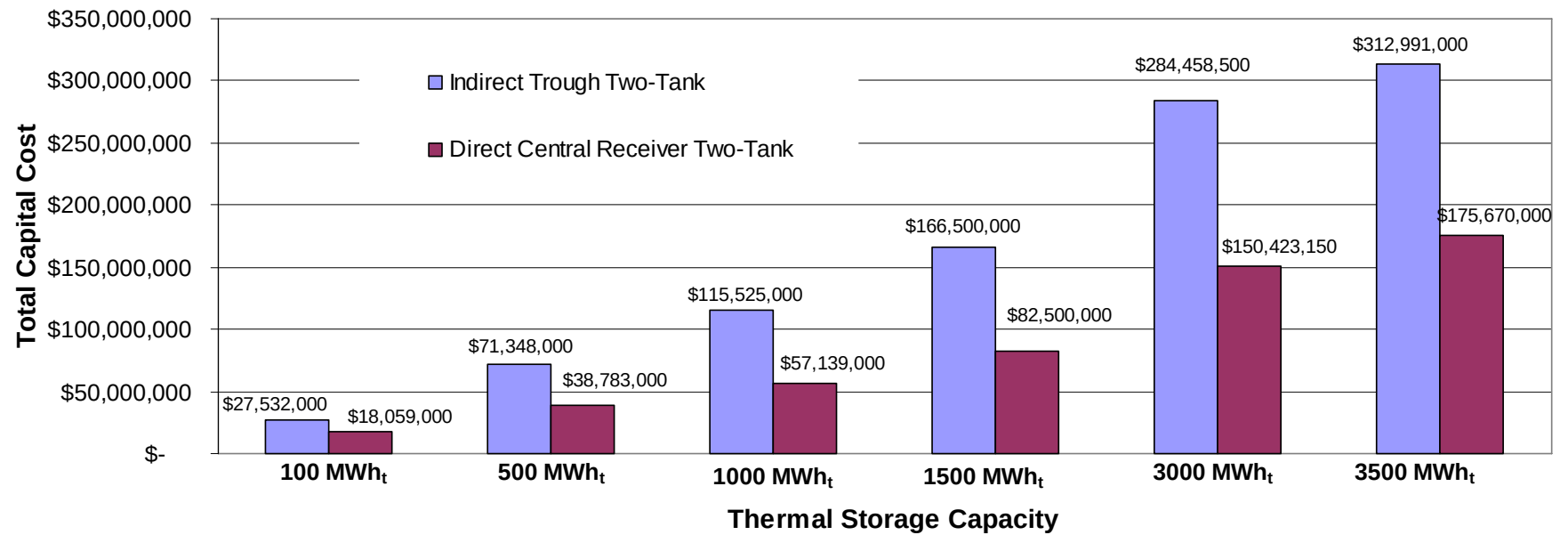


Figure G-2
Total Capital Cost (Direct & Indirect) for Two-Tank Design Cases

Table G-1
Capital Costs for Thermocline Tank Designs

Thermocline Design Estimate										
Design Case	100 MWh, Indirect Trough	100 MWh, Direct Central Receiver	500 MWh, Indirect Trough	500 MWh, Direct Central Receiver	1000 MWh, Indirect Trough	1000 MWh, Direct Central Receiver	3000 MWh, Indirect Trough	3000 MWh, Direct Central Receiver	1500 MWh, Indirect Trough	3500 MWh, Direct Central Receiver
(Thermocline Tank ID, Height - Number of Tanks)	(48' ID, 40' T/T -1)	(31' ID, 40' T/T -1)	(107' ID, 40' T/T -1)	(69' ID, 40' T/T -1)	(152' ID, 40' T/T -1)	(98 ID, 40' T/T -1)	(150 ID, 50' T/T -2)	(155' ID, 47' T/T - 1)	(160' ID, 50' T/T - 1)	(156' ID, 50' T/T - 1)
Direct										
Foundations	\$676,000	\$342,000	\$2,670,000	\$1,240,000	\$5,060,000	\$2,280,000	\$12,000,000	\$6,560,000	\$5,600,000	\$6,985,000
Platforms & Steel	\$1,117,000	\$1,095,000	\$1,195,000	\$1,145,000	\$1,255,000	\$1,182,000	\$1,700,000	\$1,641,000	\$1,400,000	\$1,700,000
Storage Tanks	\$5,835,000	\$1,715,000	\$7,765,000	\$7,900,000	\$8,560,000	\$11,700,000	\$25,162,000	\$31,130,000	\$8,700,000	\$36,900,000
Storage Tank Insulation	\$291,000	\$253,000	\$776,000	\$606,000	\$1,270,000	\$944,000	\$1,890,000	\$830,000	\$1,500,000	\$1,200,000
Heat Exchangers	\$2,930,000	\$-	\$10,100,000	\$-	\$17,500,000	\$-	\$47,500,000	\$-	\$26,300,000	\$-
Quartzite Preheating Equipment	\$395,000	\$215,000	\$1,250,000	\$636,000	\$2,490,000	\$1,100,000	\$5,070,000	\$2,360,000	\$2,700,000	\$2,500,000
Surge Tanks	\$26,000	\$32,000	\$84,000	\$95,000	\$180,000	\$240,000	\$390,000	\$450,000	\$190,000	\$550,000
Pumps & PCE	\$1,200,000	\$1,170,000	\$2,430,000	\$2,590,000	\$4,400,000	\$3,890,000	\$10,500,000	\$8,670,000	\$5,100,000	\$9,100,000
Salt Melting System	\$1,520,000	\$1,420,000	\$2,960,000	\$1,680,000	\$3,230,000	\$1,990,000	\$4,270,000	\$3,320,000	\$3,200,000	\$3,400,000
Interconnecting Piping & Valves	\$514,000	\$464,000	\$945,000	\$1,020,000	\$1,820,000	\$1,380,000	\$4,000,000	\$2,750,000	\$1,950,000	\$2,800,000
Electrical	\$238,000	\$179,000	\$238,000	\$242,000	\$316,000	\$298,000	\$1,830,000	\$732,000	\$340,000	\$800,000
Instrumentation & Controls	\$293,000	\$293,000	\$293,000	\$293,000	\$293,000	\$293,000	\$393,000	\$343,000	\$295,000	\$350,000
Quartzite (\$12.95/ton)	\$652,000	\$622,000	\$859,000	\$709,000	\$1,120,000	\$819,000	\$2,160,000	\$1,260,000	\$1,680,000	\$1,350,000
Salt (\$0.53/lb)	\$1,500,000	\$586,000	\$7,440,000	\$2,930,000	\$14,800,000	\$5,780,000	\$44,200,000	\$17,300,000	\$19,100,000	\$20,100,000
Energy Consumption for Salt Melting	\$10,000	\$4,000	\$46,000	\$18,000	\$95,000	\$36,000	\$300,000	\$110,000	\$135,000	\$145,000
Direct Subtotal	\$17,197,000	\$8,390,000	\$39,051,000	\$21,104,000	\$62,389,000	\$31,932,000	\$161,365,000	\$77,456,000	\$78,190,000	\$87,880,000

Table G-1
Capital Costs for Thermocline Tank Designs (continued)

Thermocline Design Estimate										
Design Case	100 MWh, Indirect Trough	100 MWh, Direct Central Receiver	500 MWh, Indirect Trough	500 MWh, Direct Central Receiver	1000 MWh, Indirect Trough	1000 MWh, Direct Central Receiver	3000 MWh, Indirect Trough	3000 MWh, Direct Central Receiver	1500 MWh, Indirect Trough	3500 MWh, Direct Central Receiver
Indirect										
Contingency (15% on Direct)	\$2,580,000	\$1,259,000	\$5,858,000	\$3,166,000	\$9,359,000	\$4,790,000	\$24,200,000	\$11,620,000	\$11,729,000	\$13,200,000
Sales Tax (8.75% on material)	\$1,100,000	\$944,000	\$1,660,000	\$1,990,000	\$2,860,000	\$3,580,000	\$7,010,000	\$8,190,000	\$2,970,000	\$3,400,000
Engineering - (3% of direct)	\$516,000	\$252,000	\$1,172,000	\$633,000	\$1,872,000	\$958,000	\$4,850,000	\$2,324,000	\$2,345,000	\$3,640,000
Construction Indirects Including Scaffolding)	\$1,720,000	\$839,000	\$3,905,000	\$2,110,000	\$6,239,000	\$3,195,000	\$15,000,000	\$7,750,000	\$7,819,000	\$8,790,000
Construction Management	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$3,000,000	\$3,000,000	\$1,500,000	\$1,500,000
Indirect Subtotal	\$7,416,000	\$4,794,000	\$14,095,000	\$9,399,000	\$21,830,000	\$14,023,000	\$54,060,000	\$32,884,000	\$26,363,000	\$30,530,000
Total	\$24,613,000	\$13,184,000	\$53,146,000	\$30,503,000	\$84,219,000	\$45,955,000	\$215,425,000	\$110,340,000	\$104,553,000	\$118,410,000
Capital Investment Per MW Production	\$1,476,780	\$791,040	\$637,752	\$366,036	\$505,314	\$275,730	\$430,850	\$220,680	\$418,212	\$202,989
Capital Investment Per kWh, Storage	\$246	\$132	\$106	\$61	\$84	\$46	\$72	\$37	\$70	\$34

Table G-2
Capital Costs for Two-Tank Designs

Design Case	100 MWh _t Indirect Trough	100 MWh _t Direct Central Receiver	500 MWh _t Indirect Trough	500 MWh _t Direct Central Receiver	1000 MWh _t Indirect Trough	1000 MWh _t Direct Central Receiver	3000 MWh _t Indirect Trough	3000 MWh _t Direct Central Receiver	3500 MWh _t Direct Trough
(Tank Diameter - Number of Tanks)	(37' - 2)	(23' - 2)	(82' - 2)	(52' - 2)	(115' - 2)	(73' - 2)	(141' - 4)	(126' - 2)	(136' - 2)
Direct									
Foundations	\$746,000	\$369,000	\$2,780,000	\$1,280,000	\$5,060,000	\$2,270,000	\$14,700,000	\$12,000,000	\$13,870,000
Platforms & Steel	\$1,118,000	\$1,252,000	\$1,095,000	\$1,182,000	\$1,194,000	\$2,112,000	\$1,144,000	\$1,640,000	\$2,400,000
Storage Tanks	\$5,616,000	\$2,555,000	\$13,952,000	\$8,576,000	\$21,341,000	\$13,434,000	\$41,131,000	\$37,026,000	\$45,400,000
Storage Tank Insulation	\$103,000	\$65,000	\$252,000	\$148,000	\$391,000	\$219,000	\$516,000	\$440,000	\$900,000
Heat Exchangers	\$2,930,000	\$-	\$10,100,000	\$ -	\$17,500,000	\$-	\$47,500,000	\$ -	\$-
Quartzite Preheating Equipment	\$ -	\$-	\$ -	\$ -	\$ -	\$-	\$-	\$ -	\$-
Surge Tanks	\$ -	\$-	\$ -	\$ -	\$ -	\$-	\$-	\$ -	\$
Pumps & PCE	\$1,440,000	\$1,410,000	\$2,910,000	\$2,830,000	\$5,120,000	\$4,250,000	\$12,100,000	\$9,450,000	\$10,100,000
Salt Melting System	\$1,690,000	\$1,480,000	\$3,240,000	\$2,020,000	\$3,770,000	\$2,700,000	\$3,770,000	\$3,380,000	\$3,900,000
Interconnecting Piping & Valves	\$513,000	\$463,000	\$944,000	\$1,016,000	\$1,820,000	\$1,383,000	\$6,026,000	\$2,750,000	\$3,400,000
Electrical	\$238,000	\$179,000	\$238,000	\$241,000	\$316,000	\$298,000	\$1,828,500	\$732,000	\$800,000
Instrumentation & Controls	\$293,000	\$293,000	\$293,000	\$293,000	\$293,000	\$293,000	\$393,000	\$343,000	\$350,000
Quartzite (\$12.95/ton)	\$ -	\$-	\$ -	\$ -	\$ -	\$-	\$-	\$ -	\$
Salt (\$0.53/lb)	\$3,030,000	\$1,210,000	\$15,100,000	\$6,060,000	\$30,300,000	\$12,100,000	\$90,800,000	\$36,400,000	\$42,700,000
Energy Consumption for Salt Melting	\$20,000	\$ 8,000	\$94,000	\$37,000	\$190,000	\$80,000	\$210,000	\$120,000	\$150,000
Direct Subtotal	\$17,737,000	\$9,284,000	\$50,998,000	\$23,683,000	\$87,295,000	\$39,139,000	\$220,118,500	\$104,281,000	\$123,970,000
Indirect									
Contingency (15% on Direct)	\$3,620,000	\$2,130,000	\$8,030,000	\$4,500,000	\$13,500,000	\$7,850,000	\$32,600,000	\$15,642,150	\$18,400,000
Sales Tax (8.75% on material)	\$1,500,000	\$850,000	\$3,100,000	\$1,800,000	\$5,400,000	\$3,100,000	\$13,000,000	\$11,600,000	\$13,100,000
Engineering - (3% of direct)	\$725,000	\$425,000	\$1,600,000	\$900,000	\$2,700,000	\$1,600,000	\$6,500,000	\$3,200,000	\$3,700,000

Table G-2
Capital Costs for Two-Tank Designs (Continued)

Design Case	100 MWh _t Indirect Trough	100 MWh _t Direct Central Receiver	500 MWh _t Indirect Trough	500 MWh _t Direct Central Receiver	1000 MWh _t Indirect Trough	1000 MWh _t Direct Central Receiver	3000 MWh _t Indirect Trough	3000 MWh _t Direct Central Receiver	3500 MWh _t Direct Trough
Construction Indirects (Including Scaffolding)	\$2,450,000	\$3,870,000	\$6,120,000	\$6,400,000	\$5,130,000	\$3,950,000	\$9,240,000	\$12,700,000	\$13,500,000
Construction Management	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$1,500,000	\$3,000,000	\$3,000,000	\$3,000,000
Indirect Subtotal	\$9,795,000	\$8,775,000	\$20,350,000	\$15,100,000	\$28,230,000	\$18,000,000	\$64,340,000	\$46,142,150	\$51,700,000
Total	\$27,532,000	\$18,059,000	\$71,348,000	\$38,783,000	\$115,525,000	\$57,139,000	\$284,458,500	\$150,423,150	\$175,670,000
Capital Investment Per MW Production	\$1,651,920	\$1,083,540	\$856,176	\$465,396	\$693,150	\$342,834	\$568,917	\$300,846	\$301,149
Capital Investment Per kWh _t Storage	\$275	\$181	\$143	\$78	\$116	\$57	\$95	\$50	\$50

Table G-3 provides a summary of the total installed capital costs on a per-kilowatt-hour-thermal basis.

Table G-3
Molten Salt Thermocline Storage Capital Costs

Storage Capacity (MWh)	Technology	Maximum TES Temperature	Thermocline Cost (\$)	Cost Per Unit Capacity (\$/kWh)
100	Indirect Trough	400°C (752°F)	24,613,000	246
	Direct Central Receiver	560°C (1040 °F)	13,184,000	132
500	Indirect Trough	400°C (752°F)	53,146,000	106
	Direct Central Receiver	560°C (1040 °F)	30,503,000	61
1000	Indirect Trough	400°C (752°F)	84,219,000	84
	Direct Central Receiver	560°C (1040 °F)	45,955,000	46
1500	Indirect Trough	400°C (752°F)	104,553,000	70
	Direct Central Receiver	560°C (1040 °F)	66,000,000*	44*
3000	Indirect Trough	400°C (752°F)	215,425,000	72
	Direct Central Receiver	560°C (1040 °F)	110,340,000	37
3500	Indirect Trough	400°C (752°F)	254,333,333*	73*
	Direct Central Receiver	560°C (1040 °F)	118,410,000	34

* Values were interpolated or extrapolated from other design case results

For comparison, estimates were prepared for an equivalent set of two-tank thermal energy storage systems. These estimates are provided in Table G-4.

Table G-4
Two-Tank Molten Salt Thermal Storage Capital Costs

Storage Capacity (MWh _t)	Technology	Maximum TES Temperature	Two-Tank Cost (\$)	Cost Per Unit Capacity (\$/kWh _t)
100	Indirect Trough	400°C (752°F)	27,532,000	275
	Direct Central Receiver	560°C (1040 °F)	18,059,000	181
500	Indirect Trough	400°C (752°F)	71,348,000	143
	Direct Central Receiver	560°C (1040 °F)	38,783,000	78
1000	Indirect Trough	400°C (752°F)	115,525,000	116
	Direct Central Receiver	560°C (1040 °F)	57,139,000	57
1500	Indirect Trough	400°C (752°F)	166,500,000*	111*
	Direct Central Receiver	560°C (1040 °F)	82,500,000*	55*
3000	Indirect Trough	400°C (752°F)	284,459,000	95
	Direct Central Receiver	560°C (1040 °F)	150,423,000	50
3500	Indirect Trough	400°C (752°F)	312,991,000*	89*
	Direct Central Receiver	560°C (1040 °F)	175,670,000	50

* Values were interpolated or extrapolated from other design case results

It can be seen from this information that the thermocline system offers the lowest installed capital cost at each design capacity. The average savings for the thermocline system is approximately \$25 per kWh_t, which is an average reduction of 24%. For the largest single-tank designs, the savings were 37% for the indirect trough 1500 MWh_t case and 33% for the direct central receiver 3500 MWh_t case. It can also be seen that the direct TES systems are less expensive than indirect thermal energy storage due to the higher charging temperatures, which minimizes the required storage volume. This information is presented graphically in Figure G-3.

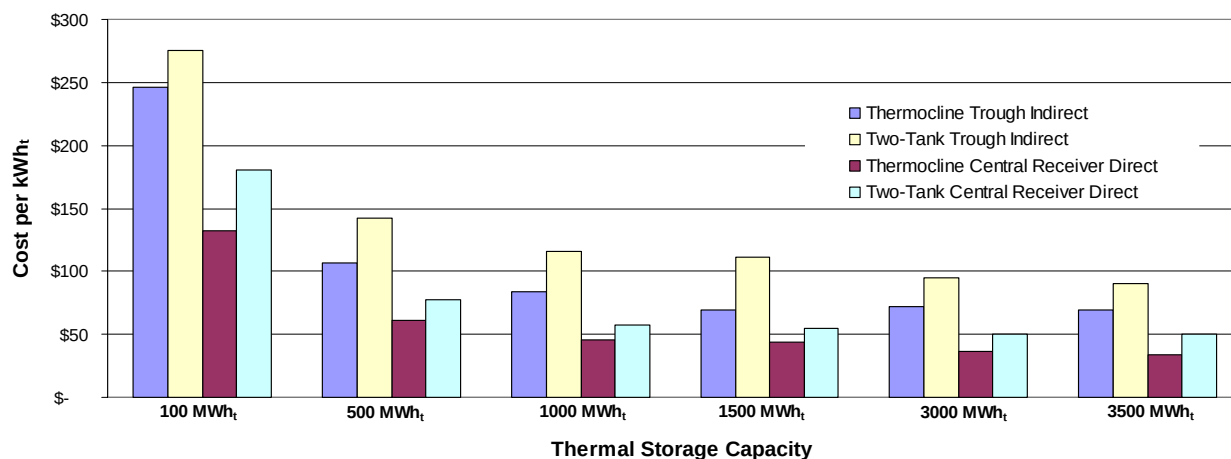


Figure G-3
Molten Salt Energy Storage Capital Cost Estimates

It can be seen from Figure G-3 that the direct thermocline central receiver offers the most favorable economics across the range of system sizes, with an installed cost of \$34-\$132/kWh_t. The second most economical option is the direct two-tank central receiver, which has an installed cost of \$50-\$181/kWh_t. The third most economical option is the indirect thermocline parabolic trough design, with a cost of \$70-\$246/kWh_t. The least economical option is the indirect two-tank parabolic trough design, with a price of \$89-\$275/kWh_t.

The main cost advantage for the thermocline system is the substitution of quartzite rock for relatively expensive molten salt. The thermoclines requires roughly half as much salt as the two-tank systems.³¹ For Case 7, this amounts to a savings of \$45 million relative to the two-tank design.

It was also found that direct TES systems are less expensive than indirect systems. This is expected because direct TES does not require oil-to-salt heat exchangers to transfer heat from the working fluid to the storage fluid, and the higher temperature of the direct central receiver system greatly decreases the size of the TES volume required for a given storage capacity. In Case 7 the heat exchangers amount to almost \$48 million, which is 30% of the total direct cost. These heat exchangers are not required in the central receiver design, which can use the molten salt directly without the need for a synthetic oil system. In addition, the cost of a TES system for a given storage capacity depends on the operating temperature of the CSP technology. The size of the storage system is directly proportional to the temperature difference. CSP technologies with a greater differential in the hot storage charging temperature and the cold return temperature will require a smaller volume to store the same amount of energy. It follows that parabolic trough storage operating at 400°C (752°F) requires roughly 2.5 times more storage volume than tower storage operating at a charging temperature of 560°C with a nominal cold return

³¹ Although the void fraction is 0.25, the salt has about 20% higher volumetric heat capacity than the quartzite filler. Thus more total volume is required to store the same amount of energy in the thermocline than in the hot tank of the two tank system. There is also some volume of the thermocline system that is free of quartzite filler, above the filler in the thermocline and in the surge tanks and interconnecting piping.

temperature of 290°C for both systems. The cost estimate results show that on average, across the range of sizes, the capital cost for the direct central receiver is 54% of the cost of the indirect parabolic trough.

A significant capital investment for the thermocline designs is the storage tank. The current material selected by EPRI for this tank is 347 stainless steel. However, a detailed material analysis may show that other metals are acceptable for this service. For comparison purposes, cost estimates have been prepared for tanks made of 304 SS, 347 SS, and Inconel 625. This comparison is presented in Appendix I.

Capital cost curves have been produced in Figure G-4 for the full set of cost data, which show the installed cost per MW_t as a function of storage size. It can be seen from this graph that the installed cost decreases as the capacity of the TES increases. This is expected, as most process equipment benefits from economies of scale. The greatest cost savings occurs between 100 and 500 MWh_t of storage. In this range the capital cost per kWh_t decreases nearly 60%. It is important not to extrapolate the costs beyond the limits of the given curve. Beyond 3500 MWh_t of production it may be necessary to build more thermocline tanks, which would increase the cost per MWh_t . Below 100 MWh_t it may be difficult to purchase equipment, which could drive up costs more significantly than a simple extrapolation may suggest. The curves presented in Figure G-4 are only a guideline based on a conceptual project, and a detailed cost estimate should be performed for actual project planning purposes.

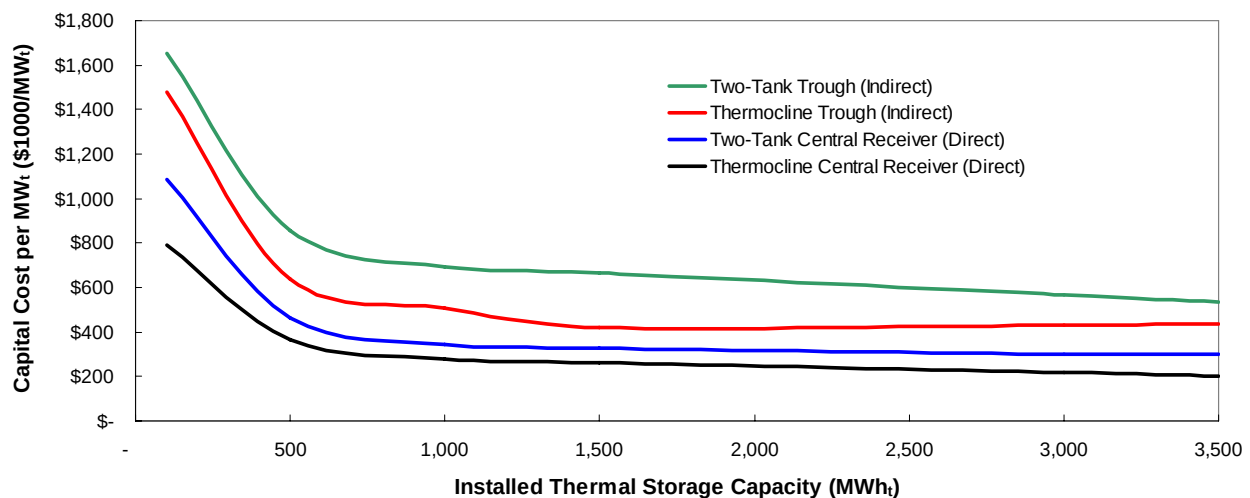


Figure G-4
Molten Salt Thermal Energy Storage Capital Cost Estimates as a Function of Installed Capacity, $\$/kWh_t$

H

THERMOCLINE SURGE TANK

Thermocline Surge Tank

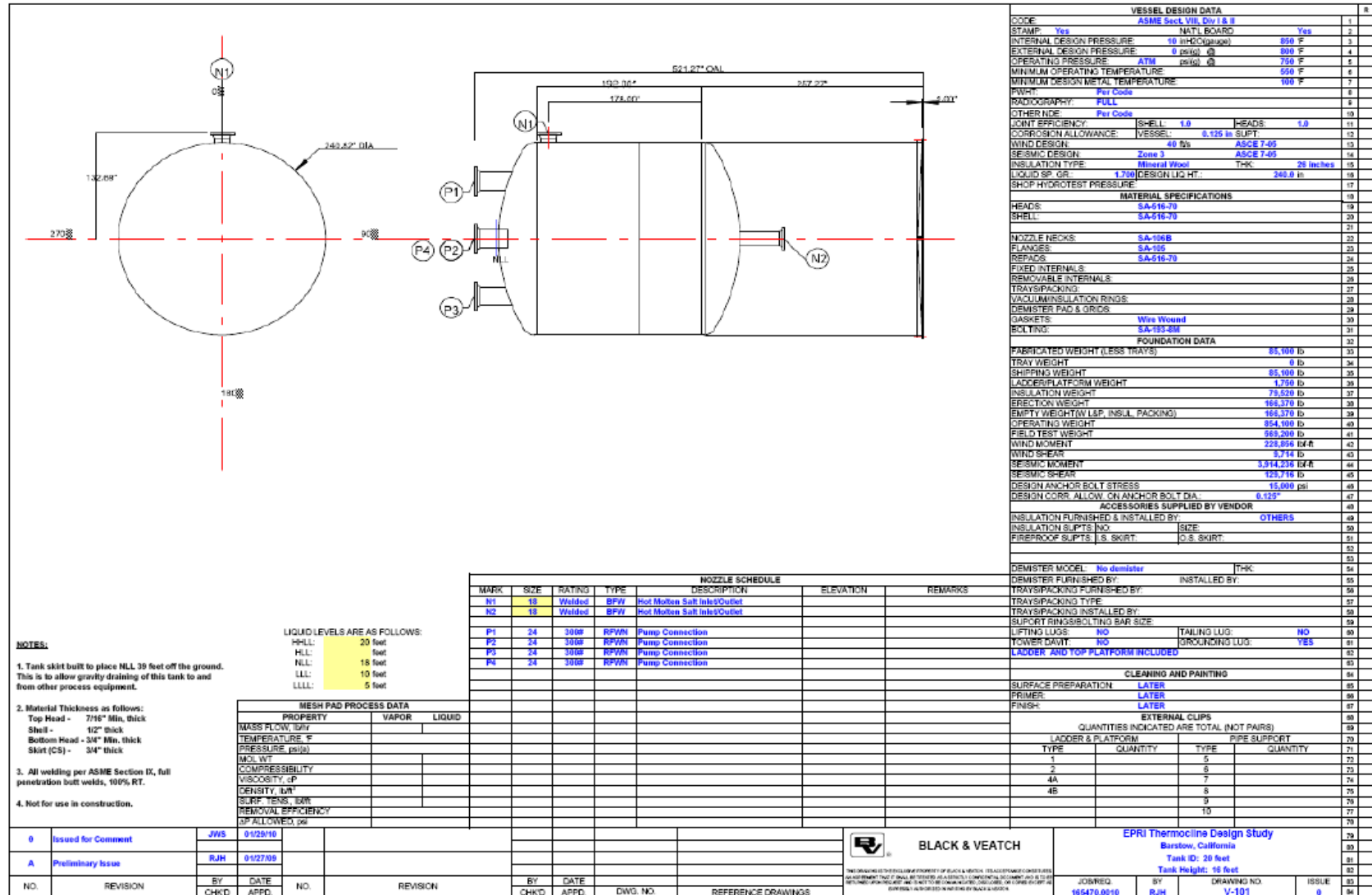


Figure H-1
Thermocline Surge Tank

TANK ALTERNATIVE MATERIAL COSTS

An important criterion in the design of the thermocline system is the selection of an appropriate material for the tank. The low temperature thermocline tanks (below 850°F (454°C)) have been designed using a high temperature carbon steel. For the high temperature thermocline tanks (above 850°F), three materials were considered, SS304, SS347, and Inconel 625. For preliminary cost comparison purposes, the estimated costs for the tanks using each of the materials is shown in Table I-1. These costs are for comparison purposes only, and final material selection should be made at the conclusion of a materials testing program.

Table I-1
Comparison of Material Costs

Design Case	Inconel 625	SS 347	SS 304
2	6,270,000	1,700,000	1,200,000
4	16,300,000	7,900,000	5,000,000
6	31,500,000	11,700,000	7,900,000
8	73,100,000	31,100,000	19,800,000

J

MAXIMUM TANK CAPACITY CALCULATIONS

The current thermocline design capacity is based on a soils analysis performed in Barstow, California. The original eight thermocline design cases assume that a standard foundation type will be used in the construction of the tanks. The soils analysis in Appendix D provided a maximum bearing pressure of the soil, which limited the maximum height of the thermocline tank to 47 ft (46 ft liquid height) for the largest tanks with standard foundations. Taller tanks exert more force on the soil, and if the bearing pressure is exceeded the foundation may not support the tank. To determine the maximum single tank capacity, oversized foundations were assumed to reduce the maximum soil bearing pressure by 10%. It was determined that the tanks could be as tall as 50 ft with the oversized foundations. This increased height would increase the maximum storage capacity of the tanks by approximately 10% to about 1500 MWh_t for the indirect trough and 3500 MWh_t for the direct central receiver. The maximum storage capacities of these optimized tanks are shown in Table J-1.

Table J-1
Maximum Storage Capacities

Tank Diameter (ft)	Tank Height (ft)	Storage Temp (°C)	Capacity (MWh)
160	46	560	3374
160	46	400	1352
160	50	560	3695
160	50	400	1481

The potential economic benefit of this option will have to be analyzed on a case by case basis considering the thermal capacity targets and costs at specific sites. For the additional design cases (9-12) located in Barstow, California, cost estimates were developed in Appendix G.

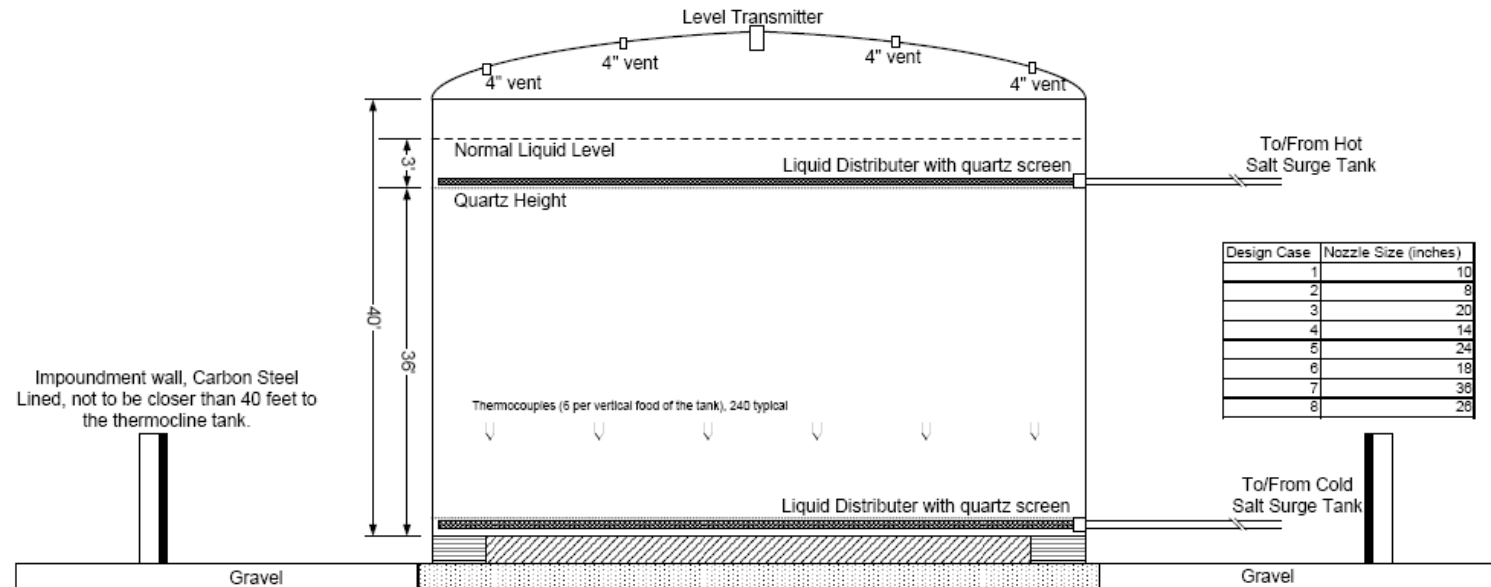
K

THERMOCLINE DESIGN DETAILS

K-2

L

TANK MECHANICAL DIAGRAM



FOR ESTIMATE ONLY
NOT FOR USE IN CONSTRUCTION

 BLACK & VEATCH	Tank Mechanical Diagram	EPRI TES Study	Revision: A	Project Number 165470	Notes: Shell thickness depends on the design case. See tank data sheets for complete design information.
			1/29/2010	Phase 0010	

Figure L-1
Tank Mechanical Diagram

M

ACRONYMS

ASME – American Society of Mechanical Engineers

BOP – balance of plant

CLFR – compact linear Fresnel reflector

CSP – concentrating solar power

DNI – direct normal irradiation/insolation

DOE – U.S. Department of Energy

EPC – Engineer – Procure – Construct

HCE – heat collection element

HTF – heat transfer fluid

kW – kilowatt

kWh_t – kilowatt-hour thermal

LCOE – levelized cost of electricity

MW – megawatt

MWh_t – megawatt-hour thermal

NREL – National Renewable Energy Laboratory

PFD – process flow diagram

P&ID – piping and instrumentation diagram

PSIA – pounds per square inch absolute

SCA – solar collector assembly

Acronyms

SEGS – Solar Electric Generating Station

SSE – Safe Shutdown Earthquake

TES – thermal energy storage

TMY – typical meteorological year

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Electric Power Research Institute

3420 Hillview Avenue, Palo Alto, California 94304-1338 • PO Box 10412, Palo Alto, California 94303-0813 USA
800.313.3774 • 650.855.2121 • askepri@epri.com • www.epri.com